

SECOND INTERNATIONAL CONFERENCE  
ON STABILITY OF  
SHIPS AND OCEAN VEHICLES

[STABILITY '82]

TOKYO, OCTOBER 1982



THE SOCIETY OF NAVAL ARCHITECTS OF JAPAN



**SECOND INTERNATIONAL CONFERENCE  
ON  
STABILITY OF SHIPS AND OCEAN VEHICLES  
  
STABILITY '82**

**OCTOBER 24-29, 1982  
SASAKAWA HALL, TOKYO**

**PROCEEDINGS**

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## *Opening Session*

*Chairman*

Dr. Kazuo Sugai  
Ship Research Institute  
Japan

## Welcome and Introductory Addresses

### Address by Masao Mizushina

*Chairman of the Organizing Committee of the Conference*

Good morning, Ladies and Gentlemen, I am honored to welcome you all to this Second International Conference on Stability of Ships and Ocean Vehicles.

We have 224 participants here today, who are all prominent experts in stability. 74 of them have come from 16 different countries outside Japan and from International Maritime Organization.

Fifty five valuable papers will be presented to this Conference. Earnest and interesting discussions over those papers are expected. We are confident that the Conference will be fruitful and will contribute to the promotion of exchange of ideas and experience related to various aspects of stability among the experts in the world, and to the development of science and technology in those fields.

We would particularly like to wish foreign delegates a pleasant and enjoyable stay in Japan. I personally hope that you could have chances of

having any new experience and observation of the culture and life of Japan. The more knowledge of the other country will help us to understand more each other over the borders.

May I record my deep appreciation and thanks to Ministry of Transport of Japan, Japan Shipbuilding Industry Foundation, Nippon Kaiji Kyokai, the Shipbuilders' Association of Japan, and the Japanese Shipowners' Association for their assistance and cooperation in holding and organizing this Conference.

I particularly extend my gratitude to Mr. Ryoichi Sasakawa, who is Chairman of Japan Shipbuilding Industry Foundation, and gave us generous financial support for this Conference.

Allow me to conclude by wishing all of you the very successful Conference

Thank you.

### Address by Seizo Motora

*Deputy Chairman of the Organizing Committee of the Conference  
Chairman of the International Technical Committee of the Conference*

Mr. Mizushina, ladies and gentlemen, on behalf of the Society of Naval Architects of Japan, I would like to express my hearty welcome to all the participants to this conference.

Stability has long been a problem of vital im-

portance ever since the first ship appeared on the Earth and will continue to represent a most important problem as long as ships and offshore floating structures are in existence.

It is of significant importance to evaluate

stability of ships at the design stage, and to reflect the results of studies on stability in the criteria or standards for making judgements on stability both in view of design and of the operation of ships.

For this purpose, international gatherings of people dealing with the stability of ships and the exchange of information and views will be very useful.

The Stability '82 is a follow up of the International Conference on Stability of Ships and Ocean Vehicles, organized by Professor Kuo and held in 1975 at the University of Strathclyde. The aim of this conference is exactly the same as that of the first conference, that is: a) to provide an opportunity for those involved in stability activities, including design, operation, research or regulatory purposes, to discuss the available research findings at an international level; and b) to see how these results can be applied in actual usage.

Seventy two abstracts were submitted for this conference and 55 papers were chosen for presentation based on the instructions from the International Technical Committee.

Simple classification of the 55 papers is as follows:

- 1) 3 on philosophy
- 2) 10 on theory
- 3) 4 on experimental techniques
- 4) 3 on damage stability
- 5) 11 on fishing vessels and special ships
- 6) 8 on offshore structures
- 7) 8 on criteria and regulations
- 8) 7 on quartering and following seas
- 9) 2 on the effects of water on-deck
- 10) 2 on roll stabilization

It can be noticed that the papers submitted to this conference have the following characteristics:

- 1) There are 8 papers dealing with stability and

the survival capability of offshore structures, showing growing attention regarding floating offshore structures reflecting attention to the disasters as Alexander Keeland and Ocean Ranger.

- 2) There are 7 papers dealing with stability in quartering and following seas reflecting increasing interest in this phenomenon.
- 3) There are 11 papers dealing with fishing vessels and special ships.
- 4) There are two theoretical papers dealing with numerical simulation of capsizing.

Such characteristics seem to indicate the recent trend of research activities in this field, and are both interesting and promising.

Since the number of papers was unexpectedly large, parallel sessions have been unavoidably arranged during the 2nd and 3rd days. However, the parallel sessions are arranged in such a manner that the theoretical and experimental papers are presented at the same time so that the participants may be able to choose either one with less inconvenience. Nevertheless, the scheduling of the parallel sessions may result in inconveniences to certain participants. The Organizing Committee would like to ask for the participants' understanding regarding this unavoidable situation.

In the afternoon of 1st and 4th days, panel discussions are provided to invite frank and constructive discussion.

The Organizing Committee is much indebted to the International Technical Committee in the selection of papers and program arrangement. On behalf of the Organizing Committee, I would like to express my sincere appreciation to the members of the ITC; namely, Prof. C. Kuo, Mr. I.A. Manum, Prof. F. Ursell, Prof. N. Rakhmanin, Prof. O. Krappinger, Prof. J. Gerritsma, Prof. J.R. Paulling, Dr. K. Kure, Prof. J. Duziak, and Prof. S. Takezawa.



Since the previous conference, progress has been made in various stability-related areas as will be seen in the presentation of papers for this conference. And the IMO, through its subcommittee on subdivision and stability, has prepared the Criteria for Small Vessels (Resolution A 167), and is currently in the process of preparing weather criteria based on more physical significance. Also, the survival capability at dangerous situations for ships is under review as one of the long-term

programs. At this conference, several papers will be presented which cover the items included in this long-term program.

I would like to conclude my remarks by expressing my hope that the results of this conference would be used effectively by the designers and operators, and would also be of assistance to the activities of the IMO.

Thank you.

*Session I*

## General Studies

*Chairmen*

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Mr. Ivar A. Manum  
Norwegian Maritime Directorate  
Norway

SI-1

## THE CAPSIZING OF "ALEXANDER L. KIELLAND"

SIGMUND RUSAAS

Det norske Veritas

Norway

### ABSTRACT

This paper deals with the following aspects related to the capsizing of the hotel platform "Alexander L. Kielland":

- A short review of the design features of the Pentagone platform and the circumstances of the accident.
- Static analysis of the capsizing. Theoretical calculations of the capsizing after loss of leg D. The calculations are carried out:
  - 1) in still water without the effect of anchor forces, wind, waves and current.
  - 2) including the effect of anchor forces and wind over-turning moment.The study is undertaken by stepwise filling of the deck structure and trunks in legs C and E. Calculations of the stability and floating position in the various phases are made.
- Study of the survival capabilities of a Pentagone rig after loss of one of the legs with particular emphasis on possible equilibrium by ballast redistribution.
- Evaluation of present stability criteria in light of the accident. Special attention is drawn to damage stability criteria. Present survival criteria deal only with "low impact" damage, and the consequences of "higher impact" damage may be serious for most of the rigs in service today.

### 1. INTRODUCTION

On 27th March 1980 at about 6:30 p.m., one of the supporting columns of the hotel platform "Alexander L. Kielland" broke off when the platform was in service on the Ekofisk field in the North Sea. The failure led to capsizing of the platform in about 20 minutes. Out of the 212 men on board, 123 lost their lives. The accident was one of the most severe in the offshore oil industry worldwide, and without comparison the most severe in the North Sea since the oil exploration started in the early 1970's. Great effort has been laid down by the governmental authorities and the classification society, Det norske Veritas, in order to investigate the reasons behind this tragic accident. In light of the findings from this work, all safety aspects of the activities on the Norwegian Continental Shelf were re-appraised, resulting in a number of new and amended rules and regulations. This paper presents some of the findings regarding stability of mobile offshore units, with special emphasis on major damage like the one which struck "Alexander L. Kielland".

### 2. DESIGN FEATURES OF THE PENTAGONE PLATFORM AND THE CIRCUMSTANCES OF THE ACCIDENT

"Alexander L. Kielland" was originally built as a drilling rig, but had always served as an accommodation platform. Since delivery from the yard in 1976, there had been four modifications in which new accommodation containers were mounted on the deck, forward of the drilling tower. After the last addition, the unit could accommodate 348 persons.

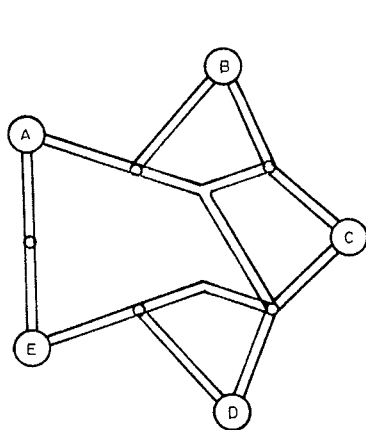
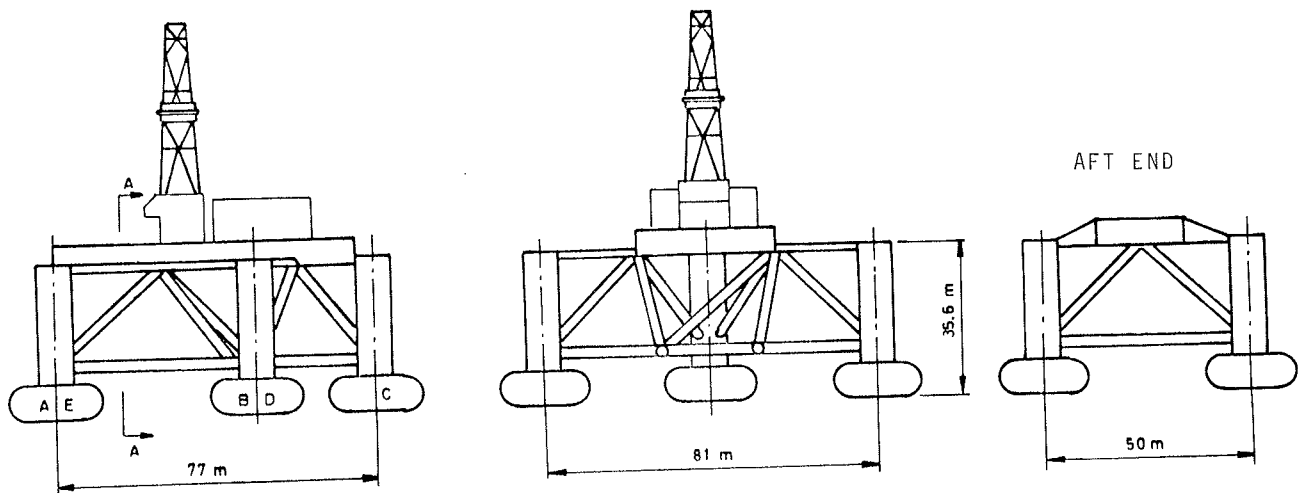


"Alexander L. Kielland" is of the so-called "Pentagone" type, developed in France between 1965-1970. The buoyancy comes from five individual columns/pontoons, supported by a system of bracings (Fig.1). The lower horizontal bracings are free-flooded, the other ones are watertight. In each pontoon there are six tanks (Fig. 2). Tank no. 1 is a void space, the others are mainly used as ballast tanks. The columns contain 5 tanks each, where the lowermost tank (tank no.7) is used for ballast. The others are mainly dry tanks. In the middle of each column there is an access trunk leading to a pump room inside of the tanks in the pontoon. The entrance to this trunk is through a watertight door on top of the column. This door was only allowed to be opened for passage. There are also ventilator inlets/

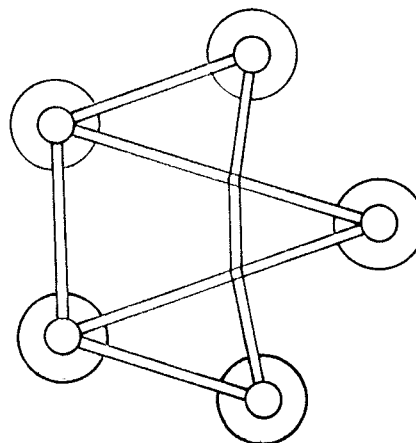
outlets fitted with weathertight covers. When the draught exceeded 18 metres, or in bad weather, the ventilator covers should be closed.

The deck structure consists of two parts, subdivided by the open moon-pool area (Fig. 3). Each of these areas is again subdivided horizontally by an intermediate deck. Access to the deck structure is from the upper deck and through doors in the transverse bulkheads. In the lower deck there are a lot of scuppers, exhaust and other ventilators, and a garbage chute. All of these openings are fitted with closing appliances which may be accepted as weathertight, but not as watertight. It is not known, however, to which extent these closing appliances were utilized at the time of the accident.

#### SECTION A-A



OBLIQUE AND UPPER BRACINGS



LOWER BRACINGS

Fig. 1 General lay-out of "Alexander L. Kielland"

In the period before the accident, "Alexander L. Kielland" was anchored next to the production platform "Edda 2/7 C". It was anchored by two anchor wires to each of the columns A, B, D and E. Column C was not anchored. Fig. 4 shows the anchor pattern used at the last anchoring at "Edda 2/7 C". Normally, the connection between the platforms was maintained by a gangway. In bad weather, however, the gangway was hoisted on board "Alexander L. Kielland" and the platform was shifted somewhat away from "Edda 2/7 C". This was also the case on the day of the accident, Thursday 27th March 1980. The visibility was poor, the wind velocity about 16 to 20 metres per second, and the wave heights about 6-8 metres. The shifting of the platform was finished at approximately 5:50 p.m. About half an hour later, a few minutes before 6:30 p.m., column D broke off.

The failure started with a fatigue crack in bracing D-6 (Fig. 5). The five other bracings connecting column D to

the platform subsequently failed from overloading, resulting in a total loss of column D. The platform almost immediately heeled over to an angle of 30-35 degrees. From this position it continued slowly to heel and sink, until it capsized after about 20 minutes. When the platform heeled, the top of columns C and E were submerged. It is thought that downflooding took place through doors/ventilators leading to the trunk in the middle of these columns. Some of the oblique bracings and the dry tank on top of column E could also be filled. This filling alone, however, is not sufficient to capsize the platform. As shown later, more than half of the deck space would then have to be filled with water. Considering the relatively short time (20 minutes) it took to capsize the platform, flooding must have occurred not only through open drain valves etc. in the lower deck, but also through doors and ventilators. There is also a possibility that flooding into the deck area took place through openings caused by structural damage.

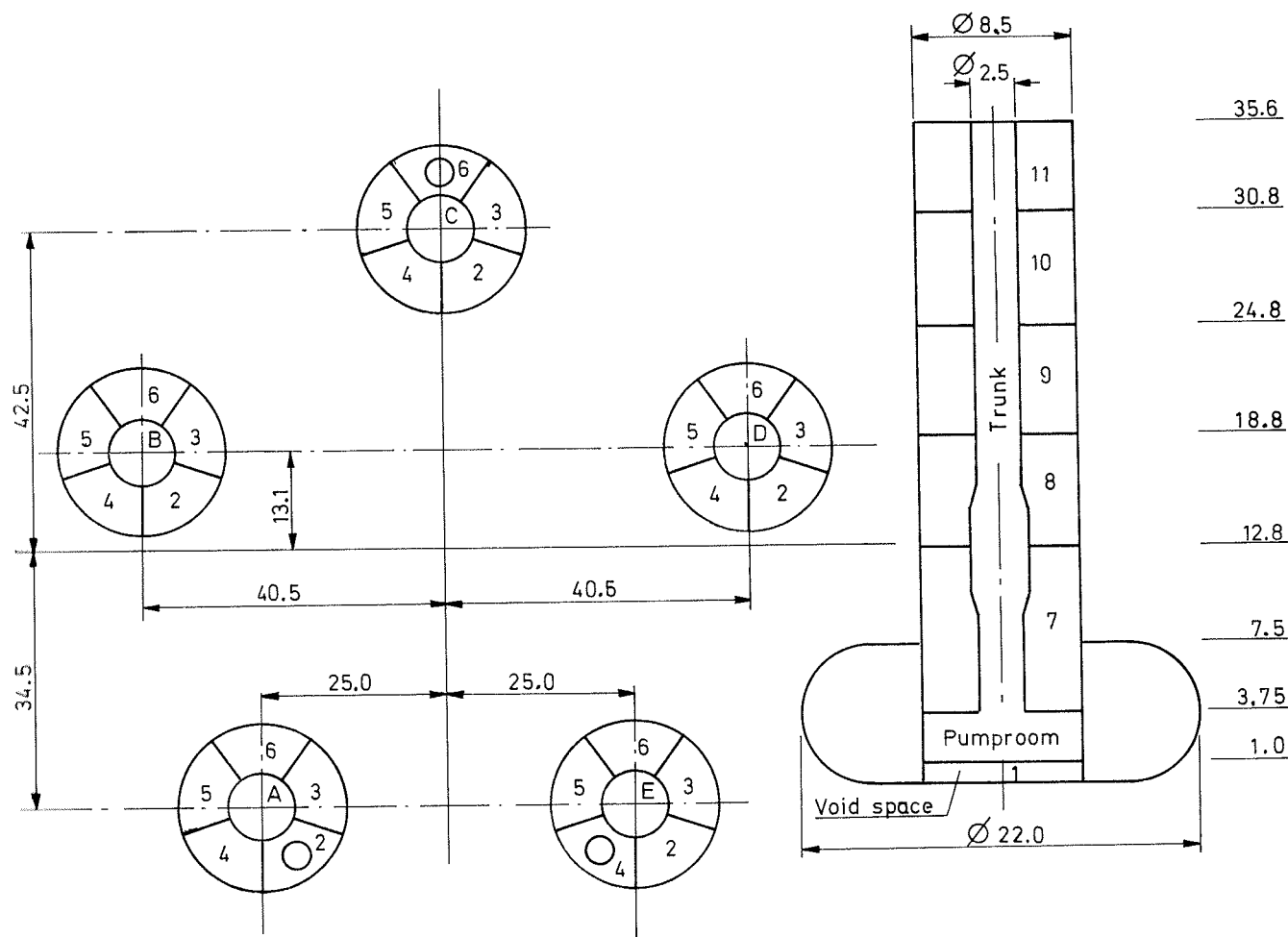
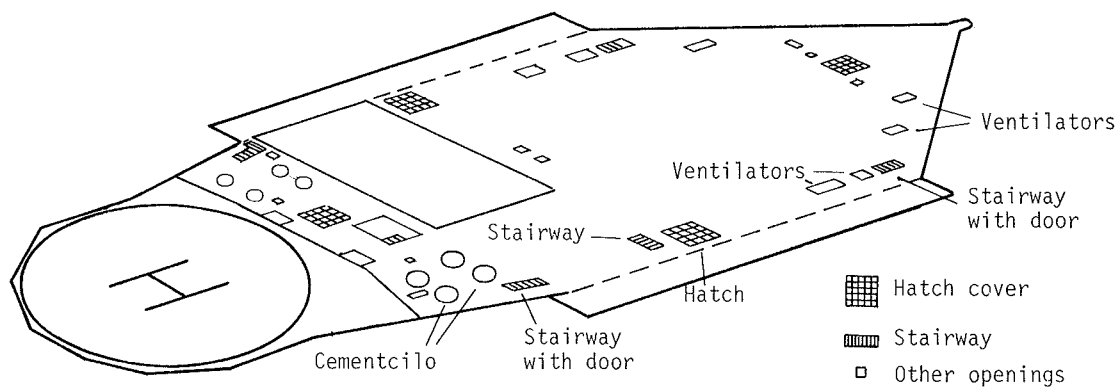
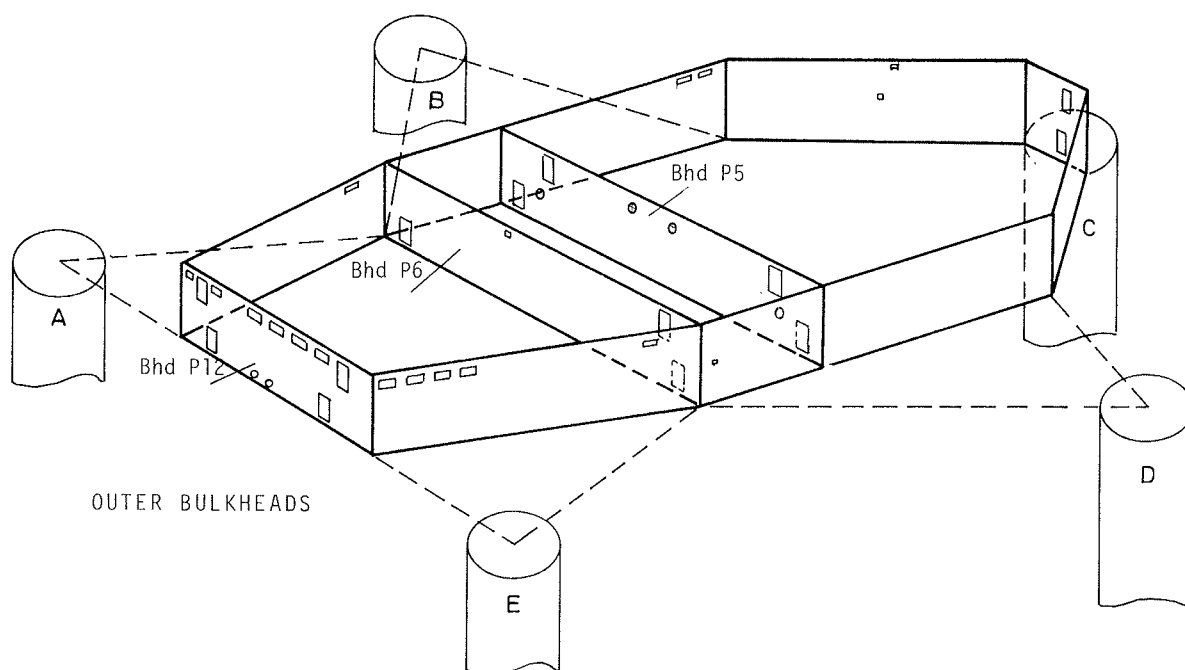


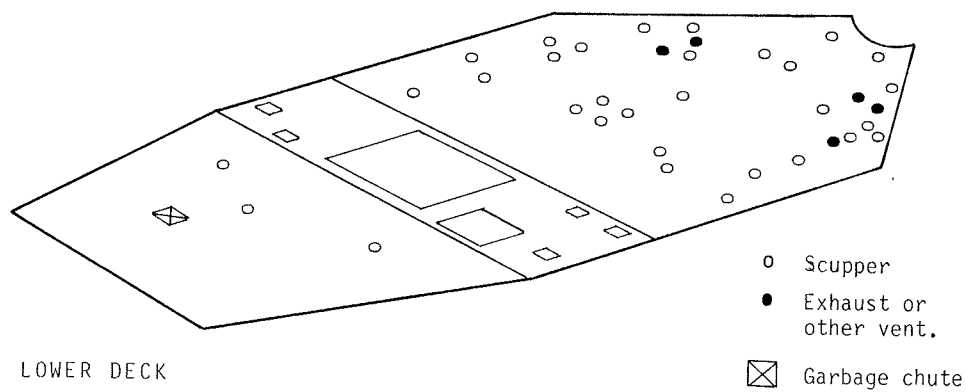
Fig. 2 Tank arrangement



UPPER DECK



OUTER BULKHEADS



LOWER DECK

Fig. 3 Deck structure of "Alexander L. Kielland"

There has been a lot of speculation whether the access doors leading into the deck structure were closed or not, especially those leading from the moon-pool area. If the doors were open, the flooding of the deck structure would certainly take place very fast. It is not clear, however, to which extent the doors should have been closed at the time of the accident. In the Operating Manual, the doors in bulkheads P5, P6 and P12 were required to be kept closed in storm condition or after damage. The wind speed on the day of the accident was close to what one could call a storm condition, but it is a question whether the words "storm condition" could be taken strictly literally. The background for the requirement to close these doors was mainly not

to maintain buoyancy, since all formal stability criteria were fulfilled without buoyancy from the deck structure. The reason was primarily to avoid water penetrating into the deck structure due to heavy sea. The access doors leading to the trunk in the middle of the columns, however, were required permanently closed at sea and only allowed to be opened for passage. The reason for this requirement was that these doors were submerged in damaged condition according to the formal stability calculations. The doors leading from the upper deck into the deck structure should preferably be closed in storm condition. This also applies to hatches in the upper deck. For the doors at column C, however, there were no requirement as to closing. The scuppers in the lower

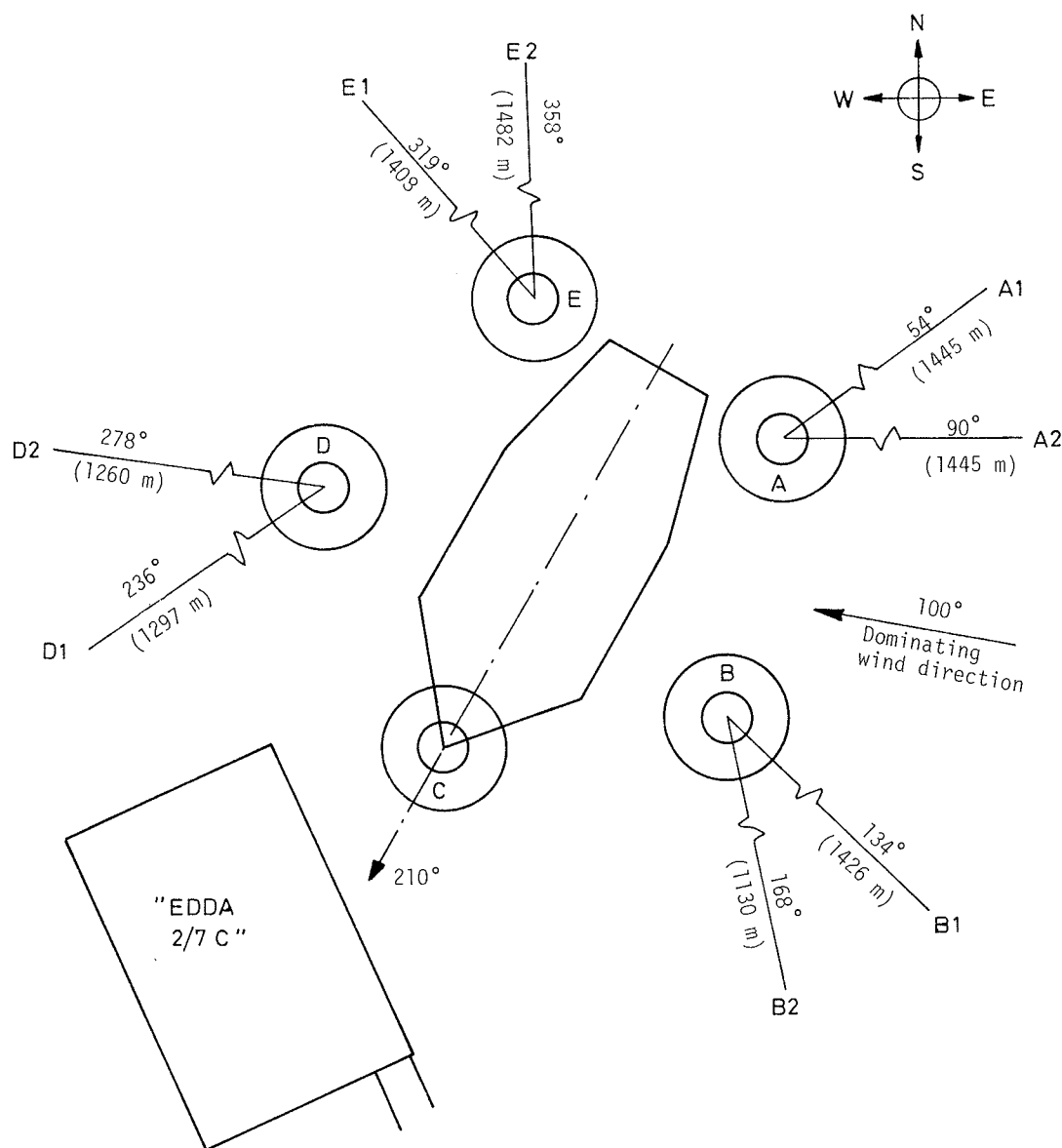


Fig. 4 Anchor pattern used at the last anchoring near "EDDA 2/7 c"

deck and two of the ventilators in bulkhead P5 should also be kept closed in storm condition or after damage. The information about the utilization of the closing appliances at the time of the accident is incomplete, and it has not been possible by diving surveys to verify the status of most of the openings leading to the deck structure. For the doors/hatches on top of the columns, however, diving surveys have revealed that some of them were open, others were indented due to the water pressure during the capsizing. The flooding through these openings, however, is found to be of minor importance for the capsizing.

The capsizing was only to a limited degree counteracted by the mooring system. It is assumed that the mooring system delayed the capsizing with only a few minutes. Nor was it possible to prevent the capsizing by redistribution of ballast, because the power supply failed immediately after the platform heeled over.

### 3. STATIC ANALYSIS OF THE CAPSIZING

The purpose of this analysis is to find the theoretical course of events when "Alexander L. Kielland" capsized due to the loss of leg D. The study is undertaken by stepwise filling of the deck structure and trunks in legs C and E. For each step, the stability and floating position of the platform is calculated. The calculations are performed:

- 1) in still water without the effect of anchor forces, wind, waves and current.
- 2) including the effect of anchor forces and wind moment.

The calculations show that the platform capsized due to progressive flooding in the deck structure. The capsizing occurred when the filling reached about 75%-80% of the available space.

During the capsizing, the following parts of the structure are assumed flooded (Fig. 6):

- Part "A": Aft part of the deck structure.
- Part "F": Fore part of the deck structure plus trunks in legs C and E.

Flooding of part "F" must have commenced immediately after the initial heeling. Flooding into part "A", however, is believed to have started somewhat later. It is found reasonable that part "F" reached a filling of 20% before flooding starts into part "A". The calculations are performed for the following stages of flooding:

Part "F"	Part "A"
0%	0%
20%	0%
40%	25%
60%	50%
80%	75%

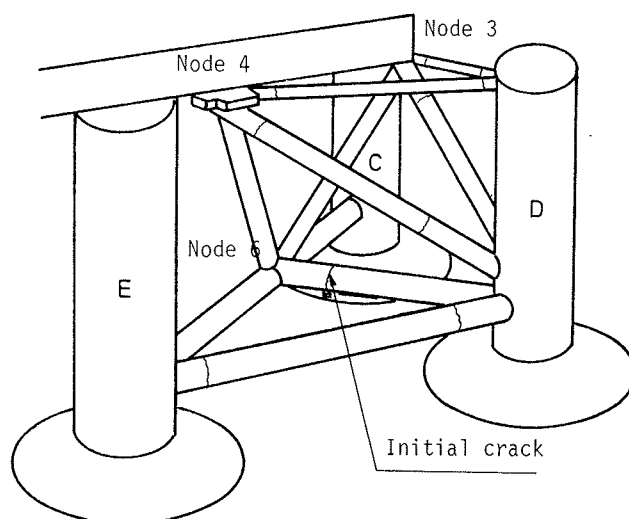


Fig. 5 Fractures in the bracings connecting column D.

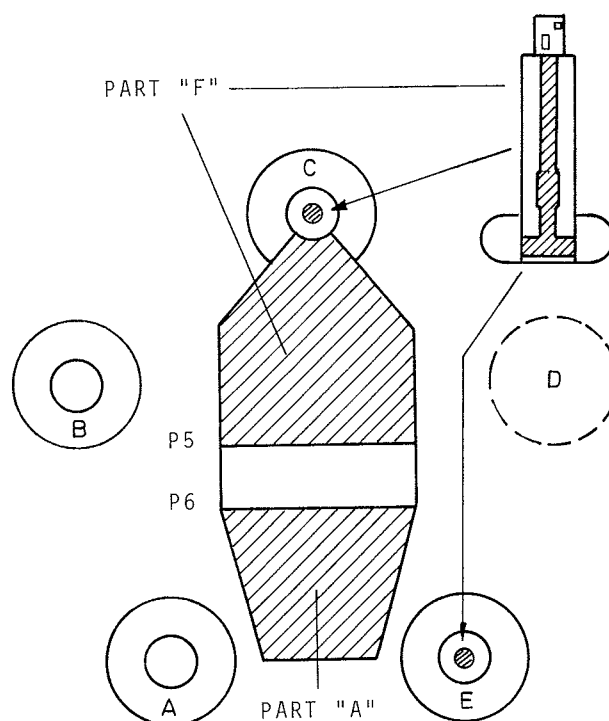


Fig. 6 Part of the structure assumed flooded during the capsizing

The results of the analysis are summarized in Table 1, showing the floating position and metacentric height for the various stages of flooding. The angle of heel and metacentric height versus flooding are shown in Fig. 7. As shown by the results, the effects of anchor forces and wind moment are rather minor. The calculations are carried out using the computer program NV223 (Ref. 2). Loss of buoyancy when column D is lost is simulated by damage with permeability = 1,0. The flooding of the deck area is simulated by damage with varying permeability factors, e.g. for a stage of flooding equal to 20% a permeability factor of 0,2 is used. The loading condition used in the calculations is taken from information regarding the operation condition on the day of the accident. These data indicate a draught of 21 metres and a vertical center of gravity of about 17,6 metres. The data are encumbered with some uncertainties, however, since the rig's master perished and the log was not found.

During the capsizing several items of equipment were lost in addition to leg D. In this calculation, however, only leg D and equipment pertaining to that leg and ballast in the pontoon are assumed lost. These items are summarized in Table 2. Fig. 8 shows the rig in equilibrium position after the loss of leg D when the deck structure and trunks are intact. Fig. 9, curve 0, shows the corresponding stability curve. Curve 1 through

4 are the stability curves for varying stages of flooding. The calculations reveal that for intact deck structure the stability is extremely good, with positive GZ almost to 90 degrees. Also for a considerable amount of water in the deck structure, the range of positive GZ remains almost the same. Even with flooding of about 50-60% of the available space in the deck structure, the rig should have been able to survive. Considering the effect of the anchor wires, the capsizing did probably not occur before 75-80% of the deck structure was filled with water.

#### Effect of anchor forces and wind moment

The anchor pattern when "Alexander L. Kielland" was anchored alongside "EDDA 2/7 C" is shown in Fig. 4. The tension in each anchor wire is assumed to be 50 tonnes. For different angles of heel, the anchor forces are calculated assuming the same draught and inclination axis as in the calculation without anchor forces. The righting moments resulting from these anchor forces, as well as the wind moments are shown in Table 3.

The wind moment, considering a sustained wind speed of 16 m/s, is estimated to 2000 tonne-m in upright condition. For the heeled conditions, the wind moment is reduced to 1700 tonne-m. Fig. 10 shows the stability curves including the effect of anchor forces and wind moment for the different stages of flooding.

Table 1: Results for various stages of flooding.

Stages of flooding		Without anchor forces and wind moment				With influence of anchor forces and wind moment
"F"	"A"	Draught	Inclination axis	Angle of heel	Meta-centric height	Angle of heel
		(m)	(deg)	(deg)	(m)	(deg)
0%	0%	29,51	10,8°	31,1°	31,36	30,5°
20%	0%	30,28	10,2°	31,9°	29,37	31,3°
40%	25%	31,48	10,0°	33,3°	25,93	32,5°
60%	50%	34,05	11,3°	36,4°	19,27	35,2°
80%	75%	----- CAPSIZING -----				



Table 2: Items assumed lost during the capsizing.

Item	Weight (tonnes)	Centre of gravity		
		LCG (m)	TCG (m)	VCG (m)
Pontoon D	413	13,10	40,50	3,40
Column D	335	13,10	40,50	19,98
Bracings	182	2,97	28,22	19,34
Pontoon & Column Equipment	170	13,10	40,50	26,72
Bracing equipment	31	18,50	26,00	35,60
Ballast pontoon D	1260	13,10	40,50	2,51
Total lost weights	2391	12,40	39,38	8,54

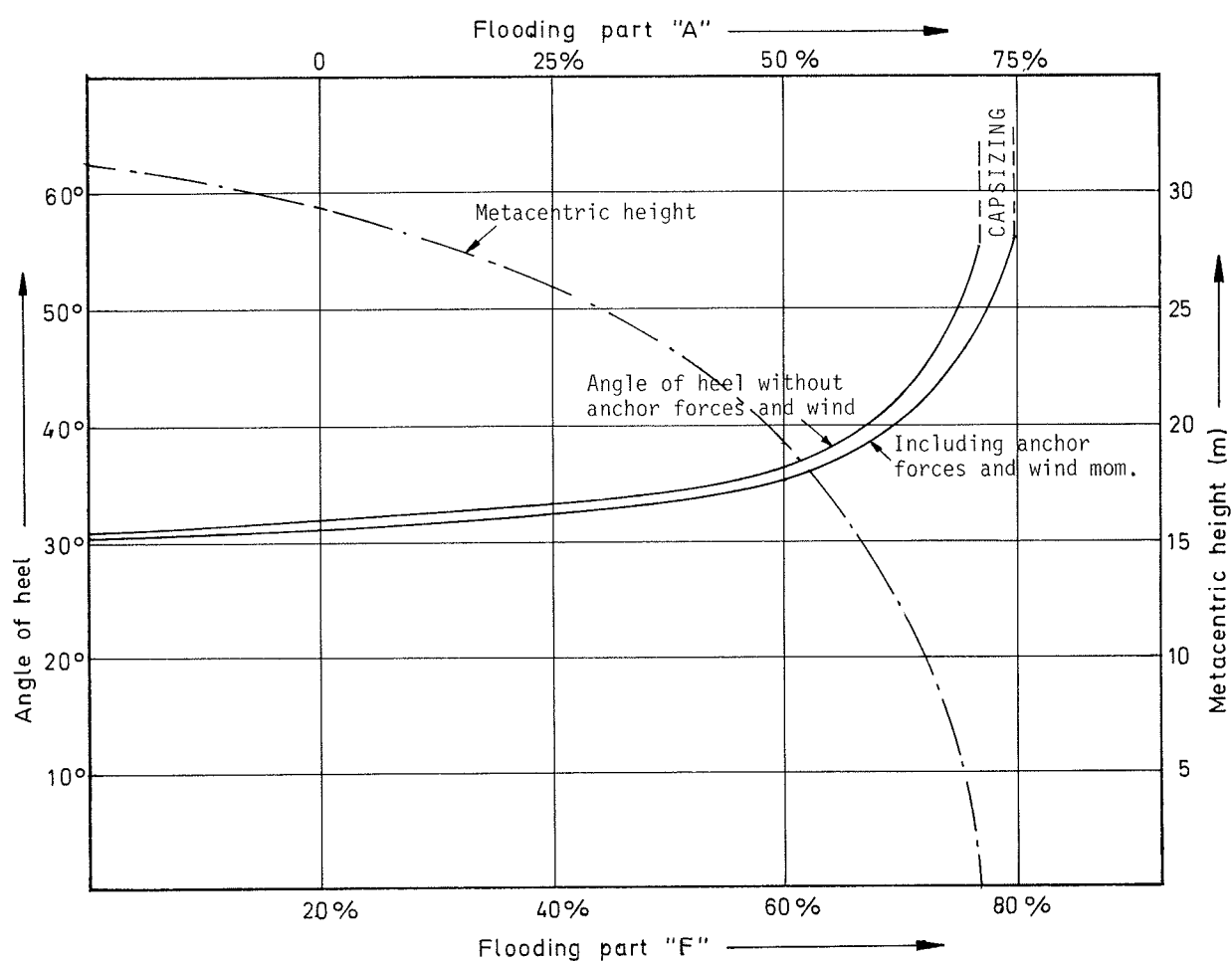


Fig. 7 Angle of heel and metacentric height versus flooding

Table 3: Righting moment due to anchor forces minus wind moment.

Angle of heel (deg.)	Anchor force moment (tm)	Wind moment (tm)	Anchor force moment minus wind moment (tm)
30 °	6 300	1 700	4 600
40 °	10 200	1 700	8 500
50 °	14 500	1 700	12 800
60 °	19 300	1 700	17 600

Table 4: Loading condition after loss of leg D, and water in trunks C and E.

Item	Weight (tonnes)	Center of gravity	
		Vertical (m)	Direction of leg D (m)
Intact cond. D=21 m	17173	17,60	- 0,02
-Lost weights	2391	8,54	41,28
+Water in C/E trunks	632	11,09	13,12
	15414	18,74	- 5,89

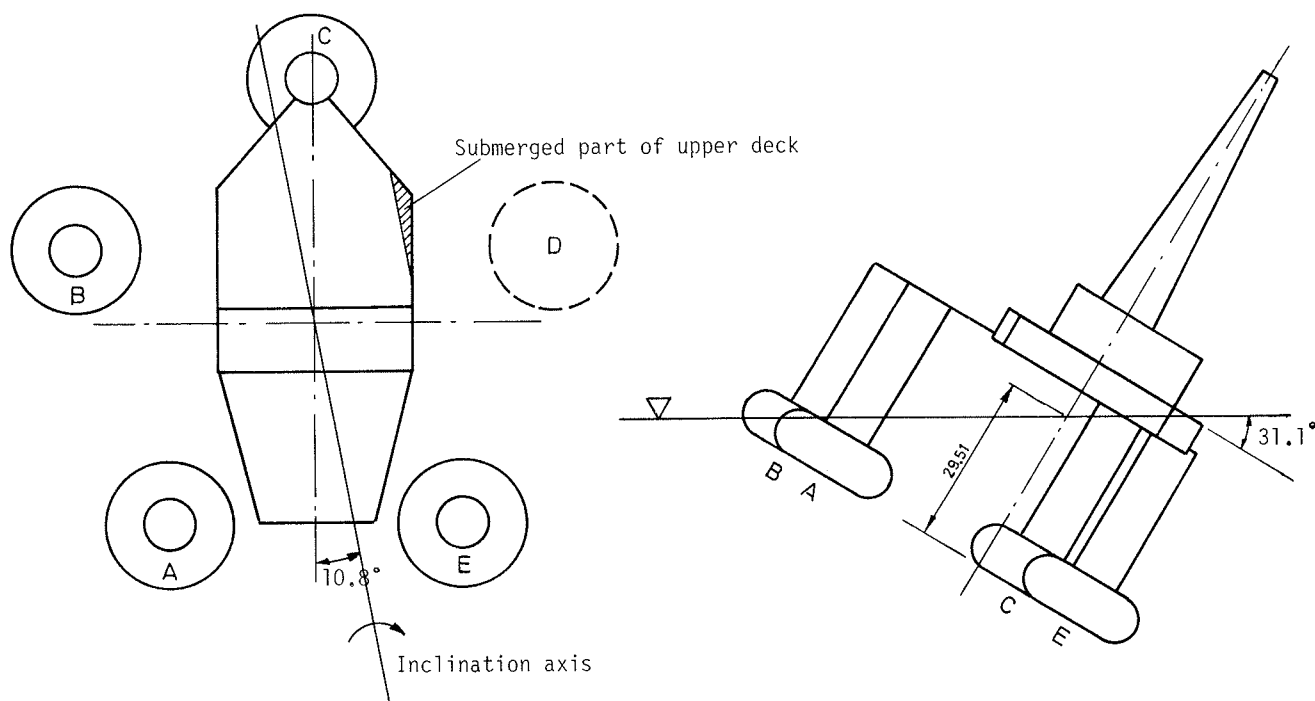


Fig. 8 Equilibrium condition in still water after loss of leg D  
(Deck structure and trunks intact)

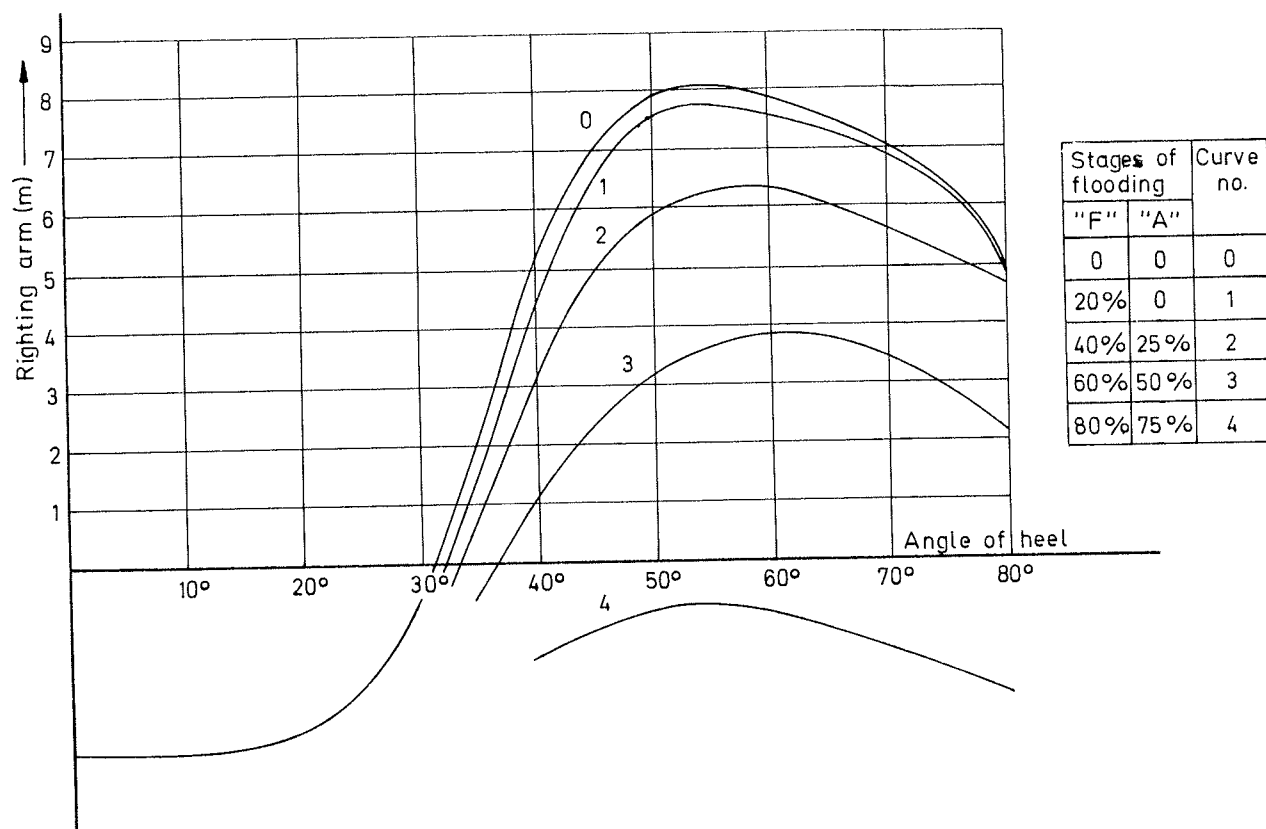


Fig. 9 Stability curves for varying stages of flooding  
(Without effect of anchor forces and wind moment)

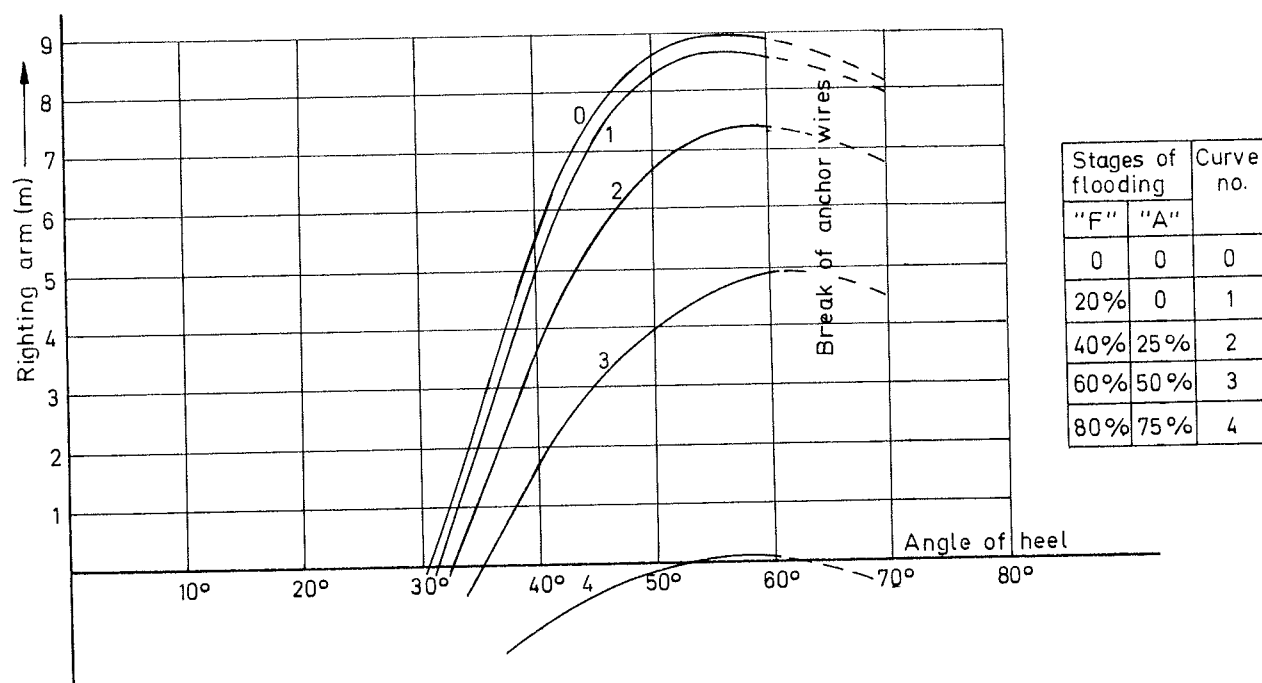


Fig. 10 Stability curves for varying stages of flooding  
(Including the effect of anchor forces and wind moment)

#### 4. SURVIVAL CAPABILITIES AFTER LOSS OF ONE LEG

When "Alexander L. Kielland" lost one of its supporting columns, a drastic redistribution of weight and buoyancy followed. As shown previously, this redistribution must lead to capsizing unless the deck structure provides some extra buoyancy.

When the buoyancy in the deck structure was lost, the only possibility to avoid capsizing would have been ballast redistribution. Since the capsizing happened very fast, and the power supply failed immediately after the initial heeling of 30-35 degrees, it was not possible to initiate any ballasting operation. A study has, however, been undertaken to see what ballasting operations are necessary in order to get a Pentagone platform to float in equilibrium condition on 4 legs. Equilibrium may be obtained

in a heeled condition, but in this study the possibility of equilibrium in upright condition only is investigated. It is assumed that the trunks/pumprooms in the columns C and E are filled with water, but no flooding in the deck structure is assumed. The loading condition for the platform after the loss of leg D is shown in Table 4. Fig. 11 shows the position of the initial metacenter and center of buoyancy versus draught. The same figure also shows the variation of center of gravity as a result of filling ballast into tanks in the A and B pontoons. The center of buoyancy and gravity are defined in the direction of leg D. From the curves it can be seen that in order to obtain equilibrium in upright condition by filling ballast into the A and B pontoons only, an amount of more than 2500 tonnes must be added. The draught will then be about 35 metres, and the metacentric height about 3.5 metres. It can further be seen

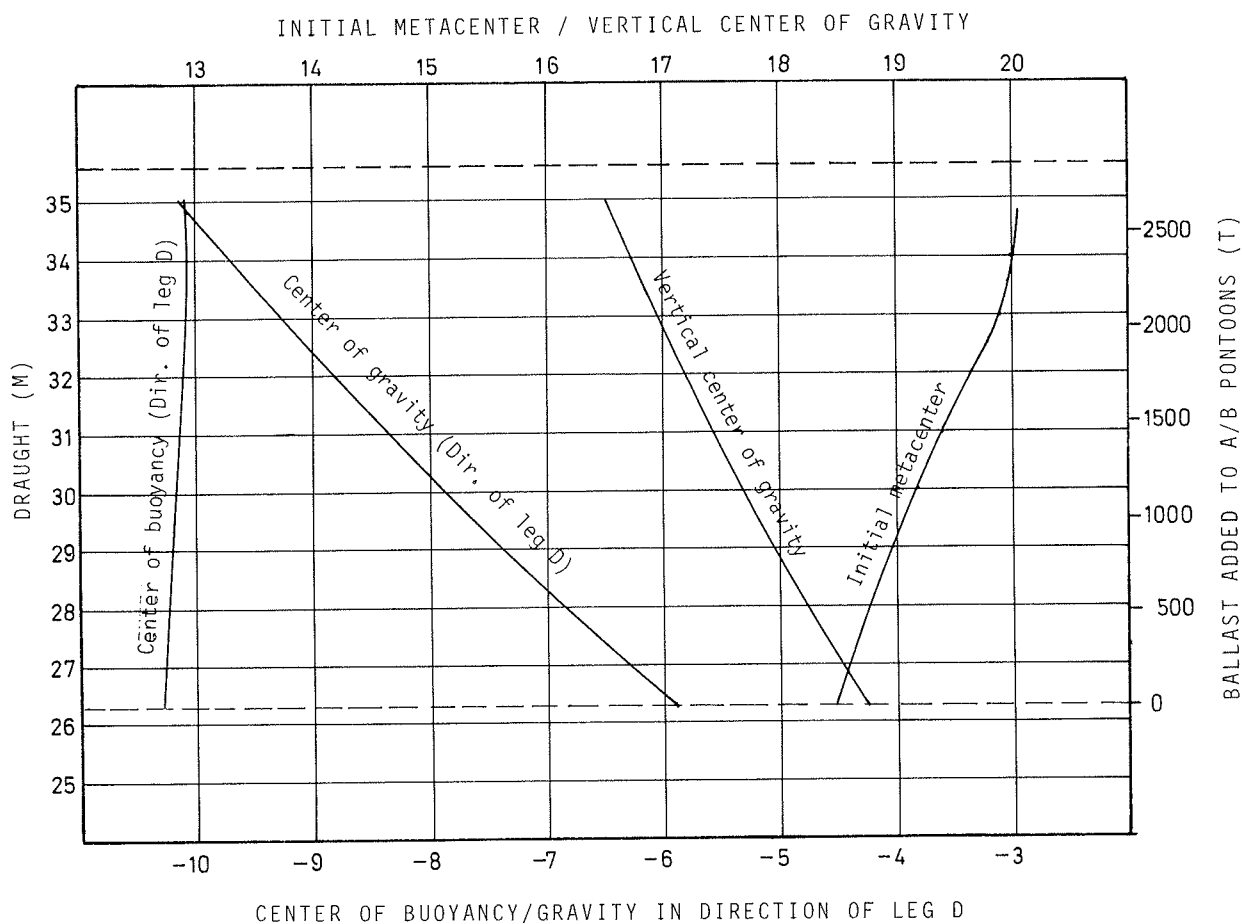


Fig. 11 Initial metacenter and center of buoyancy after loss of leg D and center of gravity versus ballast in A/B pontoons (Trunks in leg C and E are assumed filled with water)

that the minimum draught is about 27-28 metres in order to obtain positive metacentric height. If an amount of 500 tonnes is added into the ballast tanks in the A/B pontoons, this will give a metacentric height of about 0,5 metres. In addition, about 1600 tonnes must be shifted from the C/E pontoons to the A/B pontoons in order to obtain equilibrium in upright condition.

The conclusion is that it is theoretically possible to compensate the loss of one of the supporting columns by ballast redistribution, but the amount of ballast involved is considerable. Such an operation is therefore very difficult. The fact that the platform will almost immediately develop a large angle of heel, will lead to further complications to a ballasting operation.

##### 5. EVALUATION OF PRESENT STABILITY CRITERIA IN LIGHT OF THE ACCIDENT

At the time of the accident, the stability criteria in force for "Alexander L. Kielland" were found in the current rules and regulations given by the Norwegian Maritime Directorate and Det norske Veritas. Similar rules were also given by other regulatory bodies, like American Bureau of Shipping, U.K. Department of Energy, IMCO etc.

For intact stability, the different rules and regulations were to a large extent similar. The intact stability criteria were based on the assumption that there should be a sufficient excess of righting energy over that of wind energy. The righting and heeling energy were obtained from the area under the righting moment curve and the wind overturning moment curve respectively, the area taken from the upright condition to the angle of 2nd intercept between the two curves, or to any lesser angle of downflooding (Fig. 12). The required margin of excess righting energy was 1,4 for surface vessels and 1,3 for semi-submersibles. The metacentric height in intact condition was in most of the regulations only required to be positive. Only Det norske Veritas had a specific requirement to metacentric height (0,3m).

The damage stability requirements were also in principle the same in all regulations, but differed with respect to assumed extent of damage. Some regulations only required a one-compartment standard, while others prescribed a damage zone to be considered in the vicinity of the water line, and one compartment standard elsewhere. In equilibrium condition after flooding, taking into account the effect of an over-turning wind moment, openings which could lead to progressive flooding should not be immersed. In most regulations this was the only survival criterium for a platform after damage. The Norwegian Maritime Directorate had, however, a requirement with respect to the area under the

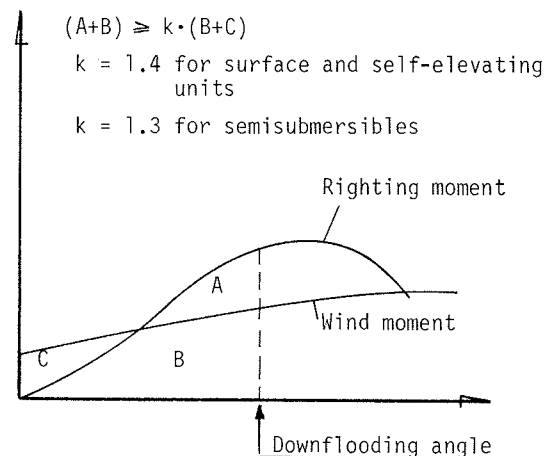


Fig. 12 Intact stability criteria

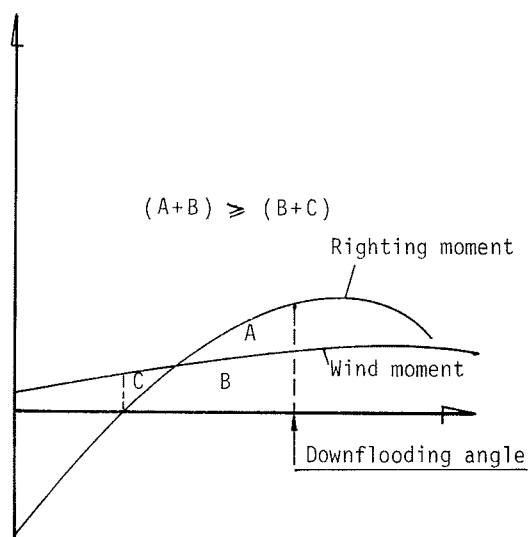


Fig. 13 Damage stability criteria

righting moment curve compared to the area under the wind overturning moment curve. This area was measured from static equilibrium without wind to 2nd intercept or to any lesser angle of downflooding (Fig. 13). In the rules of Det norske Veritas, a two-compartment standard was required for the deepest load line regardless of the position of the bulkheads. For such a damage, however, there was no requirement with respect to wind heeling moment.

The assumed extent of damage was based on the concept of "low energy" impacts caused by supply vessels, service vessels etc., which operate in the vicinity of

the platform. Collisions by bigger vessels etc. were regarded as a minor danger to be neglected. As a consequence of this, the requirement to wind speed was relaxed compared to that used with intact stability, because it was assumed that no supply vessel etc. operated near the platform in bad weather.

Usually all stability criteria are incorporated in one or more "max. KG curves", which give the highest permissible vertical center of gravity in order to fulfil the stability criteria. The max. KG curves for "Alexander L. Kielland", in order to satisfy the stability requirements of Det norske Veritas, are shown in Fig. 14. This curve shows that for draughts greater than about 19 metres, the max. allowable KG is limited by the damage stability criteria. This pattern is found for most of the rigs in service today. It means that for a rig operating with maximum allowable deck load, even a minor damage could cause heel to downflooding, since the angle of heel in damaged condition was only restricted by immersion of downflooding openings. For greater damage, the situation could be very critical. It is quite possible, within the present rules, to construct platforms which will capsize if the extent of damage only slightly exceeds the required extent, e.g. if the damage penetration exceeds 1,5 metres.

Another serious aspect is that since the angle of heel due to damage was only

restricted by the immersion of downflooding openings, downflooding could easily occur due to waves and movement of the platform in damaged condition. Inaccuracies in the calculations or in the loading condition may also easily lead to submergence of downflooding openings.

#### Consequences of structural failure with regard to stability

Data from offshore accident records (Ref. 3) indicate that structural cracking or fatigue is one of the major risks for semi-submersibles. The effect of such failures may vary from moderate leakage to structural collapse of the platform. No stability criteria seemed to pay attention to this potential danger. The reason for this was probably that the risk for serious consequences resulting from such failures were regarded as small, and that it was difficult to assess reasonable criteria for such damage. The fact that this is more a strength problem rather than a stability problem may also have caused this matter to be regarded as less significant from a stability point of view. If, however, a structural failure leads to collapse of vital parts of the platform, the consequences are very likely to be disastrous. The "Alexander L. Kielland" tragedy is a hard lesson to be learned in that respect. As previously shown, the effect of the buoyancy of the deck

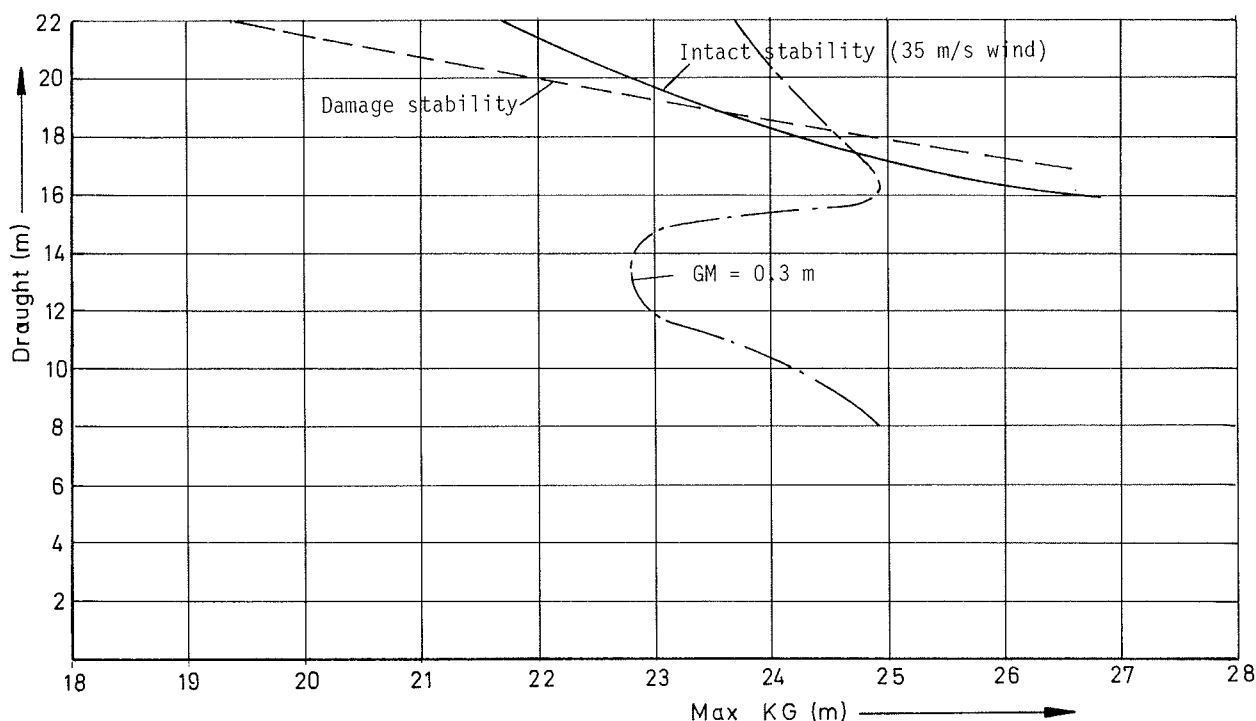


Fig. 14 Max. allowable KG curve for "Alexander L. Kielland" (DnV requirements)



structure is of vital importance with respect to survival capability of the Pentagone type rig when the buoyancy vanishes from one of the columns. Calculations of other rig concepts have shown the same trend.

By introducing a requirement to buoyancy in the deck structure, one obtains a significant improvement of safety against capsizing not only caused by structural collapse, but also from other hazards, such as collisions, blow-outs, ballasting errors, etc. An appropriate way of obtaining this buoyancy reserve suitably distributed is to require that the platform should be able to withstand the loss of buoyancy from any one of the supporting columns. Such a requirement is therefore introduced in the new rules of Det norske Veritas for accommodation units and the Norwegian Maritime Directorate for all units.

## 6. CONCLUSION

Since the capsizing of "Alexander L. Kielland" was caused by structural collapse due to a fatigue crack, most of the attention after the accident has been drawn to the strength problem. With respect to stability, no rules or regulations required that the platform should be able to withstand such damage. Calculations show that without buoyancy in the deck structure, the platform must capsize when one of the supporting columns is lost. It is therefore suggested that semi-submersibles should have a reserve buoyancy in the deck structure in order to avoid capsizing if vital parts of the buoyancy-giving elements should fail. Such require-

ments are already incorporated in the new rules and regulations of the Norwegian Maritime Directorate and Det norske Veritas. Accident records show that even if capsizing of semi-submersibles are rare events, the number of lives lost in such accidents are high.

By introducing such a "last defence line" against capsizing, we hope that a significant step forward has been made in order to improve the safety of offshore oil exploration and production.

## 7. REFERENCES

- (1) NOU 1981:11 "Alexander L. Kielland-ulykken", Report by the Commission established by the Norwegian Government. March 1981 (In Norwegian with a summary in English)
- (2) User's Manual Computer Program NV223: Stability calculation for an arbitrary floating structure.
- (3) "Safety and offshore oil", National Academy Press, Washington D.C. 1981.
- (4) "Rules for the Construction and Classification of Mobile Offshore Units" Det norske Veritas, 1975.
- (5) "Mobile Drilling Platforms. Regulations laid down by Norwegian Official Control Institutions", 1975.

## Discussion

E. Dahle (The Norwegian Maritime Directorate, Norway)

The paper gives an excellent overview of the "Alexander L. Kielland" - accident which highlighted the lack of strength, stability and adequate life-saving appliances on semi-submersible oil rigs. To deal with the stability aspect, the Norwegian Maritime Directorate has laid down new regulations for all semi-submersible rigs operating on the Norwegian shelf, requiring reserve buoyancy. Unfortunately, these requirements seem to have been misunderstood. They are not only intended for preventing disasters when substantial buoyancy parts of the structure is lost through structural failures, but are also intended for preventing disasters caused by extensive loss of buoyancy for single or coupled events like high-energy collisions and low-energy collisions continued with falling cargo etc. As such, it can be looked upon as a re-introduction of the 1966 - Loadline Convention requirement for enclosed super-

structures on ships. Could the author indicate the changes needed on A.L.K. to fulfill the new requirements mentioned, and how this would have influenced the floating position after the loss of one leg?

## Author's Reply

"Alexander L. Kielland" already has a more or less watertight deck structure, giving sufficient buoyancy to withstand loss of one column. The closing appliances, however, will have to be improved to a certain extent, and the watertight doors in the transverse bulkheads will have to be made remote controlled. Some of the inlets/outlets in the lower deck will have to be rearranged, this may also be required for staircases etc. in the upper deck. Some structural modifications in order to withstand the water pressure may also be required.

The floating position after loss of one leg will conform to fig. 8 in the paper.

M. St. Denis (U.S.A.)

There are, perhaps, two serious lessons to be learned from the "Pentagone" accident:

- a) That oceanic platforms whose buoyant elements are joined by structural assemblies of open trusses - some components of which are subject to large moments at their joints - may not be as structurally reliable as anticipated.
- b) That a greater measure of safety will be attained if the whole deck structure is made watertight.

These are merely suggestions. I cannot be more definite, since relevant structural plans as well as a hindcast of the hydrodynamic loadings to which the structure is subjected during its lifetime are not available.

Author's Reply

I do not think an open truss construction in general is less reliable than other platform concepts, but for the Pentagone design there was an obvious lack of structural redundancy, leading to structural collapse when one of the structural members failed.

Regarding deck buoyance, "Alexander L. Kielland" was in fact better than most of the platforms in service today. Without any deck buoyancy at all, the platform would have capsized almost instantaneously. The deck buoyance, even if it was insufficient to prevent the capsizing, delayed it for about 20 minutes, enough to save 89 lives.

# CONSTANT HEELING FORCES AND THEIR EFFECT UPON THE STABILITY OF A LOW-BUILT VESSEL IN WAVES

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## ABSTRACT

This report presents the main results of an experiment that was carried out with the object to measure the exciting forces acting on a fixed model of the paraller body of a low-built vessel placed beam-on to the oncoming regular waves. A calculation method is proposed for the estimation of the constant vertical force and heeling moment acting upon a rolling ship in case of systematic shipping of green water on deck. Based on the proposed method the analysis of the model experiment is performed. The experiment was carried out at Bassin d'Essais des Carine in Paris in 1966-67 with the aim to study stability and capsizing conditions for fishing trawlers of two types that are closely related as regards their architecture and dimensions, but vary as to their safety in a seaway.

It is stated that systematic green water shipping with waves of a certain frequency range causes a constant heeling moment to appear induced by water accumulating on deck since there is not enough time for the water to flow off between two sequential waves. For a conventional stability level this moment gives rise to the development of some average heel and results in an unsymmetrical ship roll relative her upright position. This average heel is called pseudostatic heel so as to underline the dynamic character of its origin. In the case of insufficient positive stability the heeling moment considered can lead to ship capsizing.

## NOMENCLATURE

### a) Ship particulars

- $L$  - length, m;
- $B$  - breadth, m;
- $D$  - depth, m;
- $d$  - draught, m;
- $F$  - minimum freeboard height, m;
- $\Delta$  - ship displacement without water on deck, tons;
- $C_W$  - waterline coefficient;
- $C_B$  - block coefficient;
- $h_w$  - bulwark height (deck well depth), m;
- $\theta_d$  - angle of deck edge immersion, deg;
- $\theta_w$  - angle of bulwark edge immersion, deg;
- $y_0, z_0$  - coordinates of inclined waterline centre, m;
- $GM$  - initial metacentric height, m;
- $GM_{cr}$  - critical value of metacentric height, m;
- $GZ$  - stability curve value without water on deck, m;
- $\delta GZ$  - negative increment arm of the stability curve in the presence of water on deck, m;
- $S_F$  - sheer value at forward perpendicular, m;
- $l_s$  - superstructure length, m;
- $l_d$  - deckhouse length, m;
- $b_d$  - deckhouse breadth, m.

### b) Main terms characterizing ship deck well according to N.B. Sevastianov

Deck well - volume open at the top; restricted on the underside by deck; on the sides - by bulwark; at the front - by deck, bulwark or aft bulkhead of superstructure; at the rear - by superstructure bulkhead. A distinction is made between closed and open deck wells.

Open deck well - a well without (partly or completely) restricting surfaces (forward or aft), impeding the backrush of water from the deck well.

Point of overflow - the lowest point of bulwark where water starts to discharge freely overboard from the deck well at ship inclination in case of closed ports and drain holes.

Depth of deck well - vertical distance between point of overflow and plane parallel to the base plane and passing through the lowest point of deck within the well. For a closed deck well its depth is defined as bulwark height  $h_W$ .

Length of deck well,  $L_W$  - maximum distance between restricting surfaces (forward and aft), m.

Breadth of deck well,  $b_W$  - maximum distance between restricting surfaces on the sides. Generally this value is coincident with the overall beam of the vessel, m.

$W = \int_0^{L_W} \omega_x dx$  - volume of water in deck well, calculated by the integration of  $\omega_x$  cross sections of deck well filled with water at a given heeling angle. An exact calculation of  $W$  is too bulky for practical use. Hence the following definition is given:

Equivalent deck well - a well of parallelepiped form with  $b_W$  breadth and  $h_W$  depth equal to respective dimensions of an actual deck well, and having a volume equal to the volume of water with its upper level at the point of overflow in the absence of heel.

Length of equivalent deck well  $L_W$  is less than length  $L_W$ . It is determined based on the following simple relations:

$$L_W = \alpha_d \alpha_s L, \quad (1)$$

where

$\alpha_d$  - horizontal deck area coefficient  
 $\alpha_s$  - coefficient taking into account the effect of sheer and superstructure length and calculated by the formula

$$\alpha_s = \left[ 1 - \sqrt{\frac{s_F}{L}} \left( 1 - \frac{l_s}{L} \right)^2 \right] \left[ 1 - \frac{l_s}{L} \right] \quad (2)$$

#### c) Main Elements of Waves and Roll

$r_{OW}$  - wave amplitude, m  
 $\alpha_{OW}$  - amplitude of wave slope, deg  
 $\alpha_{EW}$  - amplitude of effective wave slope, deg.  
 $\omega$  - wave frequency, rad/s  
 $\tau$  - wave period, s  
 $\lambda$  - wave length, m  
 $\theta$  - instant value of absolute heeling angle, deg  
 $\theta_s$  - mean value of heeling angle at

roll, deg

$n_z = \sqrt{\frac{C_W g}{2 C_g d}}$  - natural frequency of vertical ship oscillations in calm water,  $s^{-1}$

$q = \frac{\omega}{n_z}$  - relative wave frequency.

#### d) Values characterizing constant components of exciting forces

$F_{\eta W}, F_{\zeta W}, M_W$  - horizontal and vertical forces and moment acting in the athwartship plane, kg, kgm

$\rho_W$  - effective amount of water on deck, kg

$\lambda_W$  - heeling moment arm, m

$K = K_W K_d$  - empirical correction coefficient

$h_{EW}$  - effective bulwark height, m

$h_{OW}$  - height of effective water layer on deck in upright position of a ship, m

$y_W, z_W$  - CG coordinates of effective deck water amount at an arbitrary heeling angle, m

$\rho$  - parameter characterizing depth of deck edge immersion.

#### e) General symbols

$g$  - gravitational constant,  $m/s^2$   
 $\delta$  - specific water weight,  $t/m^3$   
 $\rho$  - water mass density  $ts^2/m^4$   
 $t$  - time, s.

#### I. INTRODUCTION

A statistical analysis of a great number of casualty records on fishing vessels collected by the Intergovernmental Maritime Consultative Organization (IMCO) indicated that in almost 50% of cases associated with ship losses, mainly low-built vessels, intensive green water shipping was observed [1]. To this must be added that the hazard of green water shipping is especially deepened for fishing vessels in high latitudes where they are subject to icing in winter conditions. The latter in itself leads to the reduction of stability and lowering of fishing vessel freeboard, thus facilitating the flooding of her deck. According to the data of abovementioned casualty statistics capsizing of two thirds of ships lost was due to green water shipping or icing.

The term "low-built vessel" is, in general, with no particular seagoing quality in view, to a certain degree an ambiguous term. Thus further a low-built vessel is understood to be a ship capable when rolling in a seaway of shipping of green water on deck. Of course no sharp boundary exists between high- and low-built vessels, but evidently the effect of shipping of green water will be the

stronger the lower freeboard is. As calculations show, on ships having an angle of deck immersion  $\theta_d < 15^\circ$  shipping of green water cannot be practically avoided.

In connection with the above statements rather interesting studies should be mentioned carried out in France in 1965-67 with relation to the analysis of fishing trawlers safety at sea [2], [3].

The French fishing fleet has two types of trawlers (called types "A" and "B" [2]), with insignificant disparity between them from the viewpoint of safety criteria currently in use. Nevertheless in the process of operation of "B" group ships several cases of their loss in a seaway were observed while no similar accidents with "A" group vessels occurred.

Ships of the two groups mentioned are closely related as regards their size and

the relationships of main particulars. They differ in freeboard and the stability curves. The latter are shown in Figs. 1 and 2. The main dimensions and the freeboard are also shown in the figures.

A theoretical attempt is made [2] to explain within the framework of the statics of the ship the extent to which the survivability of those ships differ, based on their stability analysis at large heeling angles. Group "A" vessels are shown to have a greater reserve of dynamic stability than group "B" vessels characterized by a lower freeboard. It is pointed out that the shape of the stability curve branch beyond its maximum may exert a significant influence on ship safety in a seaway although no special consideration was given here to the problems of dynamics. The work described in [2] was continued in an experimental investigation [3] carried out on ship models of the above mentioned groups in the Paris towing tank in 1966. Models were equipped with bulwark, deckhouses, superstructures and bulky deck structures which allowed to reproduce the stability curve in great detail. Model motions placed beam-on to regular waves were recorded by a motion picture camera. As shown by tests, with green water shipping over the bulwark, a heel occurred towards the "windward" side facing the oncoming waves. This heel is affected by wave height and wave length, as also model stability. The heel increased with the decrease of stability. At a certain critical value of metacentric height  $GM_{CR} > 0$  the models capsized as a rule in the direction of the oncoming waves. With heel due to green water shipping an insignificant roll occurred. Tests showed that for "A" group trawlers lower  $GM_{CR}$  values are allowable as compared to "B" group vessels.

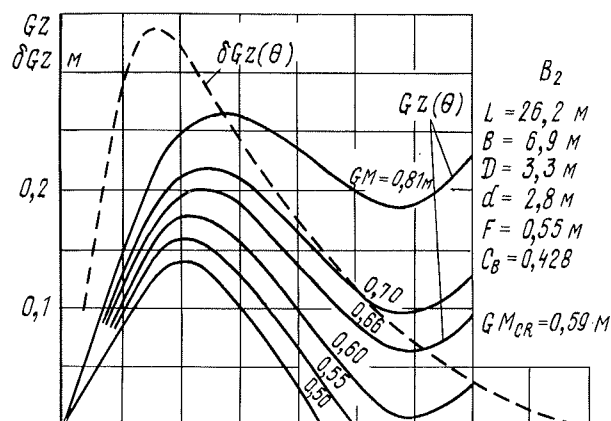
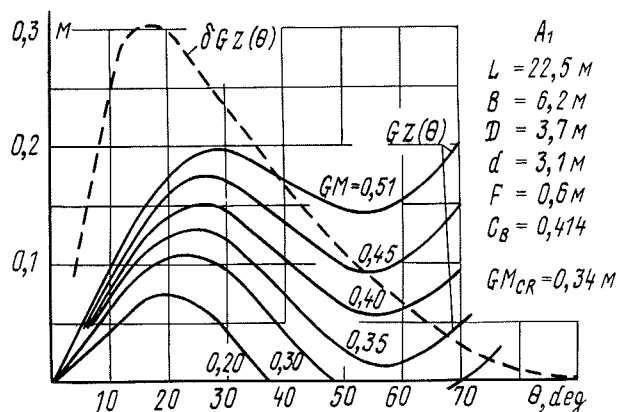
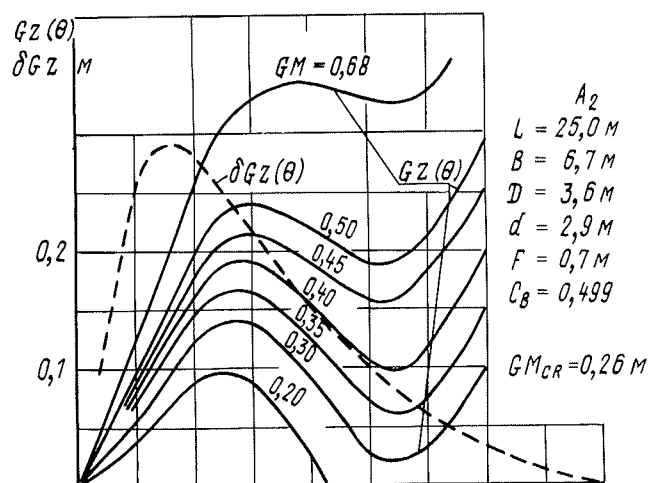


Fig. 1. Stability curves

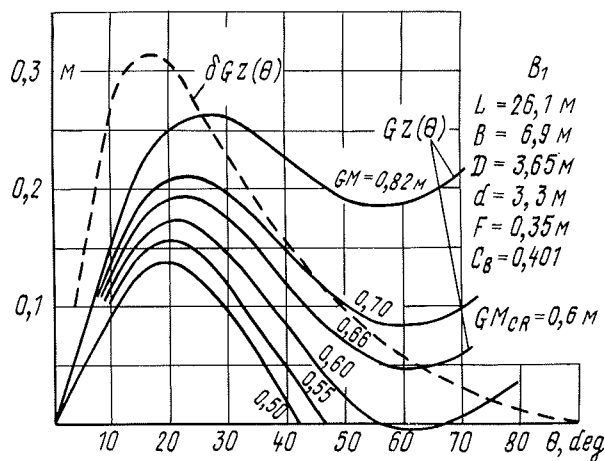


Fig. 2. Stability curves

It is interesting to notice that this feature in the behaviour of a low-built vessel was observed earlier in tests carried out in irregular waves with "Mariner" ship type model in a damage loading condition [4].

At the same time the works mentioned do not involve systematic data explaining the reasons of model heel occurring in rolling conditions followed by green water shipping.

The present work was performed with the object to explain the mechanism of ship heel in the process of systematic green water shipping in case of the ship positioned beam-on to the oncoming waves. The explanation is based upon experimental data. In the course of the tests exciting forces were measured at fixed cylindrical models of a low-built vessel in regular waves. Finally a semiempirical method is proposed for the calculation of constant components of the exciting vertical force and moment induced at ship roll under above-mentioned conditions. In conclusion analysis results of the above-mentioned French tests [3] are outlined based on the use of the method described.

## 2. CONSTANT COMPONENTS OF EXCITING FORCES DUE TO SYSTEMATIC GREEN WATER SHIPPING

According to the general methodology of the theory of ship motions the exciting forces are defined as additional forces generated on a fixed ship hull as a result of the difference between its submerged volume and hydrodynamic pressures in a seaway and in calm water. The main difference between a low- and a high-built vessel is the fact that under certain conditions her deck can be exposed to systematic green water shipping when rolling in a seaway. Systematic green water shipping is understood to be such a process of sea water flowing over the bulwark on deck when the water from the

preceding wave is not yet freely discharged from it by the moment of the deck flooded by the next wave. In consequence a certain amount of sea water  $p_w$  is accumulated on deck, its presence leading to an increase in ship displacement and affecting her dynamic characteristics.

In the framework of the statics of the ship the effect of systematic green water shipping can be described through the action of the following constant forces:

$$F_{\xi W} = 0, \quad (3)$$

$$F_{\zeta W} = p_w,$$

$$M_W = -p_w \lambda_w$$

In the last formula the sign (-) means that the moment  $M_W$  due to the presence of water on deck increases the ship heel.

Assuming the ship hull with a given heeling angle  $\theta_s$  placed beam on to the oncoming waves and using the concept of "equivalent deck well" to determine the  $p_w$  value a following relationship may be proposed

$$p_w = \gamma K \frac{h_w^2 l_w}{2 \tan \theta_s}, \quad (4)$$

and  $K$  is the correction coefficient subject to refinement on base of experimental data.

Considering the water trapped on deck under above conditions as an added load the heeling moment  $M_W$  can be defined as a moment of a couple generated by the force of load weight  $p_w$  and added buoyancy  $\gamma \delta V$  induced by the parallel immersion of the vessel down to waterline  $W_1 L_1$  due to load acceptance (Fig. 3).

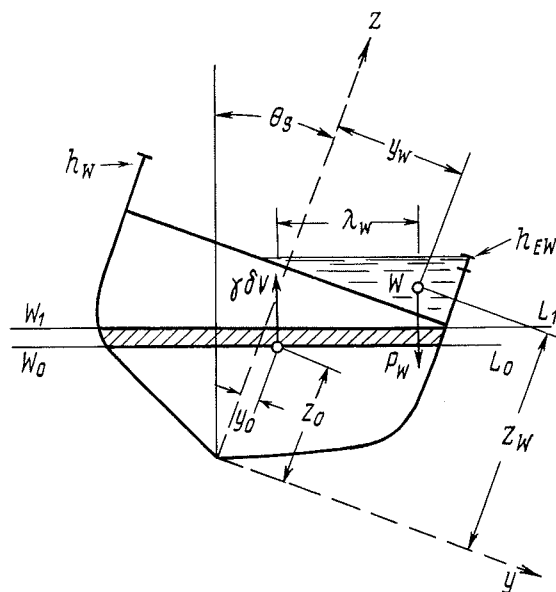


Fig.3. Scheme of forces in the presence of water on deck

The loading points  $W$  and  $O$  of these forces are centres of gravity of respective volumes. The arm of the couple is defined as the length  $\lambda_W$ . Without a large error the point  $O$  may be taken the CG of waterline  $W_0L_0$  at heeling  $\theta_s$  before green water shipping on deck. Point  $W$  coordinates in a general case should be determined by special calculations taking into account the angle of heel, as also deck well form and dimensions. These calculations are extremely simplified if the concept of equivalent deck well is applied.

With known  $O$  and  $W$  coordinates according to Fig.3, to determine the arm  $\lambda_W$  the following relation could apply:

$$\lambda_W = (y_W - y_O) \cos \theta_s + (z_W - z_O) \sin \theta_s \quad (5)$$

Green water shipping is a complex process affected by a multitude of factors which cannot be naturally taken into account while considering the problem in the framework of statics. In particular, such factors as the dynamics of water arriving on deck, the effect of wave frequency and height, freeboard and bulwark height etc. are not allowed for. Thus the problem to practically use the above relations remains open without experimental check.

### 3. EXPERIMENT

#### 3.1. Objective

A brief outline of an experimental investigation is given here, carried out with the object to measure the exciting forces acting on schematic models of a low-built vessel placed beam-on to oncoming regular waves. Experimental analysis, in accordance with the theme of the present work, is restricted by the examination of data characterizing constant components of exciting forces.

#### 3.2. Models Tested. Measured Characteristics

Tests were carried out on two models of cylindrical form fabricated of wood. The length of each model made up of 3 parallel bodies was 3 m. The middle parallel body was 0,7 m in length while each end body was 1,15 m. The main particulars of the models are given in Table 1; the shape of the cross-section is shown in Fig.4.

Table 1

Main Particulars		
Model number	1	2
Length L, m	3,000	3,000
Breadth B, m	0,370	0,370
Draught without heel, m	0,160	0,160

d, m	0,035	0,035
Freeboard F, m	2,310	2,310
Ratio B/d	0,189	0,189
Ratio 2F/B		

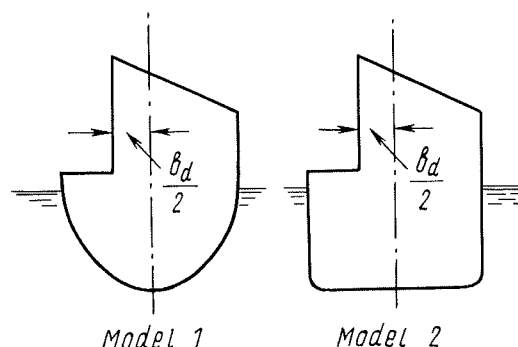


Fig.4. Model cross-sections

Model 1 frame was a rectangle its corners rounded with a radius 0,05 m. Model 2 frame corresponded to the shape proposed by Porter. On deck of the models an unsymmetrical deckhouse was placed. One of its walls was the extension of the side that rose at the heeling under study. The other wall served as the restricting surface of the deck well. Turning devices were installed on the model allowing for the variation and holding of the heeling angle on the one side and the attachment of the model to a fixed foundation on the other.

Thus in the process of testing the models had no freedom of motions. The midbody was fixed by a three-component dynamometer connected to the turning device. The dynamometer was designed so as to measure three components of hydrodynamic forces at a given angle of heel:

- $F_H$  - transverse horizontal component,
- $F_V$  - vertical component and
- $M_\xi$  transverse heeling moment.

The position of model dynamometer and wave probes is shown in Fig.5.

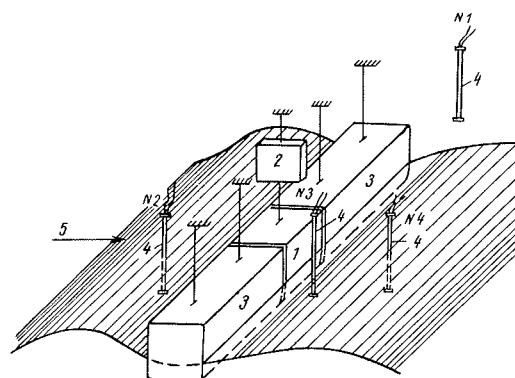


Fig.5. The position of model and wave proba, 1 - parallel middle body; 2 - dynamometer; 3 - parallel end bodies; 4 - wave probes; 5 - wave wave direction

### 3.3. Test Program. Method of Processing

Each model equipped with a given deckhouse and bulwark was tested at a fixed heel in a series of stationary regular waves. The length varied from 0,85 to 8,5 m, wave height varied from 0,016 to 0,08 m. The choice of the wave height for a given  $\lambda$  was made taking into account the kinematic features of the wavemaker based on the requirement to obtain gently sloping and regular waves. The relative bulwark height  $h_w/B$  ranged between 0 and 0,16, the relative area of freeing ports for the discharge of water from deck ranged from 3 to 8%, the relative deckhouse breadth  $b_d/B$  (see Fig.4) varied from 0 to 65%. The model angle of heel ranged from  $0^\circ$  to  $30^\circ$  in  $5^\circ$  step.

As a direct result of the experiment oscillograms were obtained of the wave profile at points where wave probes were placed, as also hydrodynamic forces  $F_x$ ,  $F_z$  and  $M_x$ . Since the variation of the registered values with time had the form of harmonic oscillations the processing of the oscillograms was reduced to the determination of the amplitudes and frequencies (periods) of these oscillations. On the records of hydrodynamic forces the mean line of the process was determined and from its displacement relative to the zero line constant components of exciting forces  $F_{xw}$  and  $M_w$  were determined; the moment  $M_w$  obtained from direct measurements was correlated relative to the longitudinal axis coinciding with the intersection line of the centerline and waterline planes for model in the upright position.

The analysis performed showed that for all wave probes the relative measurement error was about 5%. The greatest relative errors in the determination of hydrodynamic force amplitudes lie in the range of 15 to 20%. For constant force components the error did not exceed 20%.

Test results were presented as amplitudes of the processes under investigation and their mean values (constant hydrodynamic force components) related to the amplitude of the oncoming wave dependent on the frequency of the wave or the dimensionless frequency parameter

$$q = \omega \sqrt{\frac{2c_B}{c_w} \frac{d}{g}}$$

In the formulation of this parameter the frequency  $n_z$  of vertical ship oscillations in calm water was used. Processing results of data on constant components of exciting forces are

illustrated in Fig.6. Averaging of similar experimental data was performed based on the least squares method.

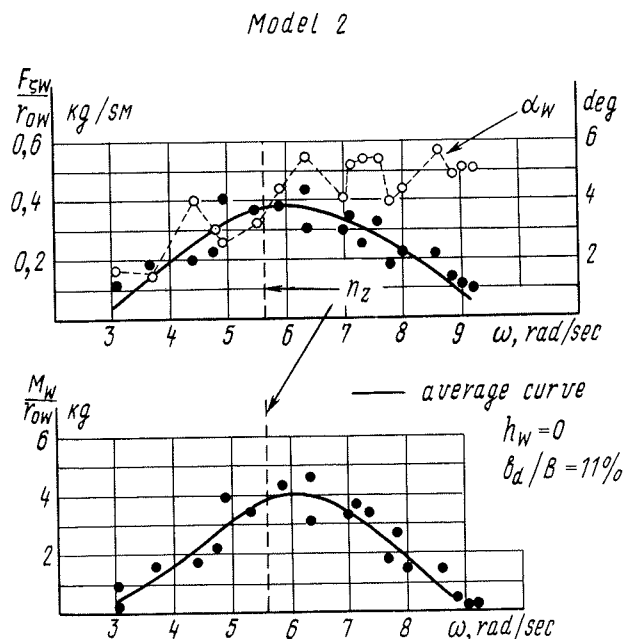


Fig.6. Measurement results for constant force components

## 4. ANALYSIS RESULTS OF EXPERIMENTAL DATA

Experimental data were analyzed to check the possibility of using the equations proposed in section 2 in order to practically estimate the constant components of exciting forces generated at systematic green water shipping.

### 4.1. Effective Bulwark

As shown in Fig.6, the vertical force  $F_{zw}$  has its maximum at  $q=1,1$ . In the absence of a deckhouse,  $b_d=0$  a satisfactory estimate of the maximum value of this force is obtained with the formula (4) if  $K=1$  is assumed and the effective bulwark height term  $h_{ew}$  is introduced. The latter differs from the design height  $h_w$  by a certain mean thickness of the additional layer of sea water on deck, due to systematic green water shipping, that exceeds the amount of water on deck in the absence of freeing ports in the bulwark.

The term  $h_{ew}$  is explained in Fig.7, where the  $h_{ew}$  value as the function of  $h_w/B$  ratio is presented.



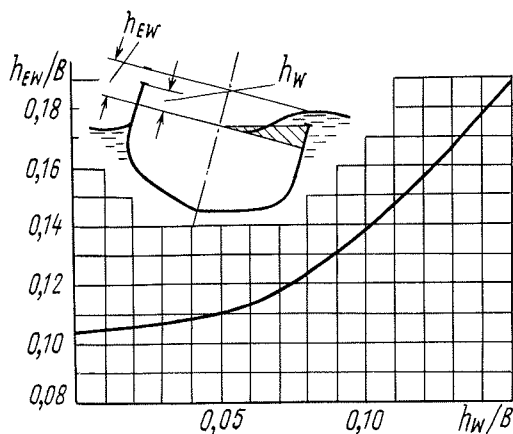


Fig. 7. Effective bulwark

#### 4.2. Deckhouse Effect

With a deckhouse on deck when estimating the maximum value  $[F_{\zeta W}]_{max}$  by the formula (4) the latter should be supplemented by a correction factor

$$K_d = \left(1 - \frac{b_d}{B} \frac{l_d}{l_w}\right) \left(1 + \alpha_d \frac{l_d}{l_w}\right) \quad (6)$$

The first factor in (6) characterizes the decrease in the volume of the equivalent deck well for the case of a deckhouse on deck. The second factor allows for the dynamic effect of the longitudinal vertical wall upon the amount of sea water trapped on deck. The empirical coefficient  $\alpha_d$  is defined from Fig. 8.

#### 4.3. Comparison of calculated and Experimental Data

With allowance made for the proposed recommendations on  $h_{EW}$  and  $K_d$

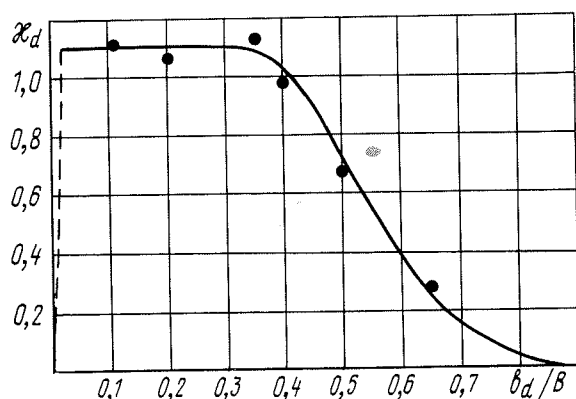


Fig. 8. Coefficient taking into account deckhouse effect

maximum values of constant vertical force  $[F_{\zeta W}]_{max} = P_W$  and constant heeling moment  $[M_W]_{max}$  in the function of heeling angle  $\theta_s$  for both models at various values of deckhouse breadth and

design bulwark height were calculated. Examples of calculated data are given in Figs. 9 and 10. In the same figures experimental values of  $[F_{\zeta W}]_{max}$  and  $[M_W]_{max}$  for the relative frequency  $q \approx 1,1$  are shown by points. As can be seen, calculated results are satisfactorily correlated with experimental data not only with heeling angles  $\theta_s < \theta_d$  or  $\theta_W$  but also in the case of deck or bulwark immersion, that is with  $\theta_s > \theta_d$ , and with the water in the deck well in permanent communication with the sea water.

Fig. 9. Results of vertical force calculation

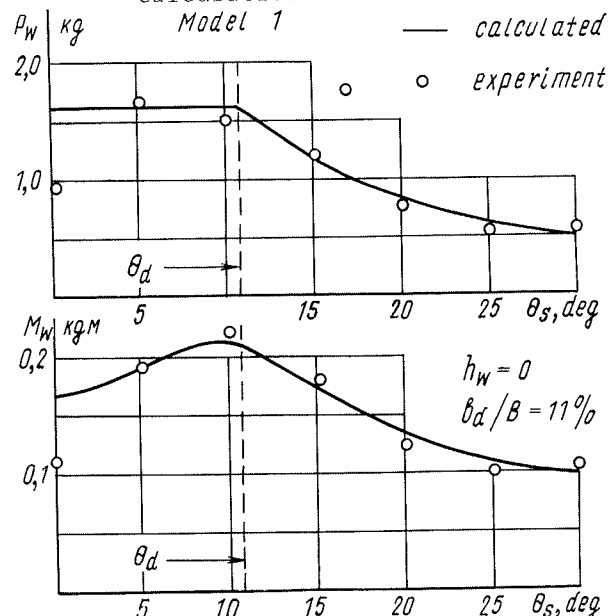


Fig. 10. Results of heeling moment calculations

Comparison of experimental and calculated data confirmed the conclusion according to which the size of the freeing ports has no effect upon the effective amount of water trapped on deck in the process of systematic green water shipping.

#### 4.4. Effect of Frequency

Practically to take into account the effect of encounter frequency on the constant components of exciting forces in Fig. 11, dimensionless curves are proposed for a corresponding corrective coefficient to the formula (4):

$$K_\omega = \frac{F_{\zeta W}}{[F_{\zeta W}]_{max}} = \frac{M_W}{[M_W]_{max}} \quad (7)$$

In Fig. 11 the abscissa is the relative frequency  $q$ . A complex dependence of  $K_\omega$  on  $q$  is recognised. It changes with the variation of  $\rho$ :

$$\rho = \frac{\alpha_{EW} + \theta_s}{\theta_d} \quad (8)$$

or

$$\rho = \frac{\alpha_{EW} + \theta_S}{\theta_W}$$

The latter characterizes the depth of deck edge or bulwark immersion under the free surface of the wave. In the absence of wave diffraction near the ship hull water on deck could accumulate only in the case of  $\rho > 1$  and if the interval between sequential deck immersions be less than the time required for the water to discharge freely from the deck through the freeing ports.

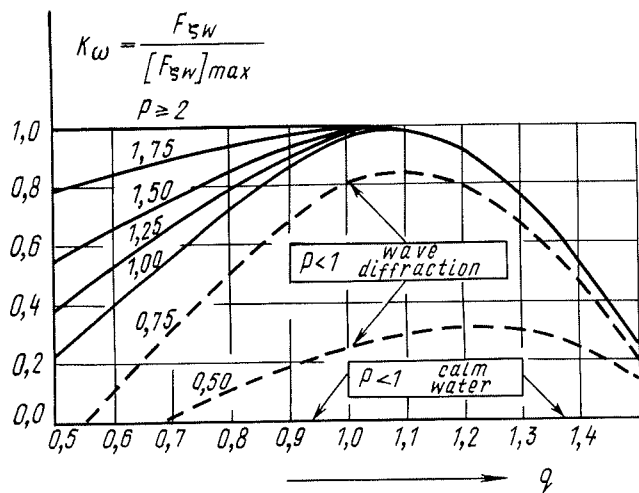


Fig.11. Coefficient taking into account frequency effect

This case of effective water amount accumulation on deck may be observed under the action of long waves of appropriate height upon the ship. Under the action of short waves with a frequency  $0.6 < q < 1.4$  on a low-built vessel the diffraction phenomenon leads to green water shipping at  $\rho < 1$ . As shown by experience, green water shipping can be neglected in the given frequency range only in case of  $\rho < 0.5$ .

## 5. FORMULAE FOR THE CONSTANT COMPONENTS OF EXITING FORCES

### 5.1. For the vertical force

$$F_{zW} = \rho_W = \gamma K_\omega K_d \frac{h_{EW}^2 l_W}{2 \tan \varphi}, \quad (9)$$

where

$$\varphi = \theta_W, \quad \theta_S < \theta_W; \quad \varphi = \theta_S, \quad \theta_S > \theta_W \quad (10)$$

Coefficient  $K_\omega$  taking into account the effect of the relative frequency of encountering waves  $q$  is defined from Fig.11. The correction coefficient  $K_d$  characterizing the effect of the

deckhouse is calculated by the formula (6).

### 5.2. For the Heeling Moment

$$M_W = -\rho_W [(y_W - y_0) \cos \theta_S + (z_W - z_0) \sin \theta_S] \quad (11)$$

The  $CG$  coordinates of the inclined waterline  $(y_0, z_0)$  and water  $CG$  on deck  $(y_W, z_W)$  are functions of the heeling angle  $\theta_S$ . The first two coordinate points are calculated based on the plotting of equal volume inclined waterlines. To define the second pair of coordinates it is reasonable to use the concept of the equivalent deck well. In this case

$$y_W = \frac{B^2}{12 h_{0W}} \tan \theta_S, \quad \tan \theta_S < \frac{2 h_{0W}}{B};$$

$$y_W = \frac{B}{6} \left( 3 - \sqrt{\frac{8 h_{0W}}{B \tan \theta_S}} \right), \quad \frac{2 h_{0W}}{B} < \tan \theta_S < \frac{h_{EW}^2}{2 h_{0W} B} \quad (12a)$$

$$y_W = \frac{B}{6} \left( 3 - \frac{2 h_{EW}}{B \tan \theta_S} \right), \quad \tan \theta_S > \frac{h_{EW}^2}{2 h_{0W} B}.$$

The  $z_W$  value, irrespective of the heeling angle, can be approximately found from the formula:

$$z_W = D + \frac{h_{EW}}{3}. \quad (12b)$$

## 6. STABILITY OF A LOW-BUILT VESSEL AT SYSTEMATIC GREEN WATER SHIPPING

The effective water amount  $\rho_W$  accumulating in the deck well of a low-built vessel due to systematic green water shipping of a vessel rolling in waves leads to the variation of her displacement and stability characteristics. Evidently, her displacement is determined by the sum:

$$\Delta_1 = \Delta + \rho_W. \quad (13)$$

The variation of stability with a known heeling moment  $M_W$  is easily obtained based on the rules of ship statics. The righting moment of the ship is defined by the sum:

$$M_i = M + M_W, \quad (14)$$

where  $M$  is the righting moment of the

ship without water on deck at an arbitrary heeling angle.  $M_W$  is defined from the relationships (3) or (11). Consequently, the stability arm corrected with allowance made for the effect of systematic green water shipping, will be defined by the relationship:

$$GZ_W(\theta) = \frac{M_1}{\Delta_1} = \frac{\Delta GZ(\theta)}{\Delta + \rho_W} - \frac{\rho_W}{\Delta + \rho_W} \lambda_W(\theta). \quad (15)$$

As follows from this relationship, systematic green water shipping may lead to a loss of stability in a certain range of initial angles of heel, if it turns out that:

$$GZ(\theta) < \frac{\rho_W}{\Delta} \lambda_W(\theta) \quad (16)$$

In this case the ship rolling in a seaway will deviate from the upright position by a certain average angle of heel  $\theta_s$  corresponding to the new equilibrium position of the ship determined from the stability curve (15) based on the known conditions

$$GZ_W(\theta) = 0, \quad \frac{\partial GZ_W(\theta)}{\partial \theta} > 0. \quad (17)$$

Essentially the angle of heel  $\theta_s$  is easily found by way of comparing the initial stability curve  $GZ(\theta)$  and the negative stability increment curve:

$$\partial GZ(\theta) = GZ_W(\theta) - GZ(\theta) = -\frac{\rho_W}{\Delta + \rho_W} [\lambda_W(\theta) + GZ(\theta)] \quad (18)$$

as shown in Figs.1 and 2. The intersection of these curves defines  $\theta_s$ .

It appears reasonable to call this angle of heel the pseudostatic angle so as to underline the dynamic character of its origin. In the absence of green water shipping this heel is equal to zero.

## 7. ANALYSIS OF THE FRENCH EXPERIMENT

The negative increment stability curve plotted in Figs.1 and 2 indicates that in case of systematic green water shipping the heeling moment  $M_W$  remains in force at angles of heel exceeding the angle of immersion of the bulwark upper edge  $\theta_W$ . This finding reflects the dynamic character of the process of water accumulation on deck in case of systematic green water shipping. As is well known, while considering the effect of water accumulation in the deck well from the viewpoint of pure statics its heeling action comes to an end at  $\theta_W$ .

Let us now direct our attention to Fig.12 where the pseudostatic heel  $\theta_s$

is plotted as a function of the relative metacentric height  $GM/B$  for four trawler models  $A_1, A_2, B_1$  and  $B_2$  tested in a seaway [3].

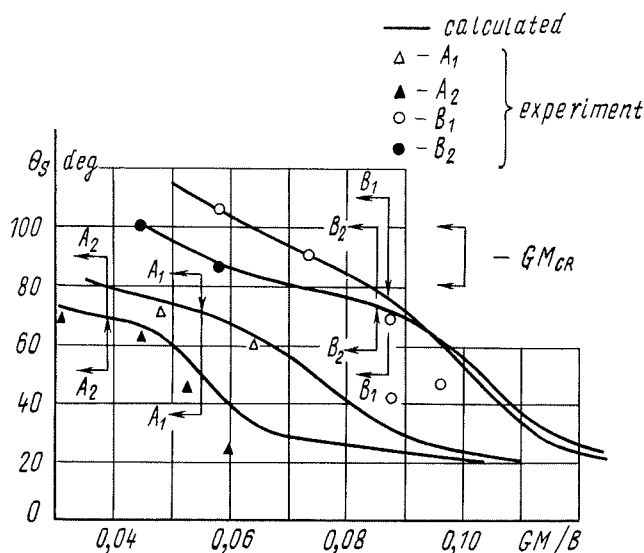


Fig.12. Results of pseudostatic heel calculations for French trawlers

The  $\theta_s$  values were determined in compliance with the recommendations of section 6. The curves  $\theta_s(GM/B)$  plotted characterize the upper limits of the pseudostatic heel since the curves (18) were calculated based on the assumption that coefficient  $K_\omega$  in (9) be unity. In the same Fig.12 for each of the four models direct measurement data of heeling angle  $\theta_s$  are given. These data are shown to correlate well with calculated results.

An illustrative example is also furnished by the plots under discussion revealing a significant difference in the response to green water shipping between the ships of "A" and "B" groups. In Fig.12 critical  $GM_{CR}$  values of the metacentric height for each of the ships are shown. For "A" group ships the  $GM_{CR}$  values are significantly lower than for "B" group ships. A higher safety level of "A" group ships is evidenced also by the fact that according to calculated results (Fig.12) for identical values of metacentric height the pseudostatic heeling angle of these ships is generally almost twice as low as for "B" group ships. It is interesting to point out that according to the data of the experiment under examination [3] capsizing of the models of both "A" and "B" groups was observed in all cases the calculated curves (18) exceeded the ordinates of the stability curve in the total range of heeling angles (see Figs.1, 2 and 12). However this finding is not sufficient to establish

the exact critical metacentric height  
 $G_{MCr}$ .

## 8. CONCLUSION

The investigation carried out made it possible to explain the mechanism of the pseudostatic heeling of a rolling vessel positioned beam-on to the oncoming regular waves in case of systematic green water shipping. The proposed method permitted the pseudostatic heel to be estimated by calculations without recourse to experiment. It is shown that at an insufficient stability level the heeling moment induced at systematic green water shipping may lead to ship's capsizing. At the same time it should be pointed out that the method developed does not establish any criterion for an unbiased judgement on the possibility of ship's capsizing in the case under discussion. With this object in view

investigations of the rolling of the vessel should be continued taking into account green water shipping and the stability of rolling in the presence of a pseudostatic heel.

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## Discussion

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The paper points out the danger of shipping green water when stability is inferior. However, when comparing Figs. 1 and 2 with Fig. 12, it becomes clear that capsizing only occurs when the GZ-values are below zero within the heeling angle range 0-70 degrees. This implies that positive GZ-values in the 0-70 degree range, even when the GZ-curve is below IMO-Standards, effectively prevents capsizing. As the Norwegian government is putting into force additional requirements for the GZ-values of new fishing vessels to be positive in the 0-80 degree range, comments from the author are appreciated.

Author's Reply

The results submitted in the report show that the range of positive stability curve is very important factor characterizing ship safety against capsizing. Considering heeling action of water on deck, one may conclude that the power stability curve values are the greater at this range than possibly seems to be. I think this conclusion generally is justified in case of any heeling forces. Referring to specific values of the stability vanishing angle, it is desirable for conventional seagoing ships to have this angle not less 50 ~ 60°. At any case, this angle, in my opinion, requires special consideration in stability standards.

## ON THE PHILOSOPHY BEHIND ASSESSING SHIP STABILITY

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### ABSTRACT

After some general considerations on mathematical modelling a definition of the expression "stability criteria" is given. Then it is shown that stability criteria can be modelled as ordinal measures of the safety against capsizing. A classification of stability criteria according to their discrimination ability is proposed. Finally some comments are made on various types of stability criteria.

### 1. MATHEMATICAL MODELLING

The famous English philosopher and mathematician Bertrand Russel has defined mathematics as something where we neither know what we are talking about nor if what we are saying is right or wrong. This statement is bewildering because mathematics is generally considered as the purest and most rigorous science. But this is only true with respect to pure mathematical exercises, where we start with axioms and definitions and are able to prove any step we are doing or any result we are getting. When we try to interpret our mathematical findings (i.e. when we use the mathematical language to talk about real facts) we are entering completely different problems. I will explain this with some very simple examples:

Let us start with the relationship:  $1 + 1 = 2$ . It is mathematically correct, but does not contain by itself a statement about any real facts. If we interpret the unit as a basket with fishes, then the relationship can be used for the following statement: If we add to a basket of fishes another one we get two baskets. But if we

interpret the unit as a school of fishes, the use of the relationship would give us a wrong result: When a school of fishes joins another one, we do not have two schools but again only one.

If we take the unit as the height of a brick, we can use our relationship to state: If we put one brick upon another one, the resulting height is two-times the height of one brick. But this does not hold in all cases: If one brick is made of burned clay and the other one of foam rubber, the result depends on which brick is put on the top. If it is the elastic one, our relationship applies; if it is the other one, the real result would be less than two. Of course we can easily find another mathematical relationship, which takes the elasticity of the foam rubber brick into account. This more comprehensive relationship would also hold in the case of two bricks both made of burned clay. But it would not make much sense to use it in such a case, because the decrease of the height of the lower brick under the weight of the upper one is much less than the accuracy with which the height of such bricks can be defined.

In engineering education the difference between mathematics and mathematical truth on the one hand and the interpretation of mathematical relationships and their validity in reality on the other hand is widely neglected: Students are taught the real facts directly in mathematical language. Thereby they do not even become aware of the problem how to make mathematical expressions (which may be considered as "empty forms") to models of real facts. This does not matter with regard to the big stock we have of well established mathematical

models of physical facts. But it is very important, if we are going to make mathematical models of real facts which are not yet sufficiently modelled.

Making a mathematical model is not a logically determined straight forward operation such as the development of pure mathematical relationships. It is rather a heuristic act to find a mathematical relationship, which might be useful as a model of reality. A heuristically created relationship must not yet be considered as a valid model of reality; at first it is only a hypothesis which still has to be proved. The proving procedure is quite different from that used in mathematics: Practically each mathematical model has deficiencies with respect to reality. In order to make it practicable it is inevitable to idealize or neglect some of the facts or conditions which are actually present. In most cases the effect of such simplifications can only be found out by repeatedly applying the hypothesis and judging the result against what really happens (If the new mathematical model is a synthesis of simpler, but already proved models this may not apply; but in this case we do not deal with an actually new model).

In judging the hypothesis two cases have to be differentiated: Firstly the merit of a model may be to explain qualitatively what really happens. E.g. it follows from a linearized equation of the roll motion that the roll amplitudes depend on the excitation frequency or it can be taken from the Mathieu equation that under certain conditions extremely heavy rolling of a ship travelling in following seas may occur (what is not obvious and was not known before the papers [1], [2] of Grim). There is nothing wrong with such explanatory models per se. But it is a misuse of mathematics if their authors sell them as "exact solutions" of a problem.

Secondly a model may have the purpose to provide quantitative predictions. Such predictions are always subject to errors. In the most favourable case the errors are only measuring errors which inevitably occur when the input and output of a model is determined in the corresponding reality in order to test the accuracy of the model. But more frequently the errors originate from idealizations and neglects in the model. Although a formal judgement of the errors could be made by wellknown statistical methods, the validation of models is mostly made informally. Thereby the practically required degree of accuracy is implicitly taken into account. Sometimes there is a choice between more or less simplified models. Then the accuracy of the model is balanced against its complexity: The practising engineer does not care much for the maximum achievable physical correctness but for results which are good enough for his purposes and can be achieved

with an acceptable effort.

With regard to the problem of capsizing of ships we do not have yet a comprehensive and valid mathematical model. But we have too many proposals which pretend to offer a solution, although they only explain some aspects of the problem qualitatively or - even worse - conclude from the mathematical complexity of the methods used that the results are valid. The latter habit has been very ably described and criticized by Kuo and Welaya [3]: "Put all your faith in mathematical theorems and terminologies and the problems of ship stability will disappear." They were only too right to reject such a naive approach.

## 2. SHIP STABILITY AND STABILITY CRITERIA (An Attempt of a Definition)

When I started my career I felt embarrassed by the use of the word "stability" in naval architecture. It did not fit with what I had been taught in engineering mechanics and automatic control at the university. At that time I was optimistic enough to believe that this situation might be changed. Therefore I wrote a paper [4] with the title: "What is ship stability?" in which I pleaded for the right use of the word stability in context with the static equilibrium of ships. Although my point of view was appreciated I had to learn during the course of time that the worldwide community of naval architects will never change their use of the word stability.

For naval architects "stability" means "safety against capsizing" in a very general sense. Combined with other words (e.g. stability calculation, stability lever, range of stability, initial stability, additional stability of form) it indicates something which plays a role in context with the safety against capsizing. Because it is common to use the word stability in naval architecture in a rather vague sense I see no chance for the acceptance of a more precise definition as given e.g. by J. Paffet in [5].

To live with this situation does not mean that the word "stability" cannot also be used in another context in a more precise sense, e.g. stability of equilibrium etc.

The word "stability criterion" is used in a more precise sense in naval architecture than just the word "stability"; and this sense is also in conformity with what we find in dictionaries (e.g. [6]): "Criterion" means "standard for making a judgement" and "standard" means (besides other things not relevant in this context) "model" or "measure". Therefore "stability criterion" means 1. the model which is the base for judging safety against capsizing and 2. a measure of the safety against capsizing. Some stability criteria do not appear as a measure but as a requirement (e.g. "the righting lever GZ

shall not be smaller than ..."). It can be shown that requirements are those values of safety measures beyond which a vessel is considered unsafe.

The definition of "stability criterion" as given above can still be expanded by asking: What is the meaning of "capsizing"? This question has two aspects: In which state is a ship considered capsized and under what circumstances shall we speak of capsizing in context with stability criteria (e.g. when a ship is careened it is clearly not considered capsized in spite of the fact that its state is the same as when capsized).

It is appropriate in context with stability criteria to consider a ship as capsized not only if it is turned bottom side up but also if the heeling angle (or if the ship is rolling the mean roll angle) assumes values which prevent the ship from further fulfilling its mission.

The state of being capsized may be effected by a wide variety of causes. They can be divided into two main groups: 1. causes which at least in principle can be controlled and 2. causes which are beyond human control.

With regard to the first group, safety against capsizing can be attained by removing or taking proper account of the causes which may effect capsizing. Examples for this group are: Shifting of insufficient secured cargo, wrong distribution of cargo (erroneous determination of KG), neglecting of free surfaces in the ship (also not closing of openings through which water may enter the ship).

With regard to the second group, safety against capsizing can only be attained by providing proper stability characteristics of the ship. Stability criteria (in a narrow sense) are concerned with the safety against capsizing with respect to this group of causes. Examples are: Effect of wind and waves (including shipping water on deck), shifting of small weights which practically remain unsecured, "random" free surfaces (which can neither be foreseen nor prevented).

Considering what just had been said one might wonder why it has been tried to define the meaning of the word "stability criterion" in place of directly defining a measure of safety against capsizing. It will be shown in the next section that there are good reasons to differentiate between stability criteria on the one hand and a measure of safety against capsizing (in a strict sense) on the other hand. There is no difference with respect to the final aim; but with regard to the means, i.e. the philosophy of modelling, it is useful to make a difference.

### 3. ON THE MODELLING OF STABILITY CRITERIA

#### 3.1 Safety Against Capsizing

Since engineers have learned that

total safety cannot be achieved in our world, the probability that certain unwanted events will not occur during a prescribed time has been widely accepted as a rational measure of safety against the occurrence of such events. While the probability concept proved very useful to understand safety and also has led to progress in many fields it soon turned out that it is not always possible to apply it practically. The reason is lacking relevant information as well as sufficient knowledge of physical relationships. This is especially true with respect to the probability that a ship will not capsize during its lifetime as was explained in some detail in a paper which was presented at the Stability Conference in Glasgow [7].

In this paper an additional problem was mentioned: Even if it would be possible to calculate the probability that a vessel will not capsize during its lifetime, this would not yet be enough. In addition to the probability of noncapsizing as a function of the ship characteristics that value of this probability would have to be determined, at which the ship is considered safe enough. Thereby it would not make much sense, after having gone through all the trouble of calculating the probability to choose a more or less arbitrary value as safety limit. But to determine a rationally based safety limit would be a task even more demanding than the determination of the probability of noncapsizing. Thereby the question of "minimizing the total mortality" as well as relationships between safety and economics would have to be considered.

Technologists are inclined to accept rationally based safety limits as normative requirements. But more recent research work on individual risk taking behaviour has shown that "rationally determined acceptable risks" do not necessarily gain public acceptance [8]. The probability of noncapsizing during the lifetime of a vessel results as an average of many probabilities of occurrence of events and situations. If we remember that a man with one foot on a stove and with the other in iced water should feel well in the average, we see the problem: A small enough probability of noncapsizing during the lifetime of a ship is not sufficient; adequate safety is judged by the public also with respect to particular situations. To provide adequate safety in this sense means that the product of the probability for the occurrence of certain (environmental) situations times the probability of capsizing in such situations must be very near to zero (as yet nobody knows the exact number). The measure of safety in this context is also a probability, namely one minus the forementioned product. Although this concept looks simpler than the probability of noncapsizing during the lifetime of a ship, our knowledge is not sufficient to practically apply it. It is clear that a safety

limit as just indicated would be interrelated with the rationally based safety limit which has been discussed before. For a particular ship the one or the other may dominate.

Summarizing this subsection it can be said that it is basically known how safety against capsizing in a strict sense should be measured; but things are too complicated to actually do it.

### 3.2 Stability Criteria as Ordinal Safety Measures

In spite of the fact that safety against capsizing cannot be measured in the strict sense, naval architects are able - in most cases - to design ships which have adequate safety against capsizing. But they have never reflected on the concept they are actually applying for that purpose. This is responsible for much fuzzy thinking about this problem.

The safety measurement in the strict sense as indicated above would be on an absolute or ratio scale. This is the highest level of measurement. It is known that when measurements on this level are not possible they often can be made on a lower level. Stability criteria may be interpreted as ordinal measures of the safety against capsizing.

In order to establish an ordinal scale for an attribute of a set of items it is only necessary to be able to place pairs of items A and B of the set in one of the following mutually exclusive categories:

- A preferred to B (a)
- B preferred to A (b)
- A and B indifferent (c)

If all items of a set can be placed in the fore-mentioned categories, they constitute a simple order. If this is not possible for some of the items (i.e. if some are incomparable), the remaining subset constitutes a partial order.

Measurement is defined as the act of assigning numbers to things according to some rules. Each function  $f$  (or rule of numbering), which is so constructed that the following three conditions hold for the above three categories respectively:

- $f(A) > f(B)$  (a')
- $f(B) > f(A)$  (b')
- $f(A) = f(B)$  (c')

constitutes an ordinal scale. Such a scale is unique only up to a monotonic increasing transformation.

Let us now consider a set of ships, all of similar type and size and especially having similar proportions. Although we have been taught that in such a case a ship with a higher GM-value is safer than one with a lower GM (of course within certain

limits) the original process to arrive at a judgement of the safety was the other way round: It were the safety records of the ships from which it could be taken that the safer ones have a higher GM. It is completely impossible to infer from the GM of a ship (without any other information) how it will behave in a random seaway which is usually accompanied by a gusty storm. In Section 4 it will be shown that essentially the same is true if in place of GM other, more sophisticated stability criteria are used.

The situation just described corresponds exactly to the procedure when establishing an ordinal scale for an attribute of a set of items: First we have to be able to state (i.e. to know from experience) for pairs of ships A and B of a set of ships if

- ship A is safer than ship B (a)
- ship B is safer than ship A (b)
- ship A and B are equally safe (c)

The next step is to find a function  $f$  of the characteristics of the ships A and B which yields for the above three cases respectively:

- $f(A) > f(B)$  (a')
- $f(B) > f(A)$  (b')
- $f(A) = f(B)$  (c')

(A and B are used here to designate particular pairs of ships as well as their characteristics.)

Any function  $f$  which complies with the above requirements constitutes an ordinal measure of the safety against capsizing. The parallelism between such a measure and stability criteria is obvious. It also covers the fact that to the infinite number of possible ordinal scales (because they are only unique up to a monotonic increasing transformation) corresponds a wide variety of stability criteria.

By the way it shall be mentioned that establishing an ordinal scale is a special category of mathematical modelling: A mathematical relationship (i.e. the function  $f$ ) is linked with real facts (i.e. the experience that a ship A is safer than a ship B etc.).

### 3.3 Assessing Safety Against Capsizing

In this context assessing safety against capsizing means to find that value of a stability criterion beyond which a ship is considered safe. Such a value will be called limiting value. Because stability criteria are ordinal measures, their values do not indicate - contrary to absolute measures - definite levels of safety. But they allow the comparison of the safety of ships. If a ship S is known to be just sufficient safe,  $f(S)$  can be taken as limiting value. A ship X will be at least as safe as the



ship S if the following relationship holds:

$$f(X) \geq f(S)$$

When deriving stability criteria as ordinal measures in the foregoing it was assumed that the ships the safety of which was to be compared are of similar type and size and that they have similar proportions. The same has to be required of ship S and ship X. Additionally, for a valid comparison of the safety of ship S and of ship X, the environment in which a ship X operates has to be similar to that of ship S.

To find a proper "calibration ship" S is not an easy task. Limiting values for stability criteria as originally proposed by Rahola and since then widely used may be interpreted as the values derived from a fictitious "average" ship S. It is quite clear that they cannot be valid for ships deviating significantly from the average. One of the most serious shortcomings of the Rahola-method is that the average ship S is not sufficiently defined. As a consequence, naval architects and shipmasters are often not aware that the limiting values may not be valid in a particular case.

It seems possible to adapt the limiting values of the average ship for other differing ships making use of model tests. This may be illustrated by the findings of Kure and Bang when they investigated the capsize of a coastal tanker in ballast condition [9]. As a matter of simplicity, only the  $e_{30}$ -criterion (i.e. the area under the righting arm curve up to a heeling angle of  $30^\circ$ ) is considered. The respective information from [9] is compiled in the following table:

#### Limiting Values of $e_{30}$ -Criterion

IMCO	0.055 m rad.
From Test for Ballast Cond.	0.141 m rad.
From Test for Loaded Cond.	0.026 m rad.

#### Actual Values of $e_{30}$ -Criterion

Loaded Cond.	0.055 m rad.
Ballast (Capsize) Cond.	0.096 m rad.

There are two interpretations possible: If it is assumed that the seaway used in the experiments is similar to that on which the IMCO limiting value is implicitly based one can conclude that for the tanker in loaded condition a lower limiting value of  $e_{30}$  would suffice than the IMCO limiting value. But the limiting value necessary to provide sufficient safety for the tanker in ballast condition is according to the test results much higher than the IMCO value.

The second interpretation is that the seaway used for the tests (which corresponded to the seaway during the accident)

was less severe than that underlying the IMCO value. In that case an even higher limiting value for the ballast condition than that derived from the tests would be valid. It is a pity that the capsizing tests have not been made before the accident. Incidentally, one of the purposes of the test program to which the paper by Blume and Hattendorff [10] refers is to adapt limiting values from ships considered safe for vessels the safety of which as yet cannot sufficiently be judged.

### 3.3 Classification of Stability Criteria

Kuo and Welaya [11] divide stability criteria into two categories which they label "statistical approach" and "physical approach". According to them the first category includes criteria based on the metacentric height and the freeboard or on values to be derived from the righting arm curve. The second category includes criteria based on the comparison of the areas under the heeling and righting moment curves (weather criteria). This division does not make much sense because both categories are based on some physical relationships (which in both cases fail to cover the most essential physical phenomenon in this context, namely the dynamic behaviour of a ship in random seas) and both rely on statistical information.

A more rational classification of stability criteria seems possible by grading their efficacy. One possibility to do this is to introduce a measure of the discrimination ability of the particular criteria as was shown in [12]. Another possibility is to state how widely the limiting values of various criteria scatter for a set of different ships. An example for this may be derived from Table 7 of [10].

It should be added here that criteria are predictive models and not explanatory ones. It is clear that for explanatory models another classification would apply.

### 4. COMMENTS ON SOME STABILITY CRITERIA

One of the most used type of stability criteria is that based on the still water righting arm curve. The relationship between characteristics of the righting arm curve and safety against capsizing is purely empirical. It implicitly takes care of all kinds of heeling moments and of the effect of the waves. Therefore it cannot be assumed that the limiting values of this type of criteria are the same for all kinds of vessels. But astonishing enough it was found possible to derive limiting values from casualty statistics which hold for a (not too well defined) sample of ships. Nevertheless, experienced naval architects would never rely blindly on such statistically determined limiting values. They would judge the stability of each particular

ship by comparing its righting arm curve with that of similar ships, existing ones and - if available - capsized ones. Sometimes the view is held that information from existing ships is of little use because they might become casualties in the future. This does not apply if severe weather experience of the existing ships is considered.

When judging the safety of new designs resort can also be taken to model tests. One wonders why this is not done more often. When predicting the propulsive power of ships in nearly each case where not data from an already investigated ship are available, model tests are carried out. Although the consequences of lack of safety are much graver than deviations from the predicted speed, naval architects are reluctant to experimentally investigate the safety against capsizing. Admittedly the procedure of doing capsizing tests is still less developed than that of tests in connection with the powering of ships. But without doubt it could be improved.

Finally, another advantage of criteria based on the righting arm curve remains to be mentioned: They lend themselves to very simple improvements. So it was shown in [12] that a criterion derived by properly combining different characteristics of the righting lever curve provides a better discrimination between safe and unsafe ships of a sample than if the characteristics are used individually. The application of the hull form factor proposed in [10] results in limiting values of criteria which are valid for a wider range of ships than the hitherto used criteria.

Another well-known type of stability criteria is based on a comparison of the heeling and righting arm curve. Sometimes just the levers at various heeling angles are compared and sometimes the areas under the levers (calculated from zero heel or from some windward angle). As criteria either the difference between righting and heeling levers or the area between them or the ratio between certain areas are used. Here too the relationship between the criteria and the safety against capsizing is purely empirical: In so far there is no difference at all between this type of criteria and those based solely on the righting lever curve. Because the latter ones are valid for similar ships only (for which similar heeling moments are to be expected) they implicitly take care of the heeling moments: It is exactly the same procedure if e.g. the limiting values of the righting levers or if the differences between righting and heeling levers have to be determined according to experience with similar ships. Another point which has to be raised is the problem of properly quantifying the moments and the windward heeling angle. Accounting for the arbitrariness so introduced, it can be said that criteria using heeling levers in addition to righting levers have no automatic

advantage against those based solely on the righting lever curve. If some quarters consider criteria making use of heeling levers as superior and physically more sound this may be caused by the misunderstanding that more complexity automatically provides better models. It would be worthwhile to carry out investigations for quantifying the discrimination ability of criteria including heeling moments in order to determine their superiority, if any.

In 1962 it was proposed by the author to use a "probability of capsizing" as stability criterion [13]. The procedure suggested in [13] was elaborated and applied to some ships by Abicht [14]. But as yet it has not been possible to prove the superiority of this method over conventional criteria. It seems that its efficiency (extension of the validity to a wider range of ships and better discrimination between safe and unsafe compared with much more calculating effort) would not be too good. This might be one reason that the probability based criterion did not become accepted.

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## Discussion

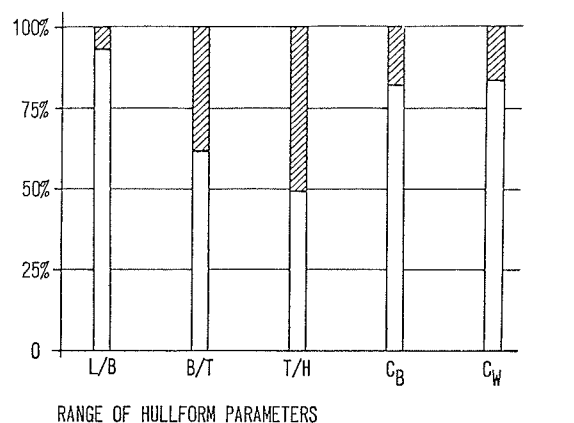
### Author's Comment

In the oral presentation of my paper the following example, which is only shortly indicated in paragraph 3.3 of the paper, was given in more detail. It shows how criteria can be classified according to the range of their validity for different ships.

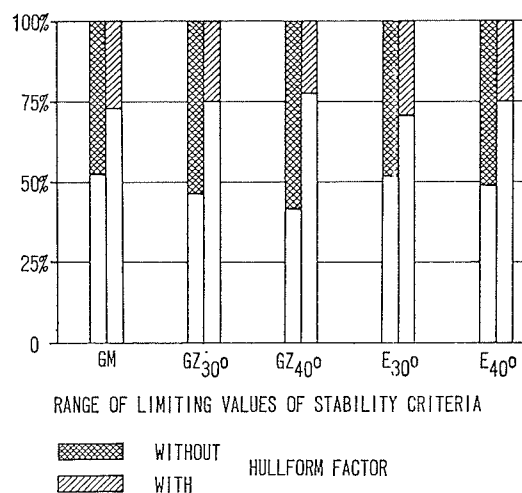
In the upper part of the following figure the range of some form parameters of a set of different ships is given. The parameters are related to the maximum ones occurring in the set. In the lower part of the figure two kinds of criteria are compared. The first kind (marked "without hullform factor") are those presently in use. Let us consider e.g. the  $GZ_{30^\circ}$ -range in the figure. For each ship of the considered set a limiting  $GZ_{30^\circ}$ -value has been determined experimentally. Of course these limiting values (which separate the safe and unsafe conditions of a ship in waves) are different for different ships of the set. If the GZ of that ship of the set which needs the highest GZ in order to be safe is taken as 100%, the GZ of the ship which needs the lowest GZ is only 45% of the highest value. The limiting GZ-values of the other ships are in the range between 45% and 100%. The situation is quite the same if we consider the  $GZ_{40^\circ}$ -values or the areas E below of the GZ-curve: In all cases the range is about 50% of the maximum values. The second kind of criteria shown in the figure (marked "with hullform factor") are a modification of the first kind according to a proposal made by Dr. Blume [10]. One can see that here the range of the limiting values of the ships of the considered set is reduced to about 25% of the maximum values. From this the following can be taken: If we would require e.g. the maximum  $GZ_{30^\circ}$  for all ships of the set, the other which actually needs a lower  $GZ_{30^\circ}$  would be penalized. It would be worst for the ship with the lowest  $GZ_{30^\circ}$ . It would be required to have a  $GZ_{30^\circ}$  twice as high as necessary. If we require the maximum value of ( $GZ_{30^\circ}$  times hullform factor) for all ships of the set, the ships which actually need lower values would be much better off. So the advantage of using criteria with hullform factor is proven. This method can be used to judge the efficacy of any other kind of criteria.

In paragraph 4 of my paper I wrote that for judging the safety of new designs resort can be taken to model tests. In this context I missed to stress the fact (as I did in the oral presentation) that beside the results of model tests the in-

formation to be derived from explanatory mathematical models might who prove very useful for that purpose. I surely consider theory also as a mighty tool - beside others - for judging criteria.



L = ship length  
B = ship breadth  
T = draft  
H = depth  
 $C_B$  = block coefficient  
 $C_W$  = water plane coefficient



GZ = righting arms  
E = area below righting arm curve

Fig. 1 Comparison of the limiting values of two kinds of stability criteria for a range of different hullforms

F. Ursell (Manchester University, UK)

Professor Krappinger is right to emphasize that mathematics by itself cannot solve a physical problem. The most important step is the modelling, i.e. a formulation which includes the most significant features of the problems to a sufficient approximation, and this must be tested by experiment in the history of ship hydrodynamics. There have been cases which mathematics has preceded experiment, or where the experiment has preceded mathematics or where the theory was obviously physically relevant. Examples are the calculation of virtual mass, the development of ocean wave spectra and more recently breaking waves, and wave motion in a rolling tank.

In engineering the insight and experience of the engineer can reduce an adequate solution in fields where mathematics is not yet developed. Professor Krappinger mentions the GZ criterion. Yet, a mathematician might note that in this fundamental problem nobody appears to have counted the numbers of independent dimensionless parameters and that the criterion is not a dimensionless relation but that GZ is expressed as a length.

#### Author's Reply

I very much appreciate the remarks of Prof. Ursell. Of course I agree with what he has said, especially with regard to the mutual relationship between mathematics and experiments (which indicate of course full scale ones too). He is also right suggesting to spend more thoughts on the question of dimensionless independent parameters and criteria. Some parameters as e.g. the height of the bulwark are actually constants. With regard to them I do not see the merit of making them dimensionless. Concerning the GZ-criterion it is shown at the end of paragraph 4 of Dr. Blumes paper [10] under which condition either a dimensional or a dimensionless magnitude might be more advantageous. Again thanks to Prof. Ursell for pointing out this problem.

A. Y. Odabasi (The British Ship Research Association, UK)

The paper searches for an answer to a question which is not defined by the author and the saying of the famous scientist Polya [D1] provides the best possible criticism:

"There is nothing more foolish than trying to answer a question that you do not understand."

The author starts by portraying a confused view of the measure theory which, in itself, is a scientific discipline. Following Stevens the scales of measurement may be considered in four groups: nominal, ordinal, interval and ratio scales. Nominal scales distinguish between objects,

individuals or categories. It associates no values or specific differences with this distinction. Ordinal scale measures qualitatively whether an object has more or less the same attribute in question, in addition to giving distinction between objects. However, an ordinal scale cannot indicate by how much an attribute is more or less than some other attribute, i.e. no quantification. Interval scales indicate the amount of differences in the attributes measured on a relative basis. The ratio scale is obtained by adding an absolute zero to the properties of an interval scale.

It is correct to say that "safety" as a concept should be measured with ordinal scales, since for example an individual may easily consider that flying the Atlantic in an aeroplane is unsafe. However, if this view is adopted no quantitative criteria can be devised and, hence, the classification and the experimentation proposed in the paper is inapplicable since master, crew, or a member of the public may feel that the proposed classification does not provide adequate safety. The same, however, does not apply to intact ship stability since for this specific element of safety a higher order measurement scale exists, even when it is in the form of A.167 recommendations.

The views of the author on models and modelling does not appear well-informed either. A model is a general term denoting any simplified representation of a system. Its objective is to identify significant factors and inter-relationships within the system and between the system and its environment which facilitates the isolation and analysis of the issues that will have the greatest influence on the system's behaviour. Depending on the aim and the level of the modelling, verbal, schematic, iconic and mathematical models may be used. Within the context of intact ship stability, iconic (i.e. scaled model) and mathematical models are commonly used. However, if one would accept the use of ordinal scales the only appropriate model would be verbal models.

Mathematical models can be derived by means of one of the three main approaches: transport principle, population-balance principle, and empirical formulation, cf. Himmelblau and Bischoff [D2]. Almost all of the presently employed mathematical models describing large amplitude rolling motion have been derived from a transport principle (not by heuristic act!) under some assumptions which may necessitate the inclusion of empirical information. However, if one remembers that the elasticity modules of steel is also an empirical quantity this is not a serious drawback provided one knows what the assumptions imply. Ideally, one would like to make the model very simple and easy to understand. This, however, is not always possible. Albert Einstein states that:

"Everything should be made as simple

as possible, but not simpler."

Therefore, if the problem demands a certain degree of complexity it has to be accepted. Here it is of the utmost importance to establish a correct and meaningful relationship between the abstraction and physics of the phenomenon; this in turn demands that the investigator should be more multi-disciplinary, enabling him (or her) to communicate with specialists and workers of other professional disciplines at a meaningful level. For example, an investigator should know that terminology and the basics of the theory of stability of motion belong to the scientific discipline called "analytical mechanics", not mathematics as claimed by the authors of Ref. [3] and the present author.

In part 2 the author still tries to equate safety and intact ship stability and attempts to provide a verbal model which adds nothing to what is already known, except confusion and misinterpretation.

The title of the next part is, in itself, a misnomer since one only models a phenomenon and derives a stability criterion. Luckily, the views expressed in this part are not accepted by the engineering profession for otherwise we should be still living in the "stone age".

The paper advises the use of model experiments as a routine to build up data to devise A.167-type criteria for differing ship types and loading conditions. Whilst appreciating the usefulness (for a large number of reasons) it is important to appreciate the difference to be expected between the behaviour of a scaled model and a ship, due to the role played by the viscous effects, which are known to be important in large amplitude rolling motion. To remedy this shortcoming, one needs to have a clear idea of viscous effect and a corresponding mathematical model which, according to the author, is too complicated and useless. The usefulness of model experiments within this frame of understanding is best demonstrated by the famous quotation of Norbert Wiener [D3].

"... things do not, in general, run around with their measures stamped on them like the capacity of a freight car; it requires a certain amount of investigation to discover what their measures are ... What most experimenters take for granted before they begin their experiments is infinitely more interesting than any results to which their experiments lead."

I propose that the author should re-evaluate the state-of-the-art in view of Fig. 1. He may then realise that a great deal of effort is being spent to improve total efficiency by improving component efficiencies.

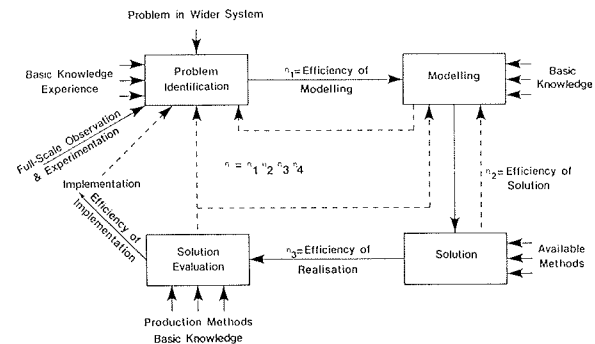


Fig. 1 A Simplified Flow-chart of Systems Approach

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#### Author's Reply

I am sorry to say that I disagree to a great extent with the approach which Dr. Odabasi is taking with respect to ship stability. Therefore, I would have felt uneasy if he would have agreed with my approach. So I only can thank him for a contribution being exactly of that kind which is to be expected from him.

J.H. Hwang (Seoul National University, Korea)

It has been reported that capsizing accidents of fishing boats and cargo vessels were occurred very often in quartering and following sea conditions.

In a quartering sea, the most dangerous situation is subjected to the dynamic effect of ship motion especially due to shipping water on a deck and cargo shifting. And in a following sea, the loss of statical stability makes a trouble.

It is well understood that these situations are occurred in a operation at relatively higher ship speed.

From the above view point, I would like to comment that ship stability criteria must be set up as a function of sea state, ship speed, heading angle of encounter wave in a proper way.

#### Author's Reply

In a particular situation the safety against capsizing is of course depending on the parameters mentioned by Prof. Hwang. There are even more parameters to be

considered (as e.g. wind, shifting of cargo). Because we are not yet able to determine the safety against capsizing theoretically we are using criteria which are based on experience and which implicitly take care of all effects which as yet cannot be dealt with in a physical correct and comprehensive manner, of course it would be possible to develop criteria which explicitly depend on the seaway, ship speed etc. (see

e.g. [13] and [14]). But as I have mentioned in my paper the efficacy of such criteria is not too good; therefore they are not worth the amount of calculating work which they require. This might change in future. But before more elaboration criteria are adopted it would have to be shown (e.g. with methods explained in [12]) that their efficacy is better than that of hitherto used criteria.

*Session IIa*

## Fundamentals of Stability

*Chairmen*

Mr. Torben Munk  
Danish Ship Research Laboratory  
Denmark

Prof. Seiji Takezawa  
Yokohama National University  
Japan

# THEORETICAL STUDY OF LINEAR STABILITY OF FLOATING BODIES IN SEA

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## ABSTRACT

It is shown mathematically that the classic sufficient condition for stability of equilibrium for a floating body in an open sea is a necessary and sufficient condition for "linear stability". Some developments including linear and nonlinear anchorings are given.

## NOMENCLATURE

$A$	linear evolution operator in (17)
$B$	linear operator in (15)
$C_0^\infty$	space of infinitely differentiable functions with compact support
$E$	$\rho E$ inertial matrix of $\mathcal{B}$
$E'$	symmetric matrix in (35)
$F$	floating surface
$G$	fluid domain at equilibrium
$H$	symmetric nonnegative matrix in (14)
$H'$	symmetric matrix in (28)
$H^1$	Sobolev space
$H_D$	Dirichlet space
$I_{00}$	moment of inertia of $F$
$J$	linear operator in (7)
$J^*$	adjoint of $J$
$K$	restoring hydrostatic matrix in (19)
$K'$	anchoring restoring matrix in (35)
$L$	linear operator in (24)
$L^2$	Lebesgue space
$M$	linear operator in (36)
$N$	nonlinear operator in (38)
$P_c$	selfadjoint projector in (23)
$S_B$	bottom of the sea
$S_F$	still free-surface
$S_I$	immersed surface of $\mathcal{B}$
$U(t)$	group of linear operators

$W$	energy form in (8)
$\mathcal{B}$	floating body
$\mathcal{B}^-$	volume of displaced liquid
$\mathcal{F}$	space of finite energy in (9)
$\mathcal{H}$	space of possible states with finite energy in (11)
$\mathcal{H}_c$	subspace of $\mathcal{H}$ in (22)
$f_\lambda$	eigenvector of $A$
$g$	gravity acceleration
$h$	functions in (13)
$h'$	functions in (27)
$k$	eigenvalue of $K$
$\vec{n}$	normal pointing outside $G$
$p$	generalized normal on $S_I$
$s$	parameters of position of $\mathcal{B}$
$t$	time
$u$	state of the body-fluid system
$v$	parameter of velocity of $\mathcal{B}$
$x=(x_1, x_2, x_3)$	coordinates
$x^0$	coordinates of $\omega^0$
$x^P$	coordinates of the center of buoyancy
$\partial_n$	normal derivative
$\Gamma$	anchoring forces on $\mathcal{B}$ in (35)
$\Phi$	potential of the flow
$\eta$	free-surface elevation
$\lambda$	eigenvalue of $A$
$\rho$	density of the fluid
$\omega^0$	center of gravity of $\mathcal{B}$ at equilibrium
$\sigma$	eigenvector of $K$

## 1. INTRODUCTION

The determination of an equilibrium position for a floating body in an open sea is a well-known subject in mechanics. The first result seems due to P.Appel [1]. In this classical theory, the body-fluid motion is disregarded and an equilibrium posi-



tion is considered as stable if the vertical coordinate of the center of gravity of the fluid-body system has a minimum. So the conditions obtained can only be considered as sufficient but not as necessary. Later F. John [2] gives an other method taking into account the motion of the liquid and the non-hydrostatic nature of the pressure. He obtains a sufficient condition for stability which is nothing but the classical static condition. However his proof is based on a somewhat intuitive consideration of the flow of energy.

Here we present a rigorous mathematical proof of the fact that this classical condition is sufficient and necessary. By studying the small motions near equilibrium of the fluid-body system, we prove that the motion of the body after a small disturbance stays small if and only if an hydrostatic restoring matrix is positive-definite. We do not consider the full non linear problem of the real motion, thus our condition can be considered as a necessary and sufficient condition of "linear stability".

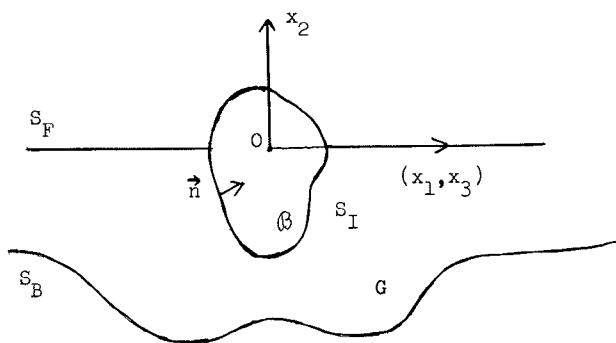
In § 2 we study the linearized problem connected with the transient motion of the (unbounded) fluid-body system through a method of functional analysis initiated by J.T. Beale [3] in order to obtain in § 3 the condition of stability. Endly in § 4 we give some developments and comments.

The essential part of this communication is extracted from [4] and for reader's commodity we practically reproduce in § 3 the proof given in this reference.

## 2. THE LINEARIZED PROBLEM OF THE TRANSIENT MOTION OF A FLOATING BODY

### 2.1 Statement of the problem

This problem involves linearized equations written on fixed domains connected with the equilibrium position (mass of the displaced liquid equal to that of the body, center of mass of the displaced liquid on the same vertical as that of the body).



Let  $x^0$  be the coordinates of the center of mass  $\omega^0$  of  $B$  at equilibrium. For commodity we assume the depth is bounded.

The mechanical state of the fluid-body system can be described by the quadruples  $u = (\phi, \eta, s, v)$ :  
 $\phi(x, t)$  potential of the flow,  
 $\eta(x_1, x_3, t)$  elevation of the free-surface,  
 $s(t) \in \mathbb{R}^3$  three parameters of position of the

body  $B$ :

$s_1(t) + x_2^0$  is the ordinate of the center of mass of  $B$  at time  $t$ ,

$s_2(t), s_3(t)$  are the  $0 x_1, 0 x_3$  - components of the rotation of  $B$ ,

$v(t) \in \mathbb{R}^6$ : the velocity parameters of  $B$ .

The classical equations of the coupled motion are :

$$\Delta \phi = 0 \quad \text{in } G \quad (1)$$

$$\phi_t = -g\eta \quad \text{on } S_F \quad (2)$$

$$\eta_t = \partial_n \phi \quad \text{on } S_F \quad (3)$$

$$\partial_n \phi = 0 \quad \text{on } S_B \quad (4)$$

$$\partial_n \phi = v \cdot p \quad \text{on } S_I \quad (5)$$

$p_i, i = 1, 3$ , are the components of the normal  $\vec{n}(M)$

$p_i, i = 4, 6$ , are the components of  $\vec{\omega}_M \wedge \vec{n}(M)$ .

$$E v_t = -g J K s - \int_{S_I} \phi_t p \quad (6)$$

$\rho E$  is the inertial matrix of  $B$ ,  $K$  is a  $(3, 3)$  symmetric hydrostatic restoring matrix and  $J$  is the operator

$$a = (a_1, a_2, a_3) \in \mathbb{R}^3 \rightarrow J a = (0, a_1, 0, a_2, 0, a_3) \in \mathbb{R}^6 \quad (7)$$

If the initial state  $u^0$  is given, the problem (P) is to determine the state  $u(t)$  at each positive instant.

### 2.2 The space $\mathcal{H}$ of possible states with finite energy

The mechanical energy of the fluid-body system is proportional to :

$$W(u) = \int_G |\nabla \phi|^2 + g \int_{S_F} |\eta|^2 + g K s \cdot s + E v \cdot v \quad (8)$$

So we introduce the functional space :

$$\mathcal{F} = H_D(G) \times L^2(S_F) \times \mathbb{R}^3 \times \mathbb{R}^6 \quad (9)$$

where  $H_D(G)$  is the closure of  $C_0^\infty(\bar{G})$ , space of infinitely differentiable functions with compact support in  $\bar{G}$ , for  $\varphi \in \mathcal{F}$ ,  $\|\varphi\|_{\mathcal{F}} = \{ \int_G |\nabla \varphi|^2 \}^{1/2}$ .

If  $K$  is positive-definite  $W(u)^{1/2}$  is an hilbert-norm on  $\mathcal{F}$ , if not we put :

$$\|u\|_{\mathcal{H}}^2 = \int_G |\nabla \phi|^2 + g \int_{S_F} |\eta|^2 + g s \cdot s + E v \cdot v \quad (10)$$

In order to take into account the stationary conditions (1), (4), (5), we define a closed subspace  $\mathcal{h}$  of  $\mathcal{F}$ : the space of possible states with finite energy of the fluid-body system. In this way we consider the functional spaces :

$H^1(G)$  Sobolev space of order 1 equipped with the norm

$$|\varphi|_{H^1(G)}^2 = \int_G |\nabla \varphi|^2 + \int_{S_F} |\varphi|^2$$

$$H_1^0(G) = \{\varphi \in H^1(G) : \varphi|_{S_F} = 0\}$$

Thus  $\mathcal{H}$  is defined by :

$$\mathcal{H} = \{u = (\phi, \eta, s, v) \in \mathcal{F} :$$

$$(\nabla \phi, \nabla \varphi)_G = (v, p, \varphi)_{S_I} \quad \forall \varphi \in H_1^0(G)\} \quad (11)$$

The elements of  $\mathcal{H}$  verify (1), (4), (5) in a weak sense.

### 2.3 Weak formulation of (P) :

According to [3] we construct an operator  $A$  with domain  $D(A)$  in  $\mathcal{H}$  so that

$$\frac{du}{dt} = Au, \quad u(0) = u^0 \quad (12)$$

may be a weak formulation of (P).

Let  $h_i$ ,  $i = 1, 6$ , the unique solution in  $H^1(G)$  of :

$$\begin{aligned} \Delta h_i &= 0 & \text{in } G \\ h_i &= 0 & \text{on } S_F \\ \partial_n h_i &= p_i & \text{on } S_I \\ \partial_n h_i &= 0 & \text{on } S_B \end{aligned} \quad (13)$$

and  $H$  be the symmetric nonnegative matrix

$$H_{ij} = \int_{S_I} h_i p_j \quad (14)$$

For  $\eta \in L^2(S_F)$ , if the elliptic problem

$$\begin{aligned} \Delta \psi &= 0 & \text{in } G \\ \psi &= \eta & \text{on } S_F \\ \partial_n \psi &= 0 & \text{on } S_I \cup S_B \end{aligned} \quad (15)$$

has a (unique) solution in  $H^1(G)$  we put  $\eta \in D(B)$ ,  $\psi = B\eta$ .

Now we can verify that  $A$  can be defined by :

$$D(A) = \{u = (\phi, \eta, s, v) \in \mathcal{H} : \partial_n \phi|_{S_F} \in L^2(S_F), \eta \in D(B)\} \quad (16)$$

$$\begin{aligned} (Au)_1 &= -g B \eta + w \cdot h \\ (Au)_2 &= \partial_n \phi|_{S_F} \\ (Au)_3 &= J^* v \\ (Au)_4 &= w \equiv (E + H)^{-1} \left\{ -g J K s + \int_{S_I} g B \eta p \right\} \end{aligned} \quad (17)$$

## 2.4 Resolution of (12)

### 2.4.1 $K$ is positive-definite

In this case Beale [3] has shown that  $A$  is skew-adjoint in  $\mathcal{H}$  with the norm  $W(u)^{1/2}$ , thus  $A$  is the infinitesimal generator of a unitary group  $U(t)$  on  $\mathcal{H}$ . Then if  $u^0 \in D(A)$ , (12) has an unique solution :

$$u(t) = U(t) u^0 \quad (18)$$

which is  $C^1[\mathbb{R}, \mathcal{H}] \cap C^0(\mathbb{R}, D(A))$ . Moreover (12-17) imply that  $\phi_t \in H^1(G)$ , then if

$\phi^0 \in H^1(G)$ ,  $\phi(t) \in H^1(G)$ ,  $\forall t \in \mathbb{R}$ , thus (2) is true in  $L^2(S_F)$ .

### 2.4.2 $K$ is indefinite

Now we consider the previous functional spaces as complex-spaces.

Let  $\mathcal{B}^- = \mathcal{B} \cap \{x_2 < 0\}$  and  $F = \{x_2 = 0\} / \overline{S_F}$ , the

expression of  $K$  is [6] :

$$K = \begin{bmatrix} I^F & -I_{x_3}^F & I_{x_1}^F \\ -I_{x_3}^F & I_{x_2}^{\mathcal{B}^-} + I_{x_3 x_3}^F & -I_{x_1 x_3}^F \\ I_{x_1}^F & -I_{x_1 x_3}^F & I_{x_2}^{\mathcal{B}^-} + I_{x_{11}}^F \end{bmatrix} \quad (19)$$

$$\begin{aligned} \text{with } I_{x_2}^{\mathcal{B}^-} &= \int_{\mathcal{B}^-} (x_2 - x_2^0) dx_1 dx_2 dx_3, \\ I^F &= |F| \left( = \int_F dx_1 dx_3 \right), \\ I_{x_1}^F &= \int_F (x_1 - x_1^0) dx_1 dx_3, \dots \end{aligned}$$

We see, for instance, that a submerged body ( $F = \emptyset$ ) induces a matrix  $K$  with a kernel,  $W(u)^{1/2}$  is no more a norm on  $\mathcal{H}$ . From now on we equip  $\mathcal{H}$  with the norm (10) and split  $A$  into  $A = A_1 + A_2$  with :

$$\begin{aligned} D(A_1) &= D(A) \\ (A_1 u)_1 &= -g B \eta + w \cdot h \\ (A_1 u)_2 &= \partial_n \phi|_{S_F} \\ (A_1 u)_4 &= (E + H)^{-1} \left\{ -g J s + \int_{S_I} g B \eta p \right\} \equiv w \end{aligned} \quad (20)$$

$$\begin{aligned} (A_2 u) &= (w_2 \cdot h, 0, 0, w_2) \\ w_2 &= (E + H)^{-1} (-g J (K - I) s) \end{aligned} \quad (21)$$

According to 2.4.1,  $A_1$  is skew-adjoint in  $\mathcal{H}$ .  $A_2$  is compact on  $\mathcal{H}$ , thus [5]  $A$  is the infinitesimal generator of a group  $U(t)$  with

$$\|U(t)\|_{\mathcal{L}(\mathcal{H})} \leq \exp \{ \|A_2\|_{\mathcal{L}(\mathcal{H})} |t| \} \quad \forall t \in \mathbb{R}$$

So we have again existence and uniqueness of a solution of (P) and  $W(U(t)) = W(u^0)$ ,  $\forall t \in \mathbb{R}$ .

In order to obtain our result about stability we, now, prove that  $A$  has a finite number of real eigenvalues. First, the spectrum of  $A_1$  is the whole imaginary axis,  $A$  is a compact perturbation of  $A_1$ , then [5]  $A$  has at most a finite number of eigenvalues with a real part not equal to 0. Next we prove that these ones are real. Let  $\mathcal{H}_c$  be the closed subspace of  $\mathcal{H}$  connected with the kinetic components in  $\mathcal{H}$  :

$$\mathcal{H}_c = \{u = (\phi, \eta, s, v) \in \mathcal{H} : \eta = 0, s = 0\} \quad (22)$$

and  $P_c$  be the projector on  $\mathcal{H}_c$ . It is immediate that :

$$\left. \begin{aligned} P_c A^2 u &= A^2 P_c u \\ P_c A_1^2 u &= A_1^2 P_c u \end{aligned} \right\} \quad \forall u \in D(A^2) \cap \mathcal{H}_c \quad (23)$$

$A_1^2$  is selfadjoint in  $\mathcal{H}$ , (23) implies that  $L_1 \equiv A_1^2 P_c$  is selfadjoint and nonpositive in  $\mathcal{H}_c$ . We put :

$$L = A^2 P_c = L_1 + L_2 \quad (24)$$

with

$$\left. \begin{aligned} L_1 u &= (-g B \partial_n \phi + z_1 \cdot h, 0, 0, z_1) \\ z_1 &= (E + H)^{-1} \left\{ \int_{S_I} g B \partial_n \phi - g J J^* v \right\} \end{aligned} \right\} \quad (25)$$

$$\left. \begin{aligned} L_2 u &= (z_2 \cdot h, 0, 0, z_2) \\ z_2 &= (E + H)^{-1} \{-g J (K - I) J^* v\} \end{aligned} \right\} \quad (26)$$

$L_2$  is compact on  $\mathcal{H}_c$  and selfadjoint because of the symmetry of  $K$ . So  $L$  is selfadjoint and arguing as previously we obtain that  $L$  has at most a finite number of positive eigenvalues

$\lambda_j^2$ ,  $j = 1, m$ , and  $(L u, u)_{\mathcal{H}}$  is not positive on the orthogonal complement in  $\mathcal{H}_c$  of the corresponding eigenvectors.

Thus (23) shows that the possible eigenvalues of  $A$  which are not purely imaginary are real. By a sample calculus, we can verify that the  $\lambda_j$  are as follows. For each real  $\lambda \neq 0$ , there exist  $h'(\lambda)_i$ ,  $i = 1, 6$ , in  $H^1(G)$  such that :

$$\left. \begin{aligned} \Delta h'_i &= 0 && \text{in } G \\ \partial_n h'_i + \frac{\lambda^2}{g} h'_i &= 0 && \text{on } S_F \\ \partial_n h'_i &= p_i && \text{on } S_I \\ \partial_n h'_i &= 0 && \text{on } S_B \end{aligned} \right\} \quad (27)$$

Let  $H'(\lambda)$  be the symmetric nonnegative matrix :

$$H'_{ij} = \int_{S_I} h'_i p_j \quad (28)$$

Then  $\lambda_j^2$  are such that there exist  $v_{\lambda_j} \neq 0$ , solution of :

$$\{\lambda_j^2 (E + H'(\lambda_j)) + g J K J^*\} v_{\lambda_j} = 0 \quad (29)$$

Finally a same calculus shows that  $\lambda_j$  and  $-\lambda_j$  are eigenvalues of  $A$  with corresponding eigenvectors :

$$\begin{aligned} f_j^\pm &= (v_{\lambda_j} \cdot h'(\lambda_j), \mp \frac{\lambda_j}{g} v_{\lambda_j} \cdot h'(\lambda_j), \\ &\quad \pm \frac{1}{\lambda_j} J^* v_{\lambda_j}, v_{\lambda_j}) \end{aligned} \quad (30)$$

## 2.5 Some remarks

It is also possible to use the previous techniques to study the forced motion of the fluid-body system (forces on the body, pressure on  $S_F$ , small motion of part of the bottom) : see [4], [6], [7].

From this approach of the problem we deduced in [4], [7] a construction of a numerical approximation of the solution.

## 3. STABILITY OF EQUILIBRIUM OF A FLOATING BODY

According to [2] "we take as definition of stability that an infinitesimal momentary disturbance causes a body displacement which stays infinitesimal at all subsequent times (except for a purely horizontal part)". So for each

$u^0 \in D(A)$ ,  $|s(t)|_{\mathbb{R}^3}$  must stay bounded at each positive  $t$ . Now we prove the

### Theorem 1 :

A necessary and sufficient condition for stability is that  $K$  is positive definite.

The following lemma of [6] is fundamental :

### Lemma 1 (Neumann's problem in $H_D(G)$ ) :

If  $f_I \in L^2(S_I)$ ,  $f_F \in L^2(S_F)$ , support of  $f_F$  is a compact in  $S_F$  and if  $\int_{S_I} f_I + \int_{S_F} f_F = 0$ , there exists a unique  $\psi$  in  $H_D(G)$  such that :

$$(\nabla \psi, \nabla \varphi)_G = (f_I, \varphi)_{S_I} + (f_F, \varphi)_{S_F} \quad \forall \varphi \in H^1(G)$$

(Such a  $\psi$  verifies in a weak sense  $\Delta \psi = 0$  in  $G$ ,  $\partial_n \psi|_{S_I} = f_I$ ,  $\partial_n \psi|_{S_F} = f_F$ ,  $\partial_n \psi|_{S_B} = 0$ )

Proof of theorem 1 : If  $K$  is positive definite,  $U(t)$  being unitary it is clear that the equilibrium is stable.

If the smallest eigenvalue of  $K$  is 0 and if  $K \sigma = 0$ , (19) implies :

$$I^F \sigma_1 - I_{x_3}^F \sigma_2 + I_{x_1}^F \sigma_3 = 0 \quad (31)$$

but we have (divergence's theorem) :

$$\int_{S_I} p_2 = I^F \int_{S_I} p_4 = - I_{x_3}^F \int_{S_I} p_6 = I_{x_1}^F \quad (32)$$

then

$$\int_{S_I} p \cdot J \sigma = 0 \quad (33)$$

According to lemma 1, let  $\phi^0$  be the solution of the Neumann's problem with  $f_I = J \sigma \cdot p$ ,  $f_F = 0$ ; we have  $u^0 \equiv (\phi^0, 0, 0, J \sigma)$ ,  $A u^0 = (0, 0, \sigma, 0)$ ,  $A^2 u^0 = 0$ . So the solution of (12) is such that  $u(t) = u^0 + t A u^0 + \int_0^t (t-\tau) U(\tau) A^2 u^0 d\tau = u^0 + t A u^0$ .

Thus  $s(t) = t \sigma$  and  $|s(t)|_{\mathbb{R}^3} \rightarrow +\infty$  if  $t \rightarrow +\infty$ : the equilibrium is unstable.

Finally if the smallest eigenvalue  $k$  of  $K$  is strictly negative, lemma 1 allows us to prove that  $A$  has a positive eigenvalue  $\lambda$  with eigenvector  $f_\lambda^+$ . Thus if  $u^0 = f_\lambda^+$ , the solution of (12) is  $u(t) = \exp\{\lambda t\} f_\lambda^+$ ; (29) implies that  $J^* v_\lambda \neq 0$  and by (30)

$|s(t)|_{\mathbb{R}^3} = \exp\{\lambda t\} |J^* v_\lambda|_{\mathbb{R}^3} \rightarrow +\infty$  if  $t \rightarrow +\infty$ : the equilibrium is unstable. It remains to prove the existence of such a  $\lambda$ : by 2.4.2 it is sufficient to display  $u \in \mathcal{H}_c$  such that  $(L u, u)_{\mathcal{H}_c} > 0$ .

If  $K \sigma = k \sigma$  we put  $C = \int_{S_I} J \sigma \cdot p$ . For each compact of  $S_F$ , with a non empty interior and lying at a positive distance of  $\partial S_F$ , we note  $\mathcal{E}_R$  the product of the characteristic function of this compact by  $-C/R$ ,  $R$  being the measure of the compact. Let  $\mathcal{E}_{R\theta}$  be the convolution product of  $\mathcal{E}_R$  by  $\chi_\theta$  the regularizing function of Friedrichs with

$$0 < \theta < \text{dist}(\partial S_F, \text{support of } \mathcal{E}_R).$$

Thus  $\mathcal{E}_{R\theta} \in C_0^\infty(S_F)$  and  $\int_{S_I} J \sigma \cdot p + \int_{S_F} \mathcal{E}_{R\theta} = 0$ .

By lemma 1, there exists  $\psi \in H_D(G)$  solution of the Neumann's problem with  $f_I = J \sigma \cdot p$  and  $f_F = \mathcal{E}_{R\theta}$ . So  $u \equiv (\psi, 0, 0, J \sigma) \in \mathcal{H}$ ,

$A u = (0, \mathcal{E}_{R\theta}, \sigma, 0) \in D(A)$  and

$$(L u, u)_{\mathcal{H}} = -W(A u) = -\int_{L^2(S_F)} |\mathcal{E}_{R\theta}|^2 + k \sigma \cdot \sigma$$

By choosing  $R$  large enough and  $\theta$  small enough we have  $(L u, u)_{\mathcal{H}} > 0$ .

F. John [2] gave a necessary and sufficient condition for  $K$  be definite-positive, so we have the

#### Theorem 2 :

A necessary and sufficient condition for stability is that  $F$  is not empty and

$$x_2^0 - x_2^p \leq \frac{I_{\theta\theta}^F}{|\mathcal{B}^-|} \quad (34)$$

$x_2^p$  is the ordinate of the center of buoyancy,  $I_{\theta\theta}^F$  the maximum of moment of inertia of  $F$  about any horizontal axis,  $|\mathcal{B}^-|$  the measure of the volume of displaced liquid.

This is nothing but the stability condition derived by statics [1]. This condition does not involve the shape of  $S_B$ .

#### 4. DEVELOPMENTS AND COMMENTS

By a sample change of some functional spaces, we can prove that the condition is again valuable in the case of infinite depth.

In order to avoid purely horizontal displacements, the floating body can be equipped with anchorings. If we suppose that the elements of reduction in  $\omega^0$  of these forces are of the form

$$\Gamma = -K' s - E' v \quad (35)$$

where  $s$  is, henceforth, all the six parameters of position of  $\mathcal{B}$ , and  $K'$  and  $E'$  symmetric (6,6) matrix, we can solve the problem as previously. We replace  $g K$  by  $g J K J^* + K'$  in (8) and  $\mathbb{R}^3$  by  $\mathbb{R}^6$  in (9), (P) can, then, be formulated by :

$$\frac{du}{dt} = A u + M u \quad u(0) = u^0 \quad (36)$$

The new expression of  $A$  involves  $-[g J K J^* + K']s$  in place of  $-g J K s$  in (17) and

$$M u = (z, h, 0, 0, z) \quad (37)$$

$$z = -(L + H)^{-1} E' v$$

By using the same kind of arguments as in §§ 2,3, we obtain that a necessary and sufficient condition for stability is that  $(g J K J^* + K')$  is definite positive and  $E'$  is not positive.

Next we can consider the case where the anchoring forces and moments are non linear functions  $\Gamma(s, v)$ . Then (P) can be formulated by

$$\frac{du}{dt} = A u + N(u) \quad (38)$$

with  $N$  a non linear operator on  $\mathcal{H}$  :

$$N(u) = (z, h, 0, 0, z) \quad (39)$$

$$z = (E + H)^{-1} \Gamma(s, v)$$

If  $\Gamma(s, v) = \Gamma(v)$  and  $\Gamma(0) = 0$ , we can deduce immediately that a sufficient condition for stability is that  $K$  is positive-definite and

$$(\Gamma(v) - \Gamma(v')) \cdot (v - v') \leq 0 \quad \forall v, v' \in \mathbb{R}^6 \quad (40)$$

$\Gamma$  is then a dissipative function of  $v$ . If  $\Gamma$  depends on  $s$  the situation is more complex.

Finally we can consider, by the same technics, the problem of the vibrations of an immersed elastic structure and we refer to [8] for the formulation of this problem.

As a natural consequence we can consider complex systems including rigid bodies and elastic structures.

We insist on the fact that we only have solved the linear problem; the full non linear problem is far more difficult in particular because of the "free-boundary" aspect.

#### ACKNOWLEDGEMENTS

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## Discussion

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Fritz John in his 1949 paper discussed mainly the linear time-periodic problem; the transient problem was not discussed except for the stability condition. Thus the present work is most welcome. A treatment of the linear transient motion of a floating circular cylinder was given by Maskell and Ursell (Jl Fluid Mech, 44, 1970, 303-313). This involved the computation of a very slowly convergent Fourier integral. Complex-variable theory was used but it was not necessary to find the complete motion of the fluid.

Dr. Licht's methods are the methods of modern abstract functional analysis which French mathematicians are now applying to industrial problems and which are powerful in proving existence theorems but not yet so powerful numerically. Progress is now being made in this direction, and I believe that these methods will become increasingly familiar to engineers.

Author's Reply

I thank Professor Ursell for his comments on my work. In the references [4], [6], [7] we deduced a numerical approximation from the theoretical formulation of the continuous problem of the transient motion of the fluid-body system that we have described in this paper. Let us give some details about the method. It uses an implicit scheme of time discretisation, finite elements in a truncated fluid domain and a radiation condition on the outer boundary. It is shown that for a given time interval the solution of the discrete problem converges towards the solution of the initial problem when first the distance of truncation goes to infinity and next the time step  $\Delta t$  and the space discretisation parameter  $h$  go independently to zero. In practical terms at each time step we have to solve always the same sparse symmetric linear system, thus the associated matrix is factorized once for all by, for example, Choleski's method; the second member involves only informations from the previous step. The 2-D numerical results agree well with experimental data and those obtained by "more classical" analysis (Fourier. Laplace transformation or intergro-differential equation).

## A MORPHOLOGY OF MATHEMATICAL STABILITY THEORY AND ITS APPLICATION TO INTACT SHIP STABILITY ASSESSMENT

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### SUMMARY

This paper aims at providing a categorised account of mathematical stability assessment methods within the context of intact ship stability. Following a brief description of the historical developments the basic concepts of stability assessment are introduced and a mathematical definition of ship stability is presented. Mathematical stability assessment methods are classified in three groups: State space methods, frequency domain methods, and topological methods. In the next part the concept of practical stability is introduced as a specific application of Lyapunov's direct method and the results of practical stability calculations for a stern trawler are presented. To demonstrate the usefulness of topological methods the resonance phenomena for Duffing's oscillator is studied and the differences with linear resonance is demonstrated. Finally, some thoughts on an improved understanding are expressed. The text is supplemented with two appendices to provide background information on topological dynamics.

### 1. INTRODUCTION

The study of stability of motion has attracted the attention of scientists and engineers since the seventeenth Century. The very first studies undertaken by Newton (1642-1727) and Euler (1707-1783) were aimed at investigating the stability of the solar system by solving the equations of motion. Lagrange (1736-1783) in his studies preferred a different approach and devised a method for assessing the stability of equilibrium positions.

He concluded that an equilibrium position is stable if the potential energy of the system assumes a minimum value at this point. His studies were obviously limited to conservative systems. Later, Routh and Hurwitz independently proposed a method for the stability assessment of linear time-invariant systems. Poincaré (1854-1912) extended Lagrange's energy method by developing phase plane techniques for second order systems and introduced the concepts of index and limit cycle. Lyapunov (1857-1918) in his celebrated doctoral dissertation[1], devised a novel method by exploiting the energy argument for any type of motion and for any order of system. After a dormant period Lyapunov's direct method has been applied and extended by numerous investigators, covering a broad range of professional disciplines from population ecology to spacecraft trajectory control. A recent book by Siljak[2] provides an excellent demonstration of this fact.

In the present-day application the mathematical stability theory follows three main lines of application:

#### (1) State Space Methods

This approach utilises the Lyapunov's direct method as anticipated within Lyapunov's original study[1]. As a result of fundamental investigations the method has been extended to discontinuous systems[3], to countable and other infinite systems of equations[4], to differential-difference equations[3], to integro-differential equations[5], and to stochastic systems[6]. In most of the extensions referred to so far scalar or functional Lyapunov functions have been

used. In 1962 Bellman[7] introduced the concept of vector Lyapunov functions in the treatment of large scale systems. The advantage of vector Lyapunov functions arises from the fact that components of a vector Lyapunov function are required to satisfy less restrictive requirements than a single scalar Lyapunov function[7]. From the ship stability point of view possibly the most important application of the vector Lyapunov functions, together with the use of differential inequalities[8], is the study of partial stability, i.e. stability with respect to a part of variables, cf. Matrasov[9] which enables one to verify the stability of motion with respect to roll when the coupling with the other modes of motion is present.

## (2) Frequency Domain Methods

During the early development of the mathematical stability theory a special non-linear equation, known as the problem of Lur'e[9], received considerable attention:

$$\begin{aligned}\dot{\underline{x}} &= \underline{A} \underline{x} + \underline{b} \tau, \quad \tau = -g(\sigma), \\ \sigma &= \underline{h}^T \underline{x}\end{aligned}\quad (1.1)$$

where  $\underline{A}$  is a  $n$  by  $n$  real matrix, the vectors  $\underline{b}$ ,  $\underline{h}$  and  $\underline{x}$  are of dimension  $n$ ,  $\sigma$  is a linear combination of the state variable  $\underline{x}$ ,  $\tau$  is a non-linear function of  $\sigma$ , superscript  $T$  denotes the transpose, and  $(\cdot)$  denotes the derivation with respect to time. Lur'e and Postnikov[10] proposed a Lyapunov function candidate

$$V(\underline{x}) = \frac{1}{2} \underline{x}^T \underline{P} \underline{x} + \beta \int_0^\sigma g(\xi) d\xi \quad (1.2)$$

and sought the stability conditions for  $g(\sigma)$  lying in the first and the third quadrants of the  $(\sigma, \tau)$ -plane, where  $\underline{P} = \underline{P}^T$  is a symmetric positive definite matrix and  $\beta$  is a non-negative constant. In 1961 the Rumanian scientist Popov[11] provided a solution to the problem of Lur'e by proving that the satisfaction of a condition in the frequency domain ensures the existence of a Lyapunov function. Later, Kalman[12] proved that the frequency domain condition of Popov was not only necessary but also sufficient for the existence of the proposed Lyapunov function. In this way the link between the frequency domain methods and the Lyapunov's theory has been fully established (see Fig.1). Further work by Yakubovich[13], Halanay[14], and many others improved both the performance and the generality of the method. A particular attraction of the frequency domain approach is due to the fact that when applicable it is equivalent to the Lyapunov's direct method and yet it allows for a simple geometric interpretation. A

fairly recent account on the subject has been made available by Vidyasagar[15].

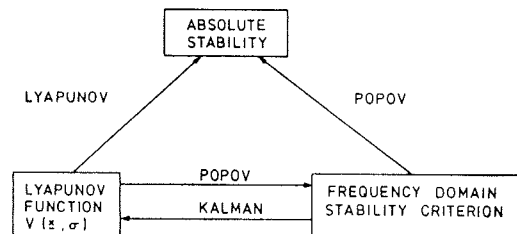


Fig.1 Relationship between Lyapunov's and Popov's Methods

## (3) Geometric (or Topological) Methods

The geometric theory of differential equations was founded by Henri Poincaré[16]. He turned his attention to the system of equations

$$\dot{x} = P(x, y), \quad \dot{y} = Q(x, y) \quad (1.3)$$

where  $P$  and  $Q$  are real polynomials in  $x$  and  $y$ . Forming the equation

$$dy/dx = Q(x, y)/P(x, y) \quad (1.4)$$

from (1.3), he introduced three important concepts. "Critical points" which are the intersection of the curves  $P=Q=0$ , "limit cycles" which are the closed (isolated) paths of the system, and "index" which defines the behaviour of critical points of a vector field. Another important contribution of Poincaré was his study of the dynamic systems depending on a parameter  $\lambda$ , in particular the behaviour of periodic orbits with respect to the variation of the parameter. Following Poincaré's death, George Birkhoff[17] provided elegant proofs for the Poincaré's conjecture and introduced the concept of surface transformations for general dynamic systems. Following the studies of Kolmogorov[18], Arnold[19] and Moser[20] this line of work has flourished and has now been linked with the bifurcation theory[21] to provide detailed insight into the transition from a stable oscillation to a chaotic motion and catastrophe, cf. Ref.[22].

While the mathematical stability theory has been developing with an ever increasing pace, the progress in the stability theory of ships has been slow in spite of a very promising start by Mosley[23] who in 1850 derived a boundedness condition for ship rolling motion under the action of conservative forces. Naval architects had to wait for nearly a century for the next important contribution. This was provided by Grim's study on the stability of ships in following seas[24] where he reduced the equation of

motion to a Mathieu-Hill type equation and studied the stability of motion with the aid of Floquet's theory[25]. In the following years, a number of techniques of the mathematical stability theory have been introduced into the literature of naval architecture, mostly by the individual effort and enthusiasm of the investigators rather than to meet a real demand from the profession. Therefore, a great majority of the new ideas received a lukewarm reception and there has been an apparent confusion in the interpretation of them. Amongst the many reasons for this indifference and confusion, one may detect the following:

- (i) The terminology used in the stability theory has been derived from analytical mechanics (not from mathematics as sometimes claimed[26]), and therefore is alien to naval architects since most of the naval architecture undergraduate courses do not include this subject.
- (ii) There is a traditionally accepted ship stability theory based on statics and users tend to stretch the application of this beyond its limits.
- (iii) A proper understanding of mathematical stability theory requires a working knowledge of analytical mechanics and of differential equations. Such knowledge, however, is not needed for the application of results and this fact has not yet been appreciated.
- (iv) Interchangeable use of simulation, solution, and stability assessment (especially by those who do not have a proper appreciation of the subject) adds to the already existing confusion.
- (v) The fact that the stability assessment depends heavily upon accuracy of the equations of motion has not been fully appreciated.

In the light of these observations, this paper aims at clarifying some of the fundamental issues in the application of the modern stability theory as they relate to the intact ship stability assessment. To achieve this end the meaning of a mathematical model, simulation, and stability analysis are introduced in the beginning and their role in intact ship stability assessment are defined. Here, various definitions of stability and their implications are also examined. The next part is devoted to the Lyapunov's direct method where some of the unjusti-

fied criticism on the method are answered. In the following part topological methods are introduced and applied for a qualitative study of the resonant solutions of the forced Duffing's equation. In the final part, some thoughts on an improved understanding are expressed, including experimental studies, and certain areas are offered for immediate research.

## 2. FUNDAMENTALS OF STABILITY ANALYSIS

A rational assessment of intact stability of ships consists of two important steps; formulation of a representative mathematical model of coupled large amplitude rolling motion, and application of a rational stability criterion. To date in almost all investigations on intact ship stability the first step has been overlooked. Some of the investigators employed linear seakeeping (or manoeuvring) equations, cf. Peters[27], Bishop et al[28], some others included ad hoc non-linear corrections for damping and restoring terms, cf. Blagovechinsky[29], Odabasi[38], whereas another group retained only restoring and excitation terms, cf. Kuo and Welaya[30]. Although these attempts may be justified on the grounds of a gradual build up of an appropriate mathematical model, available experimental data and analysis methods suggest that some of the formulations employed may actually be misleading. Figures 2 to 4 display measured hydrodynamic derivatives for three relatively small angles of roll. Variation of these derivatives both in magnitude and in character provide a clear illustration of the inappropriateness of the linearity assumption. Retaining only the linear term in the restoring moment, on the other hand, amount to the neglect of the change in the hull geometry due to heel which may result in substantial differences as demonstrated in Fig.5. As a result, the linear mathematical model becomes invalid for relatively small angles of roll making the conclusions reached by this model inappropriate for the practical range of interest, i.e. roll angles larger than, say, 6-7 degrees.

Ad hoc models attempt to account for the observed effects of non-linearity by adding corrective terms into the governing equations. They, however, fail in two important aspects; since the corrections are made mostly on intuitive grounds with inadequate data their region of validity are not known and one cannot derive a cause-effect relationship to aid more stable hull form design. A recent study by Cardo et al.[31] on various roll damping representations (using the same roll decrement data) and their consequences clearly demonstrates the shortcomings mentioned here.



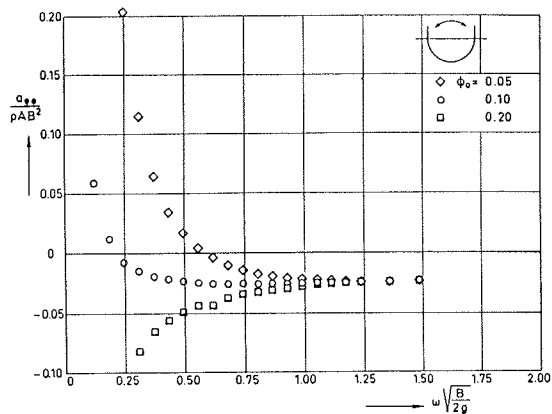


Fig. 2 Variation Added Mass Moment of Inertia Coefficient with Amplitude for a Circular Cylinder

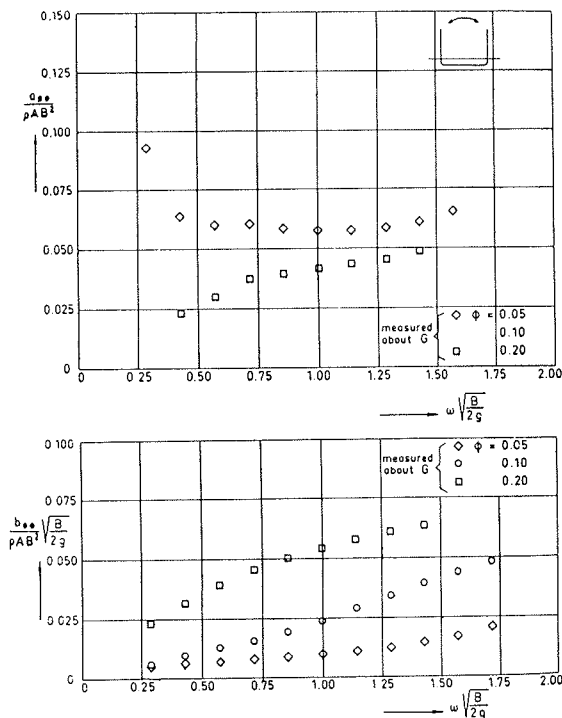


Fig. 3 Variation of Added Mass Moment of Inertia and Damping Coefficients with Amplitude for a Rectangular Cylinder

The use of only a part of the information (i.e. restoring and excitation terms) inherent in the equations of motion stems from the desire to reduce the stability problem to a quasi-static case and justification is sought on the grounds that at low frequency buoyancy terms are of the leading order. Such a justification, however, is valid only in a linear approximation since in non-linear oscillations asymptotic paradox does not hold[32] and small deviations in system parameters and in the input may produce

substantial changes in the system's response.

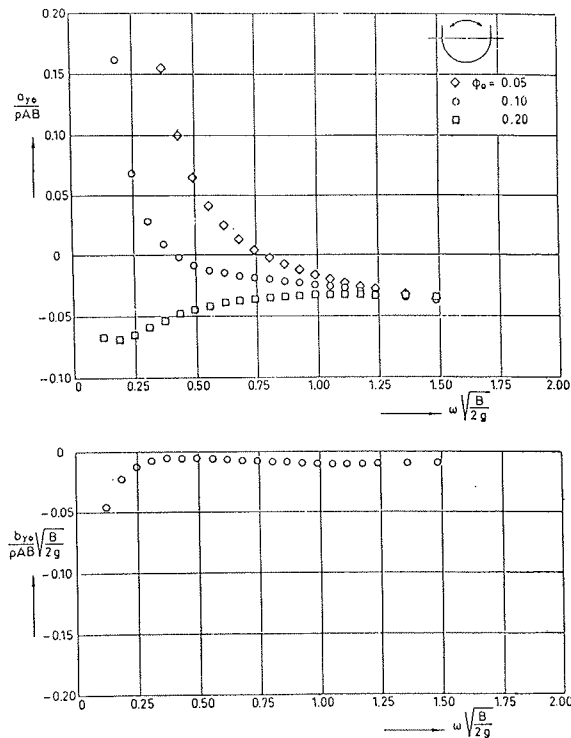


Fig. 4 Variation of Coupling Coefficients of Roll into Sway with Roll Amplitude

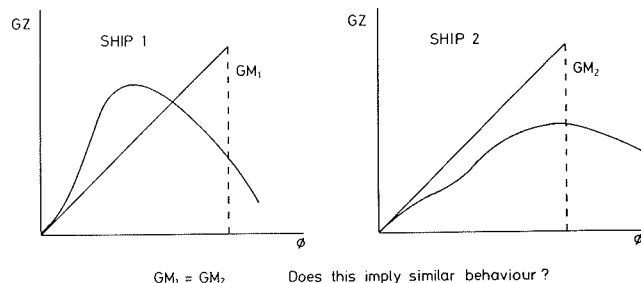


Fig. 5 Two Linearly Equivalent Righting Arm Curves

In the light of the discussion presented here the only viable alternative is the rederivation of governing equations starting from the first principles of continuum mechanics. In a recent rederivation study[33], using a variational formulation, it has been shown that the equations of motion derived in this manner display considerable differences compared to the commonly employed equations and there is a great need for further analytical and experimental research into the fluid active and reactive forces. It is therefore important to recognise that to have a representative set of equations to describe the behaviour of a ship in a seaway is of a crucial importance in the

intact stability assessment of ships and any error introduced in the formulation will necessarily be reflected in the final results.

Since the more-widespread use of the digital computers a number of investigators suggested the use of numerical simulation as a means of stability assessment. However, as proven in many other engineering applications, this proposition falls short of achieving the desired end because:

- (1) In non-linear systems solutions continuously depends on initial conditions and hence to assess the stability of motion one needs to employ all possible combinations of the initial conditions.
- (2) As a result of the modelling accuracy the parameters representing the system and the environmental behaviour contain inaccuracies and their actual values may lie anywhere in a band of possible values. Since, in non-linear systems, the response may change appreciably by small changes in the parameter values, numerical simulation needs to be repeated a sufficient number of times to cover the band of variation of each parameter.

The tasks stated above required <sup>6</sup> sets of simulations even for the ad hoc non-linear rolling equation, proving the impossibility of numerical simulation as a tool for stability assessment. The role of simulation in stability analysis, however, is by no means trivial. Through simulation one can develop better and more representative mathematical models and simulation also assists in the appreciation of various mechanisms leading to capsize or dangerous rolling motion. In addition, a further use of simulation, during the present development stage, is the verification of stability criteria derived by direct methods.

As pointed out in Ref.[34] in the stability assessment of a dynamic system one needs to define two quantities: "norm" and "measure". Norm is the quantity which indicates the state of the system and measure defines the acceptable values of the norm. For example, in the present IMO Recommendations (Resolution A167) the initial metacentric height GM, the maximum righting arm  $GZ_{max}$  and its angular location  $\theta_{max}$ , and the area under the righting arm curve up to certain angles make up the norm vector. The numerical values set out for these components, i.e.  $GM \geq 0.15$  metre,  $GZ_{max} \geq 0.20$  metre,  $\theta_{max} \geq 25$  degrees, etc., define the measure vector. However, as illustrated by Kuo and Odabasi[35] mathematical stability theory offers a variety of stability

definitions and the corresponding norm and measure vectors. The choice to be made amongst the available alternatives must be related to the problem under study. For example, the jump phenomenon as a result of the non-realizability of a branch of the solution near the ordinary resonance region of a Duffing's oscillator (see Fig.6) is a kind of instability and can take place for both a positive or a negative cubic term. However, both the direct stability assessment methods[36] and the numerical simulation results clearly demonstrate that if the cubic term is positive the motion remains bounded and within the domain of attraction of the equilibrium position.

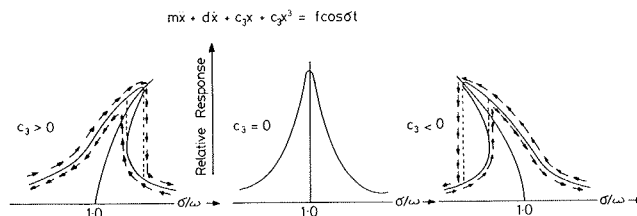


Fig.6 Forced Response of a Duffing's Oscillator

In the case of ship rolling motion the notion of stability implies that the oscillations around the upright equilibrium position should be considered stable, and hence its mathematical counterpart becomes the requirement that the perturbed motion trajectories should remain within the domain of attraction of the upright equilibrium condition. This definition in itself points the way to the solution of the problem since the only method which verifies such a definition is the Lyapunov's direct method and the methods which are either derived from it or make extensive use of it. Two such methods are Popov's frequency domain method and geometric (or topological) methods.

Since the application of Popov's frequency domain method, in its enhanced form developed by Yakubovich[13], to the intact stability assessment of ships has been dealt with in Ref.[37], here only the geometric methods will be presented after a brief discussion on the validity of some of the criticisms levelled against the Lyapunov's direct method.

### 3. LYAPUNOV'S DIRECT METHOD

Though originally introduced near the end of the last century, the direct method of Lyapunov may be broadly classified as one of the "modern" approaches to the solution of stability problems of ships. In fact a majority of naval architects tend to regard the direct method either as black magic, to be performed only by witch

doctors, or as a theoretical tool that has yet to prove itself of practical advantage. This feeling is being encouraged by some investigators lamenting that there is not a set way of determining the required Lyapunov function upon which the second method is based and the method cannot tackle problems with external forcing. This part is therefore aimed at presenting the scope and the generality of Lyapunov's direct method together with the means that are now available for generating the required Lyapunov function.

The objective of the direct method of Lyapunov is to answer questions of stability of dynamic systems, described by differential, integro-differential or difference-differential equations, utilising the given form of the equations but without explicit knowledge of the solutions. The principal idea of the direct method is contained in the following reasoning: If the rate of change  $dE(\underline{x})/dt$  of the energy  $E(\underline{x})$  of an isolated physical system is negative for every possible state  $\underline{x}$ , except for a single equilibrium state  $\underline{x}_0$ , then the energy will continually decrease until it finally assumes its minimum value  $E(\underline{x}_0)$ . In other words, a dissipative system perturbed from its equilibrium state will always return to it; this is the intuitive concept of stability. The mathematical counterpart of the foregoing reasoning is expressed by the following theorem due to Lyapunov: Consider the continuous-time, free dynamic system

$$d\underline{x}/dt = \underline{F}(\underline{x}, t) \quad (3.1)$$

where  $\underline{F}(\underline{0}, t) = \underline{0}$  for all  $t$ . Suppose there exist a scalar function of the state variables  $V(\underline{x}, t)$  with continuous first partial derivatives with respect to  $\underline{x}$  and  $t$  such that  $V(\underline{0}, t) = 0$ , and

- (1)  $V(\underline{x}, t)$  is positive definite, i.e. there exists a continuous, non-decreasing scalar function  $\alpha(\|\underline{x}\|)$  such that  $\alpha(0) = 0$  and, for all  $t$  and all  $\underline{x} \neq \underline{0}$

$$0 < \alpha(\|\underline{x}\|) \leq V(\underline{x}, t);$$

- (2) There exists a continuous scalar function  $\gamma(\|\underline{x}\|)$  such that  $\gamma(0) = 0$  and the derivative  $DV/Dt$  of  $V$  along the motion starting at  $t, \underline{x}$  satisfies, for all  $t$  and all  $\underline{x} \neq \underline{0}$ ,

$$DV(\underline{x}, t)/Dt = \partial V/\partial t + (\text{grad} V)^T \underline{F}(\underline{x}, t)$$

$$\leq -\gamma(\|\underline{x}\|) < 0;$$

- (3) There exist a continuous, non-decreasing scalar function  $\beta(\|\underline{x}\|)$  such that  $\beta(0) = 0$ , and for all  $t$  (see Fig. 7)

$$V(\underline{x}, t) \leq \beta(\|\underline{x}\|);$$

then the equilibrium state  $\underline{x} = \underline{0}$  is uniformly asymptotically stable, and  $V(\underline{x}, t)$  is called a Lyapunov function of the system (3.1).

The method as expressed by the above theorem has a "universal character" and hence its application to specific problems require further extensions and interpretations. In the treatment of intact ship stability problem two such interpretations have been suggested:

Odabaşı[38] employed the definition of eventual stability of La Salle and Rath[39] and supplemented Lyapunov's theorem with a boundedness condition to provide a stability assessment method. In this approach the domain of asymptotic stability of the unperturbed system represents the allowable states and the condition of eventual stability requires that the perturbed states of the dynamic system remain within the allowable states.

In a later study Özkan[40] employed the ultimate boundedness concept of Yoshizawa[41] and generated a Lyapunov function containing the perturbation term explicitly. In this way he was able to obtain stability boundaries in terms of system parameters and the amplitude of excitation.

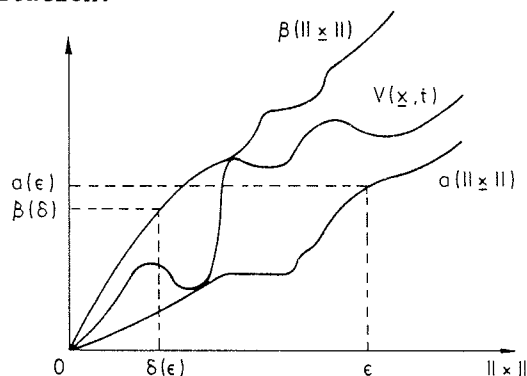


Fig. 7 Definition of Functions  $\alpha$  and  $\beta$

Although the definitions used in both approaches were quite general the criteria obtained were specific for the selected equation of motion. Therefore, application of these definitions to more general state equations require further extensions and derivations.

During the current SAFESHIP mathematical modelling studies a third, and possibly more versatile, interpretation emerged which utilises the following theorem due to La Salle and Lefshetz[42]:

Let the equations of motion of a system under the persistent perturbations

$\underline{R}(\underline{x}, t)$  (with  $\underline{R}(\underline{0}, t) \neq \underline{0}$ ) be given by

$$d\underline{x}/dt = \underline{F}(\underline{x}, t) + \underline{R}(\underline{x}, t) \quad (3.2)$$

with the unperturbed system being defined as

$$d\underline{x}/dt = \underline{F}(\underline{x}, t), \quad \underline{F}(\underline{0}, t) = \underline{0} \quad (3.3)$$

Further, let there exist for the system (3.3) a Lyapunov function  $V(\underline{x}, t)$ , which in the basic region  $\Omega(A): ||\underline{x}|| < A$  and for all  $t \geq 0$ , satisfies the required conditions by Lyapunov's asymptotic stability theorem. That is there exist three positive definite functions  $\alpha(||\underline{x}||)$ ,  $\beta(||\underline{x}||)$  and  $\gamma(||\underline{x}||)$  such that in  $\Omega(A)$  and for all  $t \geq 0$ , we have  $\alpha(||\underline{x}||) \leq V(\underline{x}, t) \leq \beta(||\underline{x}||)$ ;  $DV/Dt \leq -\gamma(||\underline{x}||)$ . Suppose, in addition, that in  $\Omega(A)$  all the partial derivatives  $\partial V/\partial x_i$  are bounded for  $t \geq 0$ ; that is there exist an  $M > 0$  such that in  $\Omega(A)$

$$|\partial V/\partial x_i| \leq M; \quad i=1, 2, \dots, n, \quad t \geq 0.$$

Then the origin (i.e. the equilibrium position) has the following kind of stability: Given any  $0 < \epsilon < A$ , there corresponds to it two numbers  $\eta_1(\epsilon) > 0$  and  $\eta_2(\epsilon) > 0$ , such that if

$$||\underline{x}(0)|| < \eta_1(\epsilon), \quad ||\underline{R}(\underline{x}, t)|| < \eta_2(\epsilon)$$

for all  $||\underline{x}|| < \epsilon$  and  $t \geq 0$ , then

$$||\underline{x}(t)|| < \epsilon \quad \text{for all } t \geq 0.$$

The definition given here is called "practical stability" by its originators and a simple demonstration of its application is provided by the following example.

**Example:** Let the equation of large amplitude rolling motion of a ship be given by

$$\ddot{\theta} + f(\theta)\dot{\theta} + g(\theta) = e_1(t) + e_2(t) \quad (3.4)$$

where  $e_1(t)$  is a transient excitation with duration  $T_*$ , and  $e_2(t)$  is the steady state excitation.

The boundary of stability of the undisturbed equation

$$\ddot{\theta} + f(\theta)\dot{\theta} + g(\theta) = 0 \quad (3.5)$$

is presented in Fig.8. Introducing

$$F(\theta) = \int_0^\theta f(\xi) d\xi, \quad G(\theta) = \int_0^\theta g(\xi) d\xi,$$

$$V = \frac{1}{2}\dot{\theta}^2 + G(\theta) + \dot{\theta}F(\theta) + \frac{1}{2}F^2(\theta),$$

$$E_1(t) = \int_0^t e_1(t) dt,$$

choosing the constant  $\lambda$  such that

$$g(\theta)F(\theta) > 0 \quad \text{for } 0 < |\theta| < \alpha,$$

$$G(\theta) < \lambda \quad \text{for } |\theta| < \alpha.$$

denoting  $\max |e_2(t)| \leq \epsilon = \eta_2$ , and rewriting the perturbed equations as

$$\dot{\theta} = y - F(\theta) + E_1(t),$$

$$\dot{y} = -g(\theta) + e(t),$$

the derivative of the Lyapunov function taken along the trajectory of the perturbed motion is obtained as

$$DV/Dt = -g(\theta)[F(\theta) - E_1(t)] +$$

$$ye(t) > g(\theta)[F(\theta) - E_1(t)]$$

$$+ \text{sign}(y)yE.$$

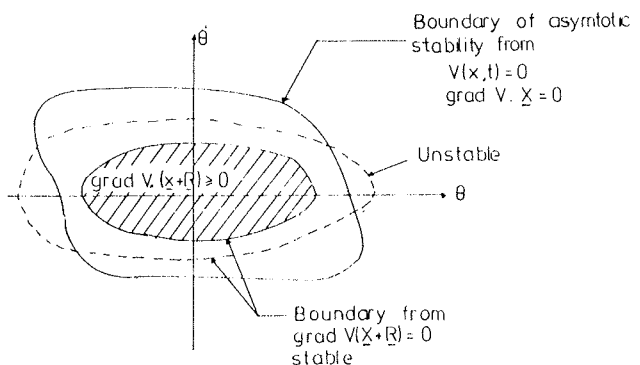


Fig.8 Definition of Practical Stability

Since the condition of practical stability requires negative semi-definiteness of  $DV/Dt$ , orientation of the curve  $\Gamma$  defined by the equation

$$-g(\theta)[F(\theta) - E_1(t)] + \text{sign}(y)yE = 0$$

or equivalently

$$[\text{sign}(y)E - g(\theta)]F(\theta) + \text{sign}(y)E\dot{\theta} + g(\theta)E_1^* \quad (3.6)$$

has a special role where  $E_1^* = E_1(T_*)$ . As shown in Fig.8 the curve  $\Gamma$  divides the phase space into two regions: In the domain  $\Omega_1$  contained within  $\Gamma$  the derivative  $DV/Dt$  is positive and hence any trajectory starting in this domain will tend to leave it, and in  $\Omega_1^c$  outside the curve  $\Gamma$  the derivative  $DV/Dt$  is negative provided this domain lies within the domain of attraction of the unperturbed system, i.e. in  $\Omega(A)$ . Hence, so long as  $\Gamma$  is contained within  $V=0$  the rolling motion of the ship is practically stable.

The result obtained in this example lends itself readily both to a systematic

evaluation by varying  $g(\theta)$ ,  $F(\theta)$ , and  $E$ , and to form the basis of a criterion by determining the limiting  $g(\theta)$  (and hence VCG values) for the given environmental conditions. A systematic variation undertaken for a stern trawler is demonstrated in Fig.9. Hull particulars for the four loading conditions considered in this example are presented in Table 1, and the method of computation is explained in Ref.[57]. Since the wave exciting force is calculated by a Froude-Krylov approximation, a change in the vertical location of the vertical centre of gravity results in a change the wave excitation amplitude.

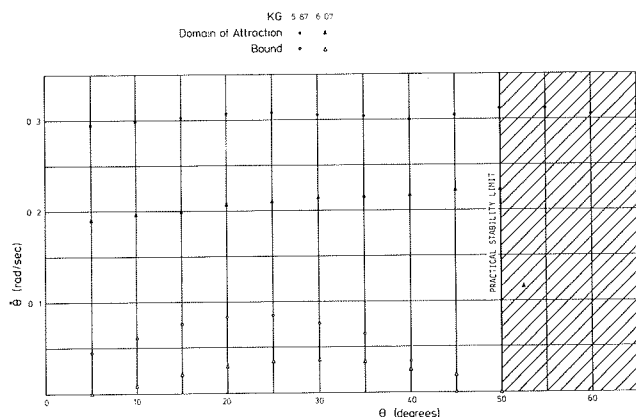


Fig.9 Variation of Practical Stability Limit with Varying KG

Table 1  
Characteristics of the Stern Trawler UK3

Lpp (Length)	56.845 metres
B (Beam)	12.192 metres
D (Depth)	6.800 metres
T (Draft)	4.323 metres
$\Delta$ (Displ.)	1582 tonnes
KM	6.329 metres
Steady Wind Mom.	1157.195 K.Newton-metres
Gust Mom.	2739.424 K.Newton-metres
Wave Excit. Mom.	2203.115 K.Newton-metres

While this example demonstrates the usefulness of the practical stability concept, the definition given by the La Salle-Leftshetz theorem can be extended to yield results in much more general circumstances. Two such uses are presented below:

#### (1) Effects of Environmental Randomness

When a ship executes its rolling motion under the action of random seawaves, the equation of motion may be written as

$$\ddot{\theta} + [f(\theta) + \xi_1(t)]\dot{\theta} + [1 + \xi_2(t)]g(\theta) + \int_0^t L(\theta; t-\tau)\ddot{\theta}(\tau)d\tau = \xi_3(t)$$

where  $\xi_1(t)$ ,  $\xi_2(t)$  and  $\xi_3(t)$  are independent random processes, and  $L(\theta; t)$  is a  $\theta$  parametric modified impulse response function. Rewriting this equation as

$$\ddot{\theta} + f(\theta)\dot{\theta} + g(\theta) = \xi_3(t) - \xi_1(t)\dot{\theta} - \xi_2(t)g(\theta) - \int_0^t L(\theta; t-\tau)\ddot{\theta}(\tau)d\tau \quad (3.7)$$

one can apply the process of practical stability evaluation without much difficulty. In this case the unperturbed system becomes

$$\ddot{\theta} + f(\theta)\dot{\theta} + g(\theta) = 0$$

and the persistent perturbation becomes expressible as

$$R = \xi_3(t) - \xi_1(t)\dot{\theta} - \xi_2(t)g(\theta) - \int_0^t L(\theta; t-\tau)\ddot{\theta}(\tau)d\tau \quad (3.8)$$

Defining statistically significant values of  $R$ , cf.  $R_{1/3}$ ,  $R_{1/10}$  etc., the corresponding level of stochastic stability can be obtained in a similar manner to the one employed in the given example, that is the equation of the curve  $\Gamma$  becomes

$$[\text{sign}(Y)R_* - g(\theta)]F(\theta) + \text{sign}(Y)R_*\dot{\theta} + g(\theta)E_1^* \quad (3.9)$$

where  $R_*$  is the selected statically significant value of  $R$ .

#### (2) Effects of Coupling

If one wishes to study the stability of the coupled sway-roll-yaw motion with respect to roll, as a result of the Cordineau's theorem[43], such a study is reduced to the stability assessment of a single equation of the form

$$\ddot{\theta} + F(\theta)\dot{\theta} + \int_0^t L(\theta; t-\tau)\ddot{\theta}d\tau + g(\theta) = e(t) + Y(\dot{\theta}, \ddot{\theta}, \text{sway}, \text{yaw}, t) \quad (3.10)$$

where  $Y(\dot{\theta}, \ddot{\theta}, \text{sway}, \text{yaw}, t)$  represents the augmented coupling terms. Again, defining the persistent perturbations as

$$R = e(t) + Y(\dot{\theta}, \ddot{\theta}, \text{sway}, \text{yaw}, t) - \int_0^t L(\theta; t-\tau)\ddot{\theta}(\tau)d\tau \quad (3.11)$$

the concept of practical stability becomes immediately applicable.

A natural continuation of the two extensions is the practical stability evaluation under combined effects of coupling and randomness, which amounts to a redefinition of the persistent perturbation term.

One of the complaints levelled by the critics of Lyapunov's direct method is the lack of a systematic method for generating Lyapunov functions, cf.[30], which seems to arise due to the lack of knowledge of analytical mechanics, and within the context of intact ship stability assessment, it is completely unjustified on at least three grounds:

- (1) Today there exist a large number of methods available for the generation of Lyapunov functions. References [44 to 49] provide a sufficient number of methods suitable for a large number of problems and Fig.10 displays the efficiency of Zubov's method for Van der Pol's equation.
- (2) Since the Lyapunov function plays the role of a generic potential, a reduction of the unperturbed equations of motion into their canonical or quasi-canonical form will lead to the derivation of an optimal Lyapunov function, cf.Santilli[50].
- (3) If one agrees on the form of governing equations for coupled large amplitude rolling motion the necessary Lyapunov function needs to be derived only once as the user needs to implement the resulting criterion.

In the light of the discussion presented here, it is hoped that the critics of the Lyapunov's direct method will spend more time in developing an understanding of both the underlying concepts and their utilisation and then comment in a constructive manner on its further development and application.

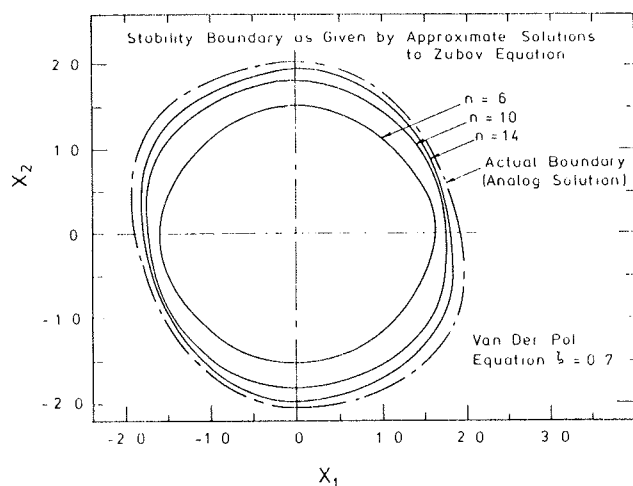


Fig.10 Efficiency of Zubov's Method for Constructing Lyapunov Functions

#### 4. TOPOLOGICAL DYNAMICS

Topological methods deal with the geometry of curves defined by differential equations in the neighbourhood of singular points or cycles. Mathematical foundations of topological methods were laid down by G.D. Birkhoff[17] who defined the basic premises of topological dynamics as:

- (1) Global study of solutions in the entire region of existence.
- (2) Study of solutions near singular points and cycles.

In this sense Lyapunov's direct method can also be considered within the context of topological dynamics.

The use of topological methods within the confines of intact stability assessment allows one to analyse various important phenomena in a complete manner. Central to the execution of such analyses are the theory of perturbations, in particular in Hamiltonian systems, and the theory of mapping. General definitions and concept on these two subjects are provided in Ref.[17]. Amongst many phenomena, which can be studied by topological methods, the problem of small denominators, i.e. resonance, provides an excellent example to demonstrate the type of information that can be acquired by a topological study. Here a complete qualitative study of Duffing's equation, including resonance, will be presented along the lines suggested by Mel'nikov[51] and Morozov[52].

Consider Duffing's equation in the form

$$\ddot{x} + x - \alpha x^3 = \epsilon (f(\Omega t) + \delta \dot{x}) \quad (4.1)$$

where  $\alpha$ ,  $\Omega$  and  $\delta$  are real constant,  $\epsilon$  is a small parameter, and the function  $f(\Omega t)$  is continuous and  $2\pi$ -periodic in  $\Omega t$ .

For  $\epsilon=0$ , (4.1) is a Hamiltonian system with Hamiltonian

$$H = (\dot{x}^2/2) + (x^2/2) - \alpha(x^4/4). \quad (4.2)$$

Phase trajectories of equation (4.1) for  $\alpha>0$  are illustrated in Fig.11. For  $H \in (0, 1/4\alpha)$  trajectories are closed. The critical value  $H=1/4\alpha$  corresponds to a separatrix contour  $\Gamma$  formed on the saddle points  $O_1(-1/\sqrt{\alpha}, 0)$  and  $O_2(1/\sqrt{\alpha}, 0)$  and the separatrices of these saddle points. For  $\alpha<0$  the entire phase plane is filled by close trajectories. Period  $T$  of motion along the closed trajectories depends on  $H$ . The energy level,  $H=H_{pq}=\text{Const.}$  is called a resonance level,  $H_{pq}$  if  $T(H_{pq})=(p/q)(2\pi/\Omega)$ . A resonance level  $H_{pq}=\text{Const.}$  is called a passable resonance if any solution of equation (4.1) at  $t=t_0$  passing through a point of the

neighbourhood after a finite time. A resonance level  $H = \text{Const.}$  is called a  $\delta$ -resonance if the equation (4.1) has periodic solutions for  $\epsilon \neq 0$  in the neighbourhood of this level for a given non-zero.

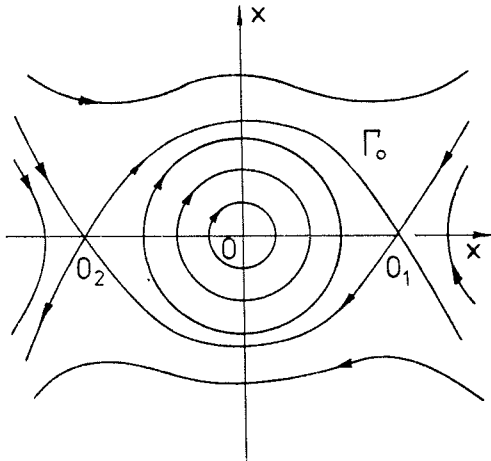


Fig.11 Phase Trajectories of Equation (4.1) for  $\epsilon=0$

Solution of the unperturbed equation can be obtained in explicit form in terms of elliptic functions. The period of oscillation is expressed as

$$T(H) = \frac{4\sqrt{2}}{x_1|\alpha|} K(k) \quad (4.3)$$

where  $K(k)$  is the complete elliptic integral of the first kind,  $k$  is the modulus of the integral

$$k = x_1/x_2 \quad (\text{for } \alpha > 0) \quad (4.4)$$

and

$$x_1 = \{(1 - \sqrt{1 - 4\alpha H})/\alpha\}^{1/2},$$

$$x_2 = \{(1 + \sqrt{1 - 4\alpha H})/\alpha\}^{1/2}.$$

For  $x(0)=0$  ( $\alpha > 0$ ), the solution can be written as

$$x(t) = x_1 \text{sn} \left( \frac{x_1 \sqrt{\alpha}}{\sqrt{2} k} t \right)$$

$$= \frac{4\sqrt{2}\omega}{\sqrt{\alpha}} \sum_{n=1}^{\infty} \frac{a^{n-1/2}}{1-a^{2n-1}} \sin(2n-1)\omega t \quad (4.5)$$

where  $\text{sn}(\cdot)$  is the elliptic sine function and

$$\omega = \frac{2\pi}{T} = \frac{\pi x_1 \sqrt{\alpha}}{2\sqrt{2}kK(k)} \quad (4.6)$$

$$a = \exp[-\pi K(\sqrt{1-k^2})/K(k)]. \quad (4.7)$$

The use of action-angle variables provides a more convenient basis for the investigation of non-linear resonance, [19]. If one writes equation (4.1) in terms of the state variables

$$\dot{x}=y, \quad \dot{y}=-x+\alpha x^3+\epsilon(f(\Omega t)+\delta y) \quad (4.8)$$

then the corresponding action  $I$  and angle  $w$  variables are written as:

$$I(H) = \frac{1}{2\pi} \oint y(\xi) d\xi = \frac{2x_2}{3\pi}$$

$$\left[ \frac{2}{\alpha} E\left(\frac{\pi}{2}, k\right) - \frac{2\sqrt{1-4\alpha H}}{\alpha} K(k) \right] \quad (4.9)$$

$$w(x, I) = \frac{\partial S(x, I)}{\partial x} \quad (4.10)$$

where the generation function  $S(x, I)$  is expressible as

$$S(x, I) = \frac{x_2}{6\pi} \left\{ \frac{2}{\alpha} E(\phi, k) - \frac{2\sqrt{1-4\alpha H}}{\alpha} F(\phi, k) \right. \\ \left. + \frac{x}{3} \sqrt{(x_1^2 - x^2)(x_2^2 - \text{sign} \alpha x^2)} \right\}, \quad (4.11)$$

$F(\phi, k)$  is the elliptic integral of the first kind,  $E(\phi, k)$  is the elliptic integral of the second kind, and  $\phi$  is given by  $x/x_1 = \sin \phi$ .

The transformation introduced by equations (4.9) and (4.10) is valid within separatrix  $\Gamma$  (i.e. in a closed region  $G = \{(x, y): \gamma \leq H(x, y) \leq 1/4\alpha - \gamma\}$  or equivalently  $V = \{(I, w): I(\gamma) \leq I \leq I(1/4\alpha - \gamma), 0 \leq w < 2\pi\}$  for any small positive  $\gamma$ ). The equation of motion in new variables are:

$$\dot{I} = \epsilon (f(\Omega t) + \delta Y) I'_Y \quad (4.12)$$

$$\dot{w} = \omega(I) - \epsilon (f(\Omega t) + \delta Y) w'_Y$$

where  $y=Y(I, w)$  and  $(\cdot)$  indicates derivative with respect to the subscript variable.

Defining the resonant values of the variables as  $I_{pq} = I(H_{pq})$ ,  $T_{pq} = T(H_{pq})$ ,  $\omega_{pq} = \omega(H_{pq})$ , and expressing their variation in a resonant oscillation as:

$$I = I_{pq} + \mu \frac{h}{\omega_{pq}}, \quad w = \omega_{pq}(t + \psi) \quad (4.13)$$

which leads to the equations for  $h$  and  $\psi$  as

$$\dot{h} = \mu \omega_{pq} [f(\Omega t) + \delta Y]$$

$$[(I_Y')_{I=I_{pq}} + \frac{\mu}{\omega_{pq}} h_Y'] , \quad (4.14)$$

$$\dot{\psi} = \frac{\omega}{\omega_{pq}} - 1 + \mu^2 [f(\Omega t) + \delta Y] \psi_Y' ,$$

where  $\mu = \sqrt{\varepsilon}$ . Equations (4.13) and (4.14) are defined in the region  $\{(I, w): |I - I_{pq}| \leq \mu R, 0 \leq w < 2\pi\}$  where the value of the positive constant  $R$  to be determined.

Equations (4.14) are in a convenient form for the application of the averaging principle, i.e. substitution of

$$h = u - \frac{i\mu p}{\Omega} \sum_{k \neq 0} \frac{1}{k} A_k(\psi) e^{ik\Omega t/p} \quad (4.15)$$

where

$$A_k(\psi) = \frac{\Omega}{2p\pi} \int_0^{2\pi p/\Omega} y_{pq}(t+\psi) [f(\Omega t) + \delta y_{pq}(t+\psi)] dt, \quad (4.16)$$

and  $y_{pq}(t) \equiv Y(I_{pq}, t)$  is a  $2\pi p/(q\Omega)$ -periodic solution of the unperturbed system.

Retaining the terms up to second order in  $\mu$  equations (4.14) becomes

$$\begin{aligned} \dot{u} &= \mu A_O(\psi) + \mu^2 P_O(u, \psi) + O(\mu^3) \\ \dot{\psi} &= \mu b u + \mu^2 Q_O(u, \psi) + O(\mu^3) \end{aligned} \quad (4.17)$$

where

$$b = - \frac{(T_H')_{H=H_{pq}}}{T},$$

$$P_O(u, \psi) = \frac{\Omega}{2\pi p} \int_0^{2\pi p/\Omega} [f(\Omega t) + \delta y_{pq}(t+\psi)]$$

$$(u_Y')_{Y=Y_{pq}} dt + \delta u,$$

$$Q_O(u, \psi) = \frac{\Omega}{2\pi p} \int_0^{2\pi p/\Omega} [f(\Omega t) + \delta y_{pq}(t+\psi)]$$

$$(\psi_Y')_{Y=Y_{pq}} dt + \tilde{b} u^2,$$

$$\tilde{b} = \frac{1}{\omega_{pq}} \left( \frac{d^2 \omega}{dI^2} \right)_{I=I_{pq}}.$$

If one considers only the first order terms in  $\mu$ , the phase portrait in the  $u$ - $\psi$  plane resembles one of the three configurations given in Fig.12, for  $\delta \leq 0$  (i.e. positive damping).

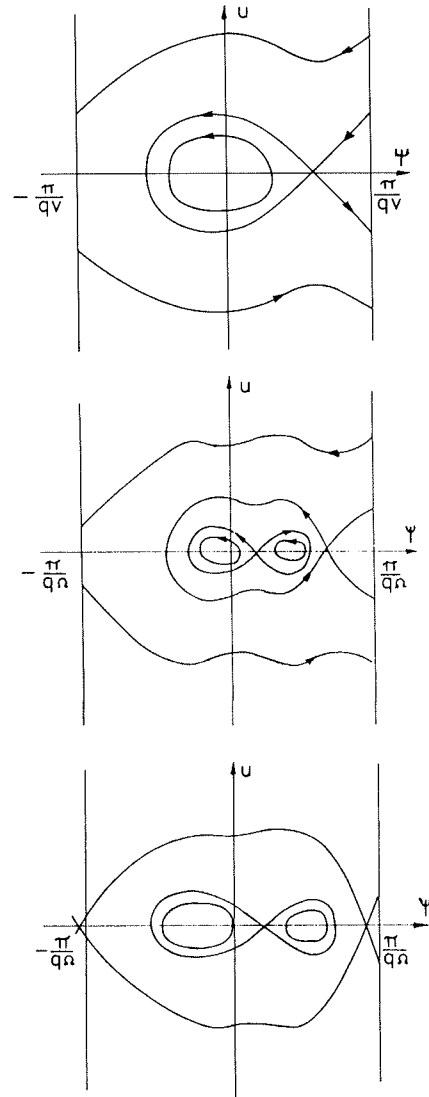


Fig.12 Phase Portrait of Equation (4.17) in Linear Approximation

When the excitation is expressible in the form of a Fourier series

$$f(\Omega t) = \frac{c_0}{2} + \sum_{k=1}^{\infty} (c_k \cos k\Omega t + d_k \sin k\Omega t) \quad (4.18)$$

$A_O(\psi)$  can be evaluated in an explicit form:



$$A_0(\psi) = \bar{A}_0(\psi) + B\delta,$$

$$\frac{2\sqrt{2}\Omega^2 q^2}{\sqrt{\alpha} p} \sum_{r=1}^{\infty} r \frac{a^{rp/2}}{1-a^{rp}}$$

$$\bar{A}_0(\psi) =$$

$$(C_{rq} \cos rq\Omega\psi - d_{rq} \sin rq\Omega\psi) \text{ for } p \text{ odd,}$$

$$0 \text{ for } p \text{ even,}$$

$$B = 16 \frac{\Omega^4 q^4}{\alpha p^4} \sum_{n=1}^{\infty} (2n-1)^2 \left( \frac{a^{n-1/2}}{1-a^{2n-1}} \right)^2.$$

where  $r$  takes odd integer values.

Substitution of (4.19) into (4.17) yields

$$\begin{aligned} \dot{u} &= \mu(\bar{A}_0(\psi) + \delta B) + \mu^2 P_0(u, \psi), \\ \dot{\psi} &= \mu b u + \mu^2 Q_0(u, \psi). \end{aligned} \quad (4.20)$$

When  $p$  is even, first order terms in equations (4.20) can be combined to give the equation of the trajectories

$$\psi - \psi_0 = \frac{b}{2\delta B} u^2. \quad (4.21)$$

To analyse the trajectories in a more general way one needs to study the behaviour of the equations

$$\begin{aligned} \bar{A}_0(\psi) + \delta B + \mu P_0(u, \psi) &= 0, \\ \mu b + \mu Q_0(u, \psi) &= 0. \end{aligned} \quad (4.22)$$

It can be seen that when  $\mu$  is sufficiently small and

$$|\delta| > \max_{\psi} \frac{1}{B} |\bar{A}_0(\psi)| = \delta_*, \quad (4.23)$$

equations (4.22) have no solutions and therefore the system (4.1) will always have a fast crossing of resonance.

For a more detailed study consider the Jacobian of the mapping defined by equations (4.20) near the equilibrium condition

$$\begin{aligned} \Delta_1 &= \begin{vmatrix} \mu^2 (P'_{ou})_0 & \mu (A'_{o\psi})_0 + \mu^2 (P'_{o\psi})_0 \\ \mu b + \mu^2 (Q'_{ou})_0 & \mu^2 (Q'_{o\psi})_0 \end{vmatrix} \\ &= -\mu^2 b (A'_{o\psi})_0 + O(\mu^3) \end{aligned} \quad (4.24)$$

where  $( )_0$  implies that the quantity within the parenthesis is evaluated at the equilibrium state. The characteristic equation is then

$$\lambda^2 - \mu^2 (P'_{ou} + Q'_{o\psi})_0 \lambda + \Delta_1 = 0.$$

Since, by definition of  $P_0$  and  $Q_0$  one can write  $P'_{ou} + Q'_{o\psi} = 2\delta$ , the characteristic values are obtained as

$$\lambda_{1,2} = \mu^2 \delta \pm \sqrt{(\mu^4 \delta - \Delta_1)}. \quad (4.25)$$

Hence, if  $\Delta_1 < 0$ , the equilibrium state is of the saddle type, while if  $\Delta_1 > 0$  it is of the focus type, stable for  $\delta < 0$  and unstable for  $\delta > 0$ . Since the function  $\bar{A}_0(\psi)$  takes opposite signs at adjacent zeros of  $A'_{o\psi}(\psi)$  the saddle type and focus type equilibrium states alternate. For sufficiently small  $\mu$ , these states resemble closely to the saddle and centre type equilibrium states depicted in Fig.12.

When the roots are not simple, i.e.  $\Delta_1 = 0$ , there exists composite equilibrium states. Using the results of Ref.[53], for  $\delta \neq 0$ , the following types of equilibrium states are distinguished:

- (1) If  $\Delta_m < 0$  and  $m$  is even-saddle-node.
- (2) If  $\Delta_m < 0$  and  $m$  is odd-composite saddle.
- (3) If  $\Delta_m > 0$  composite node.

Where  $\Delta_m = -\mu b A_{o\psi}^{(m)}(\psi_0)$  with the first non-zero value.

With these results and employing the known bifurcation properties of composite equilibrium states[54], a complete topological structure of equations (4.20) can be portrayed as a function of the parameter  $\delta$ .

- (1) For sufficiently large  $|\delta|$ , i.e.  $|\delta| > \delta_*$ , the system has no equilibrium states.
- (2) As  $\delta$  decreases, starting at some point  $|\delta| = \delta_m$  determined by the condition  $\Delta_m(\psi(\delta)) = 0$  ( $i=1,2,\dots,m-1$ ;  $m$  is odd), composite equilibrium states appear.
- (3) As  $|\delta|$  decreases further, these composite states split into  $m/2$  simple nodes and  $m/2$  simple saddles which, at some  $|\delta| = \delta_m$ , become composite equilibrium states. The process continues in this manner with decreasing  $|\delta|$ .

The stages depicted above are illustrated in Fig.13 where it is assumed that for  $|\delta| = \delta_1$ ;  $\Delta_1 = 0$ ,  $\Delta_2 \neq 0$  and for  $\delta = \delta_2$ ;  $\Delta_1 = \Delta_2 = 0$ ,  $\Delta_3 \neq 0$  ( $m \geq 3$ ). It is clear from this figure that for any fixed  $\delta$  in the region  $0 < |\delta| < \delta_*$ , the neighbourhood of  $\delta$ -resonance is divided by separatrices into two types of region: capture regions in which the trajectories remain in the neighbourhood

as  $t \rightarrow \infty$  (hatched regions), and passable regions filled by trajectories moving away from the resonance.

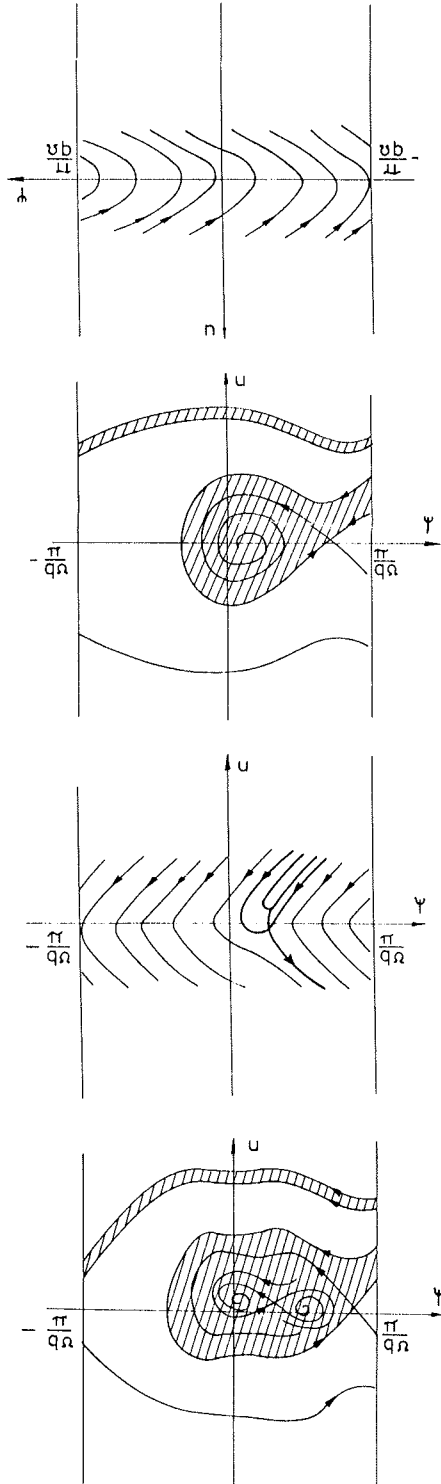


Fig.13 Phase Portrait of Composite Phase Equilibrium States

Finally, consider a specific periodic forcing function of the form

$$f(\Omega t) = b_1 \sin \Omega t + b_3 \sin 3\Omega t, \quad b_1 > 0, \quad b_3 > 0. \quad (4.26)$$

As a result of (4.19), for  $\delta$ -resonances,  $q$  can take the values 1 and 3. Let  $q=1$ . In this case, equations (4.20) become

$$\dot{u} = \mu \left\{ \frac{2\sqrt{2}\Omega^2}{\sqrt{\alpha}} \left[ -\frac{1}{p} \left( b_1 \frac{a^{p/2}}{1-a^p} \sin \Omega \psi + 3b_3 \frac{a^{3p/2}}{1-a^{3p}} \sin 3\Omega \psi \right) \right] + \delta B \right\}, \quad (4.27)$$

$$\dot{\psi} = \mu b u.$$

Setting the right hand sides of equations (4.27) equal to zero, the first equation may be expressed as:

$$z^3 - n_1 z + n_2 = 0 \quad (4.28)$$

where

$$z = \sin \Omega \psi, \quad n_1 = \frac{1}{12b_3} \left( b_1 \frac{1-a^{3p}}{a^p(1-a^p)} + 9b_3 \right),$$

$$n_2 = \delta \frac{pB\sqrt{\alpha}(1-a^{3p})}{24\sqrt{2}\Omega^2 a^{3p/2} b_3}.$$

In order that a  $\delta$ -resonance (or a capture region) exist equation (4.28) should have at least one real root with absolute value less than unity. Such regions are indicated in Fig.14 in the  $(n_1, n_2)$  parameter plane where  $D(k, \ell)$  indicates  $k$  real roots with absolute value less than unity ( $k+\ell=3$ ).

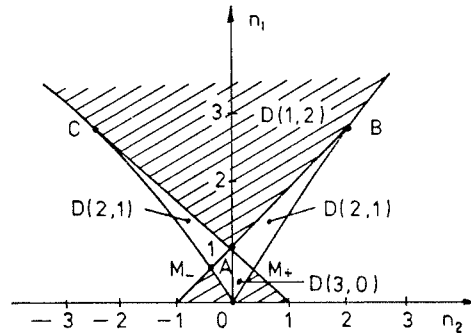


Fig.14  $\delta$ -Resonance Regions for the Forcing (4.26)

While the example presented here is, to a certain extent, elementary, it clearly demonstrates both the capability of the methods and the differences between linear and non-linear resonances. The proof of the presence of a critical  $\delta$  value ( $\delta_*$ ) depending on the forcing amplitude for the non-existence of resonant oscillations is particularly important from the practical point of view.

In the treatment of more representative equations, the only difference is the replacement of exact integration with numerical integration. A program suite, called 'BIRKHOFF' has been developed at BSRA to implement this scheme.

## 5. CONCLUDING REMARKS

This paper has been prepared with the aim of demonstrating the capabilities of mathematical stability assessment methods within the context of intact ship stability. It is hoped that the researchers and naval architects may come to the realisation that many of the shortcomings in ship stability assessment arise from the inadequacies of the equations used to describe the behaviour of a ship in a seaway. Once, a more representative mathematical model is derived and evaluated, the direct stability assessment methods will prove their value.

It is also important to take notice of deviating propositions which, in the light of the experience gained in other professional disciplines, can slow the developments in the right direction. In the view of the present author, these may be summarised as follows:

- (1) Numerical simulation is not a direct stability assessment method. Its essential usefulness lies in its ability to demonstrate various mechanisms leading to capsize, to assist in the development of a representative mathematical model, and to verify the proposed stability criteria.
- (2) Linear seakeeping (or manoeuvring) equations are not suitable to study ship stability due to the inherent non-linearities in the lateral ship motions. Since, for rolling amplitudes greater than 6-7 degrees, linearity assumptions fail and hence their use either in the form of eigenvalue analysis or in the form of polar plots may lead to erroneous conclusions.
- (3) To derive a stochastic stability criterion without a fundamental understanding of its deterministic counterpart can, at best, lead to confusion and misinterpretations.
- (4) In a governing equation, a part of it cannot contain more information than the whole and hence proposals in this direction should be received with extreme caution.

To achieve progress a great deal of effort needs to be spent for the derivation of more representative mathematical models for fluid active and reactive

forces, including wind and gust forces. In this respect, experimental studies require better planning and co-ordination with analytical research than those which have been reported in the past. Within this context it may be worth repeating Norbert Wiener's quotation[55]:

"...things do not, in general, run around with their measures stamped on them like the capacity of a freight car: it requires a certain amount of investigation to discover what their measures are... What most experimenters take for granted before they begin their experiments is infinitely more interesting than any results to which their experiments lead".

Fortunately, recent experiments and subsequent analysis reported by Ohkushu[56] clearly indicate the way towards progress and a better appreciation. It is hoped this kind of positive thinking becomes more widespread in stability research leading to the derivation of a reliable stability criterion before the Third International Conference on Ship Stability.

## ACKNOWLEDGEMENT

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## Discussion

O. Krappinger (Hamburg Ship Model Basin, FRG)

Dr. Odabasi states in his paper that most naval architects tend to regard Lyapunov's direct method either as black magic or as a theoretical tool that has yet to prove itself of practical advantage. I myself certainly belong to the second group, and it seems to me that Dr. Odabasi

also should be considered a member of this group because he says - I cite him again: A rational assessment of ship stability consists of two steps; 1. Formulation of a representative mathematical model of coupled large amplitude rolling motion and 2. application of a rational stability criterion. I do not agree with him that hitherto the first step has been overlooked. The

fact is that we have not yet been able to establish a satisfactory model of the rolling motion (and I see no chance to arrive at such a model in the foreseeable future). In this situation there is no merit in bothering about the second step. Therefore, I see no need that naval architects, who at this time are concerned with assessing ship stability take the trouble of studying all the stuff which Dr. Odabasi has compiled from the literature. He himself - in the Concluding Remarks of his paper - stresses the fact that only after a representative mathematical model will have been derived the direct stability assessment methods might prove their value.

S. Kastner (College of Engineering Bremen, FRG)

Dr. Odabasi gives us a thorough account and hints or understanding the practical application of mathematical stability theory. I agree with both the mathematical and the intuitive concept of stability by the Lyapunov direct method, as in Dr. Odabasi's words, a disturbed system perturbed from its equilibrium will always return to it. I think Dr. Odabasi has himself stated all the objections one might have against the application of the Lyapunov method. My point now is the small disturbance assumption, which underlies the further application of the method. Can stability theorems, as Dr. Odabasi is fostering, really account for the safety from capsizing of ships? As far as my understanding goes, even a solution of the motion behaviour of a ship which leads to a capsizing, might be termed to be stable in the Lyapunov sense, i.e. small disturbances in course of the process will not change the outcome. My final point: Naturally, as long as we stay within the "domain of attraction", the ship is stable, i.e. in the Naval Architecture sense she will not capsize. But this still requires us to determine, when a ship motion will cross the curves towards the outside region. Experiments, simulation, as well as criteria based on certain parameters can be used. Dr. Odabasi states the usefulness of numerical simulations. But why is, in his view, the number of simulation sets  $\infty$ ? That statement lacks the insight into probability theorems.

V. Ankudinov (Hydronautics Inc., U.S.A.)

It has been recognized that dynamics of extreme ship motions including capsizing is highly non-linear phenomena.

A new computational procedure to predict such a motions (called inertial marker particles method) completely non-linear with generalized boundary and body configurations will be briefly discussed. The general equations and assumptions will be discussed (4 viewgraphs) and one viewgraph showing the water profile around the ship hull will be presented.

As the end of the presentation a short film showing the computational prediction of motion of the ship hull in regular and irregular seas and shallow water conditions will be given.

#### Author's Reply

I would like to thank all the discussers for their interest in this paper. I am pleased to hear that Professor Krappinger is prepared to accept Lyapunov's Direct Method as a practical tool if the equations of motion formulated properly. His views on applicability, however, seems to be ill informed. Verification of a stability assessment method is quite straight forward. One chooses a mathematical model, derives the criterion and check its validity by simulation. So far as Lyapunov's direct method is concerned we undertook such exercises for a number of mathematical model, first as many other scientists and engineers did in other fields. Therefore, for us there is no need for any further proof on the validity of Lyapunov's Direct Method.

Professor Krappinger's comments on the possibility of devising a representative mathematical model seems to overlook to the ongoing activities in this direction. Provided that sufficient resources can be devoted for this purpose, its achievement to a degree, satisfactory for stability analysis, can be considered as a short term development. Furthermore, despite their shortcomings, existing mathematical models contain more information than any other purely empirically conceived views, and the use of mathematical models in a large number of papers in this Conference is a sufficient demonstration of this fact.

Professor Kastner poses two important questions. The first question is related to the definition of stability and to a large extent is answered within the paper, that is, "practical stability" definition does not make any assumption on the smallness of disturbances. The only requirement is that the disturbances should be bounded in the mean, i.e. they may take instantaneously very large values (which can include Dirac delta function type excitation). His comments on a solution being termed "stable" in the sense of Lyapunov which may lead to capsizing therefore is not valid. In fact, the method guarantees the stability of solution but may qualify some solutions unstable although they may be stable, especially if the chosen Lyapunov function is not efficient.

The comment on the number of simulations required for ad hoc rolling equation represents a theoretical limit to determine the distribution of long term extreme values, and can be replaced by the term "very large number of simulations of long duration." Unfortunately, existing methods, of probability theory provides answers for the short term responses of certain types of equations under restricted types of

excitations. Therefore, even for short term responses one may be forced to carry out numerical simulation, i.e. direct integration methods.

Finally, I would like express my appreciation for the work presented by Dr. Ankudinov and I hope to hear more about it in the near future.

*Session IIb*

## Stability of Semi Displacement Ships

*Chairmen*

Dr. Choung M. Lee  
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## ON THE STABILITY OF SEMIDISPLACEMENT CRAFTS

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### ABSTRACT

The subject under consideration is the initial static stability of given steady states of motion of semi-displacement crafts. The problem designers are facing is to check this stability at early stages of the design process. In order to solve this problem a technique based on measurements of stability derivatives of hulls with appendages, but without propellers, has been proposed and is being developed at VWS, the Berlin Model Basin.

The paper outlines the whole procedure including possible extensions and ramifications. In particular it deals with the underlying stability theory, the necessary assumptions, and the measurement procedures. Finally measurements performed on a model without and with spray rails in the VWS towing tank are described and evaluated in a format ready for use in design.

### NOMENCLATURE

#### Quantities

I	inertias
D	dampings
S, C	stiffnesses
d, a	displacements
R	rudder force per unit angle
$\delta$	rudder angle
F, B	loads, generalized forces
F, M	forces, moments
x, y, z	coordinates, Cartesian
$\phi$	heel angle
$\theta$	trim angle
$\psi$	yaw angle
D	determinants
W	weight
L	lift

V	speed
P	parameters of hull form

#### Indices, lower

G	centre of weight
N	neutral points
L, M	metacentres
K	keel
1, 2, 3	centres of lateral forces
u, v	operational, 1...6, 1...3
x, y, z	coordinate directions

#### —, upper

g	weight
h	hydrodynamic
H	hull
R	residual
r	reduced
b	body fixed

### 1. INTRODUCTION

#### 1.1 Problem Formulation

The fact, that some of the semi-displacement crafts may encounter on stability problems at higher speeds and others not, became of interest during the last years in designing slender round bilge hulls for high speed operation. Several researchers investigated the loss of roll stability and the roll induced course instability at speeds  $F_n > 0.8$  (Refs.1 to 5).

The present study is comparatively limited in scope, namely restricted to the following problems:

- to develop stability criteria applicable at early stages of the design process,
- to develop a measuring technique necessary for the experimental determina-

tion of the stability limits, and to determine, as an example, the stability limits for a given hull form without and with spray rails.

It is not considered a task of this study to describe what happens, when initial instability exists, but rather to strip the problem down to the bare essentials and develop a clear understanding of the fundamentals and the methodology underlying the experimental technique.

## 1.2 Mathematical Model

The mathematical model underlying the present study is essentially the same linear model underlying among others the paper by Baba et al. (1982). For ready reference it is restated here in operational matrix notation, which greatly simplifies the following development.

It is assumed that small generalized displacements or deviations of a craft from its average state of translatory motion may be described in terms of six linear second order differential equations

$$I_{uv} \ddot{d}_v + D_{uv} \dot{d}_v + S_{uv} d_v = -R_u \delta + F_u.$$

In these equations  $I$ ,  $D$ , and  $S$  denote the generalized inertia, damping, and stiffness components, respectively,  $d$  the components of the displacement,  $R$  the components of the generalized rudder force per unit rudder angle,  $\delta$  the rudder angle, and  $F$  the generalized disturbing forces;  $u$  and  $v$  denote operational indices, ranging from 1 to 6, equal indices indicating summation (Einstein's summing convention).

In order to be specific, a right handed Cartesian coordinate system according to ITTC recommendations is chosen with the x-axis horizontally, positive in the direction of the average required motion, the y-axis horizontally, positive to the "right", starboard in body fixed coordinates, and the z-axis vertically, positive downward (!), the origin  $O$  of the coordinate system moving with the speed  $V$  of the average motion against the calm water. The rudder angle is taken positive to starboard (!).

In the undisturbed condition the reference point for the linear displacements

$$d_1 = \Delta x, d_2 = \Delta y, d_3 = \Delta z$$

and the disturbing moments

$$F_4 = M_x, F_5 = M_y, \text{ and } F_6 = M_z$$

coincides with the origin of the coordinate system and the craft has zero heel and yaw angles, while its trim angle is that corresponding to the average motion. Angular displacements are consequently

$$d_4 = \phi, d_5 = \Delta \theta, \text{ and } d_6 = \psi,$$

the heel angle around the x-axis (!), the change of trim angle around the y-axis, and the yaw angle around the z-axis, respectively.

## 1.3 Project Definition

If no heel is occurring or "admitted" two independent, i.e. decoupled planar motions may be treated separately: the surge, heave, and pitch motions in the vertical plane and the surge, sway, and yaw motions in the horizontal plane. These are the two types of motions customarily considered in elementary seakeeping and manoeuvring theories, respectively.

In the present study only the coupling of heel and yaw is of interest, the dependence of this coupling on forward speed in particular. If only steady deviations, their equilibrium, and the initial stability of this equilibrium are being considered, heel and yaw are decoupled from the rest of the displacements and may consequently be treated separately.

In this case the general linear differential equations of motion reduce to the linear equations for the equilibrium of the static lateral forces, heeling moments, and yawing moments. If for convenience the generalized stiffness, attitude, and load matrices

$$C_{uv} \equiv \begin{vmatrix} C_{11} & C_{12} & C_{13} \\ C_{21} & C_{22} & C_{23} \\ C_{31} & C_{32} & C_{33} \end{vmatrix} = \begin{vmatrix} R_2 & S_{24} & S_{26} \\ R_4 & S_{44} & S_{46} \\ R_6 & S_{64} & S_{66} \end{vmatrix}$$

$$a_u \equiv \begin{vmatrix} a_1 \\ a_2 \\ a_3 \end{vmatrix} = \begin{vmatrix} \delta \\ d_4 \\ d_6 \end{vmatrix} = \begin{vmatrix} \delta \\ \phi \\ \psi \end{vmatrix}$$

and

$$B_u \equiv \begin{vmatrix} B_1 \\ B_2 \\ B_3 \end{vmatrix} = \begin{vmatrix} F_2 \\ F_4 \\ F_6 \end{vmatrix} = \begin{vmatrix} F_y \\ M_x \\ M_z \end{vmatrix},$$

respectively, are introduced, these three equations may be written in the concise form

$$C_{uv} a_v = B_u.$$

## 2. STABILITY THEORY

### 2.1 Stability Conditions

For any steady loads the equilibrium attitudes may be determined as solutions of

the basic linear equations

$$a_u = C_{uv}^{-1} B_v$$

Subject to given constraints, for the rudder angle in particular, the loads have to be limited in general for equilibrium to be established.

In order that the equilibrium attitudes be stable, it is necessary and sufficient, that positive work has to be done to force the craft out of its equilibrium state:

$$\Delta E = \int B_u da_u = C_{uv} a_u a_v / 2 > 0$$

In other words: the potential energy of the equilibrium state has to be a minimum.

As is well known in the theory of matrices the above condition is satisfied for all attitudes, if the (symmetric part of the) stiffness matrix is positive definite, and this condition in turn is satisfied, if all main sectional determinants of the stiffness matrix are positive:

$$D_1 \equiv C_{11} > 0$$

$$D_{12} \equiv C_{11}C_{22} - C_{12}C_{21} > 0$$

$$D_{123} \equiv \det C_{uv} > 0$$

Assuming the first condition to be satisfied, there remain two different non-trivial stability conditions to be observed.

It may be noted that although the elements of the stiffness matrix depend on the choice of the reference point, the determinants of the matrix and consequently the stability limits of course do not. They are invariant properties of the craft as they should be and is shown in the following.

Writing the stiffness matrix in the form

$$C_{uv} = \begin{vmatrix} C_1 & C_2 & C_3 \\ -z_1 C_1 & -z_2 C_2 & -z_3 C_3 \\ +x_1 C_1 & +x_2 C_2 & +x_3 C_3 \end{vmatrix},$$

i.e. defining points 1, 2, and 3 as centres of action of lateral forces in the x-z-plane, results in the stability conditions

$$D_{12} = C_1 C_2 D_{12}^* > 0$$

and

$$D_{123} = C_1 C_2 C_3 D_{123}^* > 0$$

The constants  $C_u$  are derivatives of forces with respect to attitudes and consequently invariant. The determinants

$$D_{12}^* = z_{12} \equiv z_1 - z_2$$

and

$$D_{123}^* = A_{123}$$

are the difference of the vertical coordinates of points 1 and 2, and the area of the triangle defined by the points 1, 2, and 3 in that sequence and consequently both invariant under translations of the reference point as well.

While the first determinant changes sign when points 1 and 2 are at the same height, the second determinant changes sign when the points 1, 2, and 3 are collinear, i.e. in one straight line. This interpretation of the stability conditions may be helpful in understanding the stability limits.

## 2.2 Stability Criterion

The two stability conditions may be merged into one stability criterion, when the stiffness matrix is split into its weight and hydrodynamic components

$$C_{uv} = C_{uv}^g + C_{uv}^h$$

Except for the term

$$C_{22}^g = z_G W$$

where  $W$  denotes the weight and  $-z_G$  the height of its centre, alias the centre of gravity, all other components of the "weight stiffness" vanish.

Consequently the two stability conditions may be rendered in the forms

$$-z_G < -z_{N1} \equiv + D_{12}^h / D_{11}^h$$

and

$$-z_G < -z_{N2} \equiv + D_{123}^h / D_{13}^h$$

depending on the sign of the determinant

$$D_{13}^h \equiv C_{11}^h C_{33}^h - C_{13}^h C_{31}^h > 0$$

and use being made of the equilibrium condition

$$W - L = 0$$

for the vertical forces,  $L$  denoting the total hydrodynamic "lift".

Due to the moment of this force the determinants  $D^h$  are not invariant under translations of the reference point. For that reason a modified stiffness matrix may be introduced, which differs from the original only in the component

$$C_{22}^{h'} = C_{22}^h + z_L L$$

where  $z_L$  denotes the transverse metacentre at speed  $V$ . In terms of this matrix the above stability conditions may be put into the theoretically more satisfying forms

$$-z_G < -z_{N1} = -z_L + D_{12}^{h'}/D_{11}^{h'} L$$

and

$$-z_G > -z_{N2} = -z_L + D_{123}^{h'}/D_{13}^{h'} L,$$

depending on the sign of the determinant

$$D_{13}^{h'} = D_{13}^h \geq 0,$$

where all the determinants are now invariants proper.

The critical determinant will in general be negative

$$D_{13}^h = C_1 C_3 X_{31} = C_1 C_3 (x_3 - x_1) < 0$$

due to the sign of the quantity  $C_3$  and the magnitude of the coordinate  $x_3$ . Consequently the above stability conditions may be combined into the simple stability criterion

$$-z_{N2} < -z_G < -z_{N1}$$

or, in terms of the stability range  $Z$

$$-z_G \in Z(C_{uv}^h, L).$$

For design purposes it is apparently prerequisite to know the dependance of the stability limits on speed  $V$  and hull form parameters  $P$ :

$$Z(C_{uv}^h, L) = N(V, P),$$

i.e. to know the locus of the neutral points  $N_1$  and  $N_2$  in the  $V$ - $z$ -plane, preferably transformed to body(craft) fixed coordinates (index  $b$ ). At speed zero the neutral points are identical with the transverse metacentre at rest:

$$z_{N1} = z_{N2} = z_L = z_M.$$

Fig. 1 shows schematically the essential features and implications of this simple theory. For centres of weight above the neutral point  $N_1$  transverse instability exists. For centres of weight below the neutral point  $N_2$  course instability exists, "natural" for all slow speed ships. As indicated in Fig. 1 there may exist, for centres of weight above a certain limit, speed ranges of course stability. That this type of stability in fact exists can be concluded from reports stating, that "the boat behaved as if it was moving on rails".

### 2.3 Practical Procedure

In order to establish the stability limits the hydrodynamic stiffness matrix has to be determined for given lift and hull parameters at a number of speeds. According to the basic mathematical model

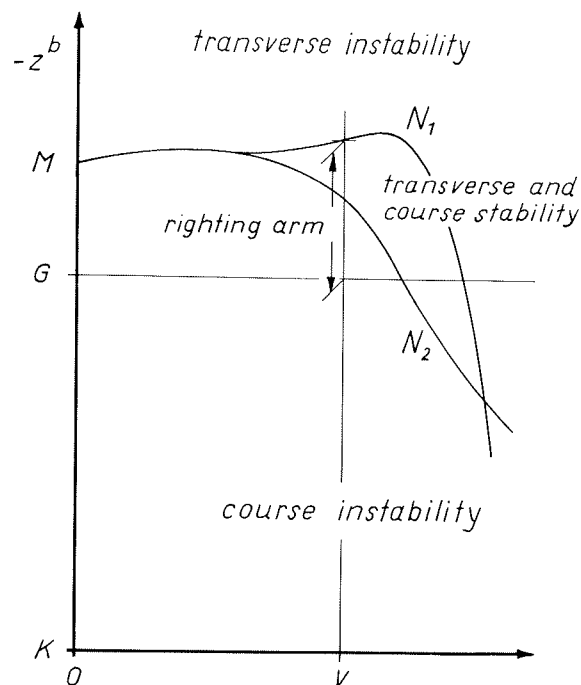


Fig. 1 Schematic diagram of stability limits

the components of the stiffness matrix are obtained as derivatives of the loads necessary to enforce given attitudes:

$$C_{uv}^h = \partial B_u / \partial a_v - C_{uv}^g$$

and are consequently called stability derivatives.

The determination of the stiffness matrix requires in principle captive tests with self-propelled models, free only to heave and pitch, at the correct Froude and cavitation numbers. So far such tests can only be performed in the large circulating tunnel of VWS, the Berlin Model Basin, in the speed range of interest.

Even if costs are of no concern, complete tests of this type are not possible at an early stage of the design process, when propeller and rudder configurations have not been decided upon. In these cases it is necessary to determine the stability limits from tests at atmospheric pressure with the hull and provisional appendices alone. The reasoning behind the procedure adopted is as follows.

The hydrodynamic stiffness may be split into the hull stiffness and the residual stiffness

$$C_{uv}^h = C_{uv}^H + C_{uv}^R.$$

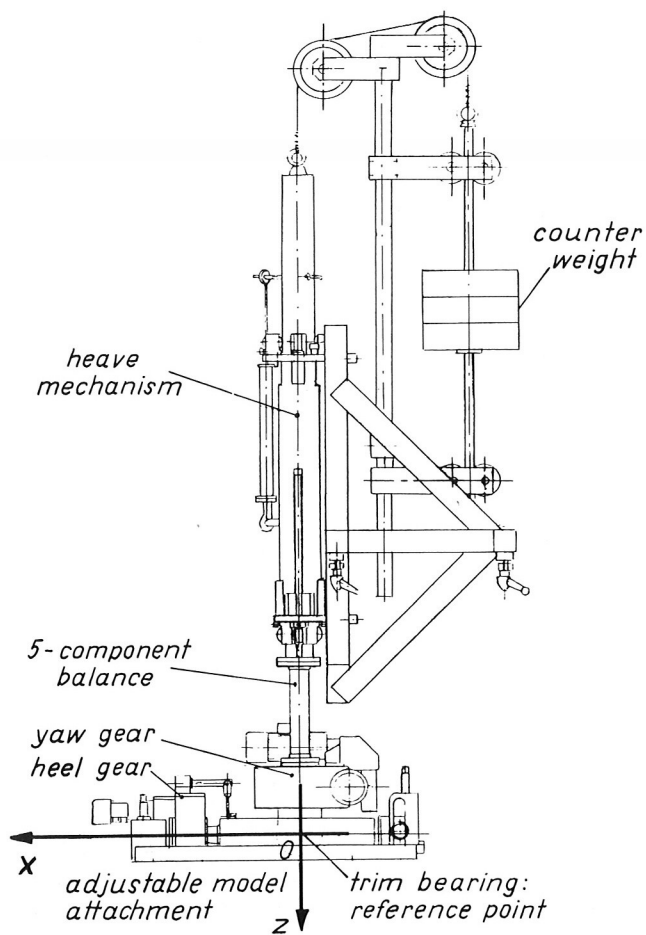


Fig. 2 Principal components of the heel-yaw-mechanism

While the structure of the hull stiffness matrix is

$$C_{uv}^H = \begin{vmatrix} 0_u & C_{u2}^H & C_{u3}^H \end{vmatrix}$$

the structure of the residual stiffness matrix is, for lack of better knowledge, assumed to be

$$C_{uv}^R = \begin{vmatrix} R_u c_1 & R_u c_2 & R_u c_3 \end{vmatrix}$$

with unknown factors  $c_v$ .

As for the stability considerations only determinants of the stiffness matrix are required, more particular, only ratios of such determinants, the values of the coordinates  $z_{N1}$  and  $z_{N2}$  remain unchanged, if instead of the original hydrodynamic stiffness matrix  $C^H$  the reduced matrix

$$C_{uv}^r = \begin{vmatrix} 1 & C_{12}^H & C_{13}^H \\ -z_R & C_{22}^H & C_{23}^H \\ +x_R & C_{32}^H & C_{33}^H \end{vmatrix}$$

is used for their evaluation. Accordingly only the hull stiffness components have to be determined and the centre of action of the resulting rudder force has to be estimated in order to establish the stability limits.

### 3. MODEL TESTS

#### 3.1 Test System

For the experimental determination of the stability limits a computer based test system has been developed at VWS. This test system can be used in the large free surface circulating water tunnel or in the deep water towing basin. The test set up, shown in Fig. 2, allows the model to pitch and heave freely.

A five component balance is attached directly to the heave mechanism. Beneath the balance follow the yaw gear and the pitch bearing in sequence. Through the centre of this bearing, which is conveniently taken as reference point, runs the axis of heel. Its inclination relative to the model may be adjusted. Heel and yaw angles can be remotely controlled.

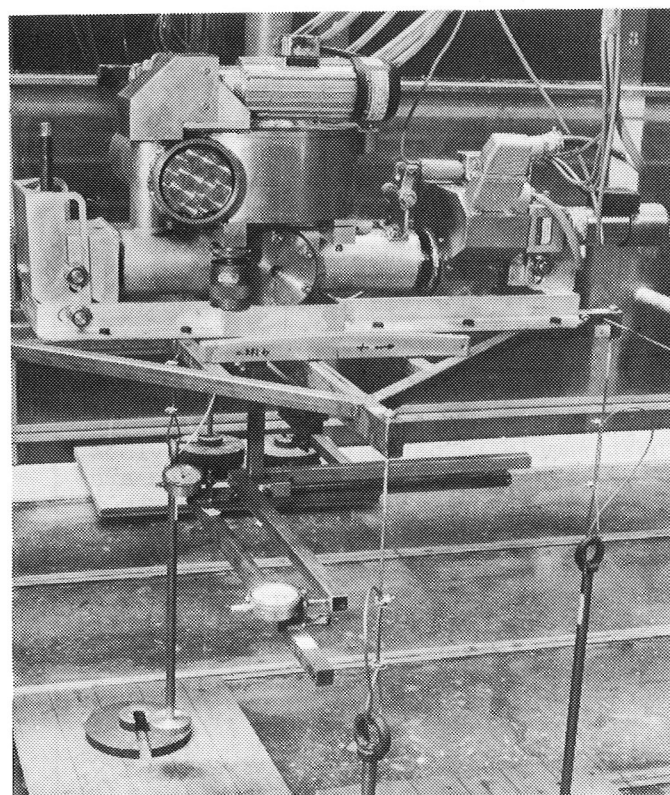


Fig. 3 Close-up view of the heel-yaw-mechanism

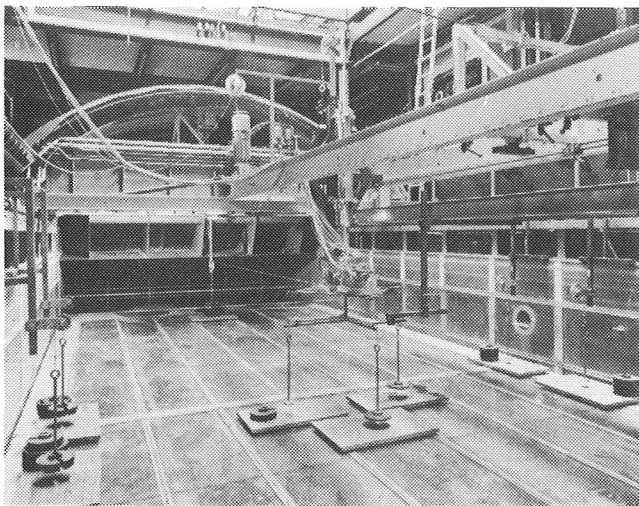


Fig. 4 Calibration set-up in the cavitation tunnel

Program and data organisation have been designed to match the test requirements with the computer (HP 9825) capabilities. The program system consists of a number of interdependent modules, reflecting the principle TASKS to be performed

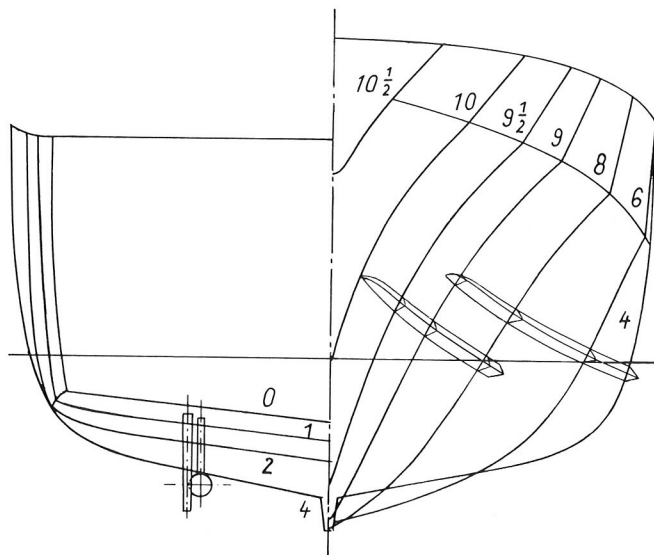
- .1 the ROOT with subroutines for DATA acquisition and matrix INVERSION,
- .2 the CALIBRATION for ATTITUDE transducers, BALANCE, and DEFLECTIONS as functions of loads,
- .3 the CHECKS of the system and last but not least
- .4 the TEST, RUNS, and LOADS INITIATION and FINILISATION.

Linear balance and deflection (tor-sion) calibrations include all inter-actions between the components. The cali-bration set-up in the test section of the free-surface cavitation tunnel is shown in Figs. 3 and 4.

In the circulating tunnel testing time is only limited in view of costs. Due to the inflow turbulence data have to be averaged, the maximum sampling period is 6 sec to avoid counter over-flow. If this turns out to be insufficient repeated samples may be taken. While the next sample is being taken the previous sample is evaluated and the results are plotted. This technique permits perfect on-line monitoring and control of the tests.

On the towing carriage on the other hand, where the tests reported have been carried out, data acquisition has top pri-ority due to the limited testing time at high speeds. One sweep of heel or yaw is performed quasisteadily in a single test run. Subsequently data are evaluated after the sweep and forces and moments are plotted for monitoring purposes.

Data obtained in the LOADS routine are combined in the RUNS routine to pro-vide the hull stiffness matrix and the stability limits at a given speed. By sev-eral test runs in the speed range of in-



Model 2401

Fig. 5 Body plan of the model hull form tested

terest, the required informations are ob-tained for the given displacement of the investigated hull.

Data reduction at the various steps follows the outline in section 2, using third order approximations for the deter-mination of the stability derivatives. Raw

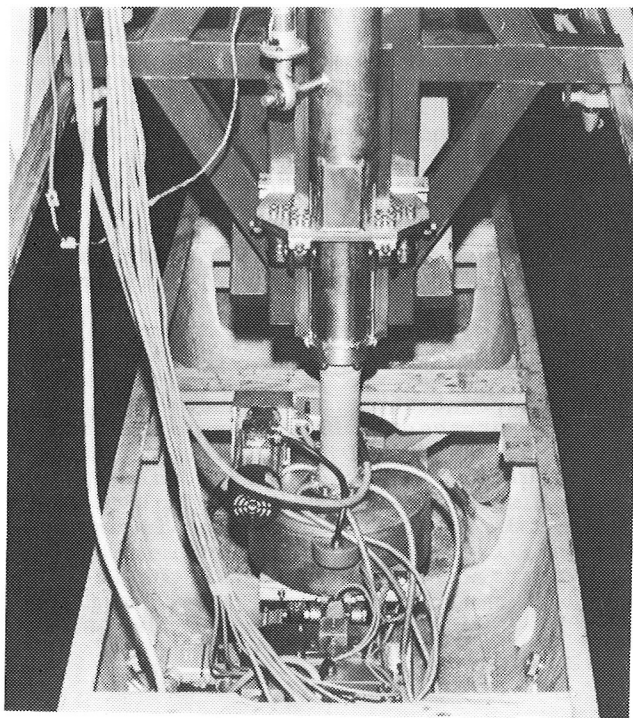


Fig. 6 Test set-up in the towing tank



data as well as intermediate results are stored for each sample, as are the final results.

### 3.2 Test Procedure

For the investigation a 1:11 scale glass fiber reinforced plastics model of a fast fishery protection vessel has been used. The body plan of the design is shown in Fig. 5, the main particulars of the model and the hull form parameters are given in the following table.

#### Main particulars of the model

Length betw. perpend	$L_{PP}$	m	3.864
Breadth, moulded	B	m	0.682
Draught of hull without keel	$T_H$	m	0.186
Displacement volume with appendages	$\nabla$	m <sup>3</sup>	0.1915
Longitud. centre of buoyancy before AP	$\overline{AB}$	m	1.716
Block coefficient	$C_B$		0.427
Midship coefficient	$C_M$		0.631
Slenderness coefficient	$L/\nabla^{1/3}$		6.70
Length-Beam ratio	$L_{WL}/B_{WL}$		6.04

The model, equipped with appendages including two shafts, four single arm struts, two rudders and a centerline skeg, was tested without and with a set of overlapping sprayrails at the forebody as indicated in Fig. 5.

The towing and reference point was located in the plane of the shaft axes above the centre of buoyancy, 0.216 m above the baseline. At all tests the heel axis was adjusted parallel to designed water line. Fig. 6 shows the test set-up on the towing carriage. The centre of the lateral rudder force has been assumed at the rudder stock, 0.1 m forward of AP and 0.15 m above the baseline.

The correct position of the longitudinal position of the centre of weight was carefully controlled by balancing the fully equipped model including the test apparatus in air. The actual height of the centre of weight  $\overline{KG}$  has been determined by a heel test in air with the fully equipped model attached to test system, in the same way as the tests have been carried out.

At the tests the towing force was adjusted to act in line with the thrust axes by applying appropriate vertical forces.

During the test runs one sweep of heel or yaw ranging up to  $\pm 6$  degrees was made, depending on the test time available. A yaw sweep at the Froude number 1.0 is shown in Fig. 7. The speed range was extended as far as possible, up to the Froude number 1.4, which may not be of practical interest.

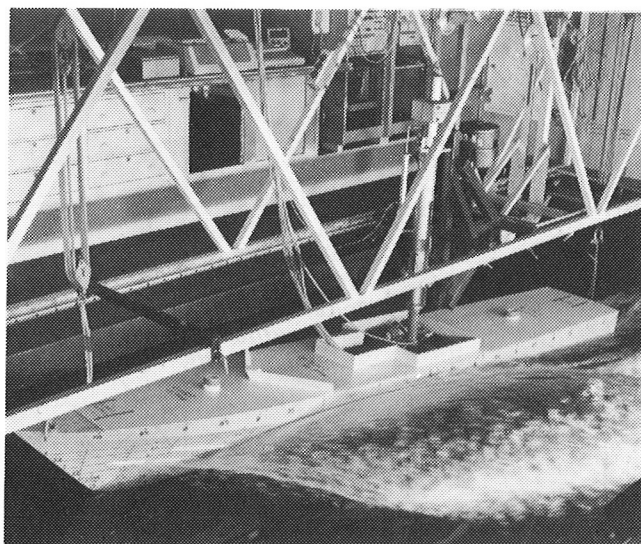


Fig. 7 Test run in the towing tank

While at slow speeds sample periods were 1 sec and 32 samples could be taken in one sweep, the sample period had to be reduced to 0.2 sec and only about half the number of samples could be taken at the top speed of 8.6 m/s. Consequently the confidence interval for the stability derivatives at the higher Froude numbers is considerably wider than at the lower ones.

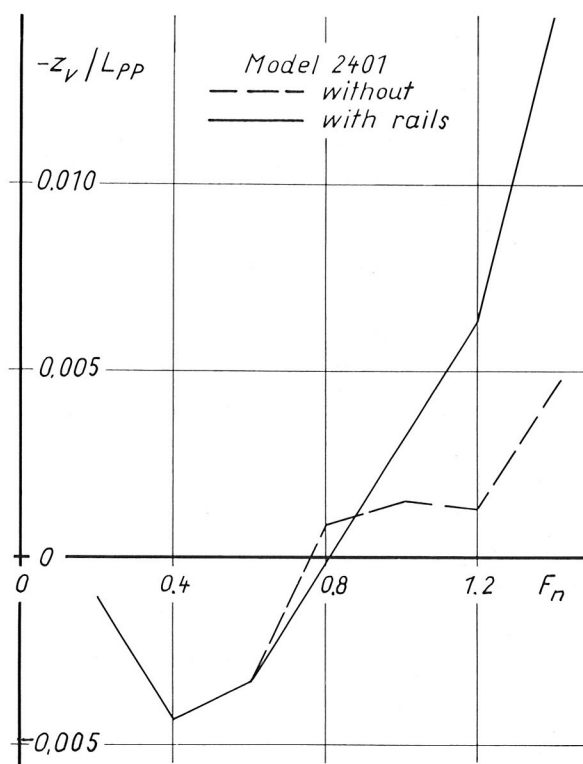


Fig. 8 Vertical displacement of the centre of buoyancy at zero heel and yaw angles

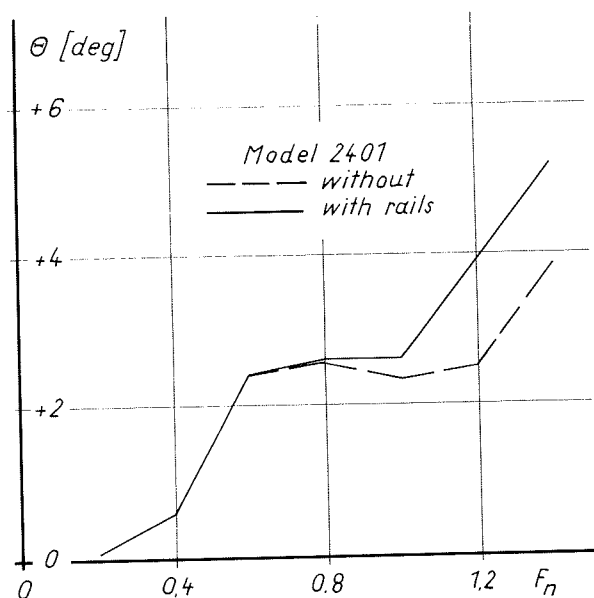


Fig. 9 Trim angle at zero heel and yaw angles

### 3.3 Test Results

Figures 8 and 9 show the vertical displacement at centre of gravity and the trim angle of the model at zero heel and yaw angles, i.e. the equilibrium states, the stability of which was investigated. Both quantities are increased remarkably by the spray rails. This fact has to be kept in mind, when the following results are discussed.

Figures 10 and 11 show the stability derivatives, which take into account the deflections of the measuring system, but no other corrections.

Figures 12 and 13 show the coordinates of the "metacentric height" and of the centres of the yaw lateral forces, respectively. Due to the moment of the lift (s. 2.2) it is not possible to construct the centres of the heel lateral forces from the data at hand, but only their longitudinal coordinate, at least in principle. Due to the signal to noise ratio even the latter may not be determined with any confidence. Consequently only the heel derivative of the hull heeling moment divided by the lift has been determined and plotted. The centres of yaw lateral forces have been determined as defined.

The final Fig. 14 shows the limits of initial static stability as determined for the model without and with spray rails. In this case the height is measured in body fixed coordinates from the keel (baseline).

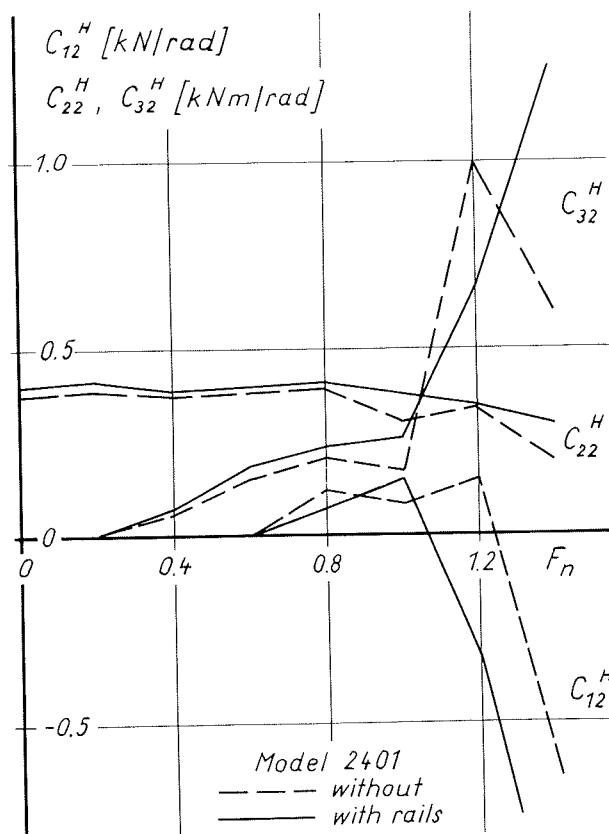


Fig. 10 Stability derivatives with reference to heel

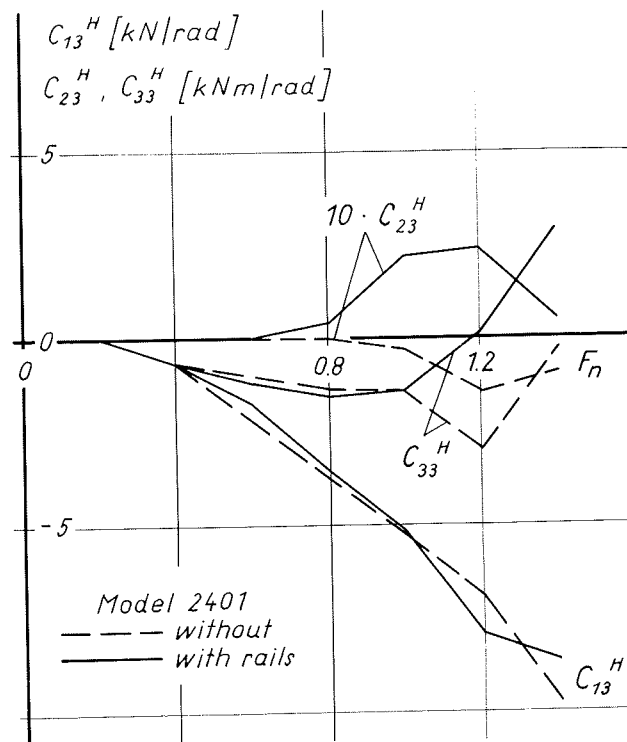


Fig. 11 Stability derivatives with reference to yaw



The tested hull form, which is characterized by straight sections and a hard chine at the afterbody and by a comparatively large value of  $\overline{KM}/T = 2.21$  and typical full scale values of  $\overline{KG}/T = 1.3 \dots 1.6$ , exhibits a very stiff behaviour. The effective metacentric height underway,  $-z_{NK}^b$ , increases for the hull without and with spray rails with speed up to the Froude number 0.8. At higher speeds the initial transverse stability begins to drop. For the hull without spray rails the limit of this stability is reached at the Froude number 0.9.

The hull with spray rails presents a larger value of  $\overline{KM}$  than without rails, because the transverse moment of inertia of the waterline is increased by the two pairs of spray rails having long intersections with the designed waterline. Due to the stabilizing lift forces, generated by the rails, the limit of initial stability is shifted to the higher Froude number 1.1. Beyond this speed the metacentric height drops rapidly. The remarkable gain in effective metacentric height underway is caused by a specific arrangement of the overlapping spray rails as seen in Fig. 5.

The other interesting fact exhibited by the results is the existence of a range of course stability before the limit of transverse stability is reached. As mentioned before this phenomenon has been observed in practice.

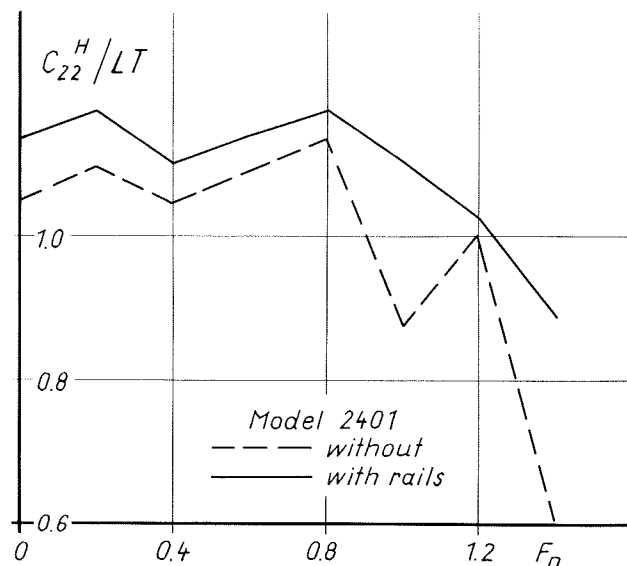


Fig. 12 Heel derivative of hull heeling moment divided by the lift

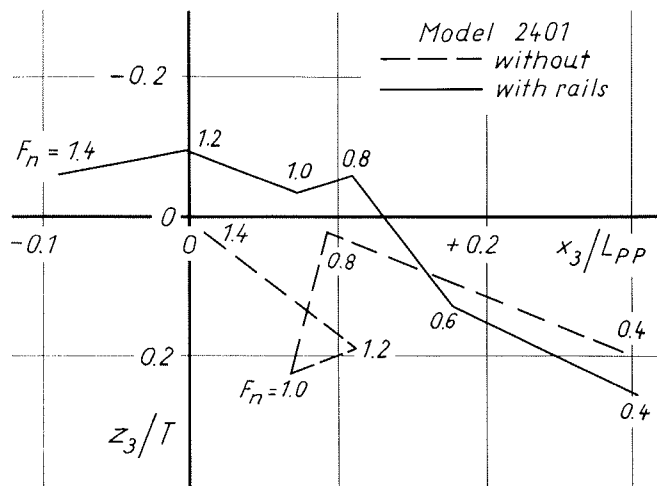


Fig. 13 Centre of yaw lateral force

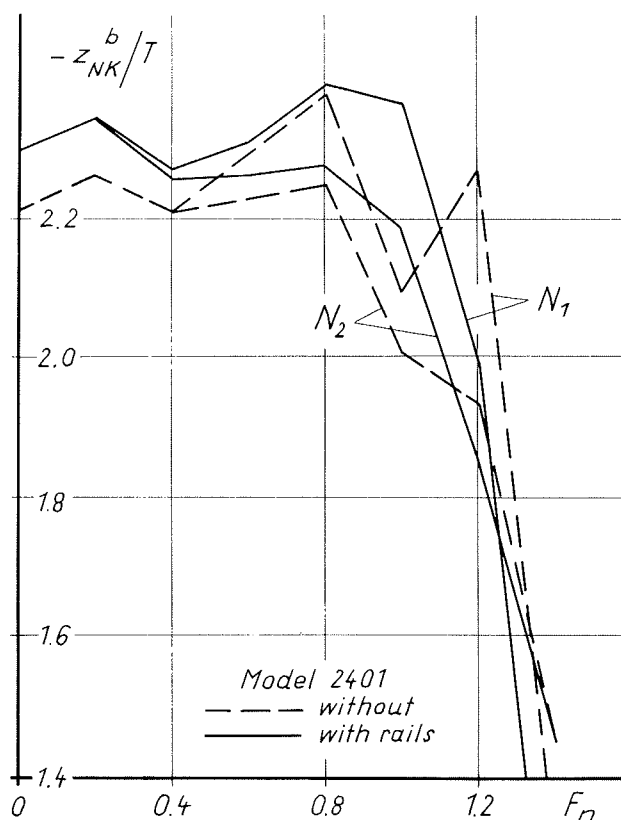


Fig. 14 Stability limits at zero heel and yaw angles

#### 4. CONCLUSIONS

##### 4.1 Mathematical Model

The scope of the present study was limited to the initial static stability of given steady states of motion under steady disturbing forces. Although the underlying mathematical model, the equilibrium conditions for lateral forces, heeling moments and yawing moments is extremely simple, it permits to describe a number of very important features of the problem.

The practical determination of the stability limits at early stages of the design process is greatly facilitated by the proposed procedure using hull and appendages only. In future the validity of the hypothesis on the residual stiffness due to the propeller interaction may be checked by testing a model with and without propeller(s).

The assumption of linearity can be tested any time without extra effort, as has been done in the tests reported. If necessary stability investigations along the lines developed have to be based on the local stiffness matrix at the equilibrium attitudes under consideration.

##### 4.2 Test Technique

The technique developed to measure the necessary stability derivatives does not only show the feasibility of the procedure proposed, but can be used now routinely at little extra testing effort along with resistance tests.

The technique will in future be used to test models without and with propellers at atmospheric conditions and at the appropriate cavitation numbers. These tests will be supplemented by measurements of the pressure distribution. When experience has been gained and data collected recommendations for the hull design may hopefully be derived.

Testing may be extended to include the determination of the hydrodynamic inertia and damping components with the existing planar-motion mechanism installed in the free surface cavitation tunnel.

In contrast to free-running model tests the technique applied here permits a more detailed study of influences of hull modifications on the stability.

##### 4.3 Stability Data

The results as they stand are of course not more than examples "proving" the effects of spray rails on a particular hull form. In order to be really useful they have to be extended to other equilibrium conditions and hull forms.

At present there appears to be no hydrodynamic theory to interpret or predict test results, even plausibility

considerations are quite difficult and not always conclusive.

So for a long time to come model tests as described will be necessary to determine the stability limits of semi-displacement crafts.

#### ACKNOWLEDGEMENTS

The present study was sponsored by the Berlin Senator für Wirtschaft, funded by special funds of the European Recovery Program, and administered by the Forschungszentrum des Deutschen Schiffbaus. The mechanical system was designed by engineers G. Schellin and A. Friedrich, and manufactured by F. Lustig, while the data system, including the A/D conversion, has been developed and implemented by engineer U. Schwer, to mention only a few members of the staff of VWS, the Berlin Model Basin, engaged in the project. All this support is gratefully acknowledged.

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## Discussion

N. Toki (Nagasaki Experimental Tank, MHI, Japan)

The authors should be congratulated for the presentation of this interesting paper. As one of the authors of the next paper, I would like to clarify the difference of the author's viewpoint from ours.

The authors presented the generalized stiffness matrix and obtained elements of it through experiments. On the basis of the stiffness matrix, the authors examined the initial stage of the instability problem. Generally speaking, we did almost the same thing. The authors might have already analysed the elements of stiffness matrix into components, however, the results are not presented so far.

On the other hand, we separated roll restoring moment in still water from the stiffness matrix, and analysed the data measured at models. As the results, we found that the hydrodynamic effects on the stiffness matrix are similar to each other in the cases of two semi-displacement type hulls, and the most dominant factor is metacentric height in still water. At the same time, we developed a numerical simulation method and obtained qualitatively good agreement between the results of simulation and free model test. On this basis, we can carry out estimations of stability at the initial design stage. Needless to say, we can perform model experiments whenever more accurate estimations are needed.

In the present paper, the authors did not show a way through which we can obtain even a rough estimation of stability without model experiments. Although the authors' model experiment technique seems to be much more sophisticated and efficient than ours, it is too time-consuming to carry out model experiments for every design.

This is my understanding of the present paper. If the authors could add further explanation, it would be very helpful.

### Author's Reply

We would like to thank the discussor for his kind appreciation of the work we did. He has correctly stated, that he and his co-workers "generally speaking" did "almost the same thing" as we did. In our view there are in fact quite important differences between our approach and the one described in the paper to be presented next.

As I have made clear, we are dealing only with the initial static stability of given states of motion. Our analysis includes forces and moments due to setting as equilibrium cannot be established without them. Finally we judge the stability directly from the stiffness of the system. As a consequence we have a problem in comparing our results with those of the following paper and understanding, what is meant by "qualitatively good agreement" with free model tests.

We agree that the metacentric height in still water is the dominant factor. Knowing this in advance, we did not study this effect, but rather concentrated on the exact determination of the neutral points. In this context I would like to draw attention to the second neutral point again. The dramatic change in the stability characteristics way upset any simulation including rudder control, if the latter is not modelled correctly, namely adaptive.

From the overall investment we put into our system we doubt that it is "needless to say" how more precise data can be obtained experimentally at the extent necessary for simulation studies. We did not intend to determine dynamic properties, nor did we intend to theoretically estimate the values of the stiffness components. The experimental technique we apply is extremely efficient and does not involve which testing time, maybe one day of tank testing for one configuration in the range of practical interest.

# A SIMULATION STUDY ON SWAY-ROLL-YAW COUPLED INSTABILITY OF SEMI-DISPLACEMENT TYPE HIGH SPEED CRAFT

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## ABSTRACT

To investigate into the roll induced instability of semi-displacement type high speed craft, simulation studies were carried out for sway-roll-yaw coupled motions. Hydrodynamic coefficients for the simulation studies were obtained by means of captive model test for a round bilge type hull form with and without spray strips, and for a hard chine type hull form. It was found that metacentric height GM rather than the difference in hull forms has a major effect on the roll induced instability of semi-displacement craft at high forward speeds. It was also found that course stability is improved by spray strips which show an effect to raise the vertical position of acting point of sway force.

## NOMENCLATURE

### [Physical constant]

$g$	Acceleration of gravity	(m/sec <sup>2</sup> )
$\rho$	Density of water	(kg.sec <sup>2</sup> /m <sup>4</sup> )

### [Hull]

$L$	Length on load waterline	(m)
$B$	Breadth	(m)
$d$	Draft at midship	(m)
$x_G$	x co-ordinate of center of gravity G	(m)
$z_G$	z co-ordinate of G	(m)
$KG$	Height of G from keel line	(m)
$\Delta a$	Displacement of hull with appendages	(kg)
$k_{xx}$	Radius of gyration about x axis	(m)
$k_{zz}$	Radius of gyration about z axis	(m)

$I_{xx}$	Moment of inertia about x axis	(kg.m.sec <sup>2</sup> )
$I_{zz}$	Moment of inertia about z axis	(kg.m.sec <sup>2</sup> )

### [Propeller]

$D$	Diameter	(m)
$P$	Pitch	(m)

### [Rudder]

$x_R$	x co-ordinate of rudder stock	(m)
$z_R$	z co-ordinate of mid point of rudder height	(m)
$b$	Breadth	(m)
$h$	Height	(m)
$A_R$	Rudder area	$A_R = b \cdot h$ (m <sup>2</sup> )
$\lambda$	Aspect ratio of rudder	$\lambda = h / b$

### [Motion]

$u$	Velocity in x direction	(m/sec)
$F_n$	Froude number	$F_n = u / \sqrt{g \cdot L}$
$v$	Sway velocity	(m/sec)
$U$	Advance speed	(m/sec)
$U_s$	Advance speed of full scale ship	(kn)
$\beta$	Drifting angle	(deg)
$r$	Yaw rate	(deg/sec)
$\psi$	Yaw angle	(deg)
$\phi$	Heel angle, Roll angle	(deg)
$n_p$	Rate of propeller revolution	(rps)
$\delta$	Rudder angle	(deg)

### [Hydrodynamic characteristics]

$X$	Force in x direction	(kg)
$Y$	Sway force	(kg)
$N$	Yaw moment	(kg.m)
$K$	Heel moment, Roll moment	(kg.m)
$R_f$	Frictional resistance	(kg)
$R_r$	Residual resistance	(kg)
$T$	Propeller thrust	(kg)
$t$	Thrust deduction factor	
$w$	Effective wake fraction	

$u_p$  Inflow velocity toward propeller  
 $u_p = (1 - w)u$  (m/sec)  
 $m_x$  Added mass in x direction (kg.sec<sup>2</sup>/m)  
 $m_y$  Added mass in y direction (kg.sec<sup>2</sup>/m)  
 $J_{xx}$  Added mass moment of inertia about x axis (kg.m.sec<sup>2</sup>)  
 $J_{zz}$  Added mass moment of inertia about z axis (kg.m.sec<sup>2</sup>)  
 $x_Y$  x co-ordinate of acting point of sway force Y (m)  
 $z_Y$  z co-ordinate of acting point of sway force Y (m)

## 1. INTRODUCTION

It is known that a high speed ship with relatively small metacentric height GM sometimes undergoes difficulty in keeping her course straight (Refs.1 to 3). This behaviour is explained in connection with hydrodynamical asymmetry of the underwater hull form due to roll and this phenomenon is called "roll induced instability".

In the case of high speed craft, the phenomenon is more critically related to the safety of operation. Therefore, studies have been made by several researchers so far, especially putting an emphasis on reduction of virtual transverse stability (Refs.4 to 7). However, detailed causes for the roll induced instability have not yet been fully explained, especially of high speed craft. In Nagasaki Experimental Tank, it has been also noticed that a semi-displacement type high speed craft has a heel angle possibly caused by the reduction of virtual GM during its resistance test at relatively high speed. For assuring the phenomenon, free-running model test was carried out. Typical examples of the test results are shown in Fig. 1, where marked port turning is observed for the condition of small GM even at zero helm. And this turning is associated with outward heel, i.e. starboard heel at port turning and port heel at starboard turning, either of which eventually invites capsizing.

Considering these situations, a basic investigation is made on the phenomenon by means of captive model test. And utilizing thus obtained hydrodynamic coefficients, simulation study is carried out on sway-roll-yaw coupled motions.

## 2. CAPTIVE MODEL TEST

### 2.1 Tested Models

Two types of semi-displacement type high speed craft model were used, i.e. a round bilge type (with and without spray strips) and a hard chine type. Principal particulars and body plans are shown in Table 1 and Fig. 2, respectively. Experiments were made in Nagasaki Experimental Tank, M.H.I. Taking operation conditions into account, speed of the model was varied in the range of Froude number  $F_n$  from 0.46

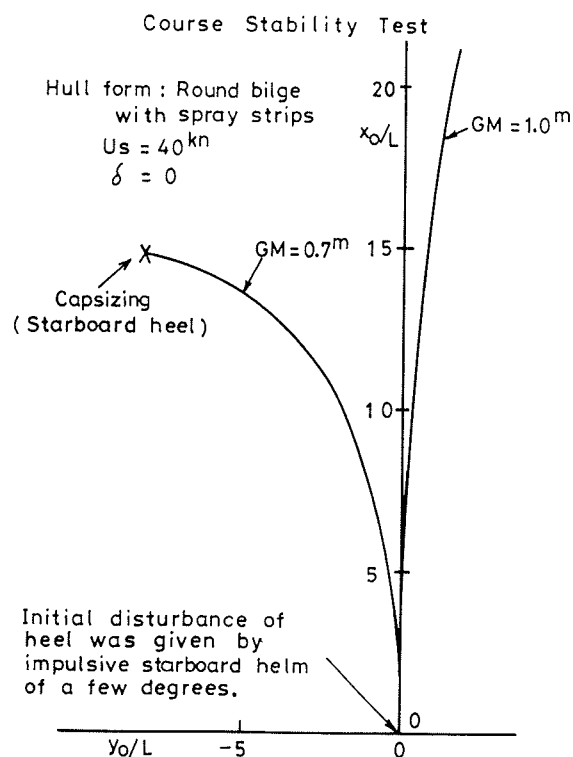


Fig. 1 Observed trajectories obtained photographically by use of the free-running model with a pinpoint flash-light

Table 1 Principal Particulars of the Hull Forms for Captive Model Tests and Simulation

Hull form	A	B
Item	(Round bilge with spray strips)	(Hard chine)
Load condition	Design load	Design load
Scale ratio	1 / 14.3	1 / 14.3
L (m)	3.600	3.600
B (m)	0.503	0.566
d (m)	0.126	0.105
L / B	7.15	6.36
B / d	4.00	5.39
$A_R / (L \cdot d)$	1 / 31.6	1 / 26.4

to 1.00 covering full scale speed ranging from 20 to 44 knots.

## Semi-Displacement Type High Speed Craft

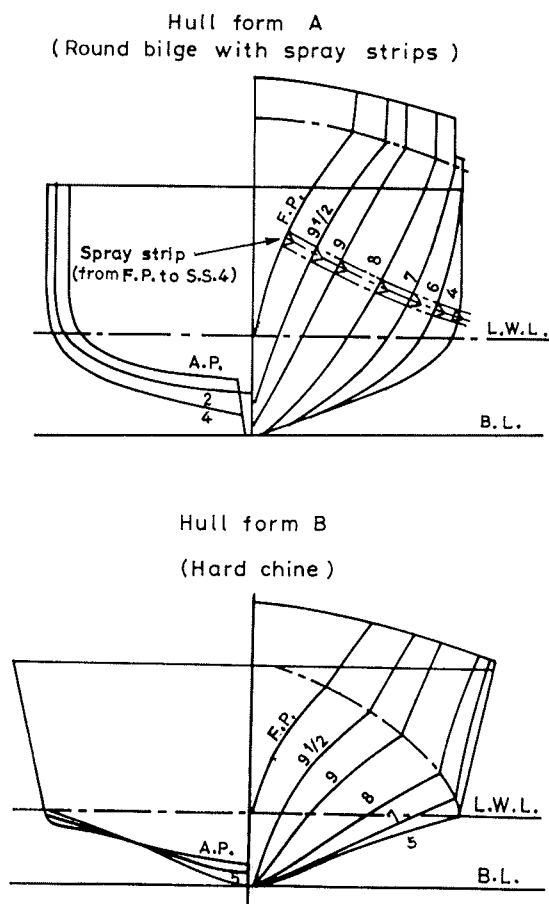


Fig. 2 Body plans of the hull forms for captive model tests and simulation

### 2.2 System of Measurement

Hydrodynamic forces and manoeuvring motions are defined in reference to the co-ordinate systems shown in Fig. 3. In the captive model tests, following hydrodynamic forces were measured, i.e. lateral forces acting on the fore and aft guides, vertical force acting on the gauge for heel moment, and rudder normal force. System of the measurement is shown in Fig. 4. In the heel free condition, heel moment was obtained from measured increase of heel angle and static GM, and in the heel restrained condition, it was obtained from measured vertical force acting on the gauge for heel moment.

### 2.3 Results of Captive Model Test

As the basic tests for steady manoeuvring characteristics, three kinds of captive model test were carried out. In these tests, sway force  $Y$ , yaw moment  $N$ , and others were obtained versus various heel angle, oblique tow angle, and rudder angle.

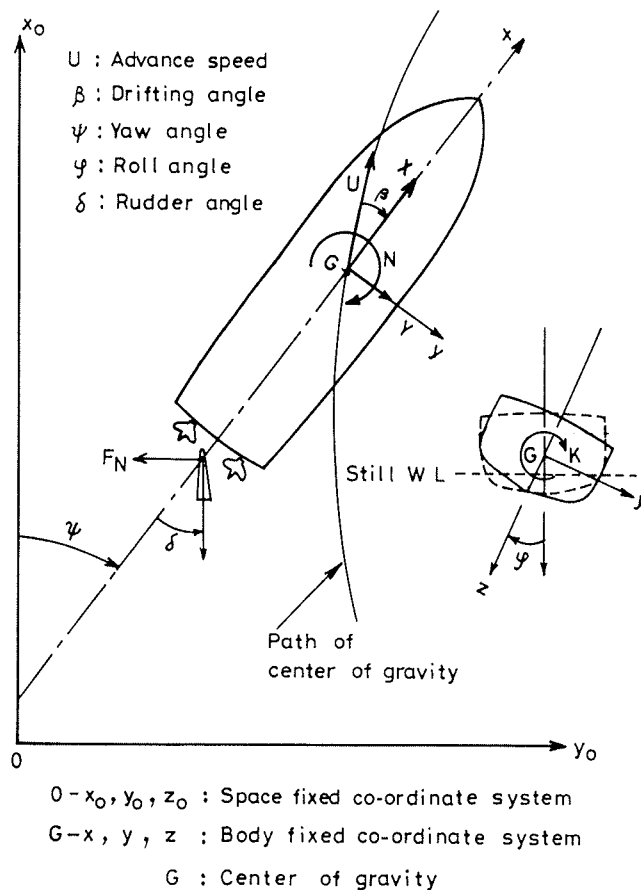


Fig. 3 System of co-ordinates and symbols

$Y$  and  $N$  were non-dimensionalized by following definitions:

$$\left. \begin{aligned} Y' &= Y / \left( \frac{\rho}{2} L d U^2 \right) \\ N' &= N / \left( \frac{\rho}{2} L^2 d U^2 \right) \end{aligned} \right\} \quad (1)$$

where  $U$  denotes advance speed.

#### [1] Heel Angle Test

Heel angle  $\phi$  of the model in straight course running was varied by transverse shift of a small ballast weight in the range of  $\phi$  from  $-10$  to  $10$  degrees. Results of the test are shown in Fig. 5, where derivative of  $Y'$  with respect to  $\phi$  of the hull form A is similar to that of the hull form B, although slight difference is observed in derivatives of  $N'$  with respect to  $\phi$  of the hull forms A and B. It is also found that  $Y'$  and  $N'$  obtained for the range of  $F_n$  from  $0.90$  to  $1.00$  show no substantial differences from the other results obtained for the range of  $F_n$  from  $0.46$  to  $0.90$ .

#### [2] Oblique Towing Test

The model was towed obliquely to a straight course, and drifting angle  $\beta$  was

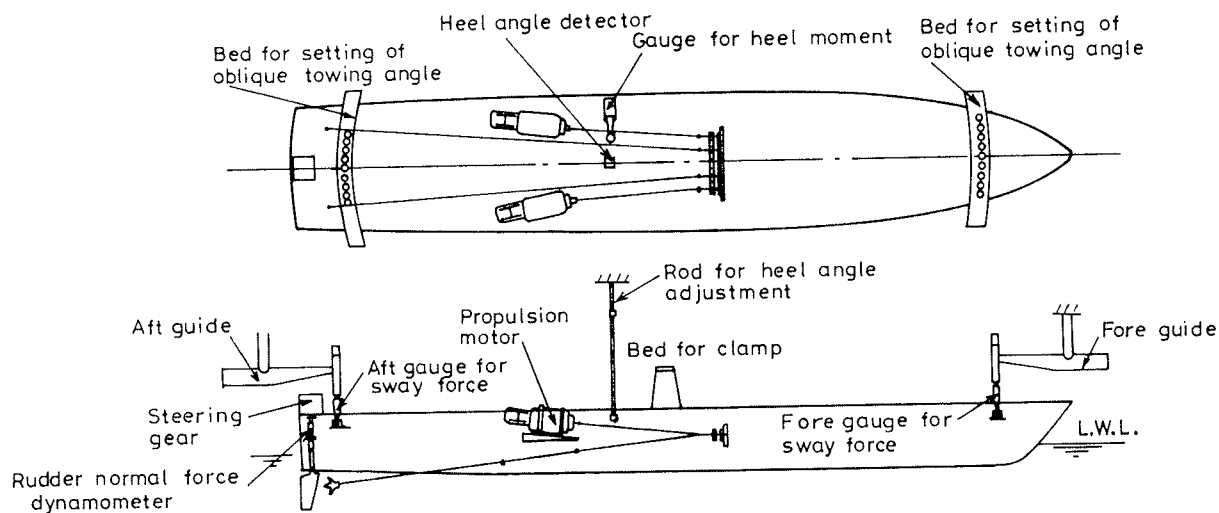


Fig. 4 System of measurement in captive model tests

varied in the range of  $\beta$  from  $-3$  to  $3$  degrees. Analysis was made for  $\beta$ -component force  $Y(\beta)$  and moment  $N(\beta)$  by subtracting  $\phi$ -component force  $Y(\phi)$  and moment  $N(\phi)$  from total force  $Y$  and moment  $N$ , respectively, where  $Y(\phi)$  and  $N(\phi)$  were obtained from the heel angle test results shown in Fig. 5. Results of the test are shown in Figs. 6 to 8. In Figs. 7 and 8,  $x$  and  $z$  co-ordinates of acting point of  $Y(\beta)$  are plotted respectively. Characteristics of both hull forms A and B are very similar to one another in these Figs. 6 to 8. From Fig. 8, it is found that acting points of  $Y(\beta)$  lie well close to the load waterline  $zy(\beta) = 0$  within the range of  $\beta$  from  $-3$  to  $3$  degrees. In Figs. 6 to 8, the results are also shown of the hull form B at which heel angle was restrained to zero. These results agree with the characteristics obtained from  $\beta$ -component force  $Y(\beta)$  and moment  $N(\beta)$  at the heel free condition. This fact indicates the possibility of linear superposition of  $\beta$ - and  $\phi$ -component forces and moments, when the tested hull form runs in the vicinity of straight course with small heel angle.

### [3] Rudder Angle Test

Rudder angle  $\delta$  was varied in the range of  $\delta$  from  $-7$  to  $7$  degrees while the model was running in a straight course. Results of the test are shown in Fig. 9, where  $Y_R$  and  $N_R$  are defined as follows:

$$Y_R = -F_N \cos \delta / \left( \frac{\rho}{2} L d U^2 \right)$$

$$N_R = Y_R (x_R - x_G) / L$$

where  $F_N$  : rudder normal force.

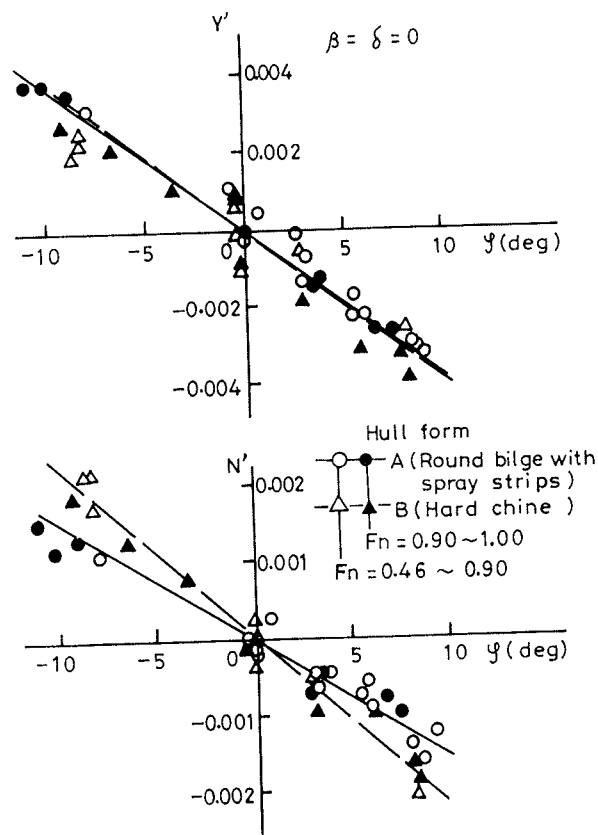


Fig. 5 Sway force and yaw moment obtained from heel angle test

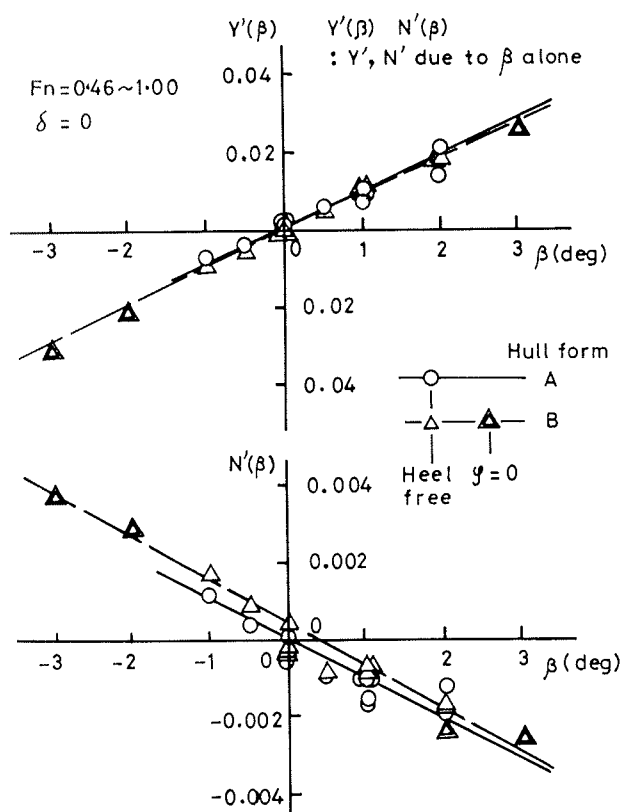


Fig. 6 Sway force and yaw moment obtained from oblique towing test

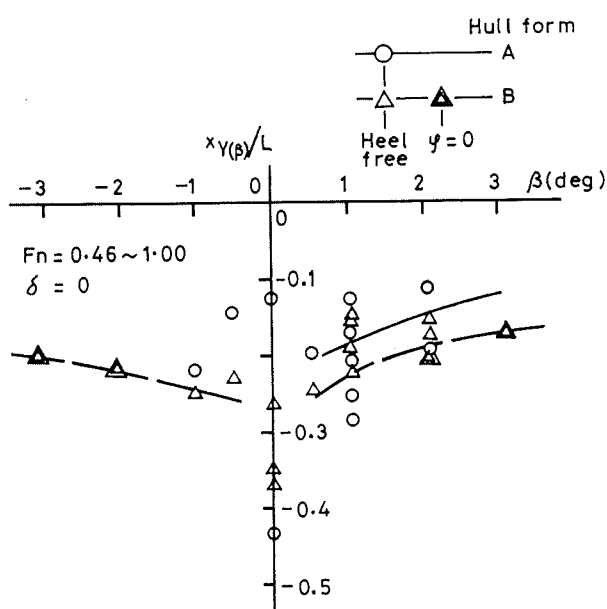


Fig. 7 Longitudinal position of acting point of sway force obtained from oblique towing test

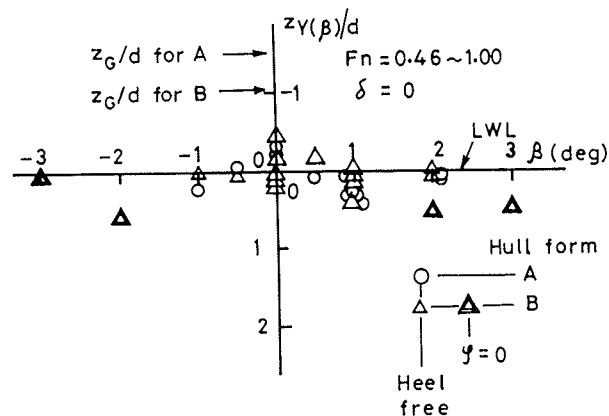


Fig. 8 Vertical position of acting point of sway force obtained from oblique towing test

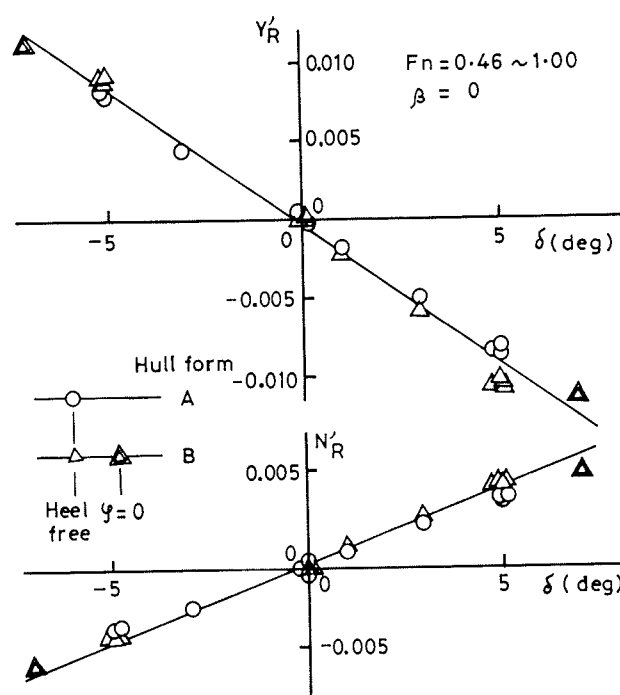


Fig. 9 Sway force and yaw moment obtained from rudder angle test

Obtained characteristics of both hull forms A and B are found very similar to one another. The results are also shown of the hull form B at which heel angle was restrained to zero. These results agree with the characteristics obtained for the heel free condition. This fact also indicates the possibility of linear superposition of  $\delta$ - and  $\phi$ -component forces and moments, when the tested hull form runs in the vicinity of straight course with small heel angle.



[4] Heel Angle Test for the Effect of Spray Strips

Heel angle  $\phi$  of the model in straight course running was varied by transverse shift of a small ballast weight.

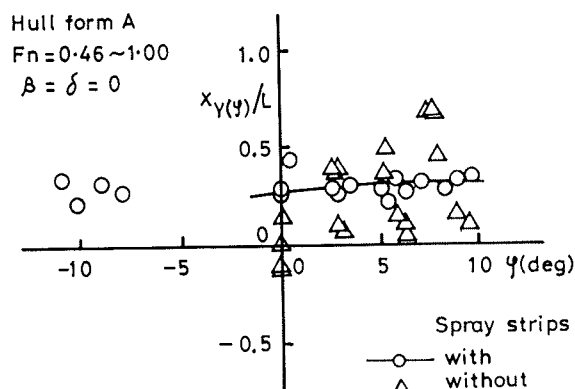


Fig. 10 Longitudinal position of acting point of sway force obtained from heel angle test

Results of the test are shown in Fig. 10, where  $x$  co-ordinates of acting point of  $Y(\phi)$  are plotted versus  $\phi$ . Considerable scattering of the results is noted for "without spray strips" condition. In other words, it can be said that the spray strips show such an effect as concentrating longitudinal acting point  $x_{Y(\phi)}$  of  $Y(\phi)$  to about  $0.3 L$ . Results of  $z$  co-ordinates of acting point of  $Y(\phi)$  are shown in Fig. 11. It is found that the spray strips also show an effect to raise acting point of  $Y(\phi)$  by as much as about  $0.5 d$ . As mentioned later, this effect results in less reduction of virtual-GM and in added stability of sway-roll-yaw coupled motions.

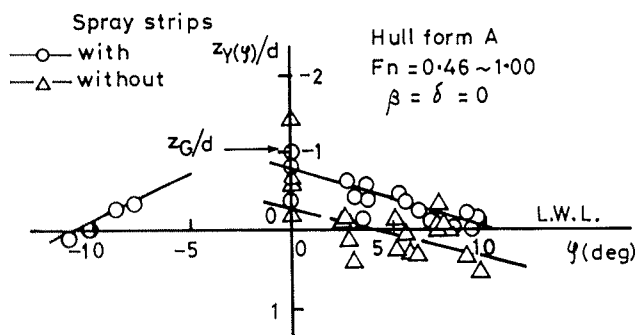


Fig. 11 Vertical position of acting point of sway force obtained from heel angle test

### 3. MATHEMATICAL MODEL FOR SWAY-ROLL-YAW COUPLED MOTIONS

In the present simulation study, a hull form is assumed to have single propeller and single rudder, both of which are in center line of the main hull, for the sake of simplicity. In reference to the co-ordinate systems shown in Fig. 3, the mathematical model expressed by the following equations is adopted for manoeuvring motions including roll:

$$\left. \begin{aligned} (m + m_x) \dot{u} - (m + m_y) v r \\ &= X_H + (1 - t) T - F_N \sin \delta \\ (m + m_y) \dot{v} + (m + m_x) u r \\ &= Y_H - F_N \cos \delta \\ (I_{zz} + J_{zz}) \dot{r} \\ &= N_H - F_N \cos \delta (x_R - x_G) \\ (I_{xx} + J_{xx}) \ddot{\phi} \\ &= K_H - F_N \cos \delta (z_R - z_G) \end{aligned} \right\} \quad (2)$$

where  $X_H = -(R_f + R_r) + \frac{\rho}{2} L d U^2 \cdot X'_{vr} v' r'$

$$Y_H = \frac{\rho}{2} L d U^2 (Y'_v v' + Y'_r r' + Y'_\phi \phi)$$

$$N_H = \frac{\rho}{2} L^2 d U^2 (N'_v v' + N'_r r' + N'_\phi \phi)$$

$$K_H = K_\phi \ddot{\phi} - \Delta a \text{ GM } \phi$$

$$- (Y_H - m_x u r) (z_Y - z_G)$$

$X_H, Y_H, N_H, K_H$ : hydrodynamic forces and moments acting on the hull

$X'_{vr}, Y'_v, Y'_r, Y'_\phi, N'_v, N'_r, N'_\phi, K_\phi$ : hydrodynamic coefficients

other detailed notations are referred to the list of nomenclature.

In regard to the left hand side of Eq. (2), several additional terms have been proposed hitherto (Refs. 1 to 3 and 8). In the present study, however, simplest mathematical model as Eq. (2) is assumed. Making reference to the experimental results summarized in the previous sections, the right hand side of Eq. (2) has been derived from the following assumptions:

[1] Non-dimensional hydrodynamic coefficients remain constant during the motions.  
[2] Sway forces  $Y(\phi)$ ,  $Y(\beta)$ , and  $Y(r)$  act at the mid point of draft, and this position does not move during the motions. And rudder normal force  $F_N$  acts at the mid point of rudder height.

Most of hydrodynamic characteristics appearing in Eq. (2) were obtained from the experiments as follows:

- \* Towing test ----->  $R_r$
- \* Propeller open test ----->  $T$
- \* Self-propulsion test ----->  $w(F_n), t(F_n)$
- \* Heel angle test ----->  $Y_\phi, N_\phi$
- \* Oblique towing test ----->  $Y_v, N_v, \delta_o(\beta_R), z_Y(\beta)/d$

where  $\delta_o$ : stern flow direction at steady turning

$\beta_R$ : drifting angle at rudder stock

$$\beta_R = -\tan^{-1}(v_R / u)$$

$$v_R = v + (x_R - x_G) r$$

- \* Rudder angle test ----->  $u_{Re}/(P \cdot n_p)$

where  $u_{Re}$ : Effective inflow velocity toward rudder

- \* Inclining test ----->  $GM$

- \* Free rolling test ----->  $I_{xx} + J_{xx}$ ; at  $F_n = 0$

- \* Free rolling test ----->  $K_\phi$ ; at running condition

where  $K_\phi$ : roll damping.

Other hydrodynamic coefficients were evaluated by the following sources or formulae:

- \*  $m_x, m_y, J_{zz}$ : from Matora chart (Ref.9)
- \*  $C_N(\delta_e) = 6.13\lambda / (\lambda + 2.25) \delta_e$  (Ref.10)

where  $C_N = F_N / (\frac{\rho}{2} A_R u_{Re}^2)$

$\delta_e$ : Effective rudder angle

$$\delta_e = \delta - \delta_o$$

- \*  $X'_{vr} = -\frac{1}{3}(m' + m'_y)$

- \*  $Y'_r = -l'_p Y'_v, N'_r = -l'_p N'_v$  (Ref.3)

where  $l'_p = l_p/L$

$l_p$ : distance from G to pivoting point.

#### 4. SIMULATION OF SWAY-ROLL-YAW COUPLED MOTIONS

Making use of the mathematical model together with the captive model test results, calculations of sway-roll-yaw coupled motions were carried out. As for operation condition, approach speed  $U_s$  is prescribed as relatively high speed, i.e.  $U_s = 40$  knots ( $F_n = 0.916$ ) in full scale. And for designed load condition, KG and GM in full scale are adjusted as follows:

KG = 3.70 meters, GM = 1.05 meters for the hull form A,

KG = 3.70 meters, GM = 3.39 meters for the hull form B.

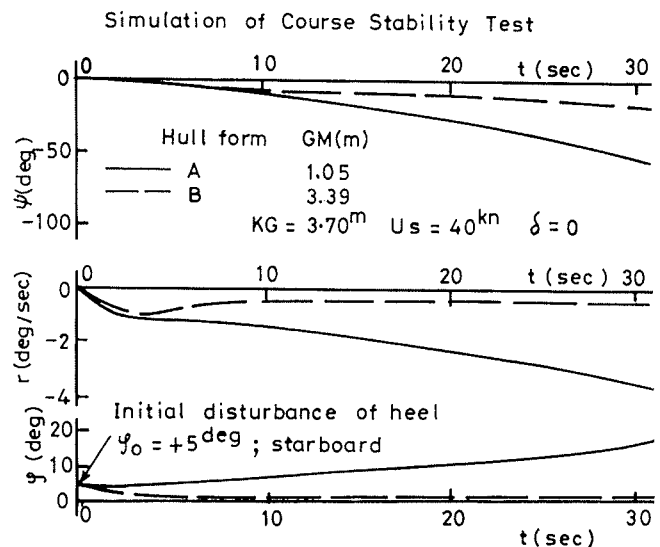


Fig. 12 Calculated manoeuvring motions of the hull forms A and B with roll coupling effect

#### 4.1 Effect of Hull Forms

As a test mode, course stability test is adopted in which rudder angle is kept at zero and initial disturbance of heel  $\phi_0 = 5.0$  degrees to starboard is given. Results of calculations are shown in Fig. 12. Yaw rate  $r$  and roll  $\phi$  of the hull form A keep increasing non-oscillatorily until eventual capsizing, while  $r$  and  $\phi$  of the hull form B non-oscillatorily tend to each steady value. Causes of these roll and yaw instabilities may be attributed to the following two effects:

[1] Effect of the hull form

Inherent course instability due to less hydrodynamical damping might have caused the apparent roll instability.

[2] Effect of GM

Relatively small GM might have caused the roll instability, and increasing roll might have induced continuous increase of yaw rate.

Considering these causes, calculations were made to obtain inherent sway and yaw, and roll motions. These inherent motions are obtained by assuming coefficients  $Y_\phi$  and  $N_\phi$  to be zero. Results of calculated trajectories are shown in Fig. 13. Non-dimensional rates of turn  $r'$  are plotted versus rudder angle  $\delta$  in Fig. 14, where  $r'$  is defined by  $r' = r L/U$ . From these results, it can be said that the inherent hydrodynamic characteristics show no substantial difference between the hull forms A and B, and that the apparent roll induced instability is more closely related to relatively small GM rather than the difference in hull forms.

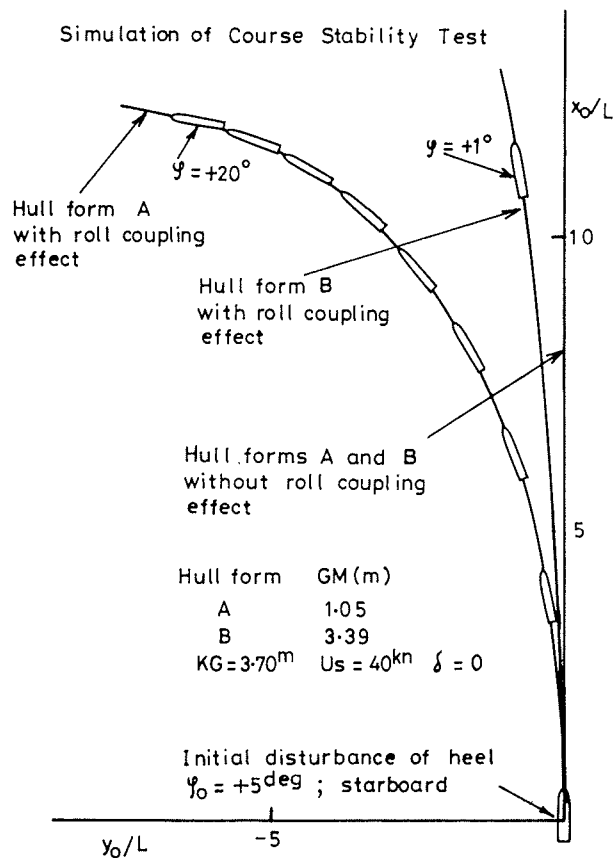


Fig. 13 Calculated trajectories of the hull forms A and B

#### 4.2 Effect of Vertical Position of Center of Gravity

Among hull forms of high speed craft, a hard chine type one is supposed to have relatively large stiffness against the roll induced instability, in general. According to the results of simulation study in the previous section, however, the effect of difference in GM is more marked than that in the hull forms in this phenomenon. To clarify the effect of vertical position of center of gravity, calculations were made for the hard chine type hull form B. The results are shown in Figs. 15 and 16. Comparing Figs. 12 and 15, and Figs. 13 and 16, respectively, it is found that the hull form B also shows the roll induced instability when GM is decreased to the value of about 2 meters. Therefore, GM is to be carefully selected in high speed craft design even for hard chine hull forms.

#### 4.3 Effect of Spray Strips

From the results of captive model test explained in the section 2.3 [4], in case of the hull form A,  $z_Y/d = 0.5$  and 1.0 are

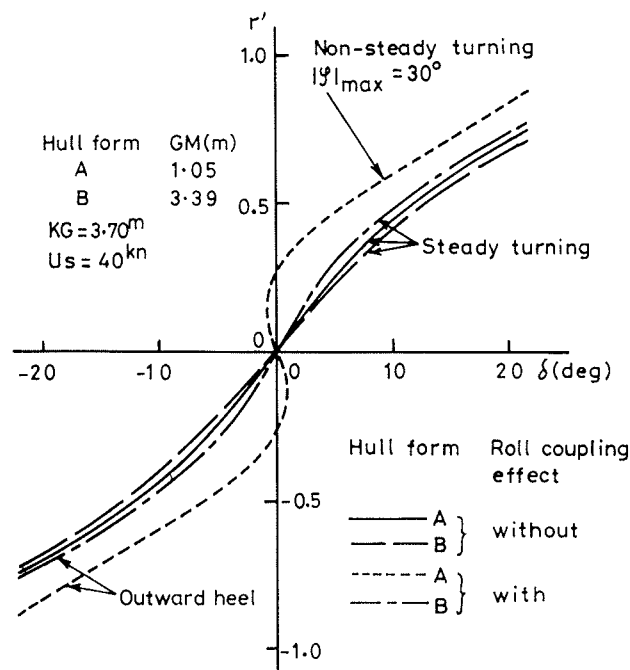


Fig. 14 Calculated non-dimensional rate of turn  $r'$  in turning test

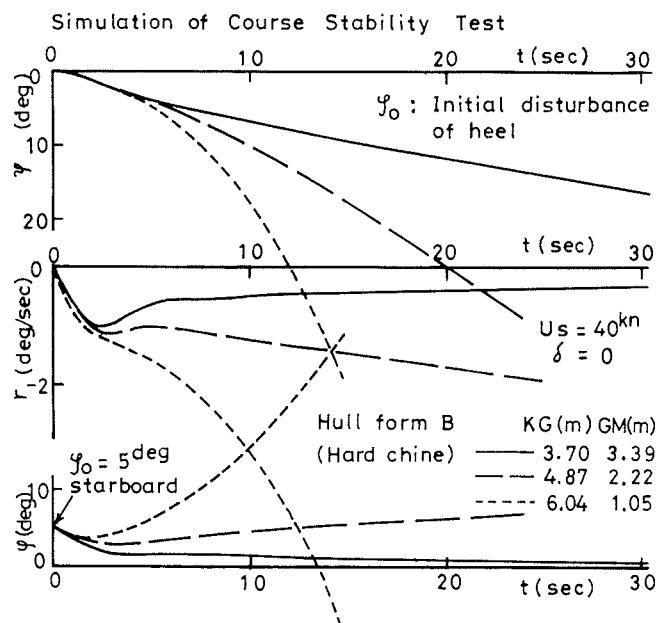


Fig. 15 Calculated manoeuvring motions of the hard chine type hull form B (Effect of vertical position of center of gravity)

# Simulation of Course Stability Test

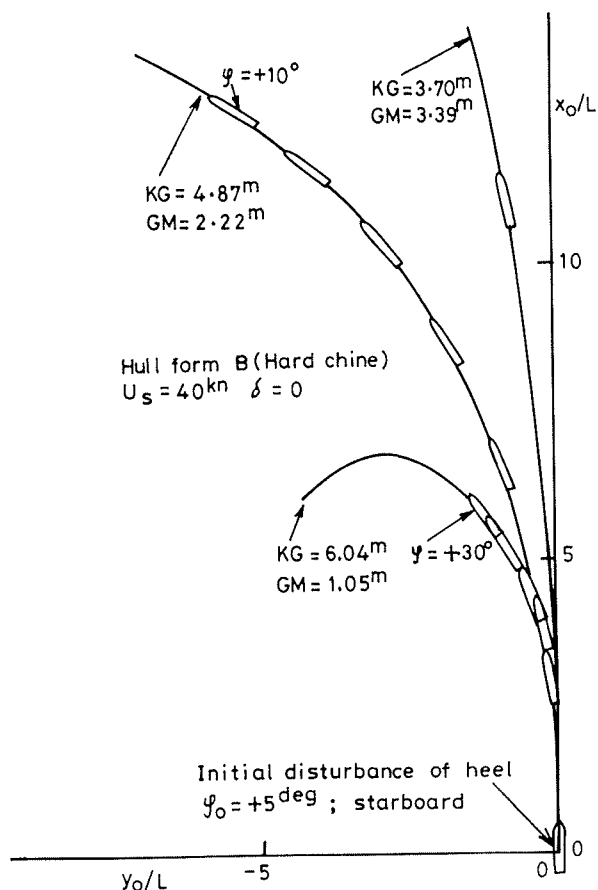


Fig. 16 Calculated trajectories of the hard chine type hull form B (Effect of vertical position of center of gravity)

considered to correspond to the hydrodynamic characteristics of "with and without spray strips", respectively. Therefore, effect of the spray strips on the roll induced instability can be calculated by varying the vertical position  $z_y$  of acting point of sway force. Results of calculation are shown in Figs. 17 and 18, where  $z_y/d$  shows considerable effect on manoeuvring motions. From these results, it is assured that apparent course stability is improved by the spray strips which show an effect to raise the vertical position of acting point of sway force. These calculated characteristics agree with the observed results of Suhrbier (Ref.6).

## 5. CONCLUSIONS

So called "roll induced instability" of high speed craft was investigated, based on the captive model test and on the simulation by use of thus obtained hydrodynamic coefficients.

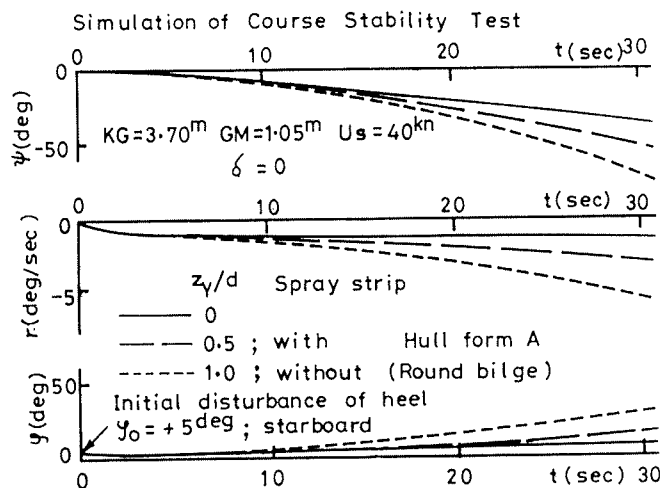


Fig. 17 Calculated manoeuvring motions of the round bilge type hull form A (Effect of spray strips)

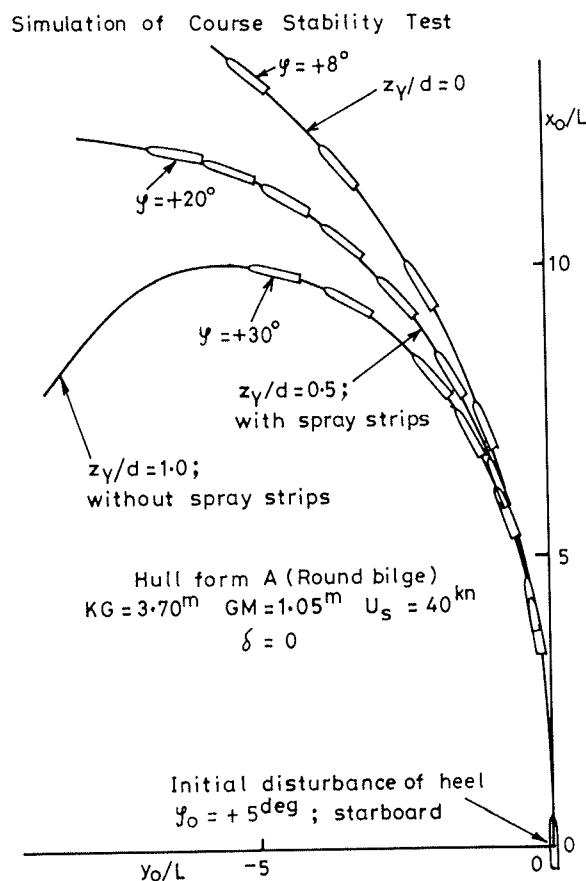


Fig. 18 Calculated trajectories of the round bilge type hull form A (Effect of spray strips)

The results of the study can be summarized as follows:

- (1) Tested two hull forms, i.e. a round bilge type and a hard chine type, show only slight difference in the characteristics of sway force and yaw moment due to heel angle, drifting angle, and rudder angle.
- (2) Variation in GM of the order of 0.5 meters, has a dominant effect on the phenomenon of the roll induced instability in comparison with the effect of the above mentioned difference in hull forms.
- (3) Apparent course stability is improved by the spray strips which show an effect to raise the vertical position of acting point of sway force induced by heel angle.

Since the roll induced instability is directly related to the safety of operation of high speed craft, such simulation study as the present one may be considered to be helpful for stability estimation in the initial design stage. Further study will be necessary on the simulation of sway-roll-yaw coupled motions by use of more accurate mathematical model. In such a model, it will be necessary to include non-linear terms due to  $\beta \cdot \phi$  and  $r \cdot \phi$ , together with variation of the vertical position  $z_y$  of acting point of sway force due to large roll angle.

#### ACKNOWLEDGEMENT

The authors wish to express their deep appreciation to Dr. Kyoji Watanabe, Former General Manager of Nagasaki Technical Institute of Mitsubishi Heavy Industries, Ltd. for his instructive discussion. The authors also wish to express their appreciation to the members of Nagasaki Experimental Tank, for their cooperation in carrying out this investigation.

## Discussion

M. Hirano (Akishima Laboratory, Mitsui Engineering and Shipbuilding Co., Ltd., Japan)

I congratulate the authors on this fine paper and would like to comment on the following points.

- (1) In general, due to changes in trim and sinkage and due to wave-making phenomenon, maneuvering hydrodynamic forces of high-speed ships are affected not a little by their advance speed. In Figs. 5 and 6, the experimental results of the maneuvering hydrodynamic forces are summarized with one straight line for each hull form, which implies that there would be no speed effects on the maneuvering hydrodynamic forces.

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Regarding this point, I feel that more data with respect to the effects of the advance speed would be needed in order to reach to such a conclusion as the above.

- (2) Regarding yaw moment  $N'(\beta)$ , Fig.6 shows negative sign of  $N'(\beta)$  for positive sign of  $\beta$ , while conventional displacement type ships generally have characteristics of positive  $N'(\beta)$  for positive  $\beta$ . Are the facts shown in Figs. 6 and 7 special features of which such semi-displacement type high-speed ships as those employed in this study are usually possessed?

#### Author's Reply

The authors wish to appreciate the dis-

cussions of Dr. Hirano. With regard to the first comment; observed difference of trim was less than 1.0% of ship length within the range of Froude number we tested, i.e. from 0.46 to 1.00. And we could not find speed effects on the non-dimensional manoeuvring derivatives.

However, as shown in the paper by Mr. Mueller-Graf and Prof. Schmiechen, some variations in hydrodynamic coefficients might have speed effects beyond Froude number 1.0.

With regard to the second comment; at the beginning of our captive model tests, we also felt it strange that variations in yaw moment  $N'$  versus drifting angle  $\beta$  were different from those of conventional displacement type ships, as Dr. Hirano suspected.

After careful examination of our experimental results, however, we confirmed that these characteristics are one of typical features of semi-displacement type high speed boats.

M. Schmiechen (VWS Berlin Model Basin, FRG)

The work done by our colleagues of MHI at Nagasaki is greatly appreciated. One point disturbing us is the fact, that two vehicles of so widely differing values of the metacentric height have been compared. In my opinion this difference is bound to blur all the other effects. A second point is that the values of most of the hydrodynamic quantities have been estimated only widely. I wonder whether the authors have carried out sensitivity tests. According to my experience this type of study is notoriously difficult if stability is marginal.

#### Author's Reply

We appreciate the discussions of Prof. M. Schmiechen. Regarding the first comment; we carried out this investigation under the specified condition. That is, from the practical viewpoint, the same KG was adopted for both hull forms of round bilge and hard chine crafts. Furthermore, we carried out simulations for the hard chine craft with the same GM as that of the round bilge one. As the results of the simulations, similar roll induced instabilities were obtained for both round bilge and hard chine crafts. And these qualitative characteristics have been confirmed by the free-running model tests.

Regarding the second comment; firstly we carried out simulations to clarify the effects of roll damping and transverse radius of gyration. And we found that they have negligibly small effects on the steady turning characteristics. We then carried out simulations for two different hull forms, for hard chine craft with various GM, and for round bilge with and without spray strips. These simulation studies are considered to be a sort of sensitivity

tests. Moreover, the free-running model tests confirmed the validity of the present mathematical model.

W.G. Price (Brunel University, UK)

My congratulations and gratitude are extended to the authors for their extremely competent experimentation and presentation of their data.

The authors in their mathematical modelling employ correctly coupled sway-yaw-roll equations and not separate the equations into coupled sway-yaw together with a single degree of freedom roll equation. From my experience it is in general impossible to derive their experimental findings using the latter simplified mathematical model and a totally coupled set of equations is essential. However, if the authors had adopted non-dimensional equations of the 'parmi' type using a suitable or modified length parameter then it is clearly seen that the coefficient most affected by speed is the term  $GM/U^2$  in the non-dimensional roll restoring expression. As  $U$  increases for fixed  $GM$  or  $GM$  decreases for fixed  $U$  the term  $GM/U^2$  decreases and from a linear stability analysis it can be shown that the ship model becomes unstable. In fact a region of stability based on  $GM$  and  $U$  can be derived and a simple stability criteria may be produced to demonstrate this. Naturally a more complicate mathematical model may be produced by including other possible dynamical influences (e.g. change in trim, displacement etc.) in the variation of the hydrodynamic derivatives and this enables further refinement and differentiation of the behaviour of the two ship models.

#### Author's Reply

Regarding to the comment by Prof. Price, we completely agree to his point. Correctly speaking,  $GM$  itself does not have dominant effect, but  $GM/U^2$  has. That is, the same hull with the same  $GM$  shows no instability at a low advance speed, and experiences instability at a higher speed. This is due to the fact that roll restoring moment provided by the initial  $GM$  is canceled by the hydrodynamic unstable moment which is proportional to the square of advance speed.

Incidentally we could not include the results of free running model test and simulation calculation showing the effect of advance speed on the roll-induced instability. In this context, we are very grateful to Prof. Price for giving us a chance to add an explanation to this point.

B. Mueller-Graf (VWS Berlin Model Basin, FRG)

I like to congratulate the authors on their very interesting paper concerning the stability of high speed craft underway. By means of simulation studies the authors

underline the dominant effect of the meta-centric height on the roll induced instability of semi-displacement craft also at high speeds. The computations confirm the present practice in designing high speed craft to overcome stability problems by selecting a high value of GM. On the basis of heeling tests the authors report that the tested hull forms despite the great differences in section shape, buttock curvature and length-beam ratio cause only slight differences in the characteristics of sway force and yawing moment. This seems to be very surprising and not in accordance with references [5,6].

In this paper, which is related to high speed craft, no figure shows results or derivatives represented versus speed. It must be assumed, that the authors are neglecting the speed only because they found out - on the basis of heel angle tests - that speed has no substantial influence on the sway force and yaw moment. This statement does not agree with the results of references [5,6] and with those obtained at VWS Berlin Model Basin in an investigation of the stability of semi-displacement crafts. We experienced, as reported in the paper before, a remarkable effect of speed on stability mainly at  $F_n > 0.8$ .

The conclusion that the apparent course stability is improved by spray rails which raise the vertical position of the centre of sway force should be underlined. But it must be pointed out that the amount of raise depends on the height of the rails above the waterline. At VWS spray rails which intersect the waterline or which are arranged close to it are preferred. In this case the height of the acting point of sway force is less than the half of the reported value.

In Fig.11 the vertical position of the acting point of sway force lies predominantly above DWL. In the simulation study the sway forces are assumed to act at the midpoint of draft. Can the authors explain the reason for departing from the test results?

#### Author's Reply

We appreciate the discussions and comments by Mr. Burkhard Mueller-Graf. For the first and second comments of his discussions, we are afraid he might have some misunderstanding. In references [5 and 6], the increase of heel angle is only measured. They never dealt with hydrodynamic forces and moments through experiments, and did not report any results which can be compared with ours. We only concluded that non-dimensionalized sway forces and yaw moments have similar characteristics for the tested two hull forms. We suppose it is partly due to appropriate non-dimensional form as shown by Eq.(1). We mentioned that there is no substantial dependency in non-dimensional sway forces and yaw moments, not sway forces and yaw moments themselves, on the basis of not only heel angle test but also oblique towing and rudder angle tests. We have already admit in the reply to Dr. Hirano's discussion that we expect Froude number dependency for  $Y'$  and  $N'$ , and that the reason why we get no substantial Froude number dependency up to  $F_n=1.0$  would be the fact that the variation of trim angle is relatively small.

Concerning the effect of spray strips, we appreciate your results for another arrangement of spray strips.

With respect to the fourth comment; as shown in Fig.11, vertical position of acting point due to roll is lowered with the increase of heel angle. The more that vertical position is lowered, the severer the roll induced instability becomes. If we assumed that hydrodynamic sway forces act above DWL by adopting the data within the range of small roll angle, we might underestimate the effects of sway force on the roll-induced instability. To obtain a stable result of simulation for a design which is unstable in reality is most dangerous and should be avoided.

*Session IIIa*

## Theoretical Calculations

*Chairmen*

Prof. Manley St. Denis  
U.S.A.

Dr. A. Yucel Odabasi  
The British Ship Research Association  
U.K.



# ON THE MAXIMUM AMPLITUDES IN NONLINEAR ROLLING

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## ABSTRACT

The rolling motion of a ship has been described by a nonlinear differential equation explicitly taking into account the nonlinearities in both the righting moment and damping. The latter is represented by a linear and a cubic term in the angular velocity and by a nonlinear term in the rolling angle. The equation has been solved by a perturbation method in the regions of the main resonance and the first ultraharmonic relative to a regular beam sea. The approximate analytical solutions allow expressions for the frequency response and moreover an estimate of the maximum amplitude forecastable as a function of the excitation intensity. These are obtained through algebraic equations from which approximate solutions can be given in an explicit form. In this manner, simple formulas are found, relating the excitation to the maximum rolling by means of the nonlinear damping coefficients. They constitute an improvement with respect to those till now used. A comparison with the results of numerical simulations shows a quite good agreement in a rather wide range of the involved parameters.

## NOMENCLATURE

$\phi$  = rolling angle with respect to the calm sea surface;  
 $t$  = time;  
 $I$  = mass moment of inertia including the added mass moment;

$D_{ij}$  = coefficients of the dissipative term;  
 $\Delta$  = displacement;  
 $\overline{GM}$  = transversal metacentric height;  
 $k_i$  = coefficients of the righting arm best-fit polynomial;  
 $E_w$  = amplitude of the heeling moment in regular waves;  
 $\Omega$  = angular frequency of wave excitation;  
 $\Omega_o$  = angular frequency of small oscillations;  
 $\phi_n$  = normalization angle;  
 $T_n$  = normalization time;  
 $x$  = adimensional rolling angle;  
 $\tau$  = adimensional time;  
 $\mu, \delta_1, \delta_2$  = adimensional damping coefficients;  
 $\alpha_i$  = adimensional coefficients of the righting moment best-fit polynomial;  
 $e_w$  = adimensional amplitude of the heeling moment in regular waves;  
 $\omega$  = adimensional angular frequency of wave excitation;  
 $\omega_o$  = adimensional angular frequency of small oscillations;

$C$	= amplitude of the resonant component in the response;
$\psi$	= phase of the resonant component;
$Q$	= amplitude of the non-resonant component in the response;
$x_m$	= maximum rolling amplitude in the steady-state oscillations;
$C_m$	= maximum amplitude of the resonant component.

## 1. INTRODUCTION

The study of the motion of a ship in the seaway represents a serious problem in naval architecture. The mathematical analysis of the phenomenon appears to be of particular complexity, due to the high number of involved parameters.

Upon a detailed examination of the available literature on the rolling motion, one can observe the following:

- 1) absolute inadequacy of model and fullscale experiments;
- 2) general uncertainty in the values of the coefficients representing the physical effects in the problem.

As far as the damping is concerned, a detailed analysis of the abovementioned situations has been carried out by the authors [1].

Most of the theoretical models used imply very restricted hypotheses so that quantitative predictions are obtained with difficulty. One has therefore the feeling that the obtained results are not strictly correlated with the real phenomenon. An example of this situation is the use of the so-called equivalent linearization technique, in treating the rolling equation. This can even bring about an accordance between theoretical predictions and experimental results but it conceals both the physical aspects and the correct understanding of the phenomenon.

On the other hand, only through a simplified mathematical model, which describes the motion without overlooking the more significant aspects, is it possible to reach synthetic results which are readily and easily applicable to practical use. The objective reality which complicates the solution of the problem is contained in the following contradiction:

- 1) a theoretical model that allows a sufficiently detailed description that

considers all the possible particularities;

- 2) but which is sufficiently easy to solve and from which a useful prediction could be obtained:
  - for deepening the research work;
  - for a more effective guide in experiments;
  - for immediate applicability, giving practical indications for the ultimate user such as the captain of the ship.

It is clear that the increased facilities in analytical and numerical computations have progressively toned down this situation during the last few years. In fact, it is now possible to obtain analytical solutions with different orders of approximation as well as exact numerical solutions for a system of coupled nonlinear equations.

Without denying the importance of the exact methods, one can observe that, in general, these are particularly suitable for checking the results and for the study of peculiar situations. Furthermore, it is evident that these methods are extremely awkward in obtaining a solution expressed in an analytical form i.e. by explicit formulas. One can realize the importance of this subject by the amount of work devoted to it in the past few years.

Among the many approximate analytical methods available, we have herewith chosen a perturbation method for the study of the rolling equation. The problem of the convergence of the perturbation series shall not be deepened also because satisfactory solutions can be obtained already from the first terms.

As will be shown in this paper, a coherent application of perturbation methods allows the handling of strong nonlinearities of the rolling equation and supplies rather simple analytical solutions which coincide with the exact numerical solutions up to relevant oscillation amplitudes.

A rational way of investigating the ship motion must proceed extensively, without leaving behind the deepening of particular aspects of the phenomenon. To this effect, an adequate stochastic analysis does not exist up to now for the rolling motion. According to the point of view of the authors, this is due to the fact that the rolling features in a regular sea are, to a certain extent, still unknown. This is true for both the steady-state and the transient oscillations. The ship's transient behaviour, the danger of which has been evidenced by several authors [2-4], is in fact still

unexplored.

The aim of the present paper is that of analyzing in detail the resonance phenomena which appear in nonlinear rolling in the case of regular beam seas. The analytical solutions describing the steady-state behaviour of the ship are obtained in Section 3 by the use of the Bogoliubov-Krylov-Mitropolsky perturbation method. As a consequence, simple formulas for maximum rolling amplitudes in terms of excitation intensity are presented in Section 5.

These formulas, on one hand, permit the understanding of the dynamic characteristics of a ship and on the other, constitute a step towards new experimental work in this field. The agreement between analytical and numerical solutions is extremely good, thus justifying the validity and applicability of perturbation methods in rolling analysis.

## 2. STATEMENT OF THE PROBLEM

This study is devoted to the nonlinear effects in rolling motion. The rolling shall be considered as an independent oscillation. In this hypothesis, for a ship in a regular beam sea, the differential equation of motion can be written as follows [5-8]

$$I\ddot{\phi} + \sum_{ij} D_{ij} \dot{\phi}^i \dot{\phi}^j + \Delta \sum_{ik} E_{ik} \phi^i \phi^k = E_w \cos \Omega t \quad (1)$$

Equation (1) has been expressed in terms of absolute angle to avoid useless complications when considering motion in regular seas from any direction or, in general, in irregular seas. Moreover, the inertial mass moment has been considered constant and the righting arm time independent. In particular, in the present analysis, a damping model shall be assumed, given by

$$D(\phi, \dot{\phi}) = (D_{01} + D_{21} \dot{\phi}^2) \dot{\phi} + D_{03} \dot{\phi}^3,$$

where an angular dependence in the linear term is also included [1]. Such a dependence is suggested when one considers the lack of rotational symmetry of the hull with respect to the rolling axis.

Equation (1) can be expressed in adimensional terms through an introduction of suitable angle and time scales,  $\phi_n$  and  $T_n$  respectively, as follows

$$\ddot{x} + (2\mu + \delta_1 x^2) \dot{x} + \delta_2 \dot{x}^3 + \omega_o^2 x + \sum_{ik} \alpha_{ik} x^i x^k = e_w \cos \omega \tau. \quad (2)$$

Here, for simplicity the differentiation is still indicated by a dot. In equation (2) the following adimensional parameters are introduced

$$x = \phi / \phi_n,$$

$$\tau = t / T_n,$$

$$\omega_o^2 = \Delta k_1 T_n^2 / I,$$

$$\omega = \Omega T_n,$$

$$\alpha_i = \Delta k_i T_n^2 \phi_n^{i-1} / I \quad \text{for } i \geq 3,$$

$$\mu = D_{01} T_n / 2I,$$

$$\delta_1 = D_{21} T_n^2 \phi_n^2 / I,$$

$$\delta_2 = D_{03} T_n^2 \phi_n^2 / I T_n,$$

$$e_w = E_w T_n^2 / I \phi_n.$$

As a convenient choice for the angle scale one assumes  $\phi_n$  coinciding with the abscissa which corresponds to the maximum of the righting arm curve. For the time scale one chooses  $T_n = 1/\Omega_o$  where  $\Omega_o = (\Delta k_1 / I)^{1/2}$  is the angular frequency of the small free oscillations and  $k_1 = GM$ .

Equation (2) represents a sufficiently general model for the study of the rolling motion as the simplifying hypotheses are verified at least for moderate rolling amplitudes and for intermediate frequencies. This allows an investigation devoted to analytical results of immediate practical use and therefore is best suited to verify the validity of the perturbation methods through a comparison with the exact numerical results.

The theoretical analysis of equation (2) by a perturbation method, can be carried out to any approximation order irrespective of the degree of complexity. However, a quantitatively extensive analysis will not be performed in the present paper in order to avoid the burdening by mathematical aspects and to allow the achievement of practical results. We shall see for physical situations in which a ship operates, how theoretical results, which settle appropriately with the exact numerical predictions, can be obtained already from a first order analysis. It is clear, however, that a higher order analysis gives way to an adjustment of the results for rolling ampli-

tudes which are ever increasing.

The system described by equation (2) possesses, apart from the synchronism, one ultraharmonic and one subharmonic resonance, respectively for  $\omega \approx \omega_0/3$  and  $\omega \approx 3\omega_0$ . This instance is due to the fact that only the cubic term of the righting moment is relevant to the calculations at this order. For this reason, the two abovementioned resonances are said to be the predominant ones.

Consequently, without causing any loss of generality, we shall further simplify our mathematical model. All theoretical results and the numerical computations, will be referred to the equation expressed in the form

$$\ddot{x} + (2\mu + \delta_1 x^2) \dot{x} + \delta_2 \dot{x}^3 + \omega_0^2 x + \alpha_3 x^3 = e_w \cos \omega \tau, \quad (3)$$

where the righting term is given by the fitting cubic relation.

The approximate analytical solutions of equation (3) are different in the various frequency regions [7,8]. For the main resonance, we have

$$x(\tau) = C \cos(\omega \tau + \psi), \quad (4)$$

whilst for the other two resonances, the solutions can be expressed in the same form and i.e.,

$$x(\tau) = C \cos(n\omega \tau + \psi) + Q \cos \omega \tau, \quad (5)$$

where

$$Q = e_w (\omega_0^2 - \omega^2)^{-1}.$$

and  $n=3$  for the ultraharmonic resonance and  $n=1/3$  for the subharmonic resonance, respectively. The solution in the intermediate frequency regions i.e. those out of resonance, is still given by (5) with  $C=0$ .

The explicit expressions for the quantities  $C$  and  $\psi$ , amplitude and phase of the resonant component of the response, are given in the following section. The steady-state solutions herewith presented concern the main and the ultraharmonic resonances only. The subharmonic resonance will not be dealt with in the present paper although it is equally important in problems concerning safety from capsizing [4]. The former has however a completely different physical origin with respect to the other two resonances, due to the fact that it can take place only after exceeding a well determined excitation threshold.

### 3. STEADY-STATE SOLUTION WITH THE NUMERICAL COMPARISON

The amplitude and the phase of the resonant component in the steady-state oscillations, for the main and the first ultraharmonic resonance, are given below. The analytical solutions have been computed with the asymptotic method of Bogoliubov-Krylov-Mitropolsky [9]. Major details on mathematical aspects are found in [7,8].

General expressions for the various functions used throughout the text are given in the Appendix.

1) Main Resonance, i.e.  $\omega \approx \omega_0$  and  $n=1$ .

$$g_0 C^6 + g_4 C^4 + g_5 C^2 - e_w^2 = 0, \quad (6)$$

$$\tan \psi = - \frac{c_4 + c_3 C^2}{D_1 + \frac{3}{4} \alpha_3 C^2}.$$

2) Ultraharmonic Resonance, i.e.  $\omega \approx \omega_0/3$  and  $n=3$ .

$$g_0 C^6 + g_1 C^4 + g_2 C^2 - g_3 Q^6 = 0, \quad (7)$$

$$\tan \psi = - \frac{c_0 (c_1 + c_3 C^2) - c_2 (Q_3 + \frac{3}{4} \alpha_3 C^2)}{c_0 (Q_3 + \frac{3}{4} \alpha_3 C^2) + c_2 (c_1 + c_3 C^2)}.$$

A comparison between the approximate analytical solutions and the exact numerical solutions has been made in order to evaluate the accuracy and the frequency range of validity. The equation of motion has been solved numerically with the Runge-Kutta-Gill method and the results obtained have been directed towards the steady-state oscillation response.

Whilst dealing numerically with equation (3) it can be immediately observed that the number of parameters involved is too large to be altered independently so as to obtain all the cases of practical interest. In particular, in comparing analytical with numerical results, we have considered a heavy situation beyond any realistic possibility. The choice of parameters has been done in order to represent a ship with very low damping in rough sea conditions. Two different values have furthermore been chosen for the nonlinearity of the righting arm, a negative one and a positive one, such to correspond to load conditions with different dynamic behaviour. The examined

situation allows a good test for the validity of the theory and also supplies a reference point for other common situations.

The maximum rolling amplitude  $x_m$  is shown in Figg. 1 and 2 as a function of the relative frequency  $\omega/\omega_0$ . The graphs quoted represent both the numerical and the corresponding analytical results, relative to the examined load conditions.

As one can see, the results obtained by means of the two different methods are in excellent agreement. From the drawings, the possibility of a theoretical reproduction of the frequency response curves emerges quite clearly. One can observe bendings, discontinuities and frequency shifts, peculiarly characteristic of the behaviour of a nonlinear system. In the main resonance region, the analytically predicted response curves are practically coincident with the numerical ones, within the limits of the graphic representation. A similar situation is verified also in the subharmonic

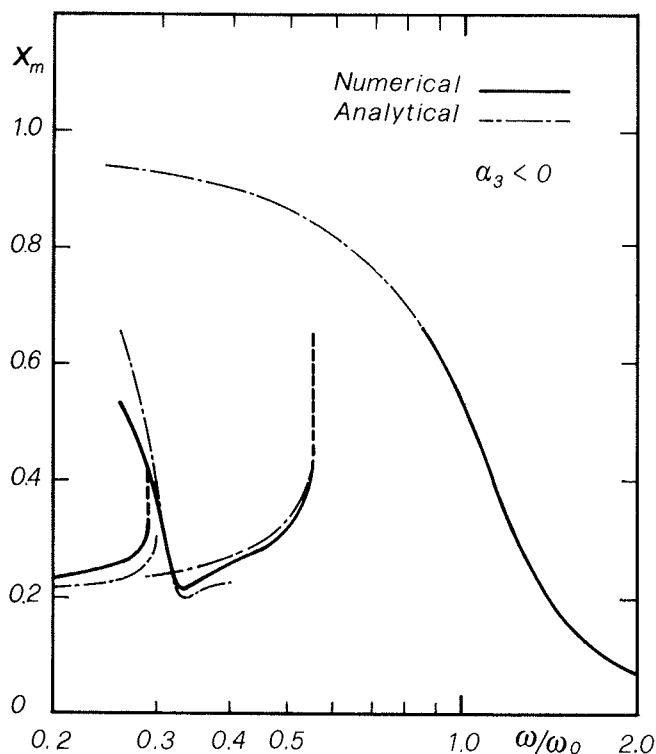


Fig. 1 - Maximum rolling amplitude  $x_m$  of the steady-state oscillations versus relative driving frequency  $\omega/\omega_0$ . The curves give a comparison between numerical and analytical computations. The following values have been used:  $\omega_0=1$ ,  $\alpha_3=-1.75$ ,  $\mu=0.005$ ,  $\delta_1=0.01$ ,  $\delta_2=0.01$ ,  $e_w=0.2$ .

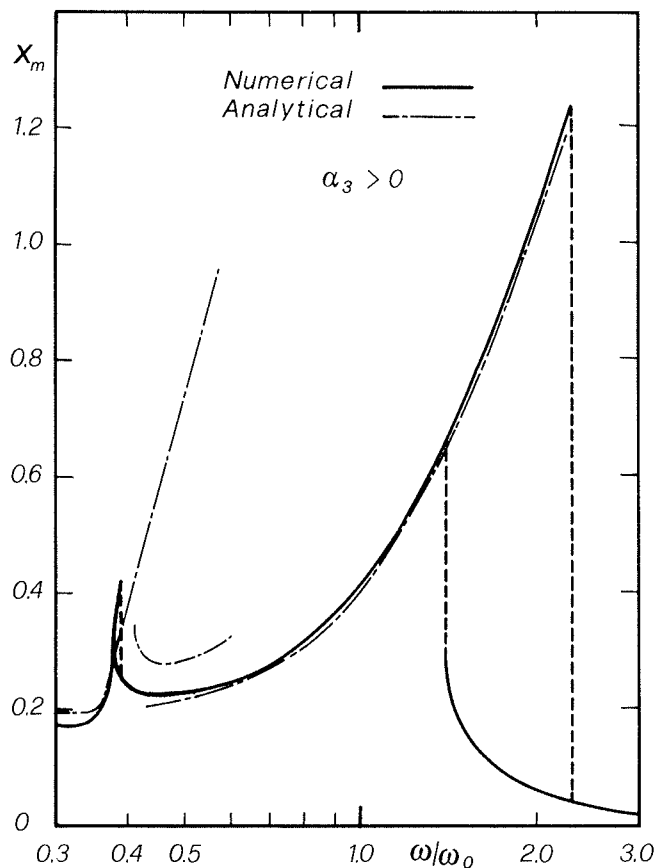


Fig. 2 - Maximum rolling amplitude  $x_m$  of the steady-state oscillations versus relative driving frequency  $\omega/\omega_0$ . The curves give a comparison between numerical and analytical computations. The following values have been used:  $\omega_0=1$ ,  $\alpha_3=4$ ,  $\mu=0.005$ ,  $\delta_1=0.01$ ,  $\delta_2=0.01$ ,  $e_w=0.2$ .

resonance region but is not shown here. Moreover, the fact that the agreement in the response goes up to and beyond  $x_m \approx 1$  appears to be quite surprising. These results confirm what has been said initially on the limits of the perturbation methods for an analytical investigation of the nonlinear rolling equation.

A situation like this, therefore, suggests a vast field of possibilities in applying the perturbation methods. The analytical results therefore represent the starting point for a deeper insight into the behaviour of a rolling ship. Moreover, they suggest a clue in obtaining approximate formulas that allow simple and immediate results, taking into account all the nonlinear effects correctly.

#### 4. MAXIMUM ROLLING AMPLITUDES

We have noticed in the previous section the validity of the formulas obtained by the perturbation method for reproducing, step by step, the frequency response curves found through the numerical integration of the rolling equation. However, their practical validity can be measured by an accurate prediction of the maximum oscillation amplitude for a given excitation intensity. This fact suggests a detailed investigation of the predictions obtained through equations (6) and (7) regarding maximum amplitudes.

The maximum rolling amplitude is shown in Figg. 3-6 as a function of the excitation intensity for the main and ultraharmonic resonance relatively to two different load conditions. The considered situations are more realistic in this case as far as the damping effect is concerned. The latter has been chosen to reproduce both ships without damping systems and ships with bilge

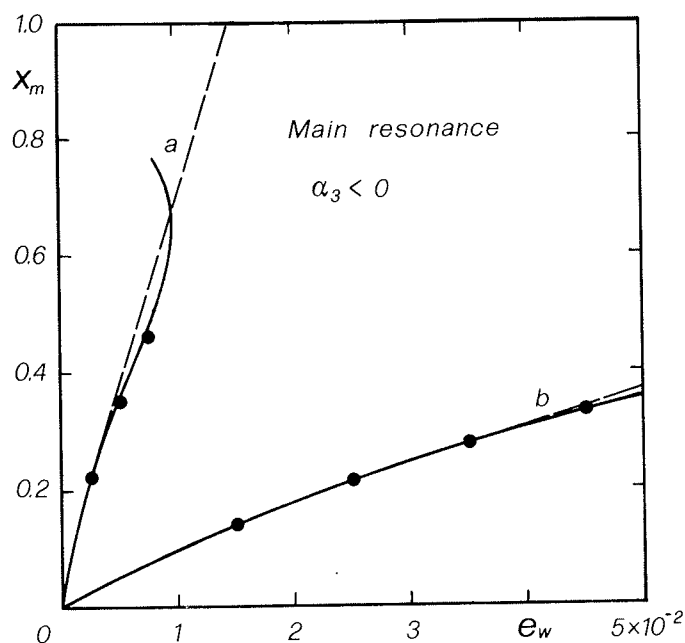


Fig. 3 - Maximum rolling amplitude  $x_m$  of the steady-state oscillations versus excitation intensity  $e_w$  in the main resonance region. Solid lines refer to analytical results and dashed lines to approximate formula (9). The points are the numerical solutions. The following values have been used: a)  $\omega_0=1$ ,  $\alpha_3=-1.75$ ,  $\mu=0.005$ ,  $\delta_1=0.05$ ,  $\delta_2=0.05$ ; b)  $\omega_0=1$ ,  $\alpha_3=-1.75$ ,  $\mu=0.05$ ,  $\delta_1=0.5$ ,  $\delta_2=0.5$ .

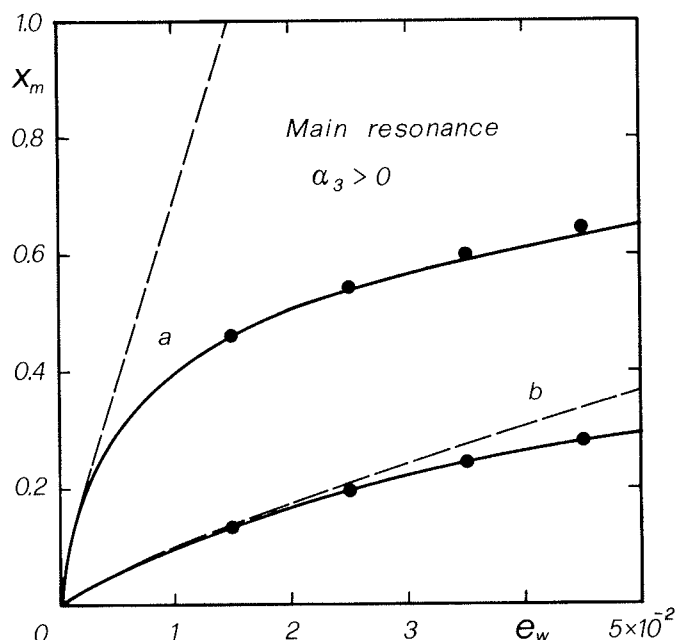


Fig. 4 - Maximum rolling amplitude  $x_m$  of the steady-state oscillations versus excitation intensity  $e_w$  in the main resonance region. Solid lines refer to analytical results and dashed lines to approximate formula (9). The points are the numerical solutions. The following values have been used: a)  $\omega_0=1$ ,  $\alpha_3=4$ ,  $\mu=0.005$ ,  $\delta_1=0.05$ ,  $\delta_2=0.05$ ; b)  $\omega_0=1$ ,  $\alpha_3=4$ ,  $\mu=0.05$ ,  $\delta_1=0.5$ ,  $\delta_2=0.5$ .

keels and other damping devices. These correspond approximately to the extreme situations of Fig. 1 of paper [10]. A few points obtained through the numerical solution of the equation of motion are shown as a comparison in the drawings. There exists an excellent agreement in all the cases considered which goes right up to extreme conditions, i.e. up to large oscillation amplitudes.

#### 5. APPROXIMATE FORMULAS FOR MAXIMUM ROLLING AMPLITUDES

The agreement between the forecasts supplied by the perturbation method and the numerical results in predicting maximum rolling amplitudes, have induced us to consider the possibility of getting simple formulas which have the same predicting capabilities. These formulas should include all the significant parameters of the system and in particular should explicitly take into account all the existing nonlinearities.

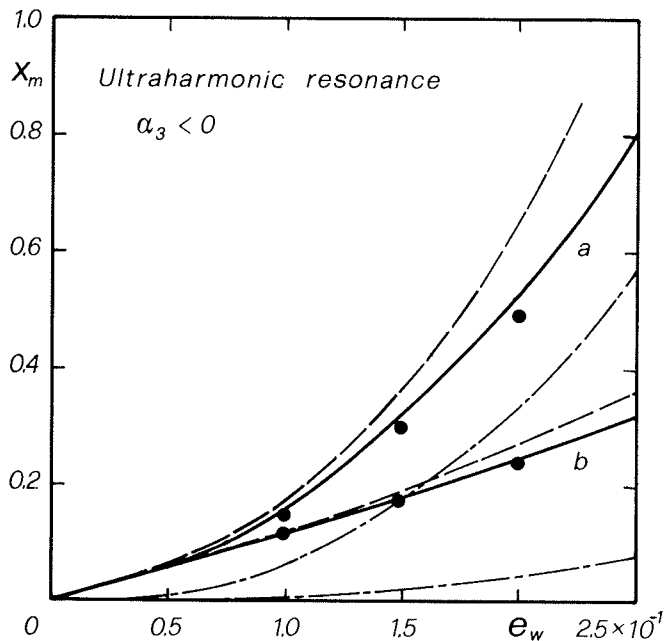


Fig. 5 - Maximum rolling amplitude  $x_m$  of the steady-state oscillations versus excitation intensity  $e_w$  in the ultraharmonic resonance region. Solid lines refer to analytical results and dashed lines to approximate formula (11). The points are the numerical solutions. As a comparison the maximum amplitude  $C$  of the resonant component in the oscillation has also been reported (dash-dot lines). The following values have been used: a)  $\omega_0=1$ ,  $\alpha_3=-1.75$ ,  $\mu=0.005$ ,  $\delta_1=0.05$ ,  $\delta_2=0.05$ ; b)  $\omega_0=1$ ,  $\alpha_3=-1.75$ ,  $\mu=0.05$ ,  $\delta_1=0.5$ ,  $\delta_2=0.5$ .

A perturbation expansion in equations (6) and (7) has been made, satisfying the abovementioned conditions. The obtained results, for the maximum amplitudes, can be synthesized as follows.

#### 1) Main Resonance

$$C_m \approx C^* \left( 1 - \frac{\bar{c}_3 C^{*2}}{\bar{c}_4 + 3\bar{c}_3 C^{*2}} \right), \quad (8)$$

where

$$C^* = e_w / 2\mu\omega_0, \quad \bar{c}_i = c_i(\omega_0).$$

In the limits of the approximations adopted, the first term of equation (8) coincides with the solution predictable in a linear investigation of the ship's motion, whereas the second gives way to a correction where the nonlinear damping is involved. To this

order the nonlinear term in the righting moment does not appear explicitly. The reasons for this procedure will be justified below.

As the rolling response of the ship in synchronism is given by (4), one can infer that equation (8) supplies also an approximate relation for the maximum rolling amplitude, i.e.

$$x_m \approx \frac{e_w}{2\mu\omega_0} \left[ 1 - \frac{(\delta_1 + 3\delta_2\omega_0^2)e_w^2}{32\mu^3\omega_0^2 + 3(\delta_1 + 3\delta_2\omega_0^2)e_w^2} \right]. \quad (9)$$

#### 2) Ultraharmonic Resonance

$$C_m \approx \frac{A}{\bar{c}_1} \left( 1 - \frac{\bar{c}_3 A^2}{\bar{c}_1^3 + 3\bar{c}_3 A^2} \right), \quad (10)$$

where

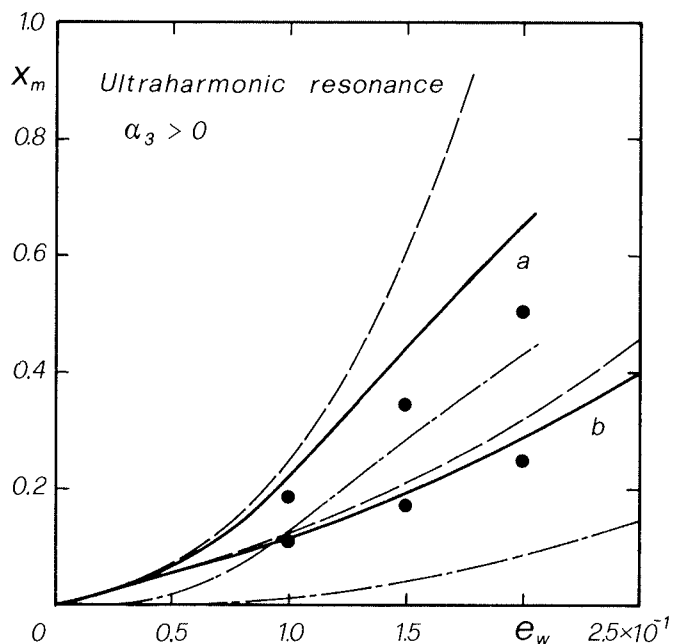


Fig. 6 - Maximum rolling amplitude  $x_m$  of the steady-state oscillations versus excitation intensity  $e_w$  in the ultraharmonic resonance region. Solid lines refer to analytical results and dashed lines to approximate formula (11). The points are the numerical solutions. As a comparison the maximum amplitude  $C$  of the resonant component in the oscillation has also been reported (dash-dot lines). The following values have been used: a)  $\omega_0=1$ ,  $\alpha_3=4$ ,  $\mu=0.005$ ,  $\delta_1=0.05$ ,  $\delta_2=0.05$ ; b)  $\omega_0=1$ ,  $\alpha_3=4$ ,  $\mu=0.05$ ,  $\delta_1=0.5$ ,  $\delta_2=0.5$ .

$$A = \omega_o^3 (\bar{c}_o^2 + \bar{c}_2^2)^{1/2}, \quad \bar{c}_i = c_i (\omega_o/3).$$

Here equation (10) includes, apart from the nonlinearity of the damping effects, also the nonlinear coefficient of the righting moment. In particular, for small amplitude oscillations, the following expression can be written

$$C_m^* \approx \frac{243 e_w^3 \left[ 9 \alpha_3^2 + \left( \delta_1 - \frac{1}{9} \delta_2 \omega_o^2 \right)^2 \omega_o^2 \right]^{1/2}}{16 \omega_o^3 \left[ 256 \mu \omega_o^4 + 81 \left( \delta_1 + \frac{1}{3} \delta_2 \omega_o^2 \right)^2 e_w^2 \right]}.$$

It must be noted that in the ultraharmonic resonance regions, the response of the ship is more complex as given by equation (5). As regards the maximum rolling amplitude, this can reasonably be estimated through relation.

$$x_m \approx C_m^* + 9 e_w / 8 \omega_o^2. \quad (11)$$

The estimations obtained with formulas (9) and (11) are quoted in Fig. 3-6. The agreement on the whole is more than reasonable if compared with the simplicity of the formulas.

It must be observed that in the drawings, for the ultraharmonic resonance, the computations of the response have been brought to excitation intensities higher than those involved in the main resonance. This is due to the fact that the ultraharmonic resonance response is of lower amplitude than the one of synchronism, i.e. the ship can withstand heavier seas in this frequency range.

## CONCLUSIONS

The main aim in investigating the rolling behaviour of a ship is that of obtaining the maximum information concerning the stability and thus the safety in the exercise of the various situations. On the other hand, stability and boundness of motion are radically different concepts although linked together. This fact, already known in other fields of applied science, has only recently entered into the theoretical knowledge of the naval architect thanks to the fundamental work done by Odabaşı [11] who was the first to underline this problem.

After this fundamental consideration, this paper falls into the research field concerning the boundness of rolling motion. It deals with a particular situation which is extremely significant. We have examined the

rolling motion due to a deterministic excitation relative to a wide range of frequencies. This shows that a whole series of resonance phenomena is predictable for the ship. These resonances, originated by the strong nonlinearity of the system, are predominantly the synchronism, the ultraharmonic 3 and the subharmonic 1/3. We have then considered the quantitative investigation of the phenomenon which gave, apart from the frequency response curves, also approximate relations for the maximum rolling amplitude in the main and ultraharmonic resonance region.

Regarding synchronism, formula (9) appears to be of immediate practical interest. It represents an improvement of the classical formulas till now proposed, because it takes explicitly into account both the linear and nonlinear damping terms. The term representing the righting arm nonlinearity does not on the other hand appear in this expression. As one can see from Figs. 3 and 4, this fact does not invalidate the theoretical predictions, not even in cases where the nonlinearity is rather strong.

The nonlinearity of the righting arm gives origin to a bending of the frequency response curve, causing the system to be more "sensitive" in a frequency interval of wider amplitude than that of the linear case and giving way to dangerous jump phenomena. Consequently, this causes a frequency shift in the maximum but does not strongly affect the corresponding maximum amplitude. The latter, is essentially linked to the energy balance in the system i.e., between the energy supply of the excitation and the energy loss due to damping.

Since, for a given excitation intensity, the maximum roll amplitude depends essentially on the damping coefficients, the problem of a realistic evaluation of the damping effects becomes particularly serious. However, such an evaluation presents a series of difficulties for both the choice of the damping model and the separation of the different damping coefficients. As observed previously [1], it is very difficult to separate the different nonlinear contributions through both the extinction decay experiments and the synchronism response analysis. This problem could, perhaps, be overcome by a quantitative investigation in other nonlinear resonance regions, where the damping terms play their role with different weights.

Formula (11) appears to be particularly interesting for model test experiments. It supplies the maximum rolling amplitude for a given excitation intensity in the ultraharmonic frequency resonance region. The term



representing the righting arm nonlinearity is explicitly present in this equation because this nonlinearity is directly responsible for the "strong" energetic coupling. However, one can observe that the ultraharmonic resonance is not as important as the synchronism and subharmonic resonance. Although the rolling amplitudes could be relevant in this region, the frequencies concerned are essentially limited to ships of stiff behaviour. In terms of safety problems, the ultraharmonic resonance should not therefore present particular risks.

#### ACKNOWLEDGEMENTS

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#### APPENDIX

$$c_0 = \frac{\alpha^3}{n+1},$$

$$c_1 = n\omega \left[ 2\mu + \frac{1}{2}(\delta_1 + 3\delta_2\omega^2)Q^2 \right],$$

$$c_2 = \frac{n+1}{16}\omega(\delta_1 - \delta_2\omega^2),$$

$$c_3 = \frac{1}{4}n\omega(\delta_1 + 3\delta_2n^2\omega^2),$$

$$c_4 = 2\mu n\omega,$$

$$c_5 = \frac{1}{2}n\omega(\delta_1 + 3\delta_2\omega^2),$$

$$D_n = \omega_n^2 - n^2\omega^2,$$

$$Q_n = D_n + \frac{3}{2}\alpha_3 Q^2,$$

$$g_0 = \frac{9}{16}\alpha_3^2 + c_3^2,$$

$$g_1 = \frac{3}{2}\alpha_3 Q_n + 2c_1 c_3,$$

$$g_2 = Q_n^2 + c_1^2,$$

$$g_3 = c_0^2 + c_2^2,$$

$$g_4 = \frac{3}{2}\alpha_3 D_n + 2c_3 c_4,$$

$$g_5 = D_n^2 + c_4^2.$$

## Discussion

R. Barr (Hydronautics Incorporated, USA)

The authors have presented an interesting treatment of maximum rolling motions, several questions are caused by the approach and the conclusions.

1. The authors base their work on equation(3) which does not consider either:

- a) A quadratic damping term, which is clearly indicated by many data (see, for example, the paper of Dr. Ikeda et al. for this Conference)
- b) Any explicit influence of roll angle on wave exciting forces

while these omissions may not be critical to the results they seem surprising in view of the authors statement of the need for

"A theoretical model that allows a sufficiently detailed description that considers all the possible particularities."

2. In the conclusions the authors state: "The non-linearity of the righting arm ... causes a frequency shift in the maximum but does not strongly affect the corresponding maximum amplitude." Figures 3 and 4, for the case of low roll damping (curves a) show a significant apparent effect on  $a_3$ , the nonlinear righting arm term, curve a of figure 3 is the only curve of figures 3 - 6 which shows a highly danger.

Author's Reply

The authors would like to thank Dr. Barr for his kind remark and his comments.

In his first question Dr. Barr refers to results which have already been published (see references 1-11). He quoted only the first part of the contradiction, presented in the introduction of our paper, and as such it is unclear as both parts are complementary to each other.

The non-linear equation considered in this paper constitutes only the first step towards the solution of the roll motion problem (see reference 7). Various non-linear damping models have been considered by us in the past (see reference 1), where we showed that they fit equally well experimental data from roll decays.

Regarding the second question we would suggest that one should take into account the physical implications related to the non-linearity of both the righting moment and the damping. The non-linearity of the system implies a resonant frequency shift directly related to motion amplitude. On the other hand, the amplitude is linked to the energy balance between excitation and damping.

The restoring and the damping effects can be considered independently for small oscillations while for larger amplitudes their interaction becomes increasingly important. This fact clearly emerges in Fig. 3-4, where the maximum rolling amplitudes which were obtained using the same damping, are different only for large amplitudes.

Furthermore, the dangerous situation of curve a in Fig.3 is due to the instability of the numerical solution for large roll amplitudes.

# ON THE ROLE OF ENCOUNTER FREQUENCY IN THE CAPSIZING OF SHIPS

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## ABSTRACT

Rather than investigate the actual process of capsizing, with its attendant non-linear phenomena, an attempt is made to predict when capsizing can become a possibility. Employing a semi-empirical theory and restricting the investigation to sinusoidal waves it is suggested that a ship which is subjected simultaneously both to linear rudder-fixed dynamic instability and to linear resonance is at risk. This is sometimes possible and results are given for particular ships.

Some brief comments are given on the standing of the theory and on the methods by which it could be made more rigorous.

## NOTATION

### Symbols

$A(\omega_e)$	Matrix of virtual mass and inertia
B	Beam of ship
$B(\omega_e)$	Matrix of hydrodynamic damping
c	Wave propagation speed
Fr	Froude number
GM	Metacentric height
g	Weight per unit mass
k	Wave number
l	Length of ship
$p(t)$	Column vector of generalised principal coordinates for antisymmetric motion of dry hull
U	Steady forward speed
$\delta$	Dimensionless wave frequency $\{ = \omega(l/g)^{1/2} \}$

$\delta_e$	Dimensionless encounter frequency $\{ = \omega_e(l/g)^{1/2} \}$
$\lambda$	Wave length; trial stability index
$\chi$	Heading angle ( $\chi=180^\circ$ for a head sea)
$\omega$	Wave frequency
$\omega_e$	Encounter frequency.

## 1. INTRODUCTION

By far the most common approach adopted in studying ship capsizing seeks, at least implicitly, to formulate a mathematical description of the process of capsize and to examine the significance of various relevant parameters in the resulting non-linear theory. There are many variants of this approach, two of the most common being

- the quasi-static analysis involving the so-called GZ-curve and
- study of the non-linear uncoupled rolling motion.

This is a very natural method of approach, but it is also an extremely difficult one. Although it dominates the literature [1] it has hitherto required the making of assumptions which are of such dubious standing that accurate predictions cannot be made with conviction. Nothing like a reliable and persuasive analysis has appeared yet despite a huge investment of effort and there is a natural tendency to fall back on drawing up fresh theories *ad hoc* in the hope that, eventually, one will be found that survives critical examination, theoretically and experimentally.

The alternative to this approach seems to be unusual in naval architecture. It is not to discuss capsizing *per se* but, rather,

to seek the condition under which the ship becomes uncontrollable while it is executing large rolling motions. It is a main feature of this second method of investigation that there is no necessity to describe the large motions themselves - only when they are likely to occur. This means that linear theory becomes defensible; being "uncontrollable" then implies linear instability and "executing large rolling motions" implies being resonant.

If it is fair to draw the analogy with aircraft flutter, we shall try to ascertain when it is that flutter can occur, using a linear theory - a familiar approach in aeronautics. The actual disintegration of an aeroplane as a result of excessive flutter cannot possibly be described in terms of linear theory and, indeed, is not normally discussed in aeronautics. Likewise, we shall not discuss the actual process of ship capsizing.

It is, of course, easy to speak of linear theory as if it is essentially simple and perfectly straightforward. In fact it does raise some basic questions which must eventually be answered unequivocally. While there do remain areas of debate, however, it is quite clear that startlingly accurate conclusions can be reached when linear theory is invoked using apparently defensible assumptions[2]. Thus it is found that, apparently, for dynamic instability to be a possibility:

- forward speed and encounter frequency are both of the essence
- a high forward speed in a following sea with low frequency of encounter is particularly dangerous
- hull shape is an important factor and, in particular, for small trawler-like forms a deep square stern has an essentially destabilising effect
- it is possible for certain types of hull to experience a combination of resonance and instability and it is those same types that appear to be the most liable to capsize in practice.

So successful does this approach employing linear dynamics appear to be, the writers believe that the full implications of linear theory should be investigated as a matter of urgency. It is this standpoint that is adopted here.

## 2. SOME PRELIMINARY COMMENTS ON LINEAR THEORY

Such has been the satisfactory outcome of every one of the preliminary linear investigations, it might be thought natural to formalise the computational techniques without further ado. It could be argued that better hydrodynamic and structural data are bound to become available and so a gradual process of refinement will be witnessed. This is not the view of the writers, however, because it will require guesswork.

It must be remembered that linear ship dynamics is by no means cut and dried. Empiricism is often the order of the day and so it now seems essential to conduct a searching re-examination of linear theory. This is now in hand and it already seems to have raised a number of questions that do not seem to have attracted attention before. It has to be admitted, though, that these questions are not easy to discuss in a single paper of strictly limited length.

In this paper, we shall first briefly recall the theory that is employed and show that encounter frequency plays a dominant role in the analysis. Following a technique described previously[2], results are presented for a transom stern trawler, a Todd Series 60 model and for a level-keel loaded and ballasted coaster. It is found that they are most encouraging. These results are admittedly assailable and they have been criticised on three distinct accounts:

- (a) that no attempt is made to describe the process of actual capsize is held to be a basic weakness of the approach;
- (b) since actual capsize is governed by non-linear equations it is possible that some crucially important behaviour is altogether missed by a linear analysis;
- (c) the mathematical standing of the approach employed is questioned.

Of these three types of objection, (a) seems to be little more than an injunction not to try what we set out to do. If an answer is needed, we suggest that it is supplied by the results we have obtained.

As regards (b), we accept the theoretical possibility that something vital is missed. But in the apparent total absence of any solid evidence on the point we can do no more than keep open minds.

It is the objection (c) that is important and one can readily show that it brings us back once more to the role of encounter frequency in the theory. Although those who have (quite rightly) raised this matter have suggested alternative approaches, these too fail to meet all the mathematical requirements of linear theory. The difficulty will be explained in simple terms in this paper, and a possible way out of it will be suggested.

## 3. HEADING ANGLE

Encounter frequency  $\omega_e$  appears to play a dominant role in ship stability and, as we have already noted, a low value of  $\omega_e$  and high speed  $U$  is an unkind combination. In deep water, these two parameters are related by the heading angle  $\chi$  through the equation[3].

$$\omega_e = \omega - \frac{\bar{U}\omega^2}{g} \cos\chi. \quad (1)$$

A study of this relationship has proved quite revealing and it very strongly suggests that a quartering sea can present

considerable difficulties. The results may be shown most easily in the form of diagrams.

Fig.1 shows the condition when a ship moves into the waves in a head sea. Here,  $\pi/2 \leq \chi \leq 3\pi/2$  and we see that, given the ship speed  $\bar{U}$  and heading angle  $\chi$ , there is a one-to-one relationship between encounter frequency  $\omega_e$  and wave frequency  $\omega$ .

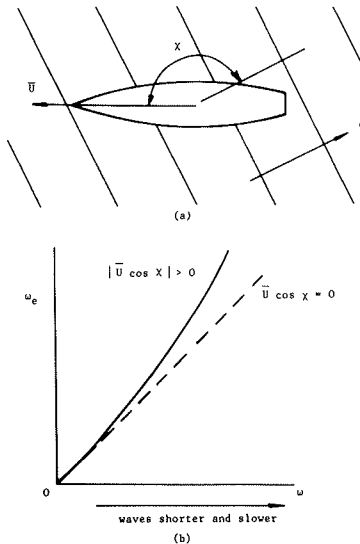


Fig.1 The relationship between encounter frequency and wave frequency for a ship moving in head seas such that  $\pi/2 \leq \chi \leq 3\pi/2$ .

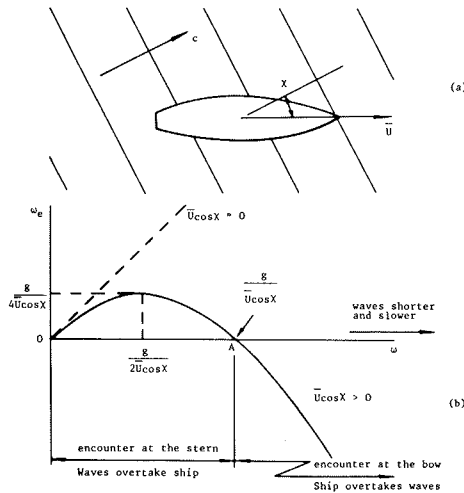


Fig.2 The relationship between encounter frequency and wave frequency for a ship moving in following seas such that  $3\pi/2 \leq \chi \leq 2\pi$  or  $0 \leq \chi \leq \pi/2$ .

Fig.2 shows that conditions are very different in a following sea, for which  $3\pi/2 \leq \chi \leq 2\pi$  or  $0 \leq \chi \leq \pi/2$ . Mathematically,  $\omega_e$  can now apparently take a negative value (though this only means that encounter takes place in the bow rather than at the stern). The matter can be thought of as indicated in fig.3(a). This last diagram can conveniently be

presented in terms of dimensionless frequency if the notation  $\delta = \omega \sqrt{l/g}$ ,  $\delta_e = \omega_e \sqrt{l/g}$  is adopted. Then we have the diagram of fig.3(b).

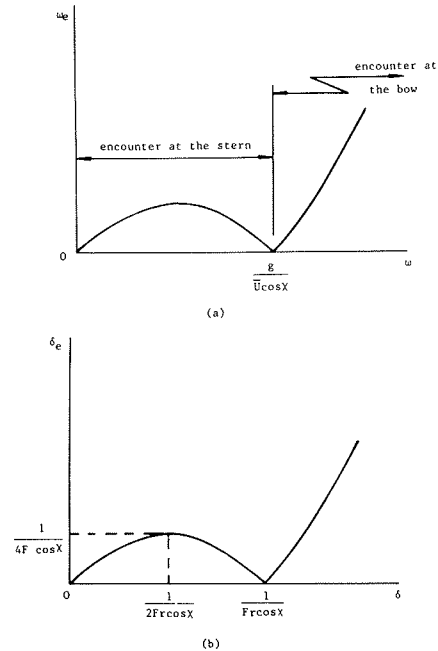


Fig.3 (a) Alternative form of fig.2(b); (b) Dimensionless form of fig.3(a).

Now since

$$\delta^2 = \omega^2 \frac{l}{g} = k l = \frac{2\pi}{\lambda} l \quad (2)$$

where  $k$  is the wave number, it follows that

$$\frac{l}{\lambda} = \frac{\delta^2}{2\pi} \quad (3)$$

In other words, if we specify the dimensionless wave frequency, then we automatically fix the ratio of ship length to wavelength. This means that the quantities

$$\delta_e, Fr, \chi, \delta \text{ (or } l/\lambda) \quad (4)$$

are interdependent so that, if we fix the value of one of these dimensionless parameters a simple relationship exists between the other three. Moreover if  $\chi$  is one of those other three, this relationship can conveniently be displayed in polar form.

To see the significance of this in the succeeding figures, imagine waves advancing towards the centre of the circles with the ship at the centre, pointing at 12 o'clock.

Fig.4 shows an example of what this implies. It is for  $Fr = 0.2$ , so it relates  $\delta_e$ ,  $\chi$  and  $\delta$  (and so  $l/\lambda$ ). Now we need no longer distinguish between the bow encounter and stern encounter regimes, since all the necessary information appears on the one diagram. Similarly, fig.5 shows the state of affairs if  $\delta_e = 2$ .

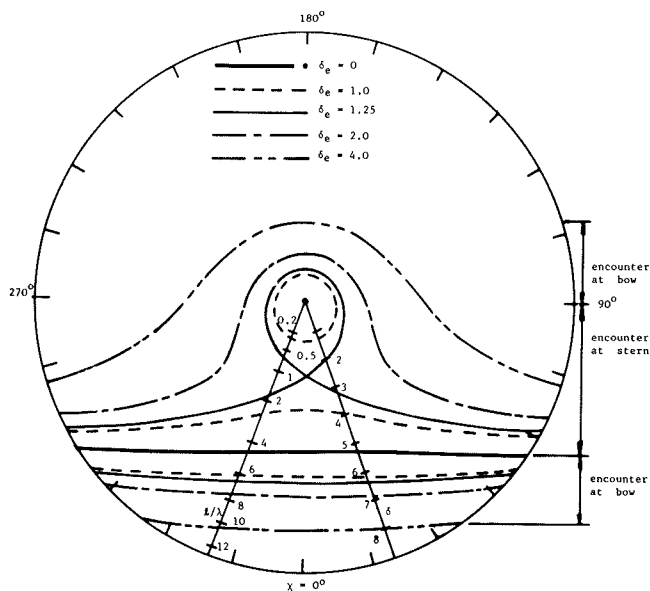


Fig.4 When  $Fr = 0.2$ , the dimensionless encounter frequency  $\delta_e$  and wave frequency  $\delta$  (or  $l/\lambda$ ) are related to  $\chi$  as shown for any ship.

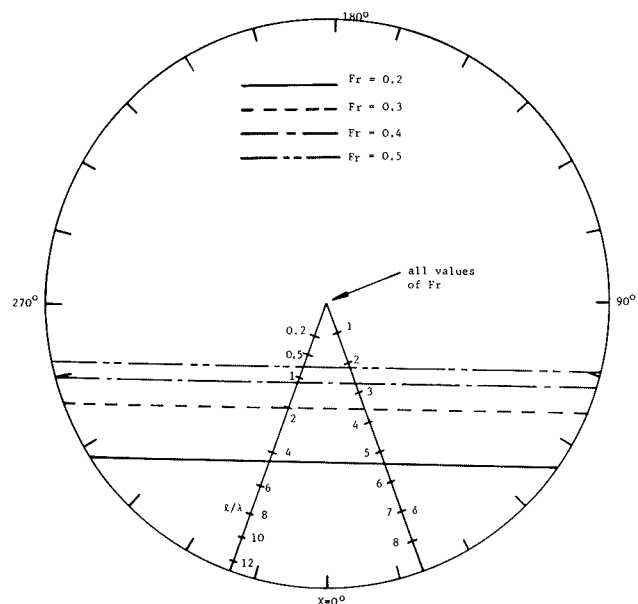


Fig.6 The case where  $\delta_e = 0$ , so that a ship sits stationary with respect to waves requires that  $l/\lambda$  (or  $\delta$ ) be related to  $\chi$  as shown for various Froude numbers. Note that the ship inevitably moves in a following sea.

#### 4. EQUATIONS OF INSTABILITY AND RESONANCE

When a ship moves in waves, the matrix of principal coordinates  $\underline{p}(t)$  of the dry hull in antisymmetric motion is governed by the equation [3]

$$\underline{A}(\omega_e)\ddot{\underline{p}}(t) + \underline{B}(\omega_e)\dot{\underline{p}}(t) + \underline{C}\underline{p}(t) = \underline{E}(\omega_e)e^{-i\omega_e t} \quad (5)$$

It will be seen that the square "system matrices"  $\underline{A}(\omega_e)$  and  $\underline{B}(\omega_e)$  are dependent on the encounter frequency  $\omega_e$ . For a given value of  $\omega_e$ , then, the equation is defined and a resonance condition of the form

$$\underline{p}(t) = \hat{\underline{p}}e^{-i\omega_e t} \quad (6)$$

may be sought.

It is implicitly assumed in this equation of motion that  $\underline{p}(t)$  will be proportional to  $e^{-i\omega_e t}$  and so, strictly speaking, a solution of the form

$$\underline{p}(t) = \underline{p}_0 e^{\lambda t} \quad (7)$$

is inadmissible. Yet it is not clear how free motion can be studied (and hence the dynamic stability assessed) unless some such trial solution can be introduced. Accordingly, in ref.2 it is suggested that this latter type of approach should be employed and it is on this basis that the results to be presented were found. That is to say, the stability of a ship under way

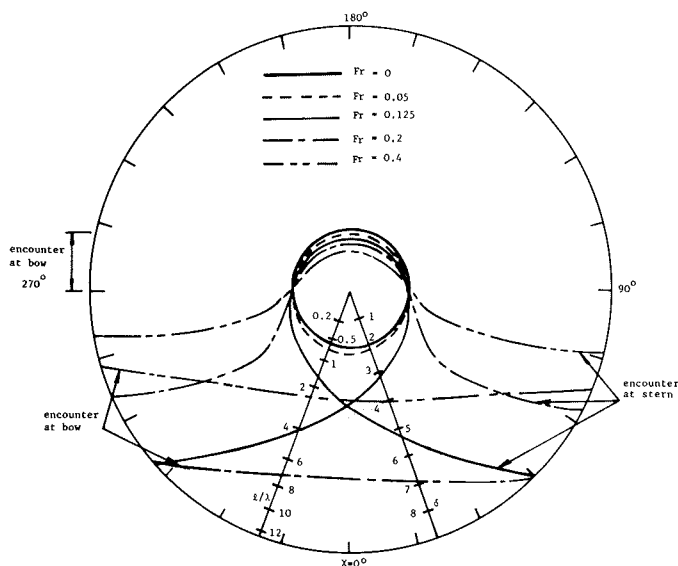


Fig.5 For a dimensionless encounter frequency  $\delta_e = 2$ , the value of  $l/\lambda$  (or  $\delta$ ) is related as shown to  $\chi$  for various Froude numbers  $Fr$ .

The important case in which  $\delta_e = 0$  is shown in fig.6. Note that, always,  $\chi < \pi/2$  or  $\chi > 3\pi/2$ : this only happens in a following sea.

was examined using the equations of anti-symmetric motion and investigating the possibility that a root  $\lambda$  of the algebraic equation

$$|A(\omega_e)\lambda^2 + B(\omega_e)\lambda + C| = 0 \quad (8)$$

shall have a positive real part (which would imply instability).

##### 5. SOME RESULTS FOR PARTICULAR SHIPS

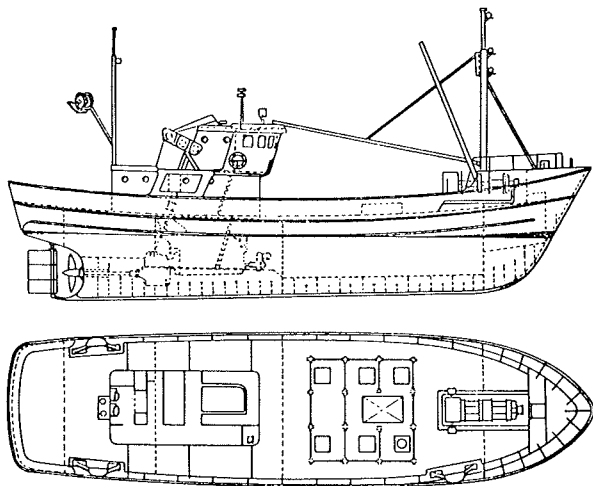


Fig. 7 Transom stern trawler. (Courtesy of Dr. A. Morrall, NMI, & of RINA)

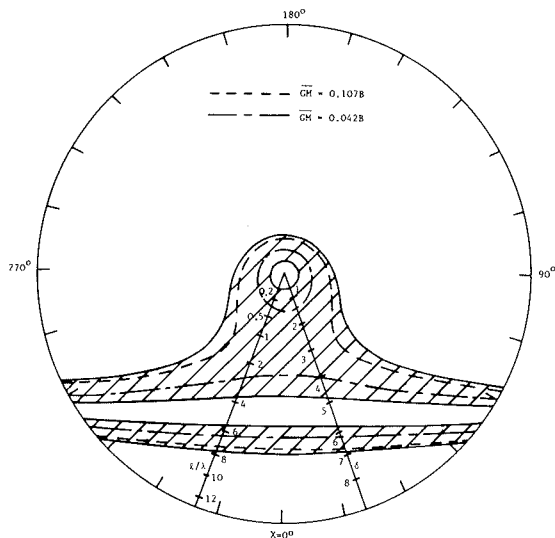


Fig. 8 Polar curves for the transom stern trawler of fig. 7 when it travels at  $Fr = 0.2$  in sinusoidal waves. The hatched area shows the region of instability and the broken lines give the resonance conditions for the two metacentric heights, 0.042B and 0.107B.

Fig. 7 shows a transom stern trawler. If this vessel is travelling at  $Fr = 0.2$ ,

the hatched area in fig. 8 is a region of instability. Thus the ship is unstable when  $\chi = \pi/4$  (i.e. in a quartering sea) if

$$0.06 < \frac{\ell}{\lambda} < 6.7, \quad \text{with encounter at the stern;}$$

$$9.2 < \frac{\ell}{\lambda} < 11.8, \quad \text{with encounter at the bow.}$$

The two broken lines represent resonance conditions for two different metacentric heights. Notice that both lie inside the hatched areas. Notice, too, that if  $\chi = 0^\circ$  resonance is much less likely with the larger GM than it is with the smaller. And this is not because the dynamic stability is the greater but because the ship is detuned.

Fig. 9 shows the results for the transom stern trawler when  $Fr = 0.3$ . Instability now arises in a much smaller range of  $\ell/\lambda$ ; but it is an important range in terms of magnitude of  $\ell/\lambda$ . What this form of plotting does not show, of course, is the intensity of the instability; this is, in fact, markedly greater for this higher Froude number. If the calculations are made for a Todd Series 60 model, the results are found to be fundamentally different. Fig. 10 is for  $Fr = 0.2$  and now it is seen that resonance and instability cannot prevail together. The inference is that a Series 60 model is likely to be much safer at this Froude number than a stern trawler in terms of capsizing.

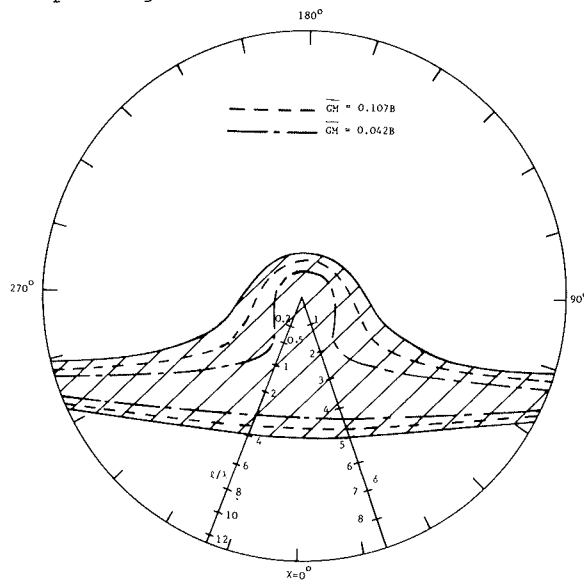


Fig. 9 Results for the transom trawler of fig. 7, comparable with those of fig. 8 but for  $Fr = 0.3$ .

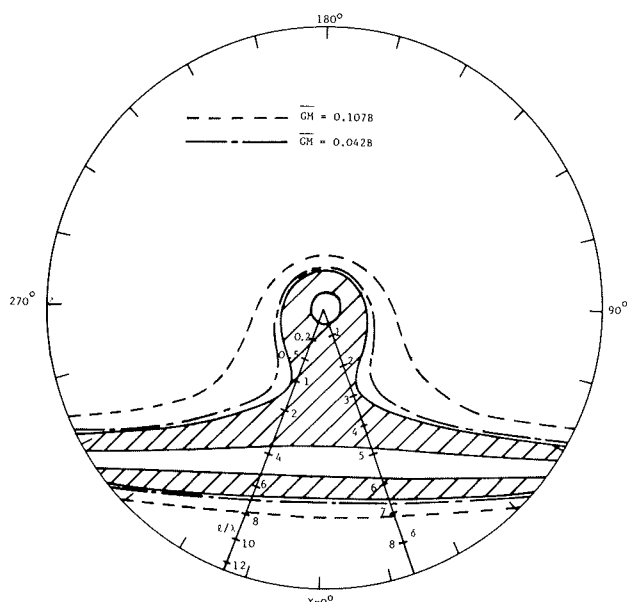


Fig.10 Results comparable with those of fig.8, but for a Todd Series 60 model travelling at  $Fr = 0.2$ . Note that the resonance conditions for the two metacentric heights taken, viz. 0.042B and 0.107B, lie outside the hatched regions of instability.

## 6. ZERO ENCOUNTER FREQUENCY

There are two distinct cases in which instability can arise with zero encounter frequency. The first is that of flat calm water and it is related to the familiar study of "directional stability", though we would suggest that the usual arbitrary suppression of rolling in the analysis is neither desirable nor necessary.

The second case is that in which a ship travels in waves and so adjusts its speed and heading as to sit stationary relative to the waves. This is a matter of special importance. The first thing to note is that it is not possible to proceed exactly as before because use of the rudder is generally necessary if the condition  $\delta_e = 0$  is to be maintained.

It would not be easy to discuss this subject in detail here because, by its very nature, the problem is essentially a mathematical one. Suffice it to say that equations have been set up and solved on the same basis as before and that comparisons have been made between a transom stern trawler and a round stern trawler. It turns out that the responses found are heavily speed dependent, those of the transom stern trawler being much the larger.

## 7. LOADING CONDITION

An attempt has been made to calculate the effects of various loading conditions on intact stability using the approach described in this paper. In a preliminary

way,, it may be said that liability to capsizing appears from that theory to depend sensitively on loading conditions. The calculations were made for a hull having the form of the EDITH TERKOL [4],[5], the main particulars being

length between perpendiculars	58.6m
beam	9.65m
depth to deck	4.15m
loaded condition:	
level keel displacement	1589 tonnes
amidships draught	3.8m
GM	1.156m
ballast condition:	
level keel displacement	595 tonnes
amidships draught	1.586,
GM	0.994m
Froude number $Fr$	0.2

Hydrodynamic data associated with each loading condition were obtained from a potential flow analysis and eigenvalues determining the stability were calculated from the homogeneous equation. The regions indicating stability and instability for these loading conditions are illustrated in fig.11(a). The dimensionless resonance frequencies associated with the complex eigenvalue are

$$\hat{\Omega}_3 = 2.18 \quad \text{for the loaded condition,}$$

$$\hat{\Omega}_3 = 1.6 \quad \text{for the ballast condition.}$$

For the loaded condition, this frequency is close to the boundary of instability, whereas for the ballast condition it is well into the instability region. This limited evidence suggests that there exists a stronger possibility of resonance and instability occurring simultaneously when the ship is in ballast with level keel than when the ship is loaded with level keel, provided no unforeseen hydrodynamic influences intervene by reason of the change in draught.

An additional investigation was made for the vessel in ballast but trimmed by the stern. This produced results similar to those found previously for the ship in ballast but with a level keel. No direct comparison can reasonably be made, as between the trimmed and level keel states, however, in the light of the data available and the calculations made.

So far, no heed has been taken of the slow motion derivatives (for which  $\omega_e = 0$ ) or for possible influences of the rudder. We have merely followed the dictates of potential flow analysis, despite its known shortcomings in the vicinity of  $\omega_e = 0$ .



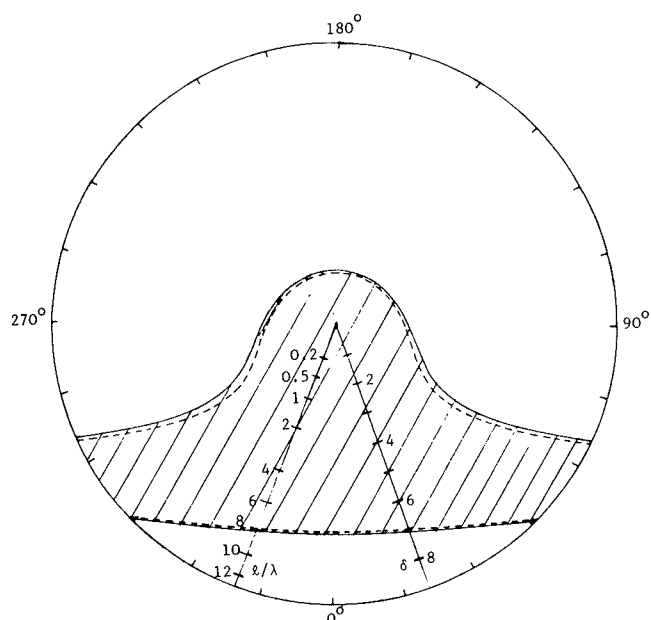


Fig. 11(a) (i) Loaded ( $\overline{GM} = 1.156\text{m}$ )

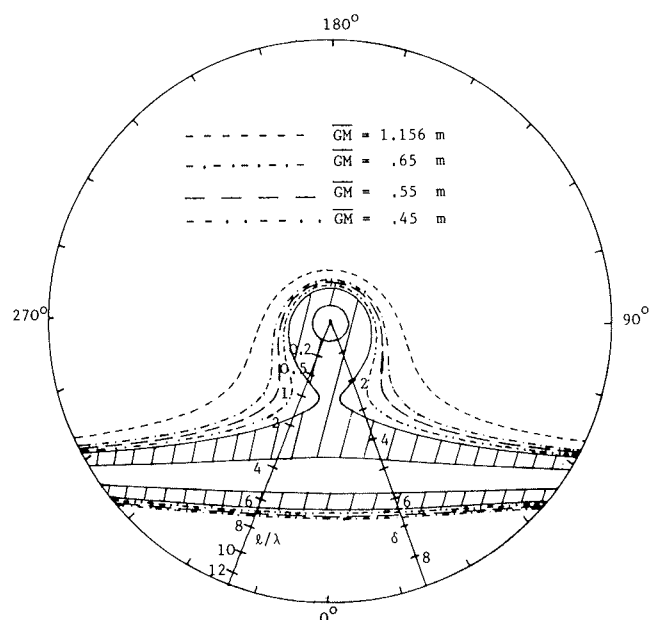


Fig. 11(b) (i) Loaded

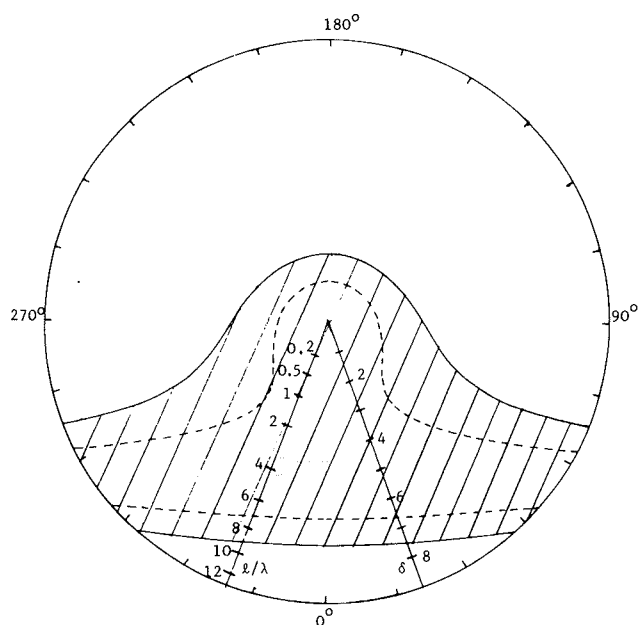


Fig. 11(a) (ii) Level keel in ballast ( $\overline{GM} = 0.994\text{m}$ )

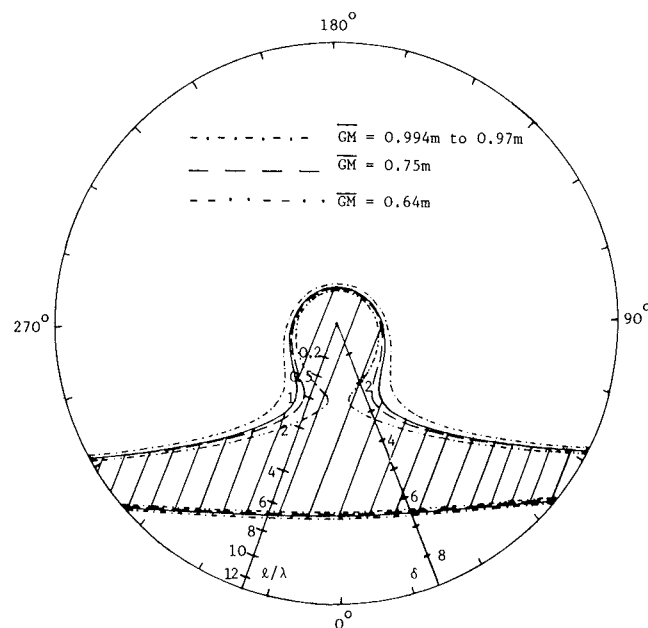


Fig. 11(b) (ii) Level keel in ballast

Unfortunately, no information is available on the type of rudder; moreover the slow motion derivatives of the ship when loaded and in ballast are not known. A more refined analysis of the ship's stability can therefore only be performed by using a patchwork of data obtained from other sources [6,7]. Then the stabilities of the vessel with level keel in the loaded condition and in ballast can be compared by applying the same approach to each case.

Using experimental slow motion derivatives [6] for a Series 60 model with rudder attached, equivalent data for the vessel under discussion were derived by adopting a simple non-dimensional analysis involving length, draught, beam and displacement parameters. Slow motion derivatives for a hull comparable with that of the EDITH TERKOL can also be obtained from the empirical formulae presented by Inoue et al [8]. Generally speaking, the data found in these ways agree more or less with the experimental findings

of Gerritsma [7]. The data derived in this way indicated that the ship in ballast suffers slight directional instability whereas the loaded vessel is directionally stable in calm water, as suggested by Gerritsma's experiments. (Derivatives may also be determined for the vessel when it is trimmed from the Inoue formulae.)

Frequency dependent hydrodynamic data were estimated on the basis of potential flow theory. When found, these data have to be combined with the slow motion derivatives in a consistent manner. This combination was achieved by comparing the theoretical and experimental hydrodynamic coefficients derived for the Series 60 model and applying correction factors so that the theoretical data agreed with the experimental results [6]. When the same correction factors are applied to the slow motion derivatives and to the potential flow calculated coefficients, the frequency dependent hydrodynamic data are arrived at. This was done for both of the level keel loaded conditions. When this information is used in the stability analysis, the regions representing stability and instability are those shown in fig.11(b).

The frequency range indicating instability of the loaded ship is now far from the resonance frequency; instability and resonance can now occur only for much smaller GM values. On the other hand, the ship in ballast with level keel is such that resonance and instability are much more likely to occur together. By reducing the GM value to what would normally be regarded as quite acceptable values, instability and resonance may easily be made to occur simultaneously.

Although the assumptions in this analysis are admittedly wide open to question, the results agree with observations made in experiments on a model of the EDITH TERKOL [4]. That is, as GM is reduced in the ballasted ship (which was also trimmed) the possibility of model capsize increased greatly; thus GM = 0.97, 0.75, 0.70, 0.64m produces  $\hat{\Omega}_3 = 1.57$  (no capsize), and 1.37, 1.32, 1.26 (capsize). By contrast in the range of GM considered for a loaded model, no capsize was predicted; that is GM = 0.65, 0.55, 0.45m produces  $\hat{\Omega}_3 = 1.67, 1.54, 1.4$  (no capsize). From the mathematical model it may be shown that instability and resonance may occur together for the model in ballast but separately for the loaded model. (Unfortunately, although slow motion derivatives can be deduced for a trimmed condition of the model no satisfactory method could be found of obtaining suitable frequency dependent data, especially when the trim angle is large.)

On the basis of this somewhat inconclusive evidence it may be suggested that a ship in ballast with level keel is more liable to capsize than the loaded ship with level keel. This is, at least, what one would expect.

## 8. IMPROVEMENT OF THE THEORY

Having seen how the stability conditions vary with encounter frequency within the terms of the foregoing theory, we now turn to the question of how the theory itself might be improved. It has been suggested [10] that, in seeking the complementary function of the equation of motion, the matrices  $A(\omega_e)$  and  $B(\omega_e)$  should be replaced by  $A(0)$  and  $B(0)$  so that the calculations become those of conventional directional stability (but with allowance made for rolling). The justification for this proposal is not clear to us since it would then appear that the complementary function would no longer be a solution of the appropriate homogeneous equation of motion.

A rather more radical approach is to start, not from the second order differential equation, but rather from a linear functional representation of the fluid actions. The equation of motion then takes the integro-differential form, first introduced by Cummings [11], in which convolution integrals are employed. This requires a substantial recasting of the stability analysis and appears to suggest that, while the matrices  $A(\omega_e)$  and  $B(\omega_e)$  are the appropriate ones, the trial solution should have the form

$$p(t) = p_0 e^{(\lambda - i\omega_e)t} \quad (9)$$

This, however, is still under investigation and will be discussed later.

## CONCLUSIONS

It appears that it may be more advantageous to seek the conditions under which uncontrollable large antisymmetric ship motions can arise, using linear theory, than to attempt to describe those motions in the terms of non-linear theory. The combination of dynamic instability and resonance in waves is thought to be the criterion that should be sought. If this hypothesis is indeed justified there remains much to be done by way of research, both theoretical and experimental.

This paper is concerned with the role played by encounter frequency. The predictions appear to be borne out by experience but they do raise the question of how unstable the ship must be and how violent the resonance must be for that ship to be in real danger. On the evidence presented in this paper it would appear that simultaneous antisymmetric instability and resonance alone may be somewhat pessimistic. Either the instability or the resonance (or both) needs to be marked for the ship to be in real danger. All one can say for certain at this stage is that much more detailed research is necessary.

The writers would emphasise the need for caution. Although, to be sure, results have so far been encouraging the theory necessarily rests on numerous assumptions as regards basic data, calculated hydrodynamic coefficients and even technique of employing linear theory.

## ACKNOWLEDGEMENT

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## Discussion

A.Y. Odabasi (The British Ship Research Association, UK)

As I have already expressed most of my misgivings on the approach proposed by the authors during the discussions of Ref.[2], I shall not repeat them here. I would however like to ask:-

(1) The authors correctly point out the inadmissibility of a solution of the form  $P_0 e^{\lambda t}$  and yet employ it, derive results of startlingly accurate conclusions! I would suggest that the authors could have written their equations either in the frequency domain in the form:

$$[-\omega^2 \tilde{A}(\omega) - i\omega \tilde{B}(\omega) + \tilde{C}] \tilde{X}(\omega) = \tilde{E}(\omega)$$

where  $X(\omega)$  is the Fourier transform of the response vector and determine poles in the complex domain, or employ a correct time domain equation, as discussed in Ref.[11], and then use  $P_0 e^{\lambda t}$ .

(2) Since a mathematical model cannot be used beyond the range of its applicability, to attach any significance to the authors' findings, even when they satisfy the criticism levelled above, one needs to know when equation (5) became invalid. Systematic experiments suggest that, with some exceptions, i.e. specially designed geometries, this range does not exceed 6 or 7 degrees, cf. [D1-D2].

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S. Kastner (College of Engineering Bremen, FRG)

It has certainly been a good idea by the authors to draw our attention again to the role of the encounter frequency. The authors introduce the polar graphs, whereas I usually look at a normalized encounter spectrum, which then can be overlapped with a probability distribution of natural roll frequencies, as I have shown at the WEMT Conference on "Safety at Sea" in London 1977. However, the authors extend the known considerations on encounter frequency resonance by including results from rudder-fixed linear instability. A further point: It has never been clear to me, why the matrices  $A(\omega_e)$  and  $B(\omega_e)$  should be replaced by the values for zero frequency, but I assume it was a matter of simpler computation. Finally: Looking at current litera-

ture, one might get the impression the authors give us, that mainly the nonlinear theory is being pursued. Again, these activities show the amount of efforts put into studies on an extremely non-linear problem such as capsizing, with all the tools available, i.e. highly developed experimental techniques, and application of computers for simulation. I am glad about this breakthrough, compared with the state of our First Conference in Glasgow, but surely we should not forget on the possibilities of linear theory. I hope the authors will continue, and their results might be compared with non-linear studies.

Y. Welaya (Alexandria University, Egypt)

Professor Bishop and his colleagues always give us something to think about in their papers and I would like to seek their opinion on two points.

Firstly, the contents and theoretical approach adapted here to deal with ship stability, is different to their other publication, e.g. the paper given in RINA Spring Meeting in 1981.

Does this mean they have abandoned the previous works in favour of the present approach?

Secondly, they use Series 60 vessels as an example to provide evidence on the soundness of their present approach and make no mention of scale. Does this mean they believe all future safe vessels should be of comparable size?

#### Author's Reply

Prof. Kastner's observations are kind and call for little reply beyond, of course, that we have not been concerned in this paper with probabilistic theory.

To Dr. Welaya we must first point out that we have not, in fact, departed from the approach outlined in the RINA Spring Meeting 1981; we are a little mystified that he should think that we have. The reference we have made to a Series 60 model as a standard of comparison has no hidden significance; it merely reflects the fact that there exists a serious shortage of reliable experimental data extending over a large frequency range and here was a case in which some moderately hard data were available.

Dr. Odabasi first suggests two methods of avoiding the problem we identified in section 4 of the paper. Unless we misunderstand him, the first of them does not provide a way out of the difficulty. We endorse his second suggestion (and indeed suggested it in section 8), but we come to a different conclusion.

The second matter raised by Dr. Odabasi rather misses the point we try to make in the paper. We are trying to find a dangerous condition and not to evaluate how dangerous it is.

# PARAMETRIC EXCITATION OF ROLL MOTION AND ITS INFLUENCE ON STABILITY

NERE G. SKOMEDAL

Det norske Veritas

Norway

## 1. ABSTRACT.

Parametrically excited roll motion of a ship and its influence on stability is studied in some detail. This is a nonlinear energy transfer process from vertical motions to roll motion.

The amount of energy transferred into roll motion is shown to depend on hull lines and loading condition. Hull forms with large change in waterplane area with draught are shown to stimulate the energy transfer the most.

The dynamic stability of a ship exposed to parametrically excited roll motion is shown to depend heavily on the roll damping.

A method to analyze the potential danger of parametric excitation of roll motion from hydrostatic calculations is presented.

Existing stability criteria for Mathieu instability are evaluated in light of the new findings and a design criterion which must be fulfilled in order to avoid the instability problem is suggested.

## NOMENCLATURE.

$a_w$	Area of waterplane	( $m^2$ )
$A_{44}$	Added moment in roll	( $kgm^2$ )
$B$	Breadth	(m)
$B_{BK}$	Breadth of bilgekeel	(m)
$B_{44}$	Damping coefficient in roll	(Nm)
$F_{44}$	Wave moment in roll	(Nm)
$g$	Gravity	( $m/s^2$ )

$GM_0$	Initial metacentric height	(m)
$GZ$	Righting arm	(m)
$h$	Exciting parameter	
$I_{44}$	Moment of inertia in roll	( $kgm^2$ )
$L_{pp}$	Length between perpendiculars	(m)
$R(\tau)$	Correlation function	
$S_h(\omega)$	Spectrum of h-parameter	(s)
$S_w(\omega)$	Wave spectrum	( $m^2s$ )
$T_1$	Zero-crossing period	(s)
$TRF(\omega)$	Transferfunction	
$z_{rel}$	Vertical relative motion	(m)
$Z_{wl}$	Distance from waterline to centre of gravity	(m)
$X_{FL}$	Ordinate for centre of flotation	(m)
$\Delta$	Displacement	(kg)
$\phi, \dot{\phi}, \ddot{\phi}$	Roll angle, -velocity and acceleration.	(rad, rad/s, rad/s <sup>2</sup> )
$\zeta$	Wave amplitude	(m)
$\rho$	Linear damping coefficient	
$\rho_c$	Cubic damping coefficient	
$\rho_{sw}$	Density of seawater	( $kg/m^3$ )
$\omega$	Wave frequency	(rad/s)
$\omega_0$	Natural roll frequency	(rad/s)

## 2. INTRODUCTION.

The problem of parametrically excited systems is as such classical and has been previously treated by many authors in a number of fields of science. For an introduction of the application to roll motion of ships, the reader is referred to Roberts 1980 /1/.

Parametric excitation of roll motion is experienced when a ship in a seaway encounters a group of waves at nearly the half of the ship's natural roll period, resulting in a significant roll motion. If the ship is sensitive to parametric excitation, the roll angles can increase to 30-40 degrees and even capsizes may occur.

The parametric excitation process is arising from a time dependent variation of the restoring moment when the ship is moving in the seaway. In this study different ways of describing the parametric exciting process qualitatively and quantitatively are shown. To the author's opinion it is important that the magnitude of the parametric excitation process is simple to calculate, giving simple expressions for the sensitivity for an actual ship to parametric excitation, avoiding extensive use of simulation methods or sophisticated hydrostatic calculations. Then it is possible to use the approach in practical design or control work in assessing the stability of a ship.

The intention with this paper is not to provide a complete solution to a complicated stability problem, but rather to show some implications of a simplified physical model of the problem.

### 3. THEORY.

The following equation is the applied mathematical model of the ship-rolling motion. Coupling from sway and yaw are neglected, as usual in previous analysis of this kind (Roberts 1980 /1/).

$$(I_{44} + A_{44})\ddot{\phi} + B_{44}(\dot{\phi}) + g\Delta_0 GM_0 \left( \frac{GZ(\phi)}{GM_0 \cdot \phi} + h(t) \right) \phi = F_4(t)$$

where

$h(t)$  is the time dependent part of the restoring coefficient.  
Can be expressed as:

$$h(t) = \frac{\Delta(t) \cdot GM(t)}{\Delta_0 \cdot GM_0} - 1$$

$I_{44}$  is moment of inertia in roll.

$A_{44}$  is added moment in roll.

$B_{44}(\dot{\phi})$  is the nonlinear damping term.  
The corresponding linear term is written as

$$B_{44}(\dot{\phi}) = 2\rho \cdot \omega_0 (I_{44} + A_{44}) \dot{\phi}$$

$\frac{GZ(\phi)}{GM_0 \cdot \phi}$  is nonlinear restoring term.  
The corresponding linear term may be set to 1.0.

The exciting parameter  $h(t)$  is an expression for the resulting time dependent variation of the metacentric height in a seaway when effect of the time dependent variation of displacement force is taken

into account as well. By neglecting the righthand side wave moment and linearize the non-linear terms, we will, after applying the following transformations,

$$\phi = y(t) e^{-\rho\omega_0 \cdot t}, y(t) = y \cdot \sin\omega t, \tau = \omega \cdot t$$

obtain the classical Mathieu equation for the harmonic case:

$$\frac{d^2 y}{d\tau^2} + (\delta + \epsilon \cdot \cos\tau)y = 0$$

$$\delta = \left( \frac{\omega_0}{\omega} \right)^2 \cdot (1 - \rho^2)$$

$$\epsilon = h \left( \frac{\omega_0}{\omega} \right)^2 (1 - \rho^2)$$

It is well known that near  $\delta = 1/4$ , the Mathieu equation has it's most important unstable region for small values of the exciting parameter.

It has been shown that the nonhomogenous Mathieu equation with wavemoment included has the same stability domains as the homogenous equation. (Rosenberg 1954 /2/ and Roberts /1/)

Assuming that the stable/instable regions may be found at small roll amplitudes where the nonlinear terms are negligible, the Mathieu equation with non-linear coefficients has the same stable/instable domains as the equation with linear coefficients. The linear coefficients are convenient to use in order to compare different hullforms, loadconditions, damping devices etc.

### Simulation method and model tests.

A brief description of the numerical procedure and model tests used in this study is given in /5/. There it is stated that the method of calculating the exciting parameter  $h(t)$  as a function of pure vertical relative motion at amidship seems to fit model test results satisfactory.

This method is correct in long crested beam sea, however it seems to be in agreement with model test results in head or following seas as well.

#### 4. METHODS TO DEDUCE THE PARAMETERS GOVERNING THE ROLL MOTION.

The effects causing the parametric exciting process of the roll motion is very complicated. Untill now most research have concentrated on pure heave-coupling into roll. This effect is believed to be of most importance and the effect was firstly treated by Froude 1861 /3/. Paulling and Rosenberg 1959 /4/ expressed the different coupling effects where they linearized the coupling effects utilizing Taylor-series expansion.

The heave coupling into roll was expressed as:

$$h = K_{z\phi} \cdot z_{rel} = \frac{d}{dz_{rel}} (\Delta \cdot GM) \cdot z_{rel}$$

where  $K_{z\phi}$  is couplingcoefficient between heave and roll.

$z_{rel}$  is average relative vertical motion

The pitch-roll coupling is expressed in a similar way:

$$h = K_{\theta\phi} \cdot \theta = d/d\theta (\Delta \cdot GM) \cdot \theta$$

and the coupling from waveprophile passing along the ship side as:

$$h = K_{\zeta\phi} \cdot \zeta = d/d\zeta (\Delta \cdot GM) \cdot \zeta$$

These parameters are only functions of hullform and mass-distribution.

##### Linear methods.

Paulling expressed the coupling-coefficients in heave and pitch as follows:

$$K_{z\phi} = 2\rho_{sw} \int_{AP}^{FP} y^2(x) \frac{dy}{dz}(x) dx + \rho_{sw} \cdot Z_{WL} \cdot a_{WL}$$

and

$$K_{\theta\phi} = -2\rho_{sw} \int_{AP}^{FP} xy^2(x) \frac{dy}{dz}(x) dx + \rho_{sw} Z_{WL} \cdot a_{WL} \cdot x_F$$

These expressions show the importance of the slope of hull side  $dy/dz$  at stillwaterline. Traditional hull forms have nearly vertical ship sides near amidship and some slope near bow and stern.

Untraditional hullforms with bargeformed stems and large flare in forebody have considerably larger average hull slope than the older traditional hullforms. In addition untraditional ships have often a larger B/d-ratio which indicates that such ships have possibilities to operate with a relatively higher position of centre of gravity, increasing the magnitude of the coupling coefficient of heave induced rolling.

##### Non-linear methods.

The time dependent variation of the h-parameter due to heave-coupling, or more correct, due to coupling from vertical relative motion is discussed in some detail by Lindemann and Skomedal 1982 /5/.

There it was shown that the h-parameter may be calculated according to:

$$h(t) = \frac{\Delta(z_{rel}(t))GM(z_{rel}(t))}{\Delta_v \cdot GM_0}$$

where

$$\Delta(z_{rel}(t)), GM(z_{rel}(t))$$

respectively displacement and metacentric height at a point of time with vertical relative motion  $z_{rel}(t)$ .  $z_{rel}(t)$  is the distance between stillwater waterline and wave elevation at amidship.

and

$$\Delta_v, GM_0$$

initial values at stillwater.

##### Wave-prophile coupling.

The h-parameter due to waveprophile passing along the ship is written as: (non-disturbed waveprophile)

$$h(t) = \frac{\Delta(\zeta(t)) \cdot GM(\zeta(t))}{\Delta_v \cdot GM_0}$$

where

$$\Delta(\zeta(t)), GM(\zeta(t))$$

are values at time t with wave amplitude  $\zeta$

$h(\zeta(t))$  may be calculated using computer programs capable to cope with hydrostatic calculations of a ship positioned in a "frozen" wave. In the present analysis the computerprogram NV210 /6/ is used. The relation between waveprofile and transverse stability are quite complex. The effect is to a rather high degree nonlinear with respect to amplitude and position of the wavecrest as well as frequency.

The form of the  $h$ -parameter function indicates that the  $h(t)$  may be expressed as:

$$h(t) = k(\omega)h'(\zeta(t)) = k(\omega)(h_0 + h_1 \cos(\omega t + \delta))$$

where the function  $k(\omega)$  takes into account the wavefrequency dependency. The factors  $h_0$  and  $h_1$  are calculated for the case of wavelength equal to  $L_{pp}$  and are dependent of waveheight.

If the actual ship has large flare or slope of hull side in the range of mean waterline, the method given above is questionable. Especially in case of a irregular seaway where the effects of a large number of small amplitude waves are added. The constant  $h_0$  seems to give an addition to the initial metacentric height reducing the sensitivity to parametric excitation. See fig. 1. Hence no detailed verification of the method is made in this report, however, the numerical simulated roll response in regular following sea seem to be in a likely range. The need for more modeltests are obvious.

## 5. THE EFFECT OF HULL FORM AND LOADING CONDITIONS.

In the preceeding chapter the calculation of the exciting parameters was discussed. In this chapter some implications of these parameters are shown.

### Hullform.

In figures 2 and 3 the  $h$ -parameter for a number of ships are plotted as function of relative motion, indicating the large differences in sensitivity to parametric excitation between different hullforms and loading conditions.

It is seen that linearization of the  $h$ -parameters are questionable for some of the ships. Simulation with these nonlinear  $h$ -parameters as input data, shows that ships with large and nearly constant slope of the  $h$ -parameter are most sensitive to parametric excitation. All the investigated ships are more sensitive to parametric excitation when they are lightly loaded, i.e. when the  $B/d$ -ratio has a large value and the average slope of hull is larger.

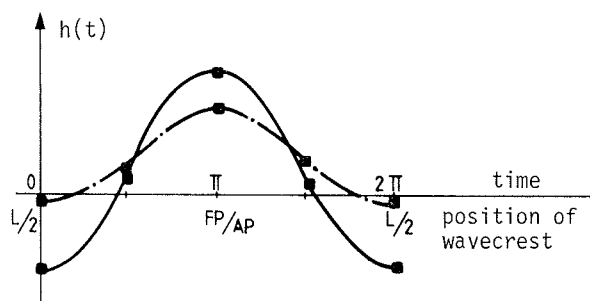


Fig. 1.

The  $h$ -parameter plotted as function of time for two waveheights.

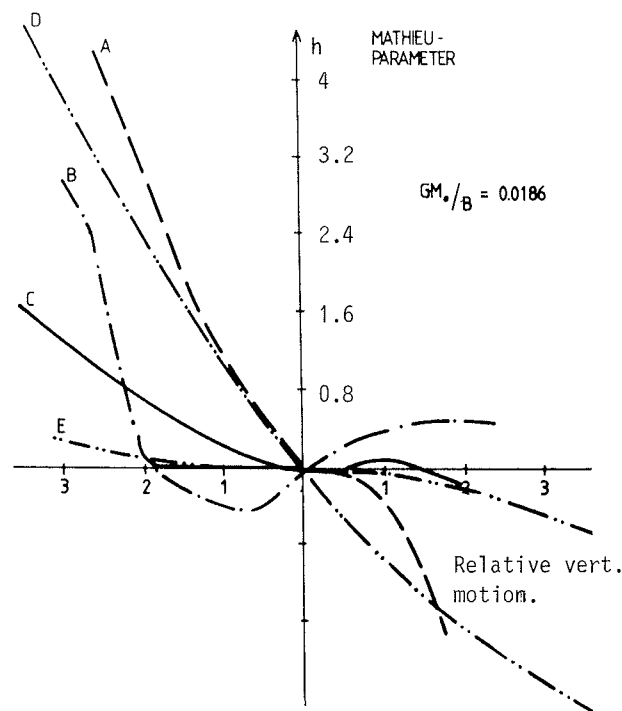


Fig. 2.

The  $h$ -parameter plotted as function of relative vertical motion. The  $GM/B$ -ratio is relatively low. A, B, C and D have untraditional hull forms. (Large change in waterplane with draught.) E is a common crude oil tanker with traditional hullform. The vessel B has large nonlinearities in the  $h$ -parameter function.



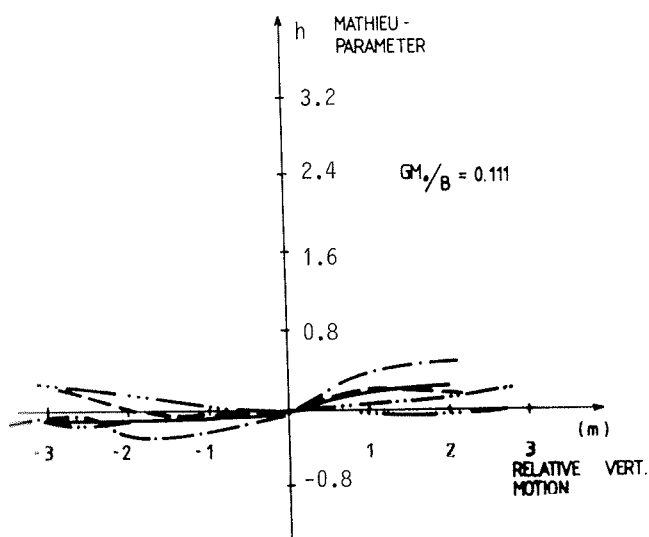


Fig. 3.

The h-parameter plotted as function of relative vertical motion for a number of ships. The GM/B-ratio is constant and relatively high. All the vessels have a small h-parameter resulting in minor risk of parametrically excited roll motion.

#### Metacentric height and IMCO-recommendations.

In reference /5/ it is shown that a ship which is sensitive to parametric excitation may capsize in a seastate with significant waveheight of 8 metres and larger, even if the IMCO recommendations to stability are complied with. In the same reference a capsize of a model in following sea was reported, however in this case the model did not fully comply with the IMCO-recommendations. This indicates that the existing stability criteria may not be sufficient for ship-types which are sensitive to parametric excitation. The IMCO stability recommendations are based on casualty statistics of ships before world-war II, and a number of new ship designs have been taken into service in the recent years indicating that a revision of the existing criteria are needed.

#### Roll damping.

The damping term in the roll equation is generally highly nonlinear with respect to roll velocity.

In order to represent this non-linearity, a linear and cubic factor is used in this report. This method seems to fit model-test result satisfactory if the coefficients are carefully calculated (Dalzell 1978 /6/).

By rolling of ship-models the decay coefficient defined as

$$\rho = \frac{1}{2\pi} \ln\left(\frac{\phi_1}{\phi_2}\right)$$

where  $\phi_i$  are subsequent roll angles.

The decay coefficient may be determined as a function of roll amplitude. A similar curve may be obtained using theoretical or empirical estimates on roll damping as for instance Tasaki 1978 /7/.

Myrhaug 1981 /8/ states that the skin friction damping is negligible in full scale related to modelscale when the ship is newly built and painted. The scale effect of the bilgekeel is believed to be nearly independent of the Reynold number due to its sharp edge making the separation point well defined. The wavedamping is related to the Froude number. Hence roll damping is believed to be somewhat overestimated by using model test results, however this error is for practical purposes believed negligible.

By a regression analysis the linearized damping coefficients as function of roll amplitude may be approximated to:

$$\rho = a_1 + a_2 \cdot \phi^2$$

Following a method similar to Blocki 1980 /9/, equating the work of the equivalent linear damping to the nonlinear (linear and cubic) damping during one roll period, the nonlinear damping coefficients may be written as:

$$B_{44}(\dot{\phi}) = 2\rho\omega_0(I_{44}+A_{44})\cdot\dot{\phi} + \rho_c/\omega_0(I_{44}+A_{44})\cdot\dot{\phi}^3$$

where

$$\rho = a_1, \quad \rho_c = \frac{3a_2}{4a_1\omega_0}$$

This method is used in the calculation of damping coefficients given in the table 1.

#### 6. STABILITY

The term stability of a ship is commonly used when discussing the ability of a ship to withstand environmental forces in an operative condition without capsizing. However it seems to be some different ways of formulating the stability criterion.

Limiting ourselves to a traditional "quasistatistical" approach the problem is of minor importance, however, extending the term to dynamical cases, some problems in the terminology may arise, especially when dealing with differential equations which have their own instability/stability domains. In order to reduce the probability for misunderstanding a short discussion of the subject will be made.

The stability of a ship may be understood in two ways using a mean square stability criterion:

i) Asymptotical stability.

The roll motion is stable when:

$$E(\phi^2) \rightarrow 0 \\ t \rightarrow \infty$$

This stability criterion may for instance be related to a mathematical system with parametric excitation, where the system is defined stable when the response dies out after a while independent of the initial condition. This criterion may be applied when discussing whether or not a system is sensitive to parametric excitation.

However, a ship may be termed stable even if it is unstable according to this stability criterion provided the roll response is limited. This leads to the following criterion.

ii) Practical stability.

The roll motion is stable if:

$$E(\phi^2) \leq \epsilon_m \\ t \rightarrow \infty$$

where  $\epsilon_m$  is correlated to a maximum allowable roll angle.

This definition is also termed "Uniform boundedness" of the roll motion and implies that roll angles, velocities and accelerations should be limited and the bounds should be independent of the initial condition.

This criterion may be related to a ship with parametrically excited roll motion. After the initial conditions have died out, the roll motion will increase until a "stable" maximum value is reached where the dissipating energy equals the input parametric excitation energy. If no such energy balance exist, the ship will capsize. This is very clearly seen in the case of regular parametric excitation shown in figure 4 where the roll motion is increasing until energy balance is reached.

In statistical terms a way of defining stability of a ship is by introducing the probability  $P_c$ , that the ship will capsize (or exceed a maximum allowable roll angle) within a predicted time interval and define

the ship to be stable when  $P_c$  becomes less than a given probability level. The time interval should at least be the estimated lifetime of the ship and  $P_c$  calculated according to the long term distribution of rolling (Vinje 1976 /10/).

A definition as above is commonly accepted as a dynamical stability criterion, however, despite of much effort, no method has until now been accepted as a correct and practical tool in the assessment of ship stability.

Believing that parametric excitation contributes significantly to the roll motion for some types of ships, considerably improvements of the ship's rolling capability and safety will be made if the probability of such excitation can be reduced or eliminated.

An investigation of this is simple to carry out, and a linear roll equation may be used in the calculation with sufficient correctness. If the roll motion is found unstable in the sense of definition i) further steps may be done in order to judge the practical stability, whether the roll response is bounded, or the roll response is unbounded in the way that the ship will capsize applying the statistical stability criterion mentioned above. When the roll amplitudes approach relative large values the nonlinear damping and restoring terms have to be taken into account. This will complicate the calculation of the stability of the ship. In addition the different loadcases the ship is operated with have to be taken into account.

## 7. Stability of a lightly damped regular system.

Parametric excitation may be considered as an energy transfer from other motions into roll motion.

Kure and Bang 1974 /11/ assumed that roll motion becomes stable when the work of damping moment is equal or larger than the work of exciting moment from other motions to roll motion during one rolling period.

Assuming the roll response to be of small magnitude, i.e. linear coefficients in the roll equation, the following result is obtained for regular waves at  $\delta = 1/4$ :

The roll motion is stable (def i)) if

$$\rho > 1/4 \cdot h$$

As shown in figure 5 this is in agreement with numerical simulation.

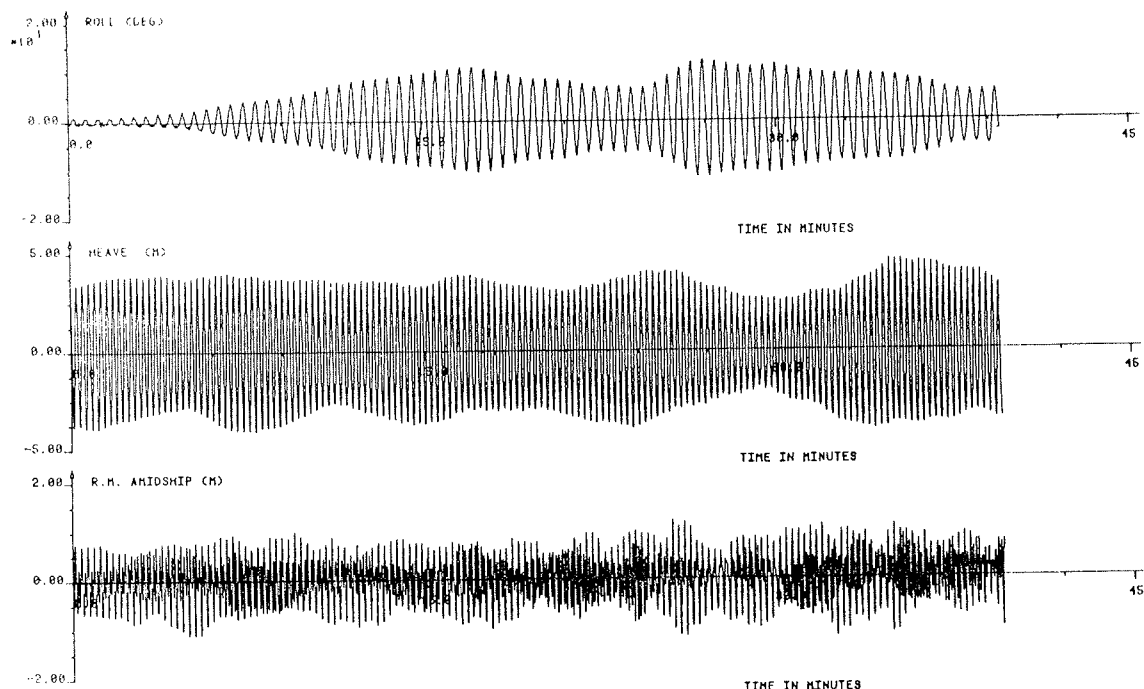


Fig. 4.

Roll motion due to parametric excitation. Regular beam sea. Medium bilgekeels.

As seen, the roll amplitudes reach a "stable" level where the roll damping work equals the input energy.

#### 8. Research in the field of stability of a randomly excited system.

Vinje 1976 /10/ has applied results from previous papers in an analogous field of electrical engineering to the ship-roll problem. Extending the results for lightly damped systems, the almost sure asymptotic stability criterion firstly introduced by Kozin /12/ is found to be:

The ship is stable (def. i)

$$\sigma < \sqrt{2\pi} \rho$$

where  $\rho$  is linear damping coefficient and

$$\sigma^2 = \text{Var}(h)$$

In his paper Vinje indicates another interesting solution of the stability due to Ariaratnam /13/, where the stability of the solution of a lightly damped system is correlated to the amplitude of the exciting process at the double of the characteristic frequency.

Two similar criteria are discussed below:

Roberts /1/ has in a recent report considered the uncoupled ship rolling motion with parametric excitation included. In his study he includes wave induced roll moment, non-linear damping and non-linear restoring coefficients. Introducing the theory of Markov-processes and standard methods of the Fokker-Planck-Kolmogorov equation (Arnold 1973 /14/) of the roll amplitude process. The solution of the stability problem is found to be for the linear roll process:

The ship is stable (def. i) if:

$$\pi\omega_0 S_h(2\omega_0)/4 < \rho$$

where  $\rho$  is nondimensional roll damping

$\omega_0$  is natural roll frequency

$S_h$  is the energy-spectrum of the exciting parameter  $h$  as:

$$S_h(\omega) = \frac{1}{2\pi} \int_{-\infty}^{\infty} R(\tau) \cos\omega\tau \cdot d\omega$$

$R(\tau)$  is the correlation function of the  $h$ -process.

Roberts has in the same paper given expressions for the roll response when the roll motion is parametrically excited. These expressions are based on the assumption that the roll response is stationary and seem to over-estimate the roll motion compared to modeltests. The estimated roll variance is larger than those obtained in model experiments. For model experiments the roll motion seems to be switched off for long periods during the test run, hence reducing the average roll response in the test run. See fig. 6. This effect stresses that this mode of roll response is dependent of the tuning factor between wave-periods and natural roll period. Hence more model test are needed in order to verify the roll response expression presented by Roberts.

Wedig 1972 /15/ has solved a analogous problem in the field of probabalistic structural dynamics. He studied the behaviour of a uniform pin-ended coloumn subjected to a time varying axial load.

Assuming the parametric excitation process as a bandpass-Gaussian process, utilizing Markov-process theory and the Fokker-Planck equation, he obtains the same formula as Roberts.

In the following the stability criterion of asymptotic stability (def 1)) is compared to modeltests in irregular beam seas, i.e. when only the vertical motion is ausing the time dependant variation of the restoring moment. Assuming the spectrum of h-parameter can be written as

$$S_h(2\omega_0) = (dh/dz_{rel})^2 \cdot (TRF_{z_{rel}}(2\omega_0))^2 \cdot S_w(2\omega_0)$$

If the ISSC-wavespectrum is applied, the threshold value of the significant waveheight for parametrically excited roll motion may be written as: (The roll motion is stable (def i)) if:)

$$H_s < 0.1213 \cdot \frac{\sqrt{\rho\omega_0}}{dh/dz_{rel} \cdot TRF_{z_{rel}}(2\omega_0)} \cdot T_1^2 \cdot (2\omega_0)^{2.5} \cdot e^{\left(\frac{345.5}{((2\omega_0)T_1)^4}\right)}$$

The ship dependant parameters is found as shown in chapter 4.

In figure 7 the threshold value for parametric excitation in beam sea of roll motion is plotted with respect to zero-crossing waveperiod  $T_1$ . In the figure modeltest results are plotted and indication of whether or not the roll motion is stable (def i)) is given. In beam seas the linear (or wave moment excited) roll motion is significant and some uncertainties in the assessment of whether the roll motion is due to parametric

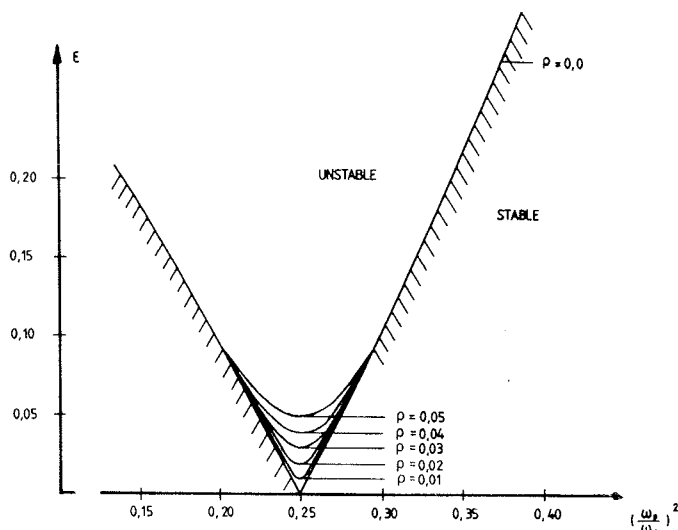


Fig. 5.

Stability chart of the Mathieu-equation with linear coefficients near  $\rho = 0.25$ . The stability borders are calculated with numerical simulation. The linear relation between exciting parameter and damping at  $\rho = 0.25$  is observed.

excitation or not, exist. In this case the roll resonance period is the double of mean waveperiod reducing the difficulty in distinguishing roll motion due to parametric excitation from linear roll motion. In fig. 8 the same function is plotted for the case of irregular following sea. Unfortunately no model test results with lower waveheights are available.

The agreement between this stability criterion and numerical simulation is generally very good as indicated in table 2. The agreement with the criterion presented by Vinje is less satisfactory.

In figure 9 a number of stability curves of the same ship is plotted varying the metacentric height in irregular beam sea at zero speed. The roll radius of gyration and displacement is kept constant. An indication of maximum possible significant waveheight with respect to wave period is given (max waveslope varying from 1/10 to 1/15).

The effect of metacentric height is seen. The threshold value seems to have a minimum value at a given metacentric height, from which the roll capability due to parametric excitation will be improved if the metacentric height is increased.

In figure 8 it is seen how the possibility for parametric excitation of roll motion is reduced when the roll damping is increased by for instance increasing the bilgekeel area. In the preceding figures a typical modern hull formed ship was used. In fig. 10 a typical traditional hull form (common oil tanker) is shown with half loaded cargo holds.

Due to large free-surfaces in cargoholds the ship has relative low GM. The large effect of draught and GM is seen, for the lightest loading condition a possibility for parametrically excited roll motion occurs, which has been experienced for a crude oil tanker loading offshore at singlepoint-mooring system. In the figure some situations are plotted where the ship experienced roll angles in the range of 10 to 20 degrees in head sea. A shift in heading reducing relative vertical motion will reduce or eliminate the problem (as has been observed in full scale).

In fully loaded condition there exist no possibility for parametric excitation of roll motion.

#### 9. PROPOSAL FOR A NEW STABILITY CRITERION.

As shown in this study some ship types are quite sensitive to parametric excitation of roll motion. For such ships this mode of rolling contributes significantly to the total roll response. Applying the methods of assessing the sensitivity of parametric excitation, which in this study are compared to model tests and numerical method, it is possible to indicate whether or not a ship is sensitive to parametric excitation. The method is described briefly in the following.

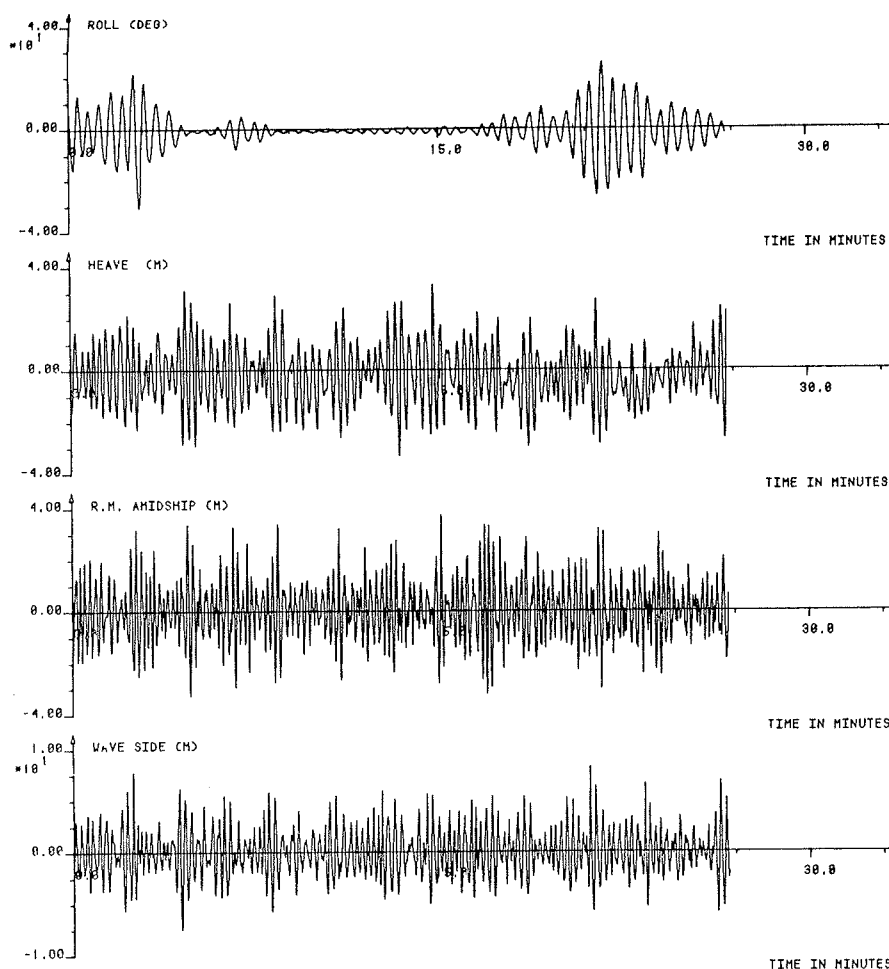


Fig. 6.

Roll motion due to parametric excitation. Irregular following sea. Medium bilgekeels. The importance of the correct tuning

between wave periods and roll periods is observed, resulting in the typical switch on/off behaviour of the roll motion.

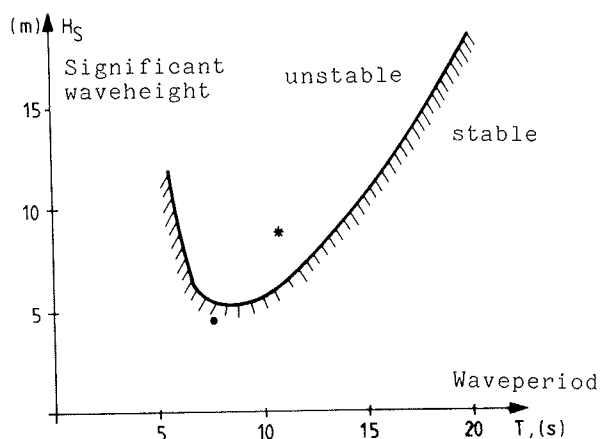


Fig. 7.

Stability criterion of Roberts /1/ compared to model test results. Irregular beam sea. Without bilgekeels.  $GM/B = 0.0434$

- \* Parametrically excited roll motion.
- No parametrically excited roll motion.

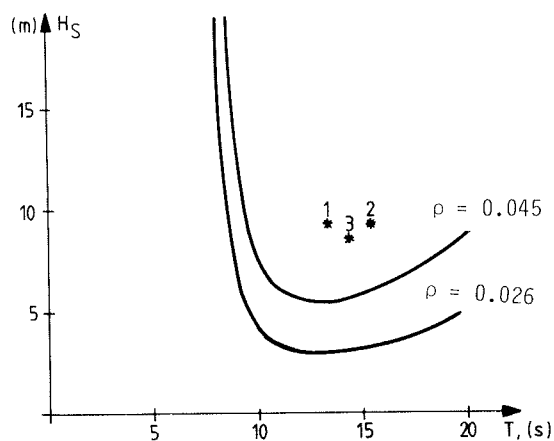


Fig. 8.

Stability criterion of Roberts compared to model test results. Irregular following sea. The effect of increased damping is seen.  $GM/B = 0.0186$

1. With medium bilgekeels,  $\rho = 0.026$
2. With large bilgekeels,  $\rho = 0.045$
3. Without bilgekeels,  $\rho = 0.014$

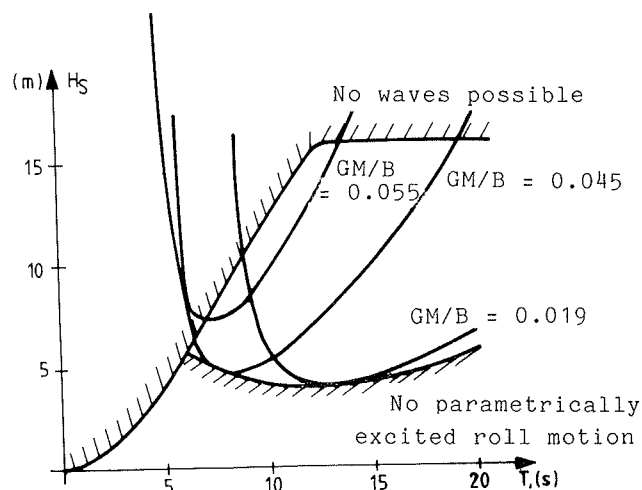


Fig. 9.

Irregular following sea, zero speed. Stability borders according to Roberts /1/ is plotted for a ship with different  $GM/B$ -ratios. The non-shaded region is where parametrically excited roll motion may occur.

Non-dimensional damping  $\rho = 0.026$

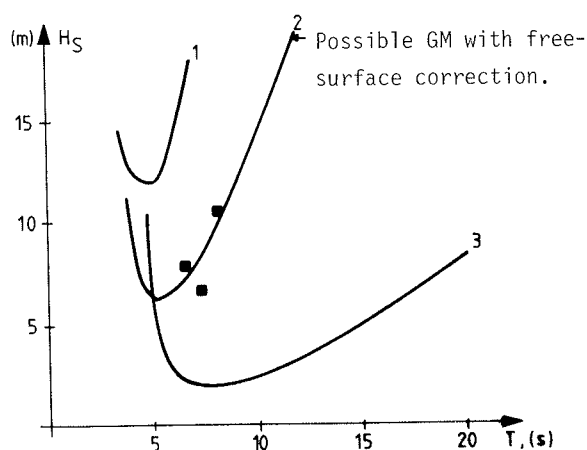


Fig. 10.

Irregular head sea, zero speed. Common crude oil-tanker, lightly loaded. The threshold value for parametric excitation of roll motion is plotted for 3  $GM$ -values.

- Results from loading operation offshore where the vessel experienced roll amplitudes in the range of 10 to 20 degrees. (waveheights and periods are visually estimated)

1. Establish design seastates and design load conditions. Calculate the natural roll frequency.

2. Calculate the Mathieu-parameter  $h$  and the slope of this function with respect to vertical relative motion. If the slope of the  $h$ -parameter is not nearly constant and have a number of local maxima or minima, the possibility for parametrically excited roll motion is reduced.

3. Calculate the transferfunction of relative vertical motion at amidship.

4. Calculate the linear damping coefficient.

5. Apply the stability criterion of Roberts.

If this method indicates that the ship may be parametrically excited, an increase in bilgekeel-area should be considered. As shown in chapter 5 a ship without bilgekeels but complying with IMCO-recommendations to stability may capsize in relative moderate seastates, however mounting bilgekeels will improve the safety of the ship considerably.

As shown in this study simple methods to deduce the sensitivity of parametric excitation for a ship exist. Model test results indicates that the method presented in this report is sufficient in the assessment of the asymptotical stability of a ship when it is parametrically excited, although more sophisticated methods could be applied. Further discussions should be made if there exist any need to add requirements to the roll damping to the IMCO-recommendations for ships which are more sensitive to parametric excitation than others. The criterion of Roberts could be applied.

Roll damping should be large enough to reduce the probability for unstable roll motion to an accepted level. This will reduce the probability for parametrically excited roll motion. This is to the author's opinion an important factor in the excitation of roll motion. For some ship-types requirements to minimum damping will improve their safety.

However, for a large number of ships in the world fleet the existing roll damping is sufficient.

## 10. CONCLUSIONS.

In this paper it has been demonstrated that some ship types in a seaway may be parametrically excited into roll motion. Such ships are characterized by a large change in water plane area and relative low GM-value when moving in waves. In addition the

roll damping has to be small which is normally the case if the ship operates without bilgekeels.

It has been shown that stability criteria given in previous papers may give a good indication whether the ship is sensitive to parametric excitation of roll motion.

Increased roll damping decreases the probability of having parametrically excited roll motion.

Believing that parametric excitation contributes significantly to the roll motion of some types of ships, a proposal for extending the existing IMCO-recommendations to stability is given. Here a requirement to roll damping is included.

Further modeltests should be carried out in order to verify the quantitative methods of calculating the exciting parameters.

A numerical simulation method which in addition to nonlinear damping, is capable to include all coupling effects should be made. This could be obtained by calculating the instantaneous centre of buoyancy and displacement force due to disturbed relative vertical motion along the ship sides.

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Table 1.

Bilge keel and damping data.  
Model experiments. Values referred  
to full scale.

	No bilgekeel	Medium bilgekeel	Large bilgekeel
B <sub>BK</sub> /B	-	0.00976	0.0190
L <sub>BK</sub> /L <sub>pp</sub>	-	0.247	0.494
$\rho$	0.014	0.026	0.045
$\rho_c$	2.99	6.56	24.2
$\omega_0$ (rad/s)	0.1970	0.1933	0.1870

Table 2.

Comparision between stability criteria  
(def i). Results from numerical  
simulation.

(dim)	Case 1	Case 2
Hs m	5.0	4.5
T1 s	9.0	9.0
$\omega_0$ rad/s	.21	.21
$S_h(2\omega_0)$ s	.16	.11
Roberts: $\pi\omega_0 S_h(2\omega_0)/4$	.027	.017
$\rho$	.026	.026
Vinje: $\sigma(h)$	.18	.12
$\sqrt{2\pi} \cdot \rho$	.046	.046
Roll motion	unstable	stable

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## Discussion

W. Abicht (University of Hamburg, FRG)

In his paper, the author points out,  
that ships having a low GM-value and a  
large change in the water plane area in a  
seaway, may be parametrically excited to  
roll motion. The possibility of capsizing  
due to this fact is well known and was  
already discussed in the first Stability

Conference (Glasgow 1975). The new investi-  
gation is to be appreciated because, up to  
now, ship designers do not observe the  
influence of hull shape on roll motion.  
This we can see when we take a look to the  
lines of modern ships. In this context I  
like to mention a paper of Dr. Burcher who  
investigated the influence of hull shape on  
transverse stability in waves. This paper



is most valuable for all designers because it shows lines which are to be preferred from the stability point of view.

Ref.: Burcher, R.K. "The Influence of Hull Shape on Transverse Stability", Trans. R.I.N.A., 1979.

Author's Reply

I want to thank Prof. Dr. Abicht for his comment on the influence of hullform

into roll motion response. I agree that this factor does not seem to be taken into consideration by recent ship designers.

As Prof. Burcher in the referred paper points out there are few problems in incorporating a less sensitive hullform with operational and design requirements. However, it seems that sufficient roll damping may cancel most of the dynamic effects due to hullforms sensitive to parametric excitation of roll motion.

*Session IIIb*

## Safety of Fishing Vessels (Part I)

*Chairmen*

Mr. Pascual O'Dogherty  
Canal de Experiencias Hidrodinámicas  
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China Ship Scientific Research Center  
China

## EFFECT OF HULL FORM AND APPENDAGE ON ROLL MOTION OF SMALL FISHING VESSEL

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University of Osaka Prefecture

Japan

### ABSTRACT

The roll characteristics of small fishing vessels of Japan are investigated. At first, the importance of the effect of sway on roll response is confirmed experimentally. The numerical solutions of the nonlinear coupled equations for sway and roll show that not only the roll damping but the added mass moment due to sway play an important role in the stability of the roll response in large amplitude.

The measurements of the roll dampings for models and full-scale vessels are carried out to reveal that the characteristics of the roll damping of such small fishing vessels are quite sensitive to the changes of hull form and the location of roll axis.

Finally, the factors affecting the roll motion and the roll damping, which are, the main hull form, the skeg, the bilge keel and the overhung deck, are investigated in detail on the basis of theoretical calculations and experiments.

### 1. INTRODUCTION

For the problem of the extreme roll and the capsize of a fishing vessel in a seaway, it is important to develop an accurate prediction method for roll motion in large amplitude. Although the ship-motion theory in six-degree of freedom based on small-oscillation assumption has fully been developed for these twenty years, the nonlinearity of the large amplitude roll motion is still remaining to be solved theoretically. Then, a single

equation for roll has often been used because of its simplicity, and a number of fruitful results have been obtained in qualitative sense. For a more detailed and quantitative analysis, however, the treatment involving coupling effects seems to be necessary. In this paper, the authors will begin the discussions with showing the importance of the coupling effect of sway on roll response and emphasize the necessity of taking the coupling effect into the roll calculation. A calculation method of the nonlinear coupled equations for roll and sway will also be introduced briefly.

For a good prediction of a ship motion, it is also necessary to obtain the accurate values of the hydrodynamic forces. The prediction of the roll motion of a small fishing vessel, however, suffers from the lack of knowledge on the hydrodynamic forces acting on the vessel, particularly on the roll damping. This is probably partly because the interest of naval architects and researchers has mainly been concentrated on large ships and partly because the wide variety of hull forms of such small vessels prevents a systematic study on the hydrodynamic forces. In the next part of this paper, attention will mainly be focused on the roll damping and the wave exciting moment, and a systematic study on the several factors of the hull configuration affecting those quantities will be made individually on the basis of theoretical calculations and experiments in order to obtain the knowledge of the hydrodynamic forces acting on small fishing vessels.

## 2. ROLL CHARACTERISTICS OF FISHING VESSEL

### 2.1. ROLL MOTION IN BEAM SEA

The resonance roll amplitude is one of the important factors on considering the ship capsize and the stability criterion.

The roll-motion equation of one degree of freedom has so far been used for the study of the roll motion in large amplitude because of its simplicity. However, the modern ship-motion theory shows that the coupling effects can not be ignored, particularly the coupling effect of sway into roll [1][2].

The measured responses for a model of a small fishing vessel in beam sea are shown in Fig.1. The roll amplitude in the case of fixed sway is about twice the value in the case when the sway is made free. The other coupling effects also appear in the sway and the heave amplitudes which have a weak peak respectively at the roll resonance frequency. The result suggests that the coupling effect of sway into roll plays an important role in the prediction of the roll motion.

Tasai et.al.[1] showed that in the

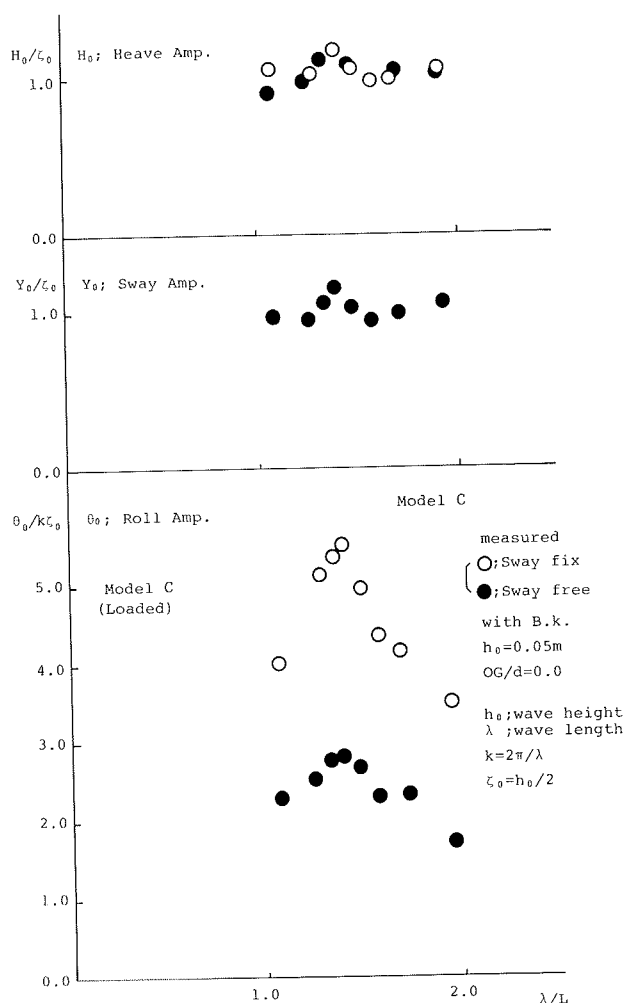


Fig. 1 Ship motions in a regular beam sea.

case of long wave-length compared with ship beam the coupling roll moment due to sway is approximately cancelled by the diffraction component of the wave exciting moment and that only the Froude-Krylov force remains as the wave exciting moment. In other words, we may use a single roll equation as a first approximation for long wave-length by taking only the Froude-Krylov force as the the wave exciting moment. For large-amplitude motion, however, we should treat the coupled equations in the quantitative analysis. The same statement is true for the situation that the coupling effect of roll on other motions appears as in the case of a small fishing vessel shown in Fig.1.

The following nonlinear coupled equation system is used for the calculation of the roll and the sway of a fishing vessel in beam waves,

$$\begin{aligned} A_{22}\ddot{y} + B_{22}\dot{y} + A_{24}\ddot{\theta} + B_{24}\dot{\theta} &= F_S \\ A_{44}\ddot{\theta} + B_{44}\dot{\theta} + F(\theta) + A_{42}\ddot{y} + B_{42}\dot{y} &= M_R \end{aligned} \quad (1)$$

where  $y$  and  $\theta$  denote the sway displacement and the roll angle respectively, and  $F_S$  the sway exciting force and  $M_R$  the roll exciting moment. In this calculation, the nonlinearities of the restoring moment  $F(\theta)$ , the roll damping  $B_{44}$ , the sway damping  $B_{22}$ , the coupling damping of roll into sway  $B_{24}$  and the coupling damping of sway into roll  $B_{42}$  are taken into account. The GZ curve shown in Fig.2 is represented by an odd-power polynomial up to the order of  $\theta^5$ , and the nonlinear dampings are replaced by the equivalent linearized forms with the amplitude-dependent coefficients. To solve the nonlinear system, a perturbation method in a similar manner to the Wright's method [3] is used. A typical outcome in Fig.3 shows that the roll response in a regular beam sea calculated by using measured hydrodynamic forces is stable. When either  $B_{44}$  or  $A_{42}$  is halved, the resonance roll amplitude increases. Moreover, if both of them are halved simultaneously, the roll and the sway responses show an unstable behaviour as shown by the dotted line in

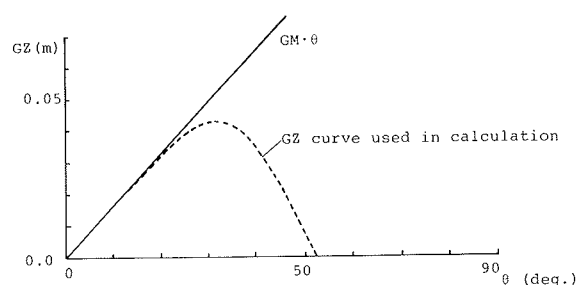


Fig. 2 GZ curve of Model C with low freeboard.

Fig.3. On the contrary, the contribution of  $B_{42}$  to the stability of roll response is negligible because the phase difference between roll and sway is about  $\pi/2$  at the roll resonance. The result shown in Fig.3 suggests that the values of  $B_{44}$  and  $A_{42}$  have significant effects on the stability of extreme roll response.

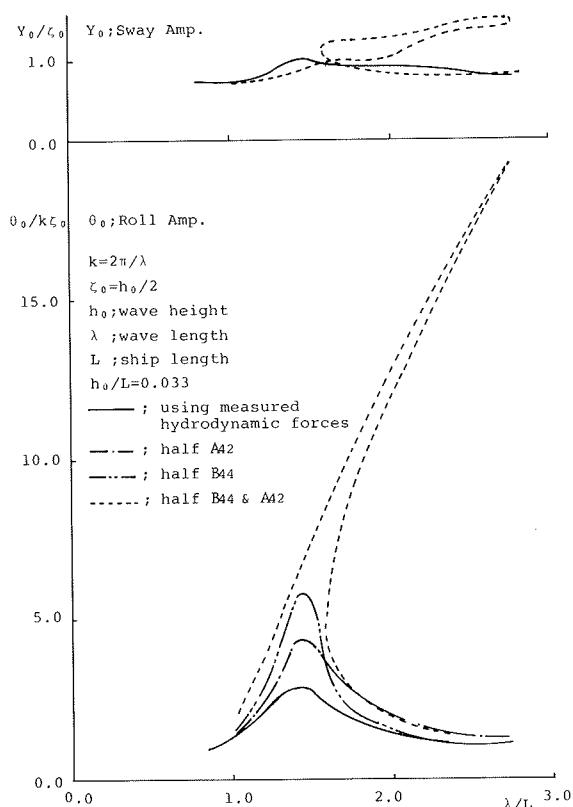


Fig. 3 Numerical solutions of coupled system of non-linear equations of roll and sway for model C with low freeboard.

## 2.2. ROLL DAMPING

It is necessary for an accurate prediction of roll motion to estimate hydrodynamic forces or coefficients of equations of motions accurately. However, the estimation of roll damping is quite difficult because it is considerably affected by the fluid viscosity. In this section, the characteristics of the roll damping of small fishing vessels are discussed through the experiments of models and full-scale vessels.

Prior to the detailed analysis, we should briefly comment on the roll-damping components and on the relationship between roll damping and wave exciting moment. The roll damping of a small fishing vessel can be assumed to consist of six components as follows,

$$B_{44} = B_W + B_F + B_E + B_L + B_{BK} + B_{SK} \quad (2)$$

where  $B_{44}$  denotes the coefficient of total roll damping,  $B_W$  the component due to wave,  $B_F$  friction,  $B_E$  eddy for a naked hull,  $B_L$  lift at forward speed,  $B_{BK}$  bilge keel and  $B_{SK}$  skeg. In these damping components, the wave component is closely related with the wave exciting moment about roll axis, which is well-known as the Haskind-Newman relation for a two-dimensional cylinder. The relationship says that the more the wave component of roll damping increases, the more the wave exciting moment increases. Generally speaking, the combination of small wave component and large other components is, therefore, favorable for the reduction of roll amplitude.

Four models, the body plans of which are shown on Fig.4, are tested to obtain the roll damping by a forced roll oscillation. The results are shown in Figs.5 thru 8. The measured roll damping for Models B and C of naked hull shown in

Table 1 Principal particulars and test conditions of small fishing vessels.

	Model A*		Model B*		Model C						Model D					
					Full-scale		Ballast		Loaded		Full-scale		Ballast		Loaded	
L (m)	1.668		1.60		16.8		1.527		1.527		13.3		1.33		1.33	
B (m)	0.374		0.339		3.7		0.336		0.336		3.0		0.3		0.3	
d <sub>m</sub> (m)	0.084		0.097		0.84		0.076		0.092		0.625		0.0625		0.0775	
V (m <sup>3</sup> )	0.03		0.0378		30.18		0.0229		0.0291		12.03		0.01203		0.0167	
Trim (m)	0.0		0.116		0.748		0.068		0.156		0.406		0.0406		0.0406	
CB	0.684		0.67		0.49		0.49		0.574		0.494		0.494		0.54	
Cm	0.821		0.85		0.78		0.78		0.886		0.7		0.7		0.73	
B.K. lxb (m <sup>2</sup> )			0.96x0.02 s.s. 1.5~7.5		5.4x0.1 s.s.2.5~5.5		0.46x0.009 s.s.2.5~5.5		0.46x0.009 s.s. 2.5~5.5							
test con.																
	OG/d	GM(m)	OG/d	GM(m)	OG/d	GM(m)	OG/d	GM(m)	OG/d	GM(m)	OG/d	GM(m)	OG/d	GM(m)	OG/d	GM(m)
	0.0	0.1097	-0.823	0.0169	-0.625	1.18	-0.625	0.107	0.0	0.089	-0.704	1.24	-0.384	0.144		
	0.238	0.0879					-0.686	0.102					-0.704	0.124		
						$\hat{\omega}_0=0.738$							$\hat{\omega}_0=0.850$			

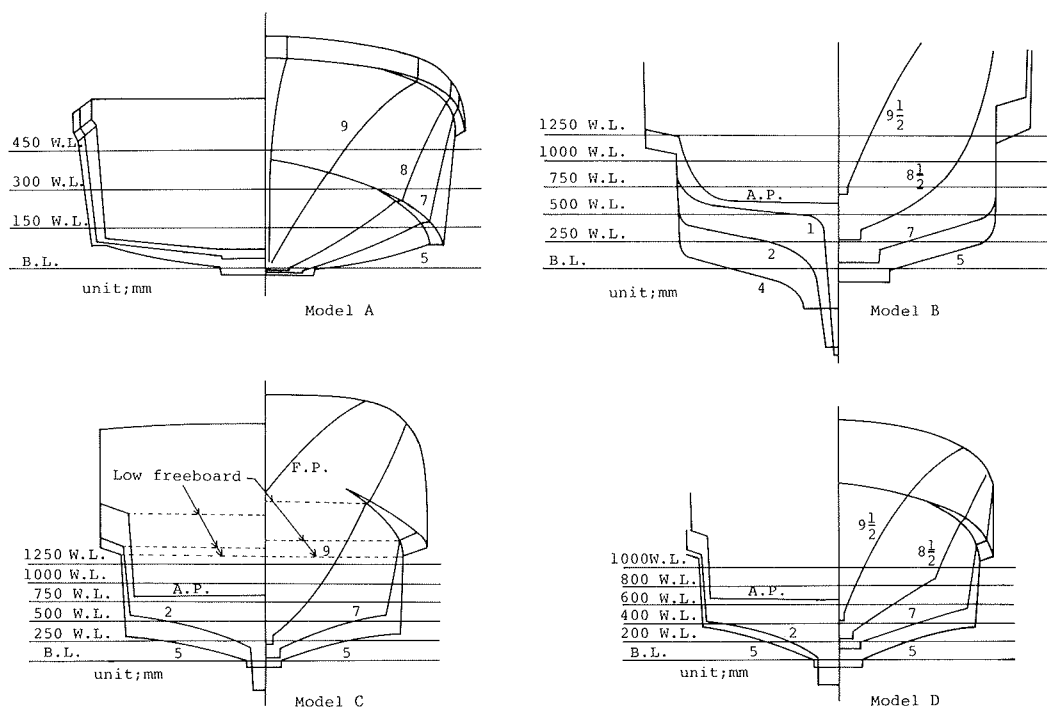


Fig.4 Body plans of small fishing vessels.

Figs.6 and 7 is much larger than the wave damping component calculated by the source distribution method. As well-known, this deviations are due to viscous effect, and will increase if bilge keels are fitted. On the contrary, the results for Model A and D are in good agreement with the calculated wave damping as shown in Figs.5 and 8 in spite of the hard chine forms. It is incomprehensible why the wave damping is dominant and the eddy damping is slight for such vessels with hard chine.

Figs.9 and 10 show the experimental results plotting the roll damping for various locations of the center of gravity or the roll axis. With lowering of the roll axis, the total roll damping gradually decreases, and the ratio of the wave damping to the total increases. At low roll axis, the wave damping is dominant.

It can be said that the characteristics of the roll damping of small fishing vessels are quite sensitive to the changes of hull form and the location of roll axis.

Finally, we discuss on the forward speed effect and the scale effect of roll damping. Figs.11 and 12 show the measured roll damping coefficients of the models and the full-scale vessels. The roll damping increases with forward speed as is generally known. The dotted line in Fig.11 is the measured lift component by the

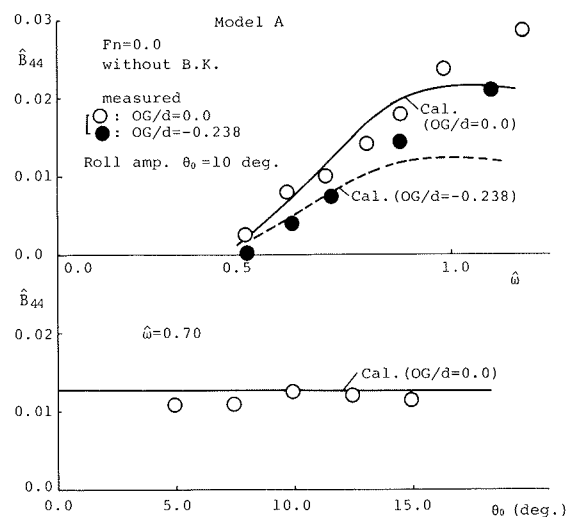


Fig. 5 Roll damping coefficient  $\hat{B}_{44}$  for model A.

free roll test at low frequency [4]. As the lift component is almost parallel to the measured one at high speed, the main part of the increase of the roll damping at forward speed is due to the lift component.

The measured roll dampings for full-scale vessels is in fairly good agreement with those for models as shown in Figs.11 and 12. Then, it can safely be said that the scale effect of the roll damping coefficient for such a small fishing vessel is negligible in practical use.

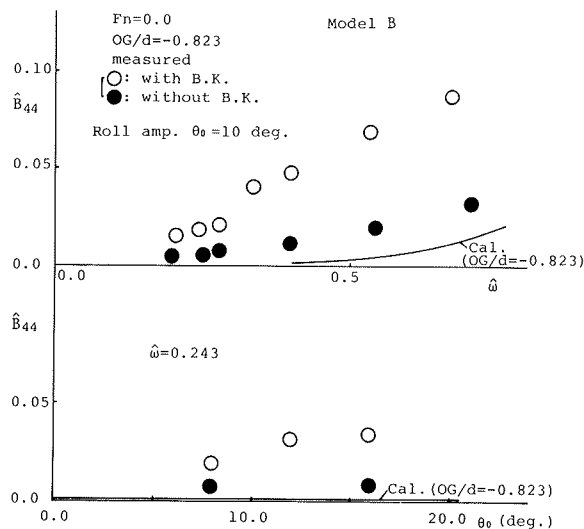


Fig. 6 Roll damping coefficient  $\hat{B}_{44}$  for model B.

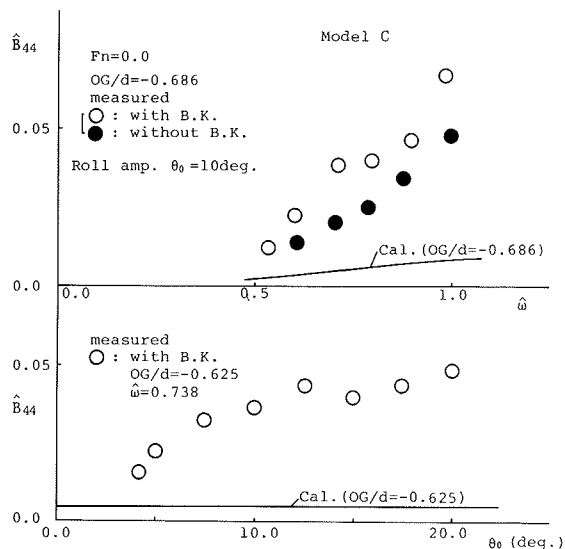


Fig. 7 Roll damping coefficient  $\hat{B}_{44}$  for Model C.

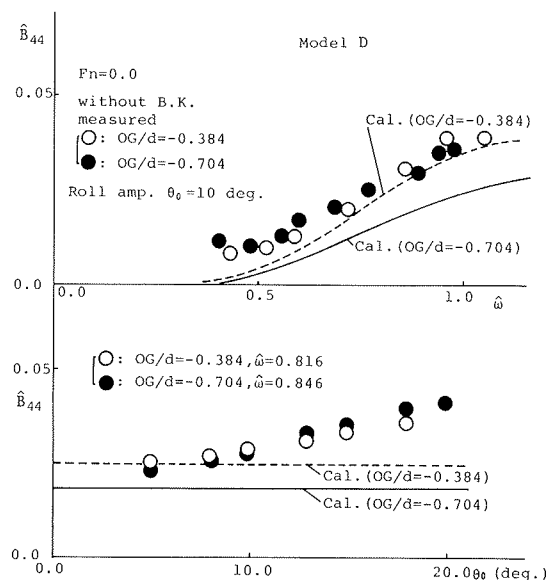


Fig. 8 Roll damping coefficient  $\hat{B}_{44}$  for Model C.

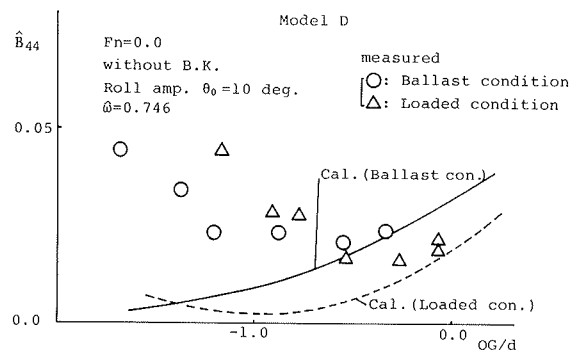


Fig.9 Effect of location of roll axis on roll damping for Model D.

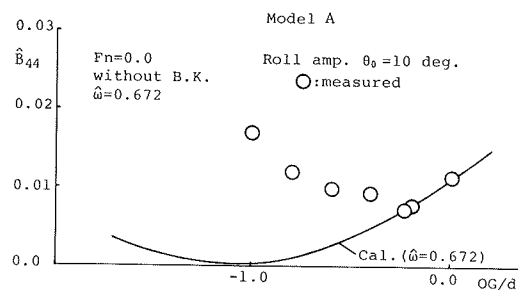


Fig.10 Effect of location of roll axis on roll damping for Model A.

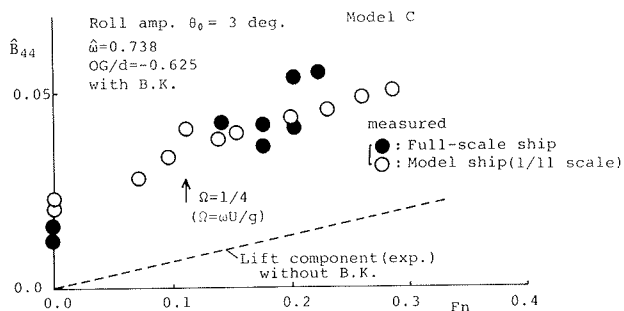


Fig. 11 Forward speed effect on roll damping for Model C and similar full-scale vessel.

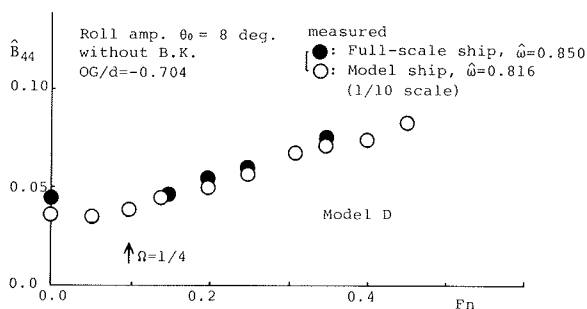


Fig. 12 Forward speed effect on roll damping for Model D and similar full-scale vessel.

### 3. FACTORS AFFECTING ROLL DAMPING AND ROLL MOTION

In this chapter, the detailed analysis on the factors affecting roll damping and roll motion of a small fishing vessel are made mainly on the basis of theoretical calculations and experiments for two-dimensional cylinders.

#### 3.1. MAIN HULL FORM

At first, the effect of main hull form on the roll damping are discussed. The distinctive feature of the main hull form of a small fishing vessel of Japan may consist of flat hull shape, large rise of floor and hard chine.

Generally speaking, a flat hull causes large wave roll damping. Fig. 13 shows the calculated radiation wave amplitude created by roll motion of Lewis form cylinders with unit length, unit cross sectional area and different ratios of beam to draft (\$H\_0\$). In the calculation, the roll frequency is kept to be constant. In the figure, the radiation wave amplitude becomes minimum at \$H\_0 = 1.2-1.5\$. As the value of \$H\_0\$ for most general cargo ship forms lies almost in this region, they have small wave component. In the region over this minimum point, the radiation wave increases with \$H\_0\$, so that the flat fishing

vessel form has a comparatively large wave damping. The calculated results of the wave component for different rise of floor in Fig. 14 show that the wave component increases with the rise of floor. Thus, a small fishing vessel with large rise of floor has a tendency to have large wave damping and it means that it also has large wave exciting moment.

However, we would get a smaller wave damping component for such a vessel by changing the location of the center of gravity, or the roll axis. Figs. 15 and 16 show the calculated wave damping versus \$OG/d\$, where \$OG\$ denotes the distance between

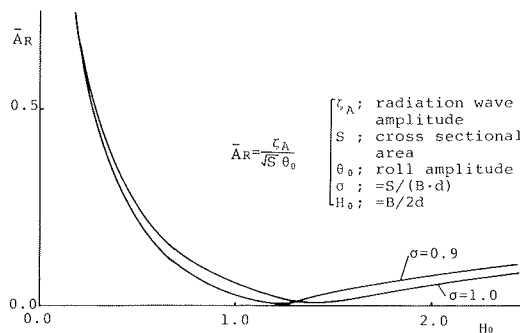


Fig. 13 Radiation wave amplitude due to roll motion for two-dimensional Lewis form cylinder with the same cross sectional area and with different \$H\_0\$.

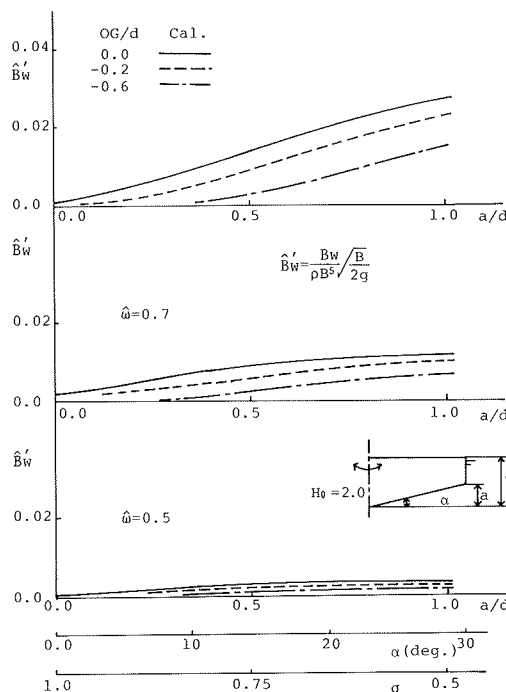


Fig. 14 Effect of rise of floor on wave roll damping for two dimensional cylinders.



the center of gravity and still water level, taking positive downward. These figures indicate that the location of the center of gravity at minimum wave damping is raised with the increase of  $H_0$  and the decrease of the area coefficient  $\sigma$ . Delicate care, however, would be necessary on raising the center of gravity to reduce the wave damping because that would also reduce the static stability of the vessel.

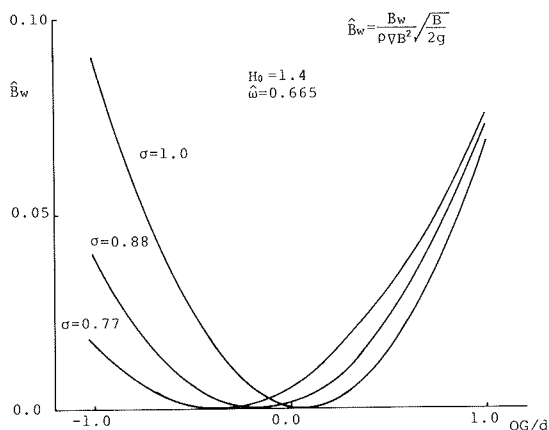


Fig. 15 Effect of location of roll axis on wave roll damping for cylinder with  $H_0=1.4$ .

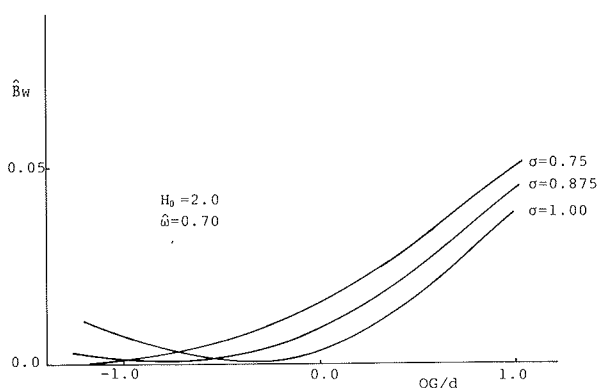


Fig. 16 Effect of location of roll axis on wave damping for cylinder with  $H_0=2.0$ .

Subsequently, we consider the variation of the eddy making component of roll damping with the change of main hull shape. Hard chine or sharp bilge, which is common to modern small fishing vessels in Japan, is expected to increase the roll damping by generating eddies. The experimental results in Fig. 17 show that the roll damping of a hard chine vessel is about twice the value for a

round-bilge vessel.

The effect of rise of floor on the eddy component of roll damping is investigated by experiments. The measurements of roll damping of cylinders with three kinds of rise of floor were carried out for two different drafts. The results are shown in coefficient of eddy component defined by the form,

$$C_R = \frac{2B_E}{\rho B^4 L \theta_0 \omega} \quad (3)$$

where  $B_E$  is obtained by subtracting the calculated wave component from the measured one,  $B$  the beam,  $L$  the length,  $\theta_0$  the roll amplitude,  $\omega$  the circular frequency and  $\rho$  the fluid density. The coefficient  $C_{R0}$  in the same figure represents the value of  $C_R$  without rise of floor. In the figure, we can see that  $C_R/C_{R0}$  decreases rapidly with increase of the rise of floor. The decreasing rate becomes large as  $H_0$  increases. This is probably partly because of the reduction of the moment lever and partly because of the reduction of the generated eddies due to the attenuation of the fluid velocity at the bilge. Although free surface may also have an effect on the eddy damping when the bilge is near the free surface, there are few studies on the effect [5].

A summary of the results obtained in this section is as follows. The small fishing vessel with flat hull form has a comparatively large wave damping which increases with the rise of floor. The eddy damping created by the hard chine increases roll damping significantly, but decreases with the rise of floor.

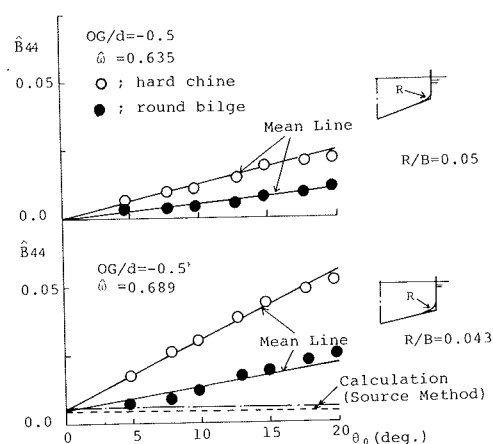


Fig. 17 Difference of roll damping between hard chine and round bilge cylinders.

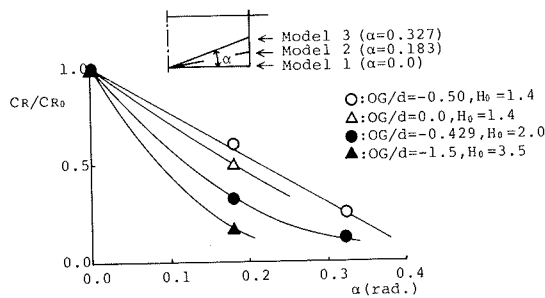
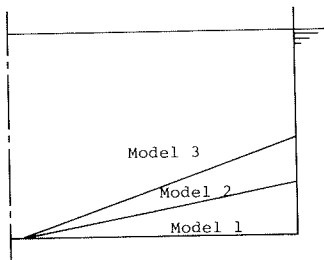


Fig. 18 Effect of rise of floor on eddy component of roll damping for two-dimensional cylinder.

Table 2 Principal particulars of cylinders.

	Model 1	Model 2	Model 3
L (m)	0.8	0.8	0.8
B (m)	0.28	0.28	0.28
d (m)	0.10	0.10	0.10
V (m <sup>3</sup> )	0.0224	0.0197	0.0173
H <sub>0</sub>	1.4	1.4	1.4
sigma	1.0	0.8795	0.7703



### 3.2. SKEG

Most of small fishing vessels have skeg and bar keel to improve their manoeuvrability performance and for the convenience when they are landed on a shipway. The skeg also affects roll damping as well as bilge keels.

At first, attention is focused on the wave roll damping. The calculated results of the wave component in Fig.19 show that the skeg installed reduces the wave component. This is possibly due to the situation that the phase of the wave created by the skeg is much different from that of the wave created by the main hull for such a flat-shape hull. It is thus possible to reduce the wave component of roll damping by fitting an appropriate skeg.

Skeg increases the viscous roll damping by creating eddies at its edge. As well as the bilge keel component, the skeg component due to making eddies divided into two components. One is the normal force component which is created by the pressure variation on a skeg, and the other is the hull surface pressure component which is created by the pressure variation on the

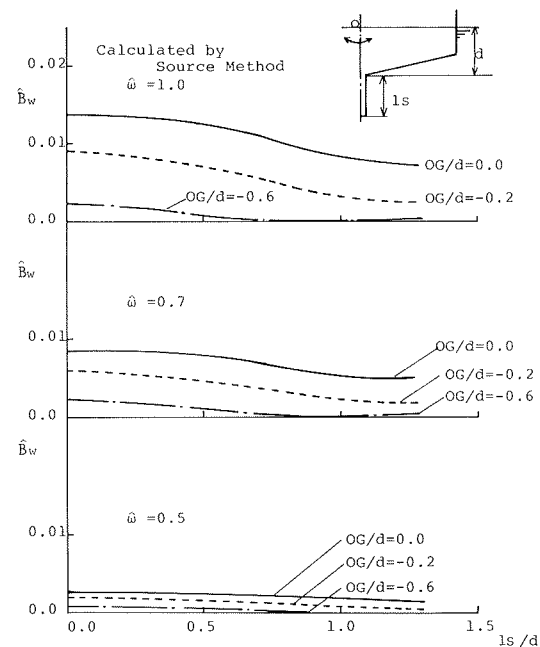


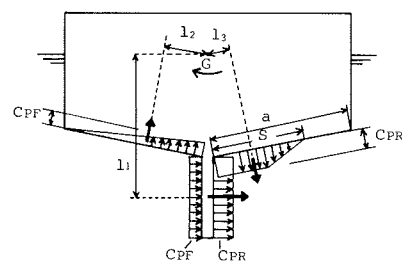
Fig. 19 Effect of skeg on wave damping.

main hull surface due to the skeg. The normal force component is always positive damping, while the surface pressure component is not always so. Actually, it becomes negative for a flat bottom ship.

The authors propose a simple prediction method of the eddy component of roll damping due to skeg as follows. A simple pressure distribution on the skeg and on the bottom of a vessel is assumed as shown in Fig.20. The pressure coefficient  $C_{PF}$  and  $C_{PR}$  on the front and the back faces of the skeg, and the length of the negative pressure region  $S$  is assumed on the basis of experimental results [4] as follows.

$$\begin{aligned} C_{PF} &= 1.2 \\ C_{PR} &= -3.8 \\ S &= 1.65 l_s K_c^{2/3} \end{aligned} \quad (4)$$

where  $l_s$  denotes the length of a skeg,  $K_c$



$$C_{PR} = \frac{PR}{\frac{1}{2} \rho U^2}$$

→: resultant force

Fig. 20 Assumed pressure distribution created by skeg.

is Keulegan-Carpenter number defined as  $U_{max} \cdot T / l_s$ , where  $U_{max}$  denotes the maximum speed of the edge of the skeg,  $T$  the period of roll motion. Although the value of  $C_{PR}$  depends on  $K_c$  number in a strict sense, it is assumed to be constant for sake of simplicity. Integrating the assumed pressure on the skeg and on the hull surface, the roll damping moment  $M_r$  for unit length of the cylinder can be obtained as follows.

$$M_r = \frac{1}{2} \rho U^2 \left\{ (C_{PF} - C_{PR}) l_s l_1 - \frac{1}{2} C_{PF} a l_2 + \frac{3}{4} C_{PR} S l_3 \right\} \quad (5)$$

where  $U$  denotes the velocity of the edge of the skeg, and  $l_1$ ,  $l_2$  and  $l_3$  the moment levers as shown in Fig.20. The agreement between the present prediction and experiments for two-dimensional cylinders seems to be rationally good as shown in Fig.21. Fig.22 shows how the roll damping changes with the length of the skeg and with the rise of floor. As shown in the figure, the roll damping increases with the length of the skeg and the increasing rate decreases with the reduction of the rise of floor. This is because the lever of the negative damping moment on the bottom decreases with the increase of the rise of floor.

The results obtained for the skeg can be summarized as follows. The skeg reduces

the wave roll damping and increases the eddy damping. The skeg component of roll damping increases with the increase of rise of floor. A prediction method of the skeg component is proposed.

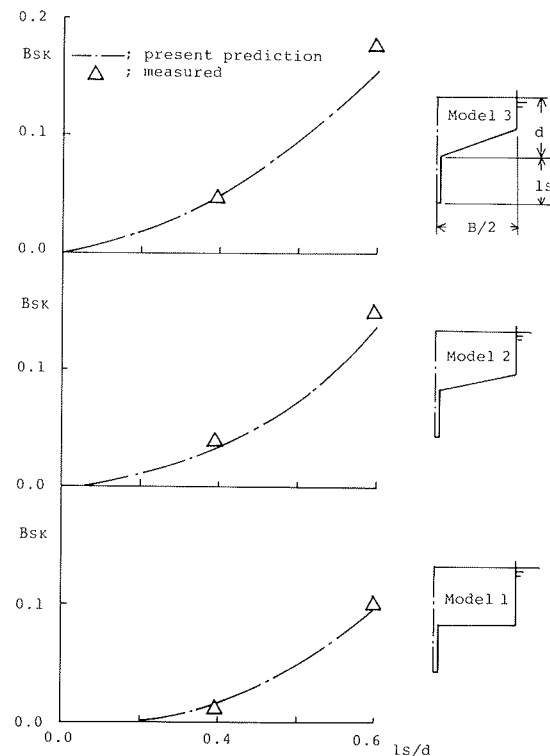


Fig.22 Effect of skeg length and rise of floor on eddy damping due to skeg.

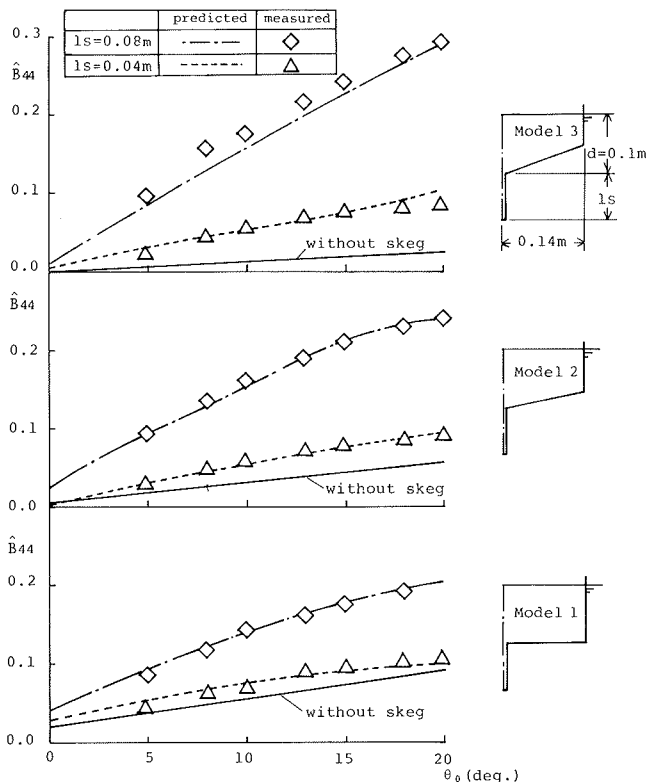


Fig. 21 Comparison between predicted and measured eddy damping due to skeg.

### 3.3. BILGE KEEL

Bilge keel is a simple and effective roll reduction tool, and is fitted on most of ships. Particularly, it is suitable for a small vessel because other roll reduction tools like a fin stabilizer and an anti-roll tank would require a wide space.

For a round bilge vessel, it is necessary to install bilge keels because the roll damping is about half of that of a hard chine vessel as shown in Fig.17. For a hard chine vessel, bilge keels are also effective as shown in Fig.7. Particularly, since the roll damping of the naked hull of a hard chine vessel with large rise of floor is small as shown in Fig.17, bilge keels are necessary for such a vessel.

The practical effect of bilge keels with different fitting angles was examined for a hard chine vessel (Model A) and the results are shown in Fig.23. The best fitting angle is 45 degree as shown in the figure, and the roll amplitude at the resonance in the presence of bilge keels is about half of that without bilge keels.

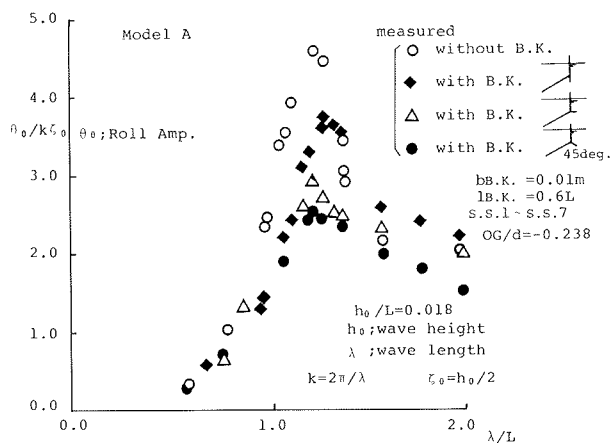


Fig. 23 Effect of fitting angle of bilge keels on roll motion of Model A in regular beam sea.

The effect of the bilge keels remains at forward speed as shown in Figs. 24 and 25 being analogous to the case of general cargo ships [6].

As mentioned above, the bilge keel is a powerful roll reduction tool even for a hard chine vessel with and without advanced speed. The rational prediction method of the bilge keel effect, however, has not been derived yet for this kind of small fishing vessel forms with hard chine.

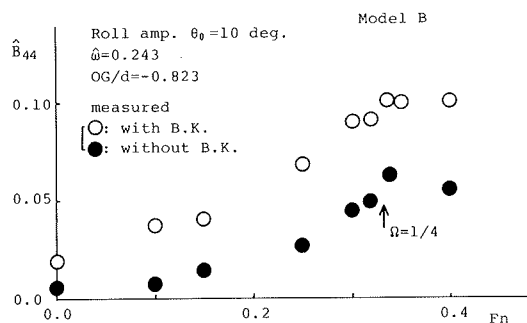


Fig. 24 Forward speed effect on bilge keel component of roll damping.

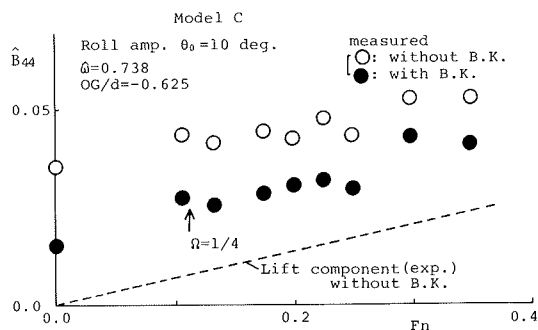


Fig. 25 Forward speed effect on bilge keel component of roll damping.

### 3.4. OVERHUNG DECK

Many Japanese small fishing vessels have an overhung deck which is a deck wider than the beam of a ship, and have a high bulwark at the edge of the overhung deck. The overhung deck gives large righting moment at large amplitude of roll motion like a flare for pitching motion. Therefore, the overhung deck is said to contribute to the safety of small fishing vessels in a heavy sea.

The discussions are made on the effects of overhung deck on the roll damping, the wave exciting moment and finally the roll motion respectively.

The roll dampings of a two-dimensional cylinder with and without overhung deck were measured by forced roll test. As shown in Fig. 26, the overhung deck makes the roll damping increase at large roll amplitude. For the actual ship form, however, the effect of the overhung deck is weakened due to the deck shear as shown in Fig. 27. As an extreme case, no effect of an overhung deck on the roll damping is recognized for a severely sheered vessel shown in Fig. 28. This is probably because only the midship part of the overhung deck runs into the water at the roll angle experimented.

The wave exciting moment acting on a fishing vessel in waves also increases by overhung deck as shown in Fig. 29. In the test, the amplitude of roll exciting moment acting on a two-dimensional cylinder with and without overhung deck fixed in regular beam seas was measured.

Fig. 30 shows the measured roll amplitude of a fishing vessel (Model C) in regular beam waves. The effect of the

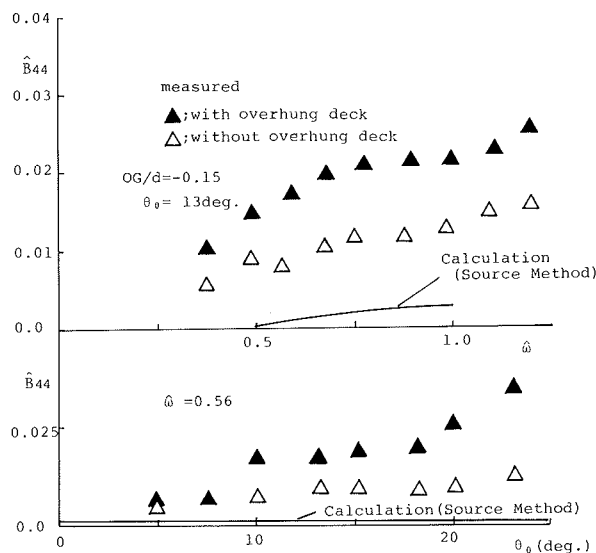


Fig. 26 Effect of overhung deck on roll damping for two-dimensional cylinder.

overhung deck on the roll amplitude of a ship is not so significant. This is probably because the increases of the roll damping and the wave exciting moment due to the overhung deck compensates each other at the roll amplitude experimented.

To summarize the results on the effect of the overhung deck, we can say that the effect of overhung deck on the reduction of the resonance-roll amplitude is not so significant though it works as a reserved buoyancy in large roll amplitude.

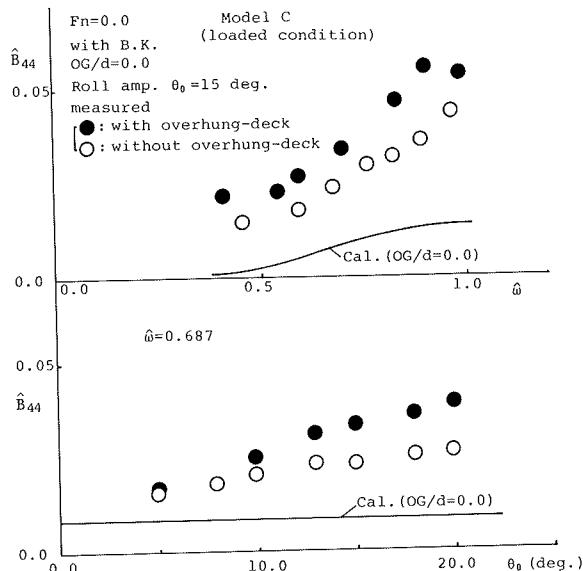


Fig.27 Effect of overhung deck on roll damping for Model C.

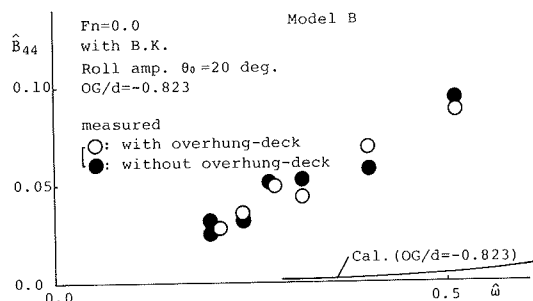


Fig.28 Effect of overhung deck on roll damping for Model B.

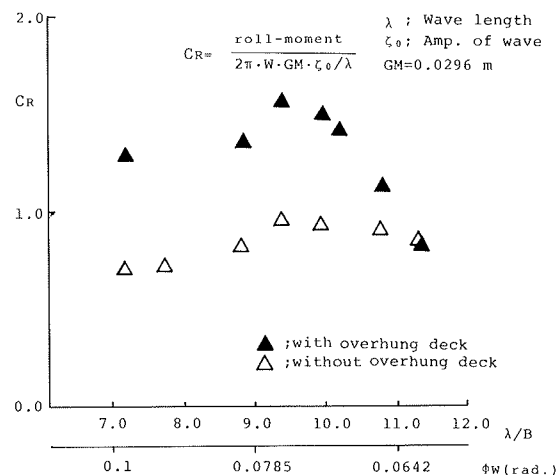


Fig. 29 Effect of overhung deck on wave exciting moment for two-dimensional cylinder.

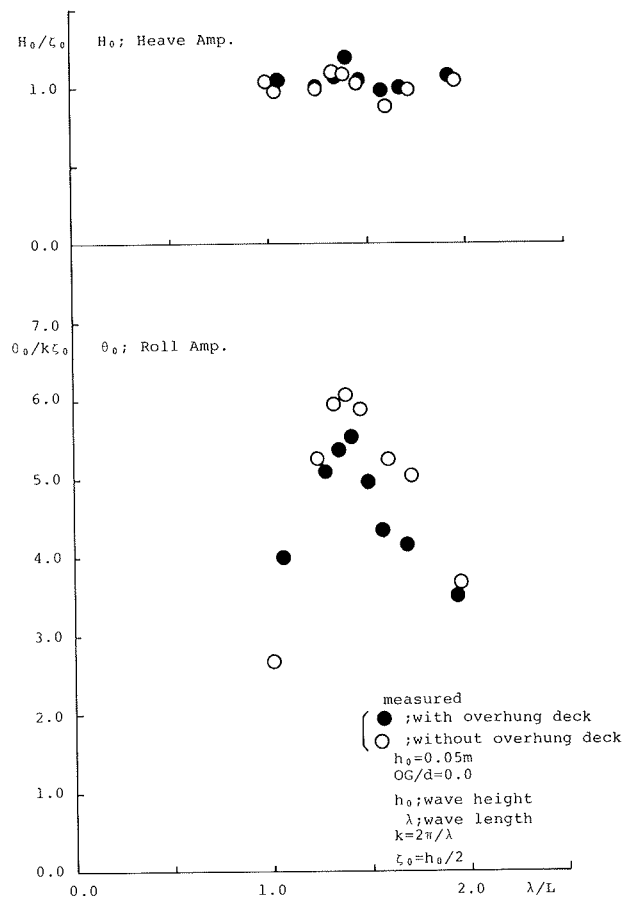


Fig.30 Effect of overhung deck on roll response of Model C in regular beam sea.

#### 4. CONCLUSIONS

Through the discussions on the roll motion and the roll damping of small fishing vessels of Japan, the following conclusions are obtained.

- 1) As the effect of sway on roll motion is significant, the coupled equations for sway and roll should be used in large amplitude.
- 2) By the numerical solutions of the nonlinear coupled equations for sway and roll, it is confirmed that the roll damping  $B_{44}$  and the added mass moment due to sway  $A_{42}$  play important roles in the stability of the roll response in large amplitude.
- 3) It is found that the ratio of the wave damping to the total roll damping depends significantly on the hull form and the location of roll axis.
- 4) The scale effect of the roll damping coefficient can be neglected for such a small fishing vessel in practical use.

The factors affecting the roll motion and the roll damping are investigated in detail, and the results obtained can be listed as follows.

- 5) The flat hull form of a small fishing vessel yields a large wave damping. This component increases with the amount of rise of floor, and can be minimized by changing the location of roll axis.
- 6) The hard chine at the bilge increases the roll damping by generating eddies. The eddy component is reduced by increasing the rise of floor.
- 7) The skeg makes the wave roll damping decrease. This is because the phase of the radiation wave created by the skeg differs from that by the main hull.
- 8) The eddy roll damping due to skeg consists of the normal force component and the hull pressure component. The former always works as a positive damping, while the latter sometimes works as negative. A prediction method of the skeg damping is proposed. The damping increases also with the rise of floor for such a flat vessel.
- 9) The overhung deck makes both roll damping and wave exciting moment increase.

Then, the reduction of the roll amplitude at roll resonance is not so significant.

#### 5. ACKNOWLEDGEMENT

The present study has been carried out as a part of RR17 Committee of the Shipbuilding Research Association of Japan. The authors would like to express their gratitude to the committee members for many fruitful discussions. The authors also wish to thank Mr. H.Okada and Mr. Y.Hada, postgraduate students of University of Osaka Prefecture, for their help on the numerical calculations and the experiments.

A part of this study was supported by the Grant-in-Aid for Scientific Research of the Ministry of Education, Science and Culture, Japan.

The computer system ACOS-700 at University of Osaka Prefecture was used for the numerical calculations.

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### Discussion

D. Vassalos (University of Strathclyde, UK)

I would like to take this opportunity to thank all three authors not only for this paper but also for a series of excellent papers on roll damping over the past few years on which we rely heavily in our research.

Much as I appreciate the role of damping on roll motion in particular, I feel that in many cases this role is overplayed and this results in omitting

another or a number of other important parameters. For example, it is a known fact that all the factors considered in the paper namely hull form, skegs, bilge keels and large roll amplitudes affect roll added moment of inertia, and hence roll motion, greatly. Yet this effect is totally overlooked in this paper and I would very much appreciate the comments of the authors on this point.

Coming next to the importance of coupling I noted that although the effect of

sway onto roll has been repeatedly emphasised nothing has been mentioned about the coupling effect of roll into sway. Is this effect equally important? And if not, why, we coupled equations at all as in this case the coupling effect of sway onto roll can be considered as an excitation term in a single degree of freedom roll equation.

It is interesting to point out that Loukakis et al. have shown by an order of magnitude approach that coupling of roll into sway is a second order quantity and he referred to rolling as a semi-coupled motion.

The view of the authors on this very interesting problem will be greatly appreciated.

#### Author's Reply

Thank you very much for your discussion.

On the added moment of inertia, the viscous effect is not so great except in the case of large size bilge keel, so that it can be predicted by potential-flow calculation. For a conventional cargo ship with normal size bilge keels, however, the effect of the added moment of inertia on roll motion is not so significant according to our experiences.

The coupling effects between roll and

sway are very important factors on considering ship motion. The effect of sway into roll is particularly significant as the authors mentioned in this paper. On the contrary, since the effect of roll into sway is not so significant usually for a conventional cargo ship in full load condition, the semi-coupled motion analysis, which was proposed by Prof. Tasai and Dr. Loukakis, is a good method as a first approximation. According to our experiences, however, there are several examples which suggest the importance of the effect. Unfortunately, we cannot make any definite conclusion on that point.

E. Dahle (University of Fisheries,  
Norway)

The paper contains interesting information on roll damping as well as coupling effects between roll and sway, thus confirming that damping is important also from a reply point of view. It would be interesting if the authors could also indicate how the various coefficients in Eq.(1) are arrived at, especially the coupling coefficients  $A_{24}$ ,  $B_{24}$ ,  $A_{42}$  and  $B_{42}$ . Possibly, also more numerical values could be given in order to facilitate calculation by other methods as the one used by the authors.

## SOME TESTS ON STABILITY OF FISHING VESSELS

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### ABSTRACT

The aim of this research was to find whether the stability of fishing vessels can be appraised in a better way by adding to the usual geometric calculations, the consideration of other factors.

In the scope of this paper no theoretical work is included but only the results of some tests carried out with the models of three ships of different types. These ships are: a side-or beam- trawler, a stern trawler and a tunna boat.

The tests consisted in: extinction tests to get the damping coefficient, both at  $V = 0$  and at the speed assumed; tests with an initial heel, in still water and with following waves; tests with a mechanical oscillator at  $V = 0$  and the speed assumed, again; and rolling tests on beam waves.

The results of these tests are shown with comments in the paper. At the end some conclusions are tentatively provided for designers.

### NOMENCLATURE

$A_R$  .. Rudder Area  
 $B$  .. Beam of ship  
 $b$  .. damping coefficient  
 $C_B$  .. Block coefficient  
 $GM$  .. Metacentric height  
 $GZ$  .. Righting arm  
 $H$  .. Depth of ship  
 $J$  .. Moment of Inertia, with added mass  
 $k$  .. wave number  $2\pi/\lambda$   
 $L$  .. Length of ship  
 $M$  .. Moment, exciting  
 $M_a$  .. Amplitude of exciting moment  
 $n$  .. damping coefficient  $n = b/2J$

$T$  .. Draft of Ship  
 $V$  .. Speed of Ship  
 $\alpha$  .. wave slope, corrected:  $2\pi\zeta/\lambda \exp(-kT)$   
 $\Delta$  .. Displacement  
 $\phi$  .. Roll angle  
 $\phi_0$  .. heel angle in still ship  
 $\zeta$  .. wave amplitude  
 $\Lambda$  .. Tuning factor  
 $\lambda$  .. wave length  
 $\omega$  .. circular frequency  
 $\omega_0$  .. circular free frequency for rolling

### INTRODUCTORY REMARKS

Spain, though not so much as Japan, which is a country made up of many islands has always paid great attention to fishing and is one of the countries in which the fish consumption is high and the fishing fleet is a large one.

The vessels used for this purpose are not big ones and since the righting moment  $\Delta GZ$  grows with  $L^4$  and the meteorological conditions are not on the same scale, these vessels are exposed to greater risks than cargo ships, which are usually larger.

Furthermore the fishing vessels are more affected by the sea than most of the other types of ships because,

- To do their work they have to be in rougher seas than those to be found on the usual trade routes and they always do their work at a low speed or zero speed.
- Their freeboard is often small, not only because of their size, but also because of the fishing gear they use. For this reason they are prone to ship water. In any case they have to load their decks with sliding catches.
- Those that have the nets on the side



are subjected to heeling moments which may be important.

Usually the crews are not exactly university trained and are more interested in the catch than in the vessel's sea worthiness.

These well known circumstances pushed the IMCO into this problem. It established some criteria at the Torremolinos meeting and is still engaged on the subject.

On the other hand, designers have to cope not only with security requirements - but with those ensuring an efficient use - of the ships while in commission. Therefore, besides criteria more or less based on casualty statistics, stability aims should follow different lines according to the type of ship and the type of gear. This is usually well done thanks to experience based on previous ships, but results in quantitative and the margins available for each type of ship are not common.

These reasons induced us to perform tests with models of some present-day ships. Nothing that has not already been done will be shown, but a comparison of the results obtained with three types of ship which may be considered as representative of those in the present fishing fleet in Spain. Inasmuch as it is oriented to design offices, not to scientists, the work has a limited range and is essentially experimental. It is hoped, however that, it will be of some interest.

#### THE BASIN AND THE MODELS

The tests were carried out by the Spanish Ship Research Association (ASINAVE) in the basin of the Naval Architects School in Madrid. This basin is 98 m. in length, 3.8 m. wide and about 2 m. deep. It has a flap wave generator, home made, which is capable of producing regular waves within the limits permitted by the 2 KW power of its motor. Some irregular or impulse waves were sometimes produced, but in order to ensure the repeatability and direct comparison of the results obtained, only regular waves were used in this work.

It is clear, too, that only longitudinal waves were available. Therefore, the tests with waves were limited to traverse waves and some stern waves. The need to fit the length of the model into the 3.8 m. width of the basin, limited the size of the models to about 2 m. in length, hoping that wall effects would be sufficiently small for the purpose. Athwart the basin were also carried out extinction and other roll excited tests.

At the time of these tests the instrumentation was not the best: Pen recordings were used and the analyses were performed by hand. Roll angles were measured by a steel strip with a weight in one end and strain gauges glued on the other one. No sway movements were measured. As regards roll tests excited by mechanical means, a two counter-rotating weights device was

installed, in such a way that the C. of G. of the weights, when they are opposite - each other in relation to the vertical - shaft, coincide with the C. of G. of the model.

The models are those of ships built - about 10 years ago:

- a tunna boat with a length of 56.1 m.
- a side trawler of 34.26 m. in length
- a stern trawler 45.42 m. in length

The body lines, main dimensions and other particulars of these ships are shown in Fig. 1, 2 and 3, as well as the speeds assumed. The draughts which have been indicated do not really correspond to full load and ballast conditions but to those the models had during the tests. The values given for  $\overline{GM}$  are the design ones. Besides, all models were tested with the minimum metacentre radius  $\overline{GM} = 0.35$  according to IMCO.

All models were tested with rudder -- but without propeller. Therefore only -- towing tests were carried out with stern -- waves.

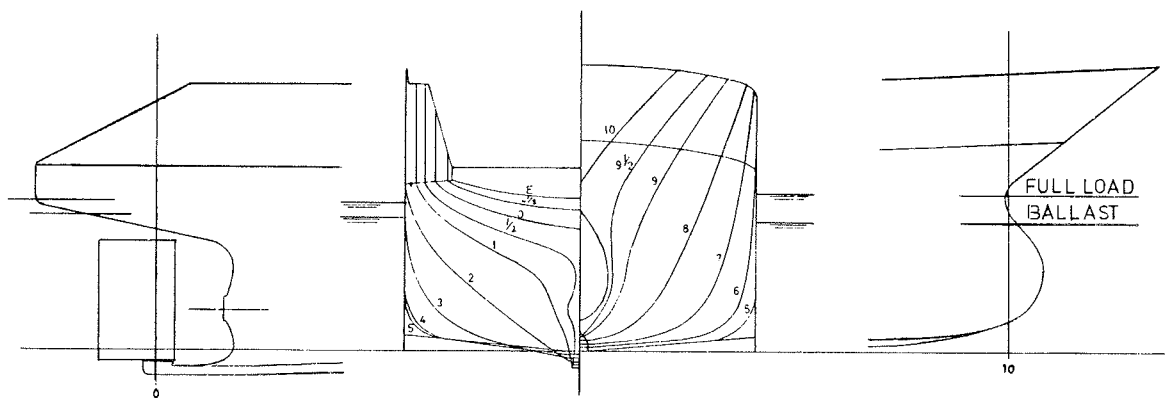
Some care was taken to reproduce the radii of gyration, but in such wood made small models correct values are difficult to obtain.

Though not deemed to be necessary, it may be commented that, since tunna boats have refrigerated brine tanks, they have both high mass moments of inertia and  $\overline{GM}$ . They should not be allowed to have large motions when they are still, which happens for long times during fishing. From the point of view of stability these ships are, therefore, a special case which leads i.a. to the installation of passive tanks.

Side trawlers are no longer the most common fishing craft they used to be, but are in any case a typical boat, with small freeboard and a long side gangway. Furthermore, they are prone to get considerable loads on one side when hoisting the catch.

Stern trawlers have the higher freeboards and do not need to stop while fishing. The danger may come more from shipping water through the stern into the sort of half bath tube built on the main deck rather than over the side in so far as it is intact. But this type of ship is important enough to be included in this choice.

As shown, the requirements of these ships are different from each other and this group of models has a minimum of systematization, if any. Therefore the criteria for these tests were to ascertain the angle of roll the ships may experience - when subjected to a given load.



$$L_{pp} = 56.1 \text{ m.}$$

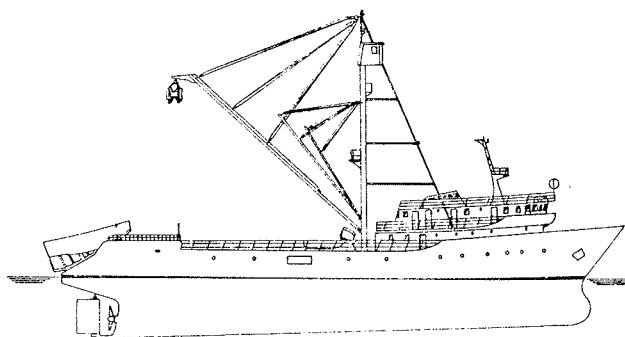
$$B = 11.6 \text{ m.}$$

$$H = 5.70 \text{ m.}$$

$$A_R = 9.79 \text{ m}^2.$$

$$\overline{GM} = 0.90 \text{ m.}$$

$$V = 14.9 \text{ Kn.}$$



$$T_m = 5.19 \text{ m.}$$

$$\Delta = 2033 \text{ t.}$$

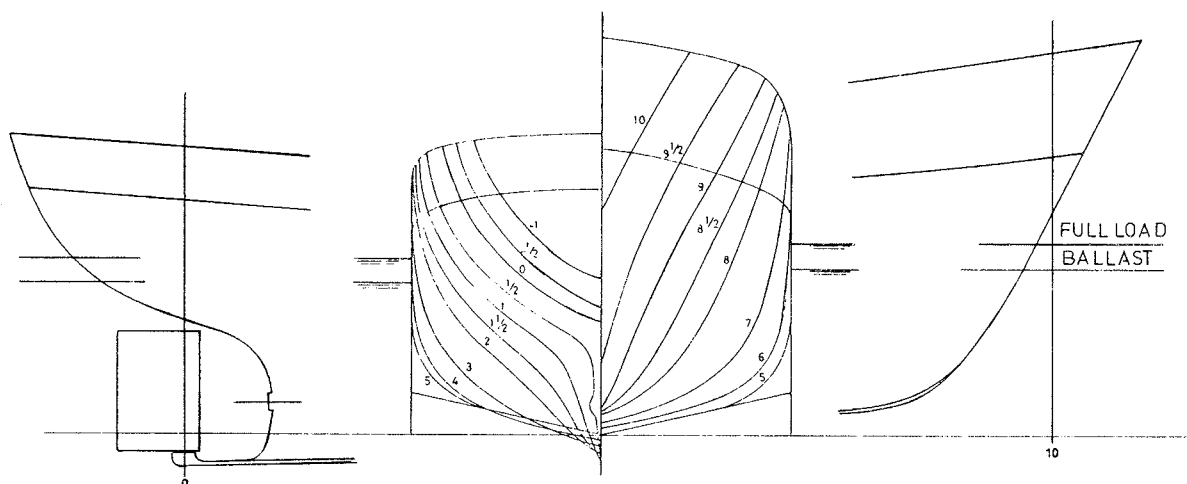
$$C_B = 0.602$$

$$T_m = 4.54 \text{ m.}$$

$$\Delta = 1696 \text{ t.}$$

$$C_B = 0.574+$$

Fig. 1 - Tuna boat



$$L_{pp} = 34.26 \text{ m.}$$

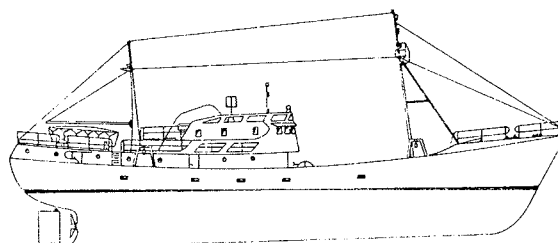
$$B = 7.20 \text{ m.}$$

$$H = 3.95 \text{ m.}$$

$$A_R = 3.45 \text{ m}^2.$$

$$\overline{GM} = 0.58 \text{ m.}$$

$$V = 12 \text{ Kn.}$$



$$T_m = 3.37 \text{ m.}$$

$$\Delta = 431 \text{ t.}$$

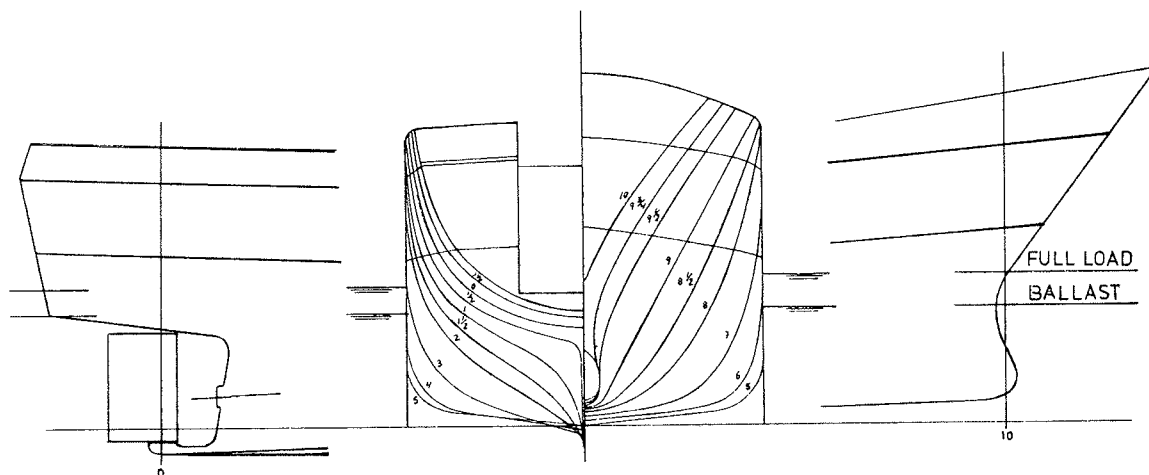
$$C_B = 0.508$$

$$T_m = 2.89 \text{ m.}$$

$$\Delta = 332 \text{ t.}$$

$$C_B = 0.461$$

Fig. 2 - Side trawler



$$L_{pp} = 37.66 \text{ m.}$$

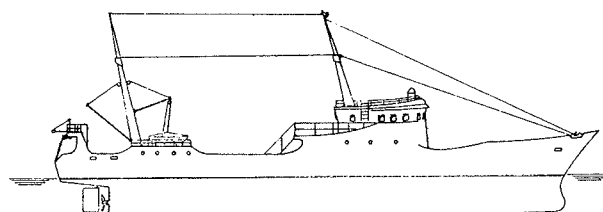
$$B = 9.50 \text{ m.}$$

$$H = 4.30 \text{ m.}$$

$$A_R = 5.24 \text{ m}^2.$$

$$\overline{GM} = 1.57 \text{ m.}$$

$$V = 12 \text{ Kn.}$$



$$T_m = 3.82 \text{ m.}$$

$$\Delta = 814 \text{ t.}$$

$$C_B = 0.596$$

$$T_m = 3.08 \text{ m.}$$

$$\Delta = 605 \text{ t.}$$

$$C_B = 0.549$$

Fig. 3 - Stern trawler

#### THE DAMPING COEFFICIENT

Since the roll amplitudes depend on -- the damping coefficients, several tests were carried out in order to determine its values. These tests consisted mainly in measuring the amplitude reduction while the model was freely rolling. This was done both in load and ballast conditions and in some cases with the speed assumed, besides zero speed.

Figures 4, 5 and 6 give some of these results as  $n/\omega_0(\phi)$ , where  $n$  and  $\omega_0$  are the coefficients of the linealized equation of roll

$$\ddot{\phi} + 2n\dot{\phi} + \omega_0\phi = \frac{M}{J} \quad (1)$$

where  $\omega_0 = \sqrt{\frac{\Delta \cdot \overline{GM}}{J}}$  is the free roll frequency without damping

$M$  is the exciting moment, which is zero during the extinction tests and  $\phi$ , the angles of heel or roll, which were limited in all tests to those that ensured that no water being shipped into the models.

As can be seen, in the range of angles tested, the experimental spots for  $V = 0$  spread more or less about straight lines. Since  $\omega \approx \omega_0 \approx \text{constant}$ , this result is not against the Proude damping formulation --  $b = c_1 + c_2 |\phi|$ , which agrees with the usual assumption that hydrodynamic resistance -- (the part of the damping due to eddies in this case) is proportional to the square of the relative speed of a body. Therefore a --

small program was written in order to make the computer fit, by minimum squares, the lines shown.

The results of  $V = 0$  confirm that the damping measured on models is small and -- does not change very much with the forms of the ship or draught. The side trawler seems to be, however, an exception. Probably due to its shortness, its small block coefficient and the extreme V forms. All these factors enhance the effect of the -- ends of the ship on the damping, which is important, as can also be seen in fig. 4b, where the results of extinction tests with the tunna boat in ballast and sailing on -- stern waves are given. Though the number of spots is rather low (some are missing -- because these figures were cut out at the right end), the lines clearly show that -- the damping is much larger when the crests of the waves are near the bow and stern -- than when the ends of the model get the -- troughs of the same regular waves. It may be added that similar results are obtained with small war ships (patrol boats) with -- low  $C_B$  and V forms in the forebody.

Both lines as well as the lines corresponding to the model sailing in still water show a large increase of the damping coefficient in relation to the same models with  $V = 0$ . If this increment, which is -- well known and generally observed, is due to the angle of attack of the relative -- speed which results from the sum of the -- ship advance and the transverse speeds due to roll, the damping coefficient should --

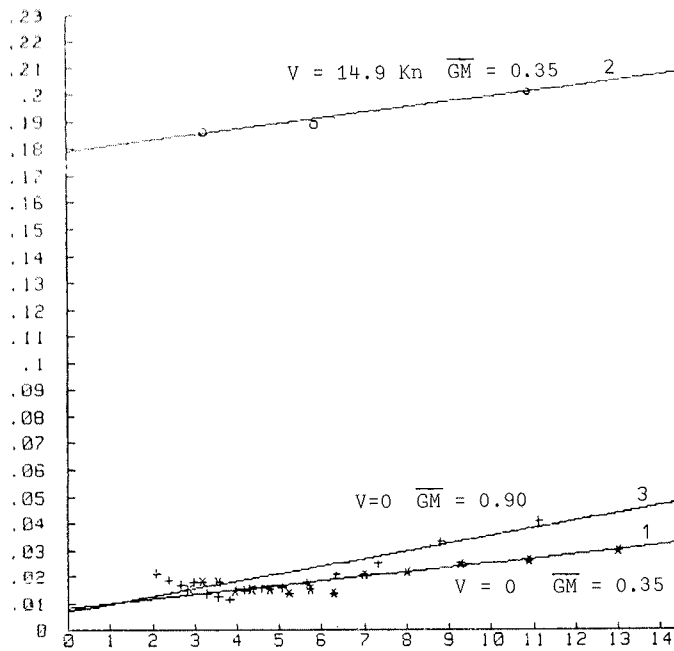


Fig. 4a - Tunna boat, full load.

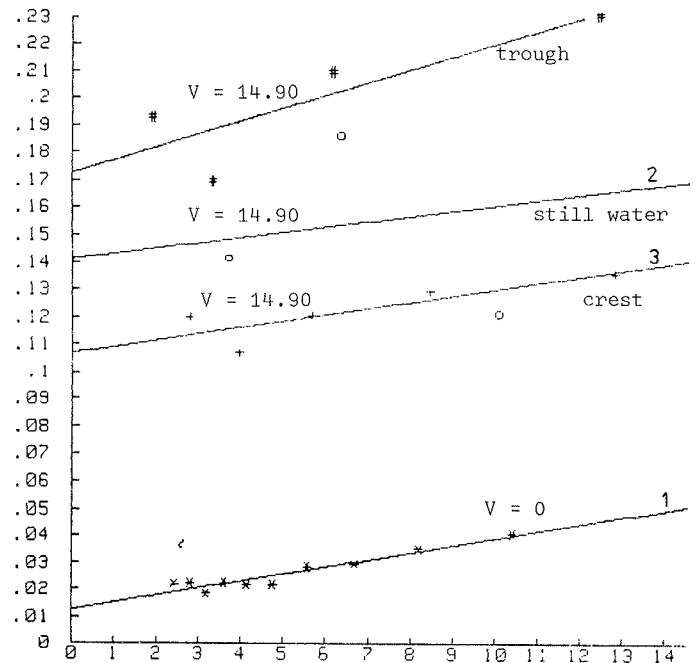


Fig. 4b - Tunna boat, ballast.  
 $\overline{GM} = 0.35.$

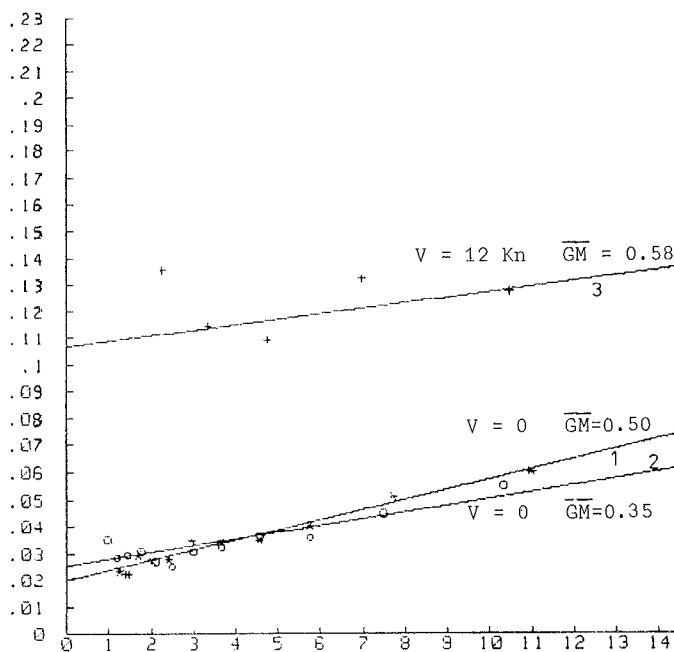


Fig. 5a - Side trawler, full load.

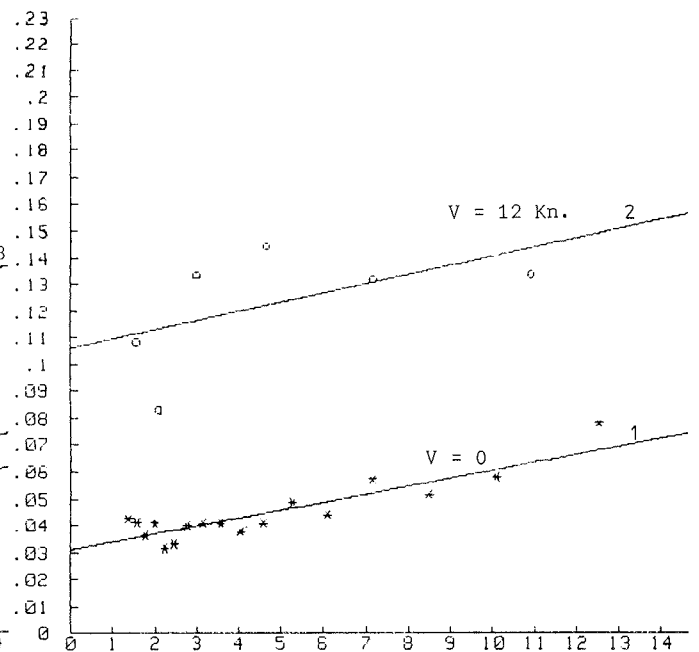


Fig. 5b - Side trawler, ballast.  
 $\overline{GM} = 0.35.$

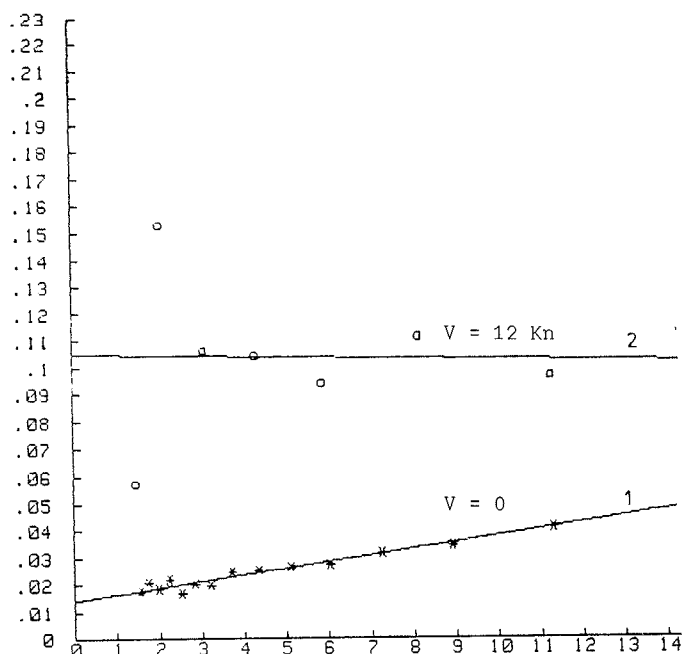


Fig. 6a - Stern trawler, full load.

$\overline{GM} = 0.35$ .

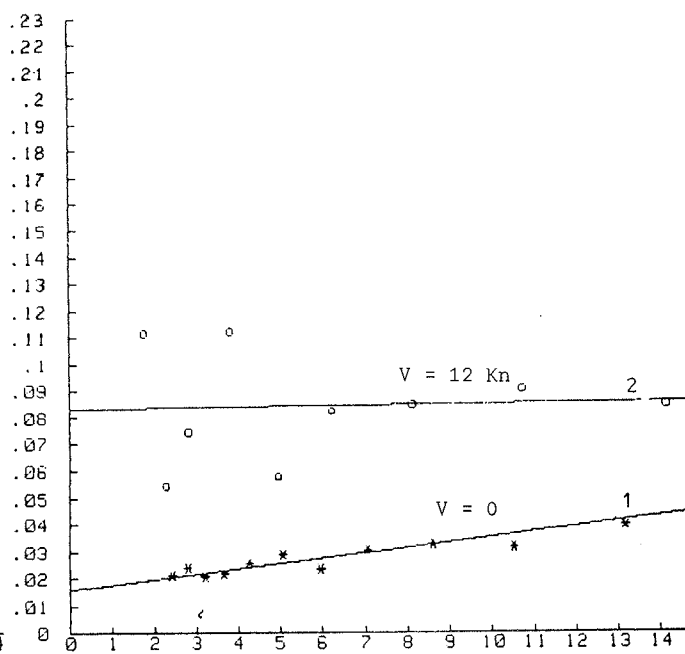


Fig. 6b - Stern trawler, ballast.

$\overline{GM} = 0.35$

grow with the angle of roll, as other results (Ref. 1) have shown.

This does not happen, however, in the present case-maybe because the scarcity -- and the spread of the experimental points while the model is at speed, did not make it possible to draw the correct lines. But other results from tests carried out by -- ASINAVE show the same trend as those given in this paper. Maybe the reduction of the eddy making part of the damping with -- growing speed (Ref. 2) provides the explanation for the small inclination of the lines corresponding to the models at speed. On the other hand, it appears that the increase of damping with speed grows with draught, which means larger "profile" surface and a more efficient aspect ratio.

In any case, the increase of damping with the roll angle is clearly confirmed -- when  $V = 0$ . A direct consequence of this fact is that, even if sway coupling and -- other secondary effects be disregarded, -- equation (1) can only be applied as a -- first approach. It is however so simple, -- that it is used very often and so was done with the results of forced roll tests, excited by the mechanical oscillator.

The aim of these calculations was to ascertain how the damping changes with the frequency of oscillation, which cannot be determined by the extinction tests, in -- which the model moves with its own free -- frequency. Of course, this frequency may be changed -- see fig. 4a and 5a -- shifting --

up or downwards the C. of gravity of the -- model. But the inconvenience of this procedure leads us to analyse, as already said, the oscillator tests. This was done through the solution of (1), which gives the amplification factor:

$$\frac{\phi_a}{M_a/J\omega_0^2} = \left[ (1 - \Lambda^2)^2 + \left( \frac{2n}{\omega_0} \right)^2 \Lambda^2 \right]^{-1/2} \quad (2)$$

where

$\Lambda = \omega/\omega_0$ , and

$M_a/J\omega_0^2 = \phi_0$  is the static heel angle the model would get under  $M_a$ , which is the amplitude of the exciting moment

The growth of  $n/\omega_0$  with the frequency appears to be confirmed both by these and the extinction tests. However, no results are given, because the accuracy of the -- tests was not sufficient to obtain reliable results for values of  $\Lambda$  differing by -- some amount from unit.

#### THE SEA FOLLOWING

The changes of stability experienced by a ship when sailing on long crested waves of a length of the same order of magnitude as the ship's length are considered -- as a cause of foundering. This may happen because of a loss of stability when the waves advance with the same speed and direction as those of the ship or by self-exci-

tation, due to the variation of the stability arms, when the difference of speeds -- leads one to encounter frequencies related to those of the free oscillations of the -- ship. Although not touched on in this paper, broaching may also occur.

Since self-excitation is due to periodic changes of stability, not only its minimum values but also the difference of -- these values with the maxima -- which are -- larger than the static stability -- are of -- interest.

A first approach to coping with this problem may be to compute the metacentric radius, or other stability data, for different positions of the wave assumed and -- find the extreme values. Indeed the semi-static calculations as usually performed don't lead to results agreeing with the experimental data and, therefore, they cannot be directly used as a criterion for -- assessing the behaviour of different hulls on following waves. But at least, these -- calculations provide some useful indications, as for instance the wave positions and the ship displacements with which these waves are most dangerous.

For example, the values of GZ at  $10^\circ$  are given in fig. 7, for the tunna boat, -- in several positions of the trough of a wave of which the length is  $\lambda = 1,2L$ , as a function of the displacement. The difference between the maxima and minima diminishes with growing displacement and this -- change takes place also for small displacements in the other ships. Though not always the case -- since it depends on the inclination  $dz/dy$  of the frames near the water line -- that variation suggests the desirability of considering more specially the ballast conditions.

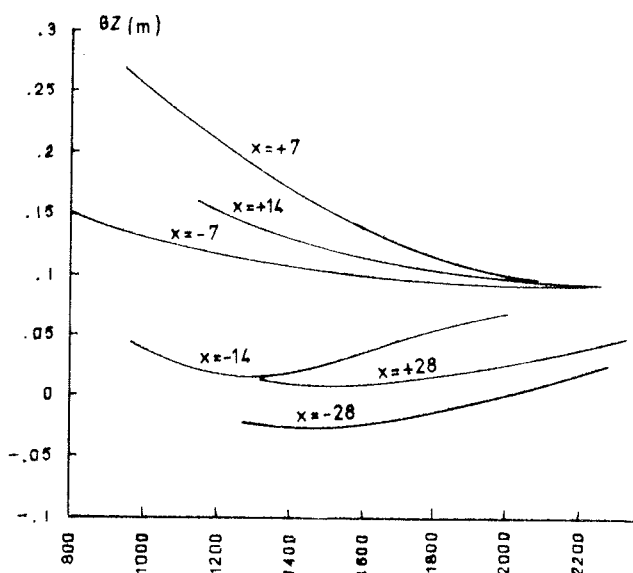


Fig. 7

With a vertical cut, the  $(GZ)_{10}$  values for a given displacement can be obtained. Fig. 8 shows this cut for the ballast condition of the tunna boat. The inflexion point near the maximum can not always be found, but it is not abnormal either for this type of curves not to be necessarily smooth and sinusoide-like. A more interesting feature is that the maximum does not occur exactly when the trough is amidships. Though not in the present cases, it has already occurred that the same stability is obtained with a wave of the length  $\lambda = L$  in the two conventional positions (amidships and on the perpendiculars) but very different results with a small shift of the wave.

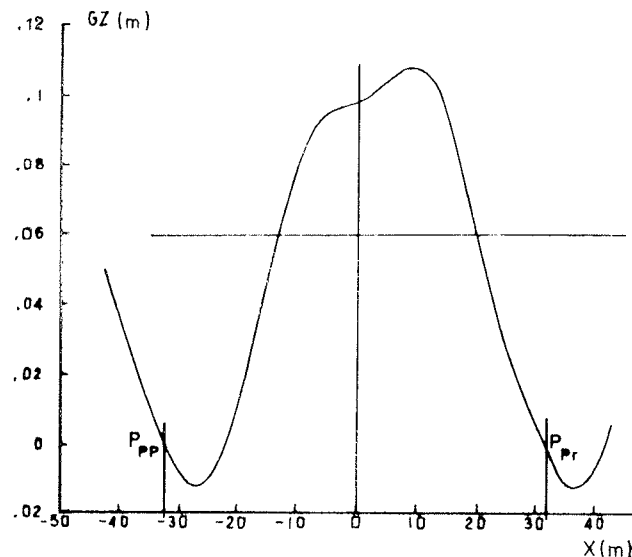


Fig. 8

I am sorry to say, owing to a change to a computer with another language, our program, which includes the Smith effect, was not available when these calculations were carried out. Therefore the real changes in stability would probably be much less than those shown. This also applies to the stability curves given in Fig. 10 to 13.

In these conditions no direct comparison can be made between computer and experimental results. However, some tests were carried out in this connection. The procedure was similar to that quoted in (Ref. 3), i.e. to shift athwart the center of gravity of the model in order for it to get an initial static heel ( $15^\circ$  in this case) and then run it in the waves following.

As the wave Froude number,  $(2\pi)^{-1/2} \approx 0.40$ , is greater than those of ships', to avoid too large a generation of waves which would juggle with what we are seeking, wave lengths shorter ( $\lambda = \frac{2}{3} L$ ) --

than those of the models were chosen. Only with the tunna boat was a test carried out with a longer wave ( $\lambda = 1.2 L$ ). The height of the waves was 4.5% of its length and the speed of the models was kept about that of the waves, which when  $\lambda = 2/3 L$  is not far from the ship's speed assumed.

When towed in still water the heel angle  $\phi_v$  was less than the heel  $\phi_0$  at rest. This reduction means an increase of the stability which, since these angles are not large, may be approximated by

$$\overline{GM}_v - \overline{GM}_0 \approx \overline{GM}_0 \frac{\text{tg} \phi_0 - \text{tg} \phi_v}{\text{tg} \phi_v} \approx \overline{GM}_0 \frac{\phi_0 - \phi_v}{\phi_v}$$

With  $\overline{GM}_0 = 0.35$  this increase amounts as much as to 30% in the faster tunna boat, 10% in the side trawler and 8% in the stern trawler, in ballast conditions. With full load the results are different but similar, and no trend is shown. With larger static stabilities the percentage is, of course, less but the absolute increase is maintained.

While the center of gravity is kept constant, all points representing these tests should be on the line  $M_{15} \cos \phi / \cos 15^\circ$  passing through the point S on the static stability curve at  $15^\circ$  (Fig. 9). If the run in still water gives an angle  $\phi_v$ , we shall get a point W for the virtual stability curve for these conditions. The increment  $\overline{WO}$  suggests another "effective" stability curve, which might be drawn through points such as

$$\overline{GZ}_0 + \frac{\overline{WO}}{\sin \phi_v} \sin \phi$$

The dotted line shows this approximation.

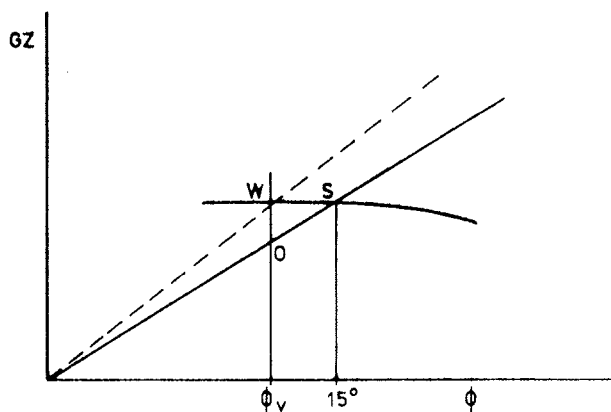


Fig. 9

There is no special reason to suppose that the difference in righting arms  $\overline{WO}$  will be equally assumed by the ship while sailing on waves. But it seems reasonable that a similar effect will occur. On this

basis the higher and lower stability curves calculated for the ship positions -- about the trough and the crest of the wave, would get the same increment. The figures 10 to 13 show such curves.

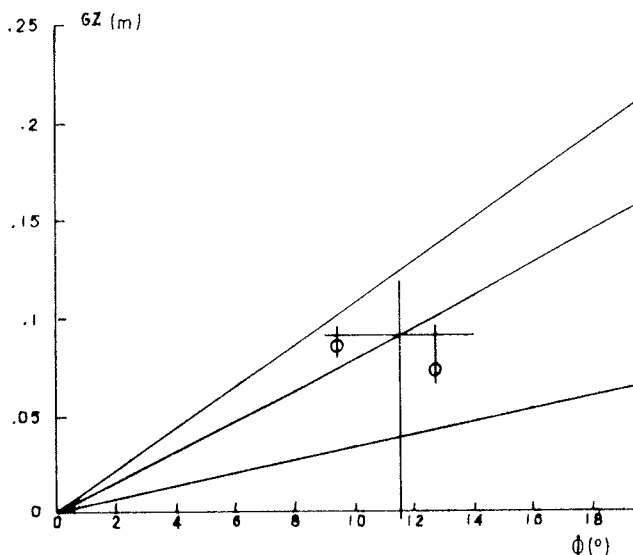


Fig. 10 - Tunna boat  $\lambda = \frac{2}{3} L$

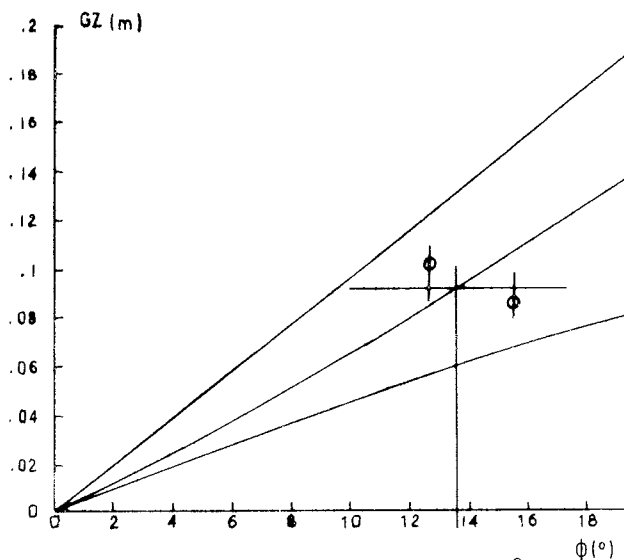
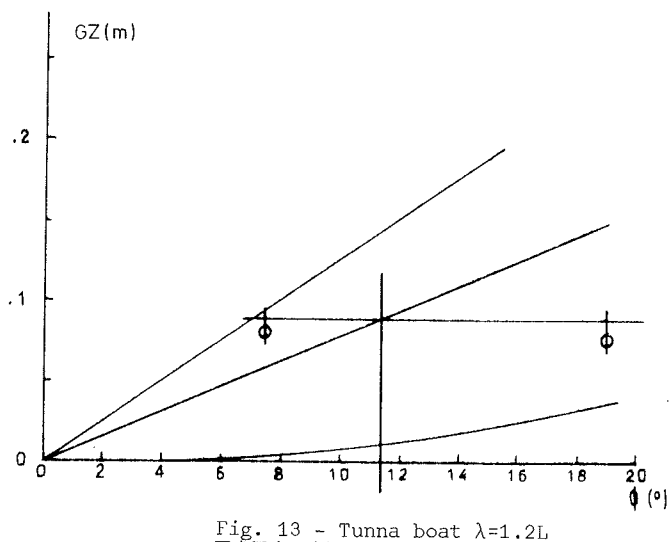
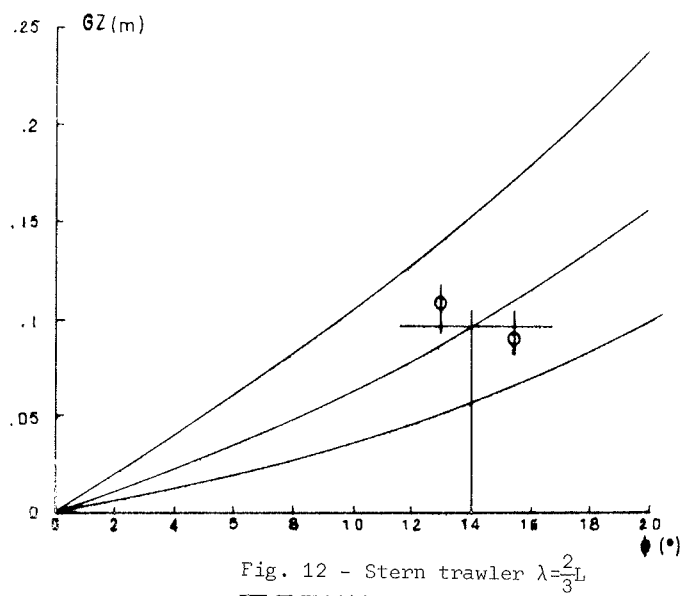


Fig. 11 - Side Trawler  $\lambda = \frac{2}{3} L$



It may be seen in these figures that the points where the vertical lines, which represent the angles measured, cross the corresponding higher or lower computed curves, are very far away from their cross points with the heeling moment curve (WS in figure 9) - this last line being approximated by a straight horizontal line. This result might be expected since the rotation of the water particles was not taken into account. In expectation of further calculations and aware of the non linearity of the stability arms with wave heights and of the fact that the effective depth differs from the draught in the inclined hull, the usual correction  $\exp(-kT)$  was applied to the difference of the righting arms provided by the curves obtained for still water and wave calculations. This corrections gives the points signaled by -

small circles in the drawings. They show a better agreement, though still some differences are to be found.

From this approach it seems that the stability variation can be approximated by simple calculations. As expected, this variation is not great for  $\lambda = \frac{2}{3} L$ . The differences between ships are not large either. Only in the tunna boat does the loss of stability appears to be more important.

On this basis and taking advantage of her highest speed, a test and calculations as before were undertaken for this ship -- and  $\lambda = 1.2 L$ . Results are shown in Fig. 13. The considerable loss of stability "on the crest" and its reduced increase in the trough were also confirmed by mechanical oscillator tests carried out on those wave positions.

The results of calculations for full load conditions did not show differences -- significant enough to allow more tests to be undertaken.

#### THE ROLL RESPONSES

In order to show the roll responses of the different models, several tests were carried out with the mechanical oscillator. Through these tests, both the oscillating weights and the arms were unchanged, the intention being to keep constant the initial exciting moment. It can be said -- that this was the case, since the moment of the centrifugal force is negligible and the same applies to  $(1 - \cos \phi_0)$  -- where  $\phi_0$  is the initial heel produced by the weights -- since this angle varied between  $1^\circ$  and  $3.5^\circ$ , depending on the values of  $\Delta GM$ .

This choice, based on the fact that all models were of a similar length, is open to criticism, since the displacements were quite different. But length is the common reference in fishing craft. Anyways, and though non-linear effects cannot be eliminated in this way, the results are given as  $\phi/\phi_0$ .

As already mentioned, the radii of gyration do not agree with the recommended standard  $0.35B$ . But they do not differ either much from it. For this reason and in order to keep well apart the curves related with the different ships in the same figure, the frequencies were not adimensionalized.

The results obtained from the tests of the models with full load and the design  $GM$  are shown by the full lines of Fig. 14. Great differences in  $\phi/\phi_0$  are found. The real amplitudes were, however, not so different as shown, since the values of  $\phi_0$  differ. For example, for the tunna boat  $\phi_0 = 1.37$  and for the stern trawler  $\phi_0 = 1.02$ . It may be commented that these results differ from the predictions drawn from the linear equation (1) in the lower frequencies, but in their largest part --



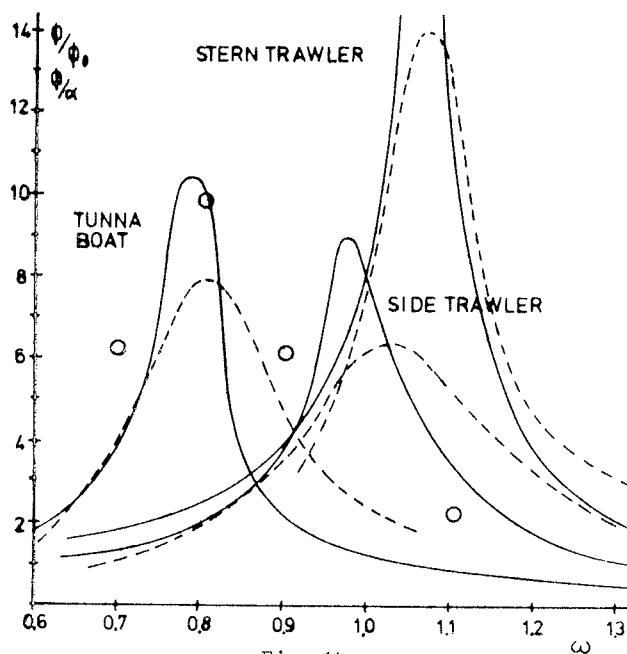


Fig. 14

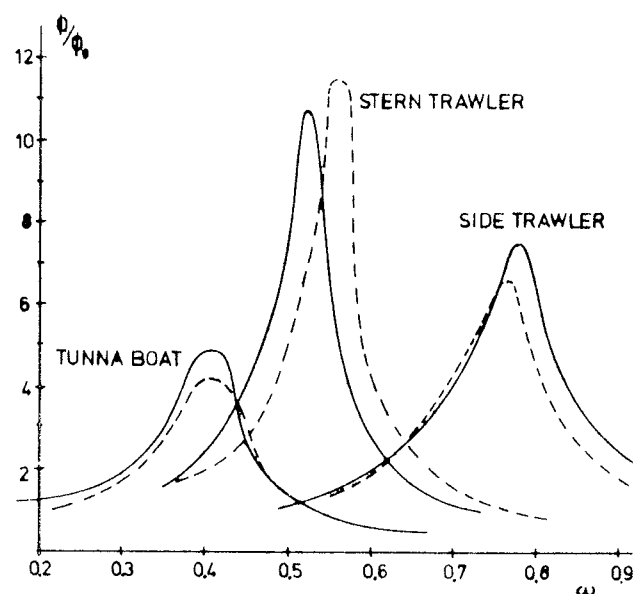


Fig. 15

agree more or less with them if an increase of the damping coefficient with  $\omega$  is introduced. Therefore it seems that such a simple equation may be used if only a small sway excitation is present.

The dotted lines show the results of tests carried out with transverse waves with the same models and conditions. Instead of  $\phi/\phi_0$ , these results are plotted on the, say, equivalent values of  $\phi/\alpha$ , where  $\alpha$  is the wave slope  $\alpha$  corrected by  $\exp(-kT)$ . Though lower and broader, these curves show the same trends as those obtained from the oscillator tests.

These results were also compared with those obtained by computation. For this purpose, the usual coefficients of the combined equations of roll and sway (which should not be ignored in this case) were computed with a program (Ref. 4) based on the strip theory. No attention was paid however to the damping coefficients calculated for rolling, and instead the values obtained from the extinction tests were introduced in the equation of roll. Some results of these calculations for the tunna boat were plotted in the same figure. The form of the curve is well forecasted but the agreement is not good as regards to the amplitudes. A somewhat better agreement is obtained for the other ships, but computed in this way, the amplitudes seem to be always larger than those obtained in the tests.

The different  $\overline{GM}$  might obscure the differences in the peculiar behaviour of the hulls. Therefore, further tests were carried out, now with  $\overline{GM}$  values equivalent to 0.35 m. in the real ships. Fig. 15 shows the results of these tests with the models

at full load (full lines) and ballast (dotted lines). The differences between these lines are not large but, of course, the real angles were much larger in ballast, since in this condition the values of  $\phi_0$  were bigger, as a result of the lesser displacement. The frequency shift of the roll maxima at full load and ballast might have its origin in differences of the radii of gyration after changing the displacements. As regards to the side trawler curves, which are shifted to the right and show amplitudes which are larger than those of the tunna boat, it may be commented that, while not proportional, roll amplitudes vary with  $\omega_0/2\pi$  as it comes from (2). Anyhow both the values of  $\phi/\phi_0$  and the differences between those obtained with the different models were diminished by the adoption of a common and lower  $\overline{GM}$ . As might be expected.

In Fig. 16, the curves for hull load condition and  $\overline{GM} = 0.35$  were repeated, but compared now with those obtained in the same conditions with the models advancing at the speeds assumed, instead of being at rest as the other oscillator tests were made. In this case the radii of gyration were constant in each model and the shifting of the maxima are due to the increase of stability, already mentioned, in the running ship. The reduction of the roll is, of course, due to the increase of damping with the speed, as was already shown in the results of the extinction tests.

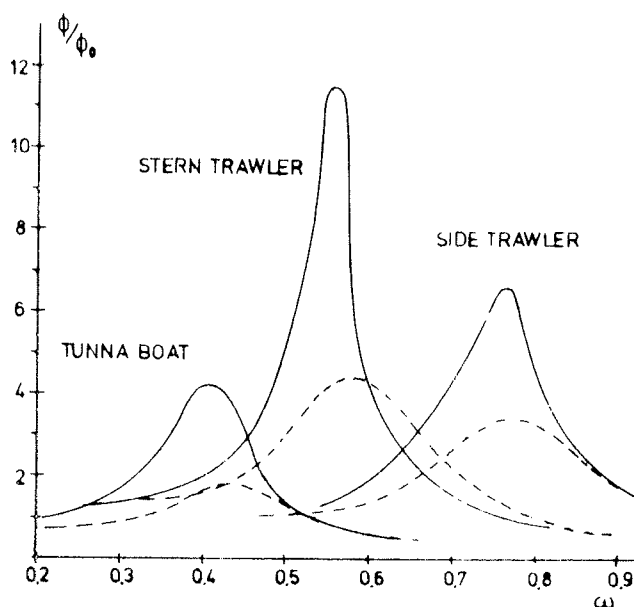


Fig. 16

#### FINAL REMARKS

Not many conclusions may be drawn - - from these tests. However it seems that, - for the fishing vessels that were tested, the following can be said,

- Since the changes of draught are not important, only a draught ought to be considered if additional factors affecting the stability of the forms are sought. - Preferably the minor draught in service should be chosen.
- Both stability and damping increase with speed. Therefore the worst situation is found when the ship is still or sailing at low speed -either while fishing or in bad weather.
- The loss of stability on following waves should be investigated when designing -- the ship, since vessels of this size and work are prone to encounter waves equal or longer than their length. At least, - this should be done with fast ships - - which have forms similar to those of the tunna boat.
- Equation (2) may be called naive at the present state of the art. However it - - shows clearly that roll responses are a function of  $\Lambda$  -which depends on the sea and the skipper- and of  $n/\omega_0$ , which depends mainly on the design of the ship.

The importance of the last mentioned - - coefficient is enhanced, since the dynamic angle of roll plays a primary role - in the service and security of the ship. Though not treated in this paper, it may be commented that while yawing was considered without any influence on roll -because its angles are small- the coupling coefficient is not small and broaching may occur with heavy rolling in some - ships. This may be a further reason to - increase the damping coefficient.

- Only results of two models with bulb bow are presented and only one without it. - But other tests carried out in ASINAVE - confirm that this type of bow diminishes the damping. Therefore and specially in these hulls, the old conventional keel, a skeg or bilge keels -even if skippers dislike them- are recommended.
- Finally and now from the point of view - of the experimental work it was concluded that this type of tests requires - - higher accuracy than others normally - - carried out in the basin. Though this -- was known from earlier tests (Ref. 5), - still a greater care had to be taken in some of the present ones.

#### ACKNOWLEDGMENT

The author is grateful for the help - received from the personnal of ASINAVE and specially from Antonio García who carried out a large part of the tests and computer calculations.

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## Discussion

Y. Takaishi (Ship Research Institute, Japan)

I would like to express my appreciation to the author who has just shown us very interesting results on roll damping of fishing vessels from the view-point of practical use.

Of my opinion, the data on roll damping should be accumulated in order to be able to not only contribute to developing rational estimation method of roll damping but also predict ship responses in oblique waves, as rolling or relative wave elevations.

The data presented here must be useful for this purpose and such experimental data should be welcome for various types of ships.

My question concerns with the damping coefficients variation in waves. It decreases on the wave crest amidship (Fig.4b). What is the main reason of this phenomena? Does it be induced from variation of ship form due to wave profile or from change of relative velocity of advance due to the orbital velocity of waves? In SRI the model test was done for measurement of distribution of roll damping force with ship speed and it was found that the fore and aft body ends contribute more. Therefore, I presume that the relative velocity variation contributes to this phenomenon.

I would like to hear the author's opinion on this problem.

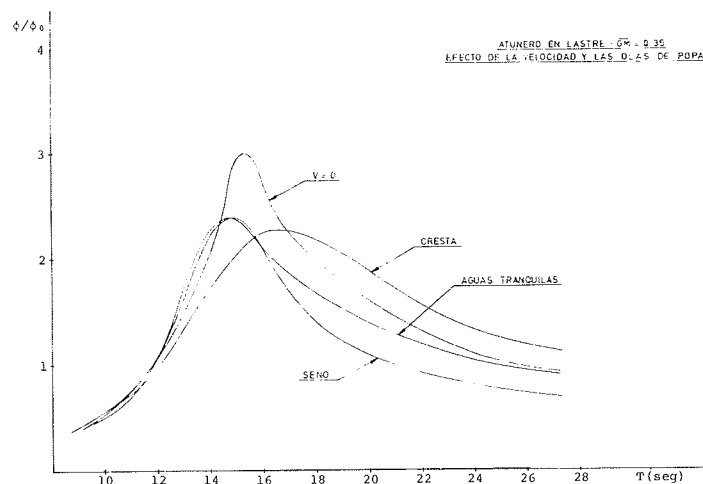
### Author's Reply

Thanks for your kind contribution. The data I have given may be accumulate with other already available to help to build a

damping formulation. However, since the aim of these tests was not a scientific one, but to call the attention of designers on the behaviour of different ship types, caution should be applied to some of the lines. This is particularly true in the case of the model on waves in which the number of points is very scarce.

However, though not quantitatively, the trend shown by these tests was confirmed by oscillator tests; as can be seen in the attached figure, the maxima do not differ as the lines  $n/\omega_0(\omega)$  do, but for longer periods (t) it appears that the damping is larger, when the through (seno) is amidships, than in still water (aguas tranquilas) and this one, larger again then when the ship is on the crest. I am glad to hear that you found that the ends of the ship have a big contribution. My opinion is that the eddies generated by the bow and the stern are very important and therefore those ships with small run and entrance angles will have more damping and those with bulbs, lesser. The same applies to the ship on waves, since the inversion of the ends change with the position of the wave and is maximum when they both get crests.

I want to add that though the accuracy of the tests was not excellent, it was neither as bad as the comparison with calculated results, given in fig. 14, may suggest, I gave these results in order to be honest, but I were wrong, because the computer program was also wrong - due to change of language already mentioned in the paper.



## SAFETY OF FISHING VESSELS BY MEANS OF EXPERIMENTS IN WIND WAVES

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### ABSTRACT

The experiments were performed with the use of 2 meters long self-propelled and radio controlled models for fishing vessels in wind waves at Lake Ohnuma, Japan. The object of this study was to clarify the behaviours of fishing vessels in wind waves with regard to intact stability of fishing vessels, especially from the view point of seamanship.

In this paper, describing briefly about the experimental field, facilities and the system for measuring wind waves and motions of models, the results of analysis for cruising experiments of 3 types of fishing vessels were detailed and discussed.

In experiment with a 124 GT big off-shore trawler, the shipping water on deck was defined and analysed and the longitudinal motion of the ship was examined by comparing between observed and calculated motion spectra.

The cruising experiments were performed to examine the intact stability criteria for small fishing vessels. In experiment with 19.9 GT small salmon fishing vessel, lurching phenomena were observed within the rolling motions and analysed by measuring relative wave elevations and observing parametric resonance phenomena.

### 1. INTRODUCTION

Concerning the intact stability of fishing vessels fishing and navigating in wind waves, the observation of their behaviours on the actual sea is an important consideration for ship safety in rough waters.

As for capsizing incidents of fishing vessels in rough waters, even the Maritime

Accident Inquiry has had considerable difficulty in providing reasonable explanations for this phenomenon.

Generally speaking, in spite of their small size, fishing vessels usually navigate far offshore engaging in various fishing operations. For example, in salmon drift net fishing, the fishing operation includes casting net into the sea, hauling the net in, and unloading catch. A fishing operation will cause changes in seakeeping qualities, and this is a noteworthy problem for fishing vessels in regard to stability and the safety of the vessels.

It is necessary for a seaman to have the ability to judge how to insure the safety of a fishing vessel in rough weather and to have knowledge concerning the stability of the vessel.

The results of recent studies on the safety of fishing vessels on the seaway, supported by the remarkable development of seakeeping qualities should be used not only for research but for seamanship as well.

From this point of view, in order to get a good understanding of the behaviour of fishing vessels in rough sea, it would be quite necessary to know the relation between actual phenomena and theoretical studies of seakeeping qualities. The authors believe that the discrepancy between full scale and model ship experiments will continue to exist despite present studies, and that it will be necessary to find some method of explaining the relation between the former and the latter, other than by full scale ship experiment.

Considering these matters, the authors performed the model experiments on wind waves of a lake and have presented the results below.

## 2. EXPERIMENT

### 2.1 Experimental Field

The experiments were performed on the northern area of Lake Ohnuma in Hokkaido. The shadowed portion shown in Fig.1 is the experimental field. The maximum fetch in this field was about 4 km for wind direction of SSW. The experiment was begun when the wind was blowing from this direction, with velocity of over 8 m/sec. The experiments were performed in October, which has a high frequency occurrence of wind blowing from this direction.

### 2.2 Model Fishing Vessel

3 kinds of model fishing vessels were used. Body plans for these models are shown in Fig.2 and their principal dimensions in Table 1.

### 2.3 Measuring and Data Processing Systems

The experiments were performed by using the following 3 systems:

- (1) System for measuring wind waves, wind direction and velocity
- (2) System for measuring motions of model fishing vessels
- (3) Data processing system

For the measurement of wind wave, a steel pole 8 meters long was set near the center of the experimental field shown in Fig.1. Apparatus for measuring wind velocity and direction were set on the top of this pole. The wave capacitance sensors were also fitted on this pole.

In the experiments, these measured analog data for wind waves were telemetered to the observation station by cables. The model fishing vessel was equipped with motion measuring apparatuses, self-propelled system and power supply system. The measured analog data were transmitted from the model to the observation station by the use of a multi-channel FM-FM radio telemetering system with 169.65 MHz of main carrier wave. At the observation station, a multi-channel data processing system was installed, which received the data.

### 2.4 Cruising the Model Fishing Vessel

When the SSW wind started to blow with wind velocity of more than 8 m/sec and the wind-generated wave height grew to over 15cm the model cruising experiment was begun.

- (1) Things measured: wind direction, wind velocity, wave heights, wave direction, motions of model, velocity of model, positions and angles in which model is moving.
- (2) Observation of behaviours of model: The behaviour of the model in wind waves was recorded by the use of a 35 mm motor driven camera and an 8 mm movie camera.

- (3) Cruising of model in waves:

The relative angles between direction of movement of model and wave direction were as follows: Head, bow, beam, quartering and following. Cruising model experiment was repeated for each relation. The motions, wind direction and velocity, and wave heights were measured and recorded, respectively. The model fishing vessel was controlled by radio control system.

### 2.5 Example Records from the Experiment

In October 1981, the capsizing experiment for a small salmon drift net fishing vessel was performed by the above method. The weather charts are shown in Fig.3. Cruising experiment was done on October 31 and November 1. Example of the cruising track is shown in Fig.4, the analog data of motions are shown in Fig.5 and the rolling motions are shown in Fig.6.

## 3. ANALYSIS

During the last 10 years, 6 kinds of model fishing vessels have been used for these experiments. The length of these models was 2 meters. These model experiments were performed, using previously mentioned method. In this paper, the results of 3 kinds of model experiments are described:

- (1) Capsizing experiment with 96 GT salmon drift net fishing vessel
- (2) Experiment in regard to the shipping water on deck for 124 GT big offshore trawler
- (3) Experiment concerning stability criteria for 19.9 GT small salmon drift net fishing vessel

3 kinds of model fishing vessels shown in Table 1 were used for this experiment. All measured analog data for wave heights and motion of model vessels converted into digital data and analysed statistically. Their power spectra and statistics were obtained, and the wave direction was calculated from the wave heights. Based on the behaviours recorded in movie, the relative wave profiles were obtained. The results of the analysis for these cruising experiment are as follows:

### 3.1 Wind Waves on Lake Ohnuma

- (1) Depth of experimental field: Using ultrasonic depth recorder, the depth along maximum fetch direction, which was about 4 km long for SSW wind direction, was measured and shown Fig.7. The average depth of this field was about 7.5 meters.
- (2) Wind direction at Ohnuma: The frequency distribution of wind direction in October according to meteorological statistics of Japan is shown in Fig.7 as the rose chart.

- (3) Wind waves in experimental field:  
Based on data for wave height over the last 10 years, the relation between significant wave height and wind velocity is shown in Fig.8 and the empirical formula is given as follows;

$$H_{sig} = 2.538 * V * 0.01 \text{ (m)}$$

where  $H_{sig}$ : significant wave height  
 $V$ : wind velocity in m/sec

- (4) Wind wave spectra:  
By analyzing wave data in the experiments, the following wave spectra were obtained;  
a) rapidly developed wind wave spectra  
b) wave spectra due to wind blowing with constant velocity  
c) numerical model of wind wave spectra  
Their spectra are shown in Figs. 9,10 and 11a,b.  
In the above section c), Darbishire's coastal wave spectrum and Pierson-Moskowitz's wave spectrum were compared with measured spectra and shown in Fig. 11a,b.

### 3.2 Capsizing Experiments with Fishing Vessels

- (1) Capsizing experiment in wind waves:  
Ship condition in the experiment was described in Table 1. The wave statistics and the wave spectra are shown in Tables 2,3 and 4 and Figs.12, 13 and 14. Based on the results of motions record for 3 kinds of model experiments, the relation between rolling and pitching angles for each model and significant wave height is shown in Fig.15a,b.  
In these capsizing experiments, the capsizing of the model vessel did not occur many times. The capsizing of a 96 GT model salmon fishing vessel navigating in beam sea with beam wind, for example, occurred due to high wind waves and gust, and the power spectrum at this time was shown in Fig.12. Also, the power spectrum for rolling angles before capsizing is shown in Fig.16.
- (2) Shipping water on deck in rough water for 124 GT big offshore trawler  
In recent studies (Ref.1,2 and 3), it has been confirmed experimentally by many authors that the occurrences of capsizing of vessels accompanied by the shipping water on deck has been observed in following and quartering sea. In these experiments, using the relative wave heights measuring apparatus which consists of capacitance sensors, the relative wave heights were measured continuously at the side of the model. Also, the behaviour of the model was recorded on film using an 8mm movie camera. Based on these data, an analysis was made. The results are as follows:  
a) wave height statistics and wave spectra are shown in Table 3 and Fig.13.

- b) Condition of the model in the experiment is shown in Table 1.  
c) Spectra of relative wave elevations are shown in Fig.17a,b.

- (3) Rolling and lurching in wind waves for 19.9 GT small salmon fishing vessel:  
Condition during this experiments are shown in Table 1. Wave statistics and spectra are shown in Table 4 and Fig.14. Spectra for model ship are shown in Fig.18, and their statistics are shown in Table 5, for example. In order to indicate the irregular motions of the fishing vessels due to wind waves, expressions of three dimensional spectra and rolling motion by phase plane were used and are shown in Figs. 19 and 20. From these figures, it is clear that the lurching phenomenon was conspicuous in the beam and quartering seas. In the phase plane, the relation between rolling angles and rate were described, and they show peculiarity of the rolling motion.

By analyzing the 8 mm movies, the relative wave elevations were obtained from each frame, once every 1/9 second. From these records of relative wave elevations, 4 principal wave surface profiles were obtained and are shown in Fig.21. Based on these profiles, the stability curves were calculated and are shown in figure.

## 4. DISCUSSION

the results of the analysis of the cruising experiments are shown as follows:

### 4.1 The Equivalent Probability Level for Shipping Water on Deck

From the measurement of relative wave elevations, the probability density functions were obtained and ascertained to be Gaussian applying Kai's test. From these probability density functions, the probability of water rising above the established height level was calculated and conversely, for given probability values, the equivalent probability level was obtained.(Ref.4 and 5) These levels are shown in Fig.22.

### 4.2 Comparison between the Measured and Calculated Longitudinal Motion Spectra

The longitudinal motion spectra were computed on the basis of linear theory. The frequency response function computed by OSM from the observed wind wave spectrum was used, with long-crested and short-crested waves as input. These spectra are shown in Fig.23a,b and c.

A comparison between observed spectra and calculated spectra was made, and it was found that the results obtained from calculation using the short-crested wave spectra coincided with observed ones. That is, for the means for short-crested waves, 1/3 hei-

ghest means and 1/10 heighest mean values observed coincided with the calculated values. These are shown in Fig.24.

These coincidences were remarkable for head, bow and following seas.(Ref. 6)

#### 4.3 Parametric Resonance Phenomena

The authors assumed the lurching phenomenon so often observed in the experiments to be low cycle resonance. In the above mentioned experiments, the non-linear Mathieu's function was used as Dr. Blocki (Ref.7) had use it, to describe the rolling motion in the following sea. The authors solved the differential equations numerically by means of the Runge-Kutta-Gill method and compared them with the observed motions. Then, it was found that the observed time series for rolling rate and angles coincided with the calculated output series from a computer. They are shown in Fig.25.

Based on these results, the authors ascertained that the parametric resonance phenomena had occurred.(Ref.8 and 9)

#### 5. CONCLUSION

The object of this study was to clarify the behaviours of fishing vessels in wind waves, with regard to intact stability of fishing vessels, especially from the view point of seamanship. The main points of this study are summarized as follows:

- (1) The model experiments for 3 types of fishing vessels were performed and analysed by the use of system specially devised for wind waves.
- (2) In experiment with a 124 GT big off-shore trawler, the shipping of water on the deck was discussed and the longitudinal motion of the ship was examined by comparing between observed and calculated motion spectra.
- (3) In experiments with a 19.9 GT small salmon fishing vessel, the intact stability criteria for small fishing vessels was examined. Lurching phenomena were observed within the rolling motions and analyzed by measuring relative wave elevations and observing parametric resonance phenomena.

The conclusions are as follows:

- (a) The relation between wave surface and ship motion was defined according to relative wave elevations and also according to shipping water on deck. The statistical properties of the water on deck were defined from the measured data. Based on these data, the probability density function of this was deduced and equivalent probability levels were obtained.
- (b) The longitudinal response functions, calculated by OSM method, and their motion spectra were estimated as input forces to be the calculated spectra having short crested directional spectra. It was found that the

observed and calculated spectra coincided.

- (c) Lurching phenomena within the rolling motion;  
The authors did not discussed the criteria of intact stability of fishing vessels in this paper, because the experiment will continued.  
The parametric resonance phenomena were observed in this experiment in cruising the following seas, and the non-linear Mathieu's function was deduced by Dr. Blockie's method. The authors calculated the rolling motion by solving this equation numerically the Runge-Kutta-Gill method, and the results coincided with measured motion records.
- (d) Wind wave spectrum when the model capsized in wind waves;  
The capsizing in wind waves is a natural phenomena, which occure accidentally. It would be almost impossible to observe the capsizing of a model in the same conditions as in the basin experiments. The authors could not observe the capsizing of the model as often as in the basins, but they could obtain the wind wave spectra when the capsizing occurred and they are shown in Fig.26.

#### ACKNOWLEDGEMENT

The authors wish to thank many people who assisted and advised during the period of study. First, they would like to thank Dr.Tsuchiya and Dr.Takaishi. The experiments conducted at Lake Ohnuma performed with the cooperation and assistance of the Faculty of Fisheries of Hokkaido University and the National Research Institute of Fisheries Engineering. Finally, the authors would like to acknowledge the considerable assistance for people associated with the aforementioned institutions.

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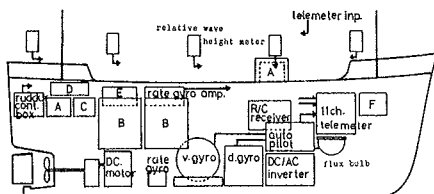
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## APPENDICES

### A.1. Schemetic diagram of arrangement of measuring apparatus for ship's motion and wave elevation

A  
dryfit battery 12V7.4Ah x1 (motor)  
B  
dryfit battery 12V20Ah x4 (gyro and autopilot)  
C  
dryfit battery 12V7.4Ah x1 (rate gyro)  
D  
dry battery 3V x2 (cudder cont. box)  
E  
dry battery 1.5V x12 regulated 12V (relative wave height meter)  
F  
battery 12V (transmitter)



### A.2. Probability of occurence of shipping water $P(x(t) > H_{bw})$

$$P[x(t) > H_{bw}] = \int_{H_{bw}}^{\infty} p(x(t)) dx$$

$$= \frac{1}{\sqrt{2\pi}\sigma_x} \int_{H_{bw}}^{\infty} e^{-\frac{(x(t) - \bar{x})^2}{2\sigma_x^2}} dx$$

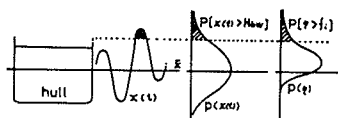
$$P[\xi > H_{bw}] = P[\eta > f_i]$$

$$= \int_{f_i}^{\infty} p(\eta) d\eta$$

$$= \frac{1}{\sqrt{2\pi}} \int_{f_i}^{\infty} e^{-\frac{\eta^2}{2}} + \sqrt{1-\epsilon^2} \eta e^{-\frac{\eta^2}{2}} \int_{-\infty}^{\frac{\eta\sqrt{1-\epsilon^2}}{\epsilon}} e^{-\frac{y^2}{2}} dy d\eta$$

$$F_i = H_{bw} - \bar{x}$$

$$f_i = F_i / \sqrt{m_0}$$



### A.3 Non-linear Mathieu's Equation on the rolling motion of fishing vessel

$$\ddot{\phi} + K_1 \dot{\phi} + K_2 |\dot{\phi}| \cdot \dot{\phi} + \omega_{\phi}^2 (1 - \epsilon_1 \phi^2 - h \sin \omega_e t) \phi = K(t)$$

$$h_{\phi 1} = \frac{K_{\phi 1} \cdot a_z}{2\omega_{\phi}}; K_{\phi 2} = -\nu \cdot A_w \left( \frac{dI_T}{dv} - b \right)$$

$$h_{\phi 2} = \frac{K_{\phi 2} \cdot a_z}{2\omega_{\phi}}; K_{\phi 2} = -\nu \cdot A_w \cdot x_{cr} \left( \frac{dI_L}{dv} - b \right)$$

$\phi$  ; roll angle

$K_1$ ; linear damping coefficient

$K_2$ ; non-linear damping coefficient

$\omega_{\phi}$ ; characteristic frequency

$\epsilon_1$ ; coefficient of non-linear restoring moment

$h$ ; amplitude of parametric excitation

$\omega_e$ ; frequency of encounter

$a_z$ ; amplitude of heav

$\nu$ ; specific gravity of water

$A_w$ ; waterplane area

$I_T$ ; transverse moment of inertia of waterline

$I_L$ ; longitudinal moment of inertia of waterline

$v$ ; displacement valume

### A.4 Coefficients on Non-linear Mathieu's Equation of the Rolling Motion in 19.9 GT Small Salmon Fishing Vessel for Following sea

$$FNA(A99, B99, C99) = C99 + 0.0 \cdot A99 + 0.0 \cdot B99$$

$$FNB(A99, B99, C99) = x4 \cdot B99 + x5 \cdot \sin(x6 \cdot A99) \cdot B99$$

$$+ x7 \cdot B99 + x8 \cdot C99 + x9 \cdot C99 \cdot \text{ABS}(C99)$$

PARAMETER E1	(x4)	19.6670
PARAMETER H	(x5)	2.0000
(GM)	(x7)	-6.0634
LINEAR DAMP. COEFF	(x8)	-0.0137
QUARD. DAMP. COEFF	(x9)	-0.0232
ROLL RATE INITIAL	(YY2)	0.4000
ROLL INITIAL	(YY1)	0.0000
STEP OF INTEGRATION	(H9)	0.1000
FREQUENCY OF ENCOUNTER	(FREQ)	0.5000
STEP OF ENCOUNTER FREQ.	(DPREQ)	0.0500
CAUTION FOR ROLL ANGL	(ANGL)	32.0000
NUMBER OF TIME SERIES DATA	(M)	400
MAX LAG	(L1)	40
DOLOOP OF ENCOUNTER FREQ.	(N)	10



Table 1. Principal dimension of fishing vessels and ship conditions in experiments

Items	H maru		E maru		M maru	
	Model	Ship	Model	Ship	Model	Ship
G.T. (ton)	-	19.99	-	124.0	-	96.0
Lpp (m)	2.00	15.20	2.00	30.20	1.986	28.30
B (m)	0.50	3.80	0.47	7.155	0.428	6.10
D (m)	0.195	1.48	0.175	2.650	0.187	2.65
Scale	1/7.6		1/15.1		1/14.25	
Disp.	129.05kg	56.65ton	125.0kg	441.1ton	89.3kg	264.86ton
df (m)	0.157	1.195	-	-	0.112	-
da (m)	0.201	1.530	-	-	0.190	-
dm (m)	0.179	1.362	0.178	2.690	0.151	2.15
Trim (m)	0.044	0.345	0.098	1.490	0.078	1.12
Fb (m)	0.016	0.118	-0.003	-0.040	0.036	0.50
GM (m)	0.049	0.372	0.053	0.800	0.052	0.74
Tr (sec)	2.10	5.79	1.545	6.004	1.56	5.89
Kxx/B	0.463	-	0.377	-	0.413	-
Kyy/Lpp	0.275	-	0.256	-	0.275	-

Table 2. Wave statistics in experiment with 96 GT salmon fishig vessel

		H <sub>1/3</sub> (cm)	H <sub>1/10</sub> (cm)	Tr (sec)
Exp. 1	Head	16.2	20.6	1.21
Exp. 2	Follow	13.5	17.2	1.18
Exp. 3	R.Quarter	13.0	16.6	1.12
Exp. 4	L.Bow	17.8	22.7	1.24
Exp. 5	R.Bow	17.6	22.7	1.24
Exp. 6	Head	16.5	21.0	1.25
Exp. 7	R.Beam	14.9	19.0	1.19
Exp. 8	L.Quarter	17.6	22.4	1.30
Exp. 9	R.Beam	16.3	20.7	1.22

Table 3 Wave statistics in experiment with 124 GT big offhsore trawler

	H (cm)	H <sub>1/3</sub> (cm)	H <sub>1/10</sub> (cm)	Tr (sec)
Exp A-1	7.33	14.05	17.59	0.89
	8.50	15.40	19.61	0.97
Exp A-2	9.14	17.62	22.71	0.96
	9.46	16.96	21.08	1.02
Exp B-2	8.45	17.28	22.80	0.90
	9.11	18.98	26.37	0.87
Exp A-3	10.82	21.23	27.78	0.99
	10.09	20.24	26.16	1.00

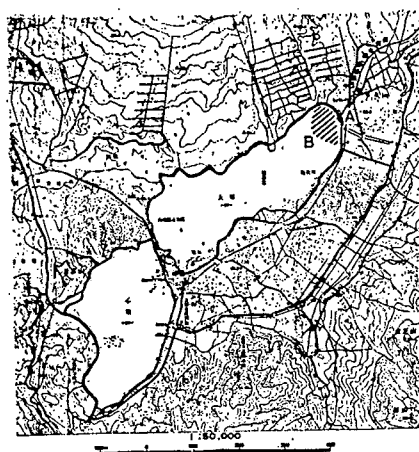
Table 5. Ship motion statistics in experiment with 19.9 GT small salmon fishing vessel

	Roll				Pitch				Roll-rate			
	Angle (deg)	Period (sec)	Angle (deg)	Period (sec)	Angle (deg)	Period (sec)	Angle (deg)	Period (sec)	Amplitude (deg/sec)	Period (sec)	Amplitude (deg/sec)	Period (sec)
	Mean	Sig.	Max	Mean	Mean	Sig.	Max	Mean	Mean	Sig.	Max	Mean
L.Bow	12.5	20.3	30.7	1.75	6.5	10.7	19.7	1.53	37.7	67.7	101.1	1.78
R.Bow	9.6	13.9	18.1	1.99	6.7	11.9	20.9	1.60	27.6	41.0	57.5	1.98
L.Beam	20.6	29.8	35.0	2.01	5.3	8.9	11.8	1.41	68.6	96.9	117.0	2.01
R.Beam	14.5	21.4	29.3	2.00	6.0	9.2	12.7	1.36	52.4	77.7	97.6	1.99
R.Quart.	16.2	22.7	25.6	2.52	4.4	6.8	8.0	1.70	46.5	70.9	79.5	2.22
Follow	14.1	23.1	31.2	2.58	5.8	8.7	10.6	2.75	25.7	36.2	47.9	2.66

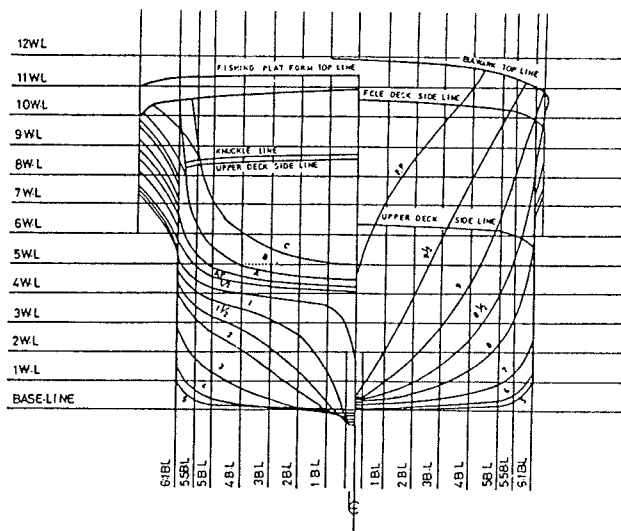
Table 4 Wave statistics in experiment with 19.9 GT small salmon fishing vessel

	H (cm)	H <sub>1/3</sub> (cm)	H <sub>1/10</sub> (cm)	Tr (sec)
L.Bow	17.5	28.3	37.1	1.63
R.Bow	15.4	24.8	30.5	1.46
R.Beam	14.5	24.0	31.7	1.56
L.Beam	16.8	27.3	36.2	1.64
R.Quarter	17.7	27.9	37.3	1.62
Follow	13.4	21.8	26.1	1.42
R.Beam	13.7	21.7	27.7	-

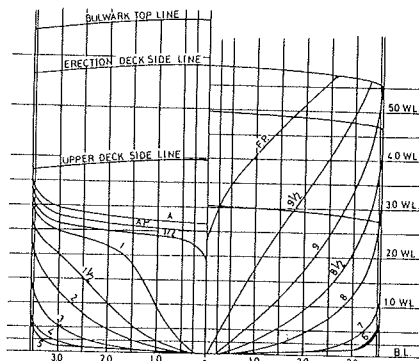
Fig.1. Experimental area at Lake Ohnuma



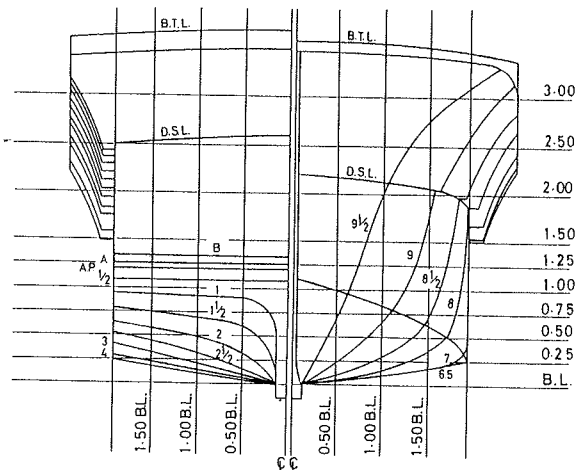
A: Observation station  
B: Experimental field



(a) Body ines for 96 GT steel made salmon fishing vessel



(b) Body lines for 124 GT big offshore trawler



(c) Body lines for 19.9 GT FRP made small salmon fishing vessel

Fig.2 Ship body lines for 3 kinds of fishing vessels used in this experiments

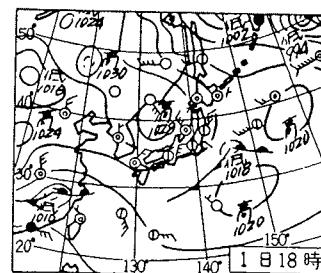
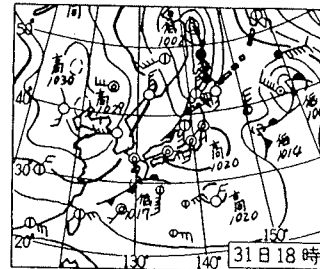
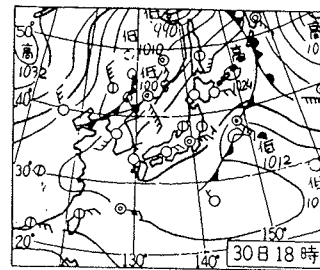


Fig.3 Weather charts at the experiment performed, for example

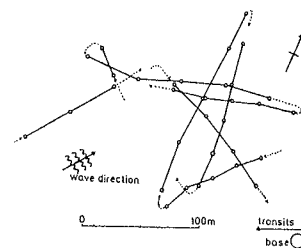


Fig.4 An example of cruising track of model

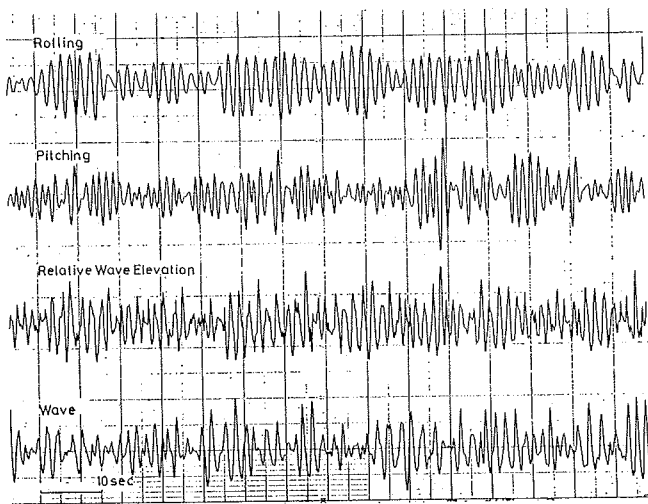


Fig.5 Record of rolling, pitching, relative wave elevations for model and wave height

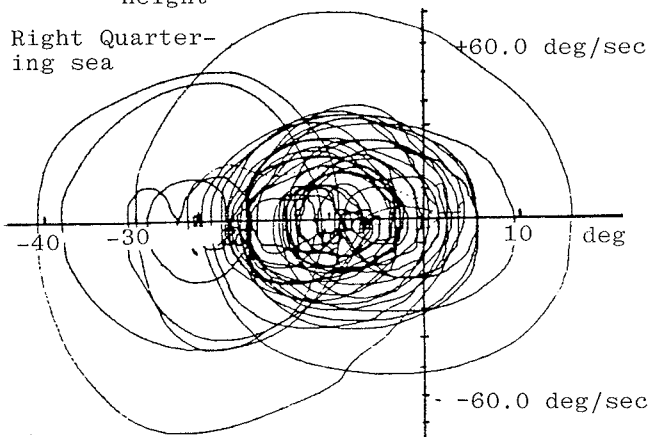


Fig.6 Rolling motion in phase plan  
abscissa: rolling angles  
ordinate: rolling rate

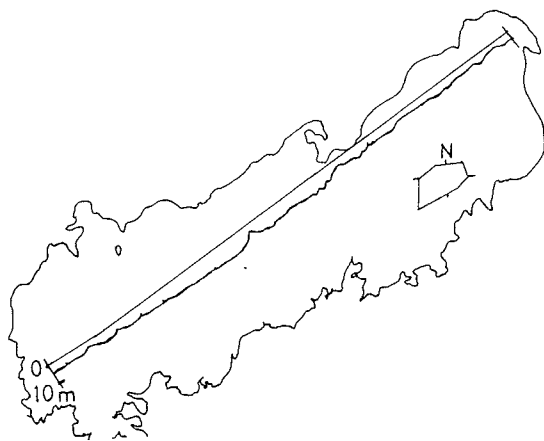


Fig.7 Depth recorded along maximum fetch and wind rose chart in Ohnuma

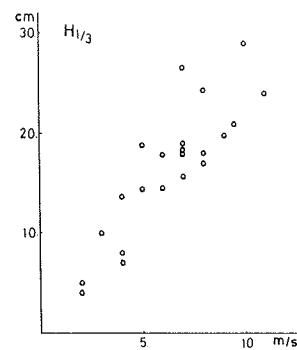


Fig.8 Relation between wind velocity and significant wave height in Lake Ohnuma

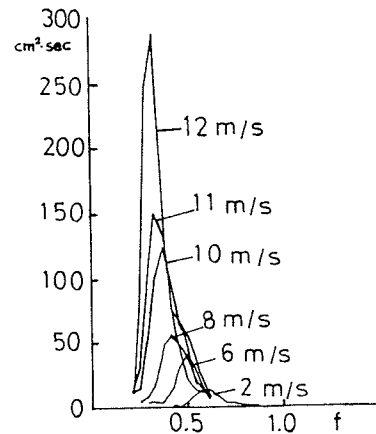


Fig.9 Rapidly developed wind wave spectra

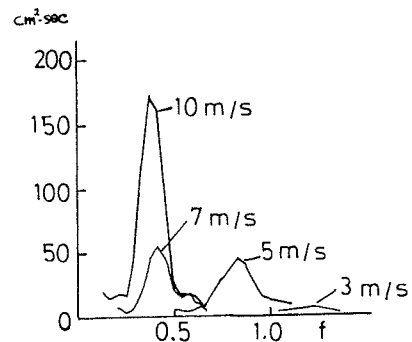


Fig.10 Wave spectra due to wind blowing constant velocity

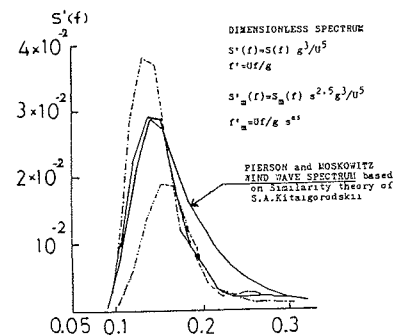


Fig.11 Numerical model of wind wave spectra

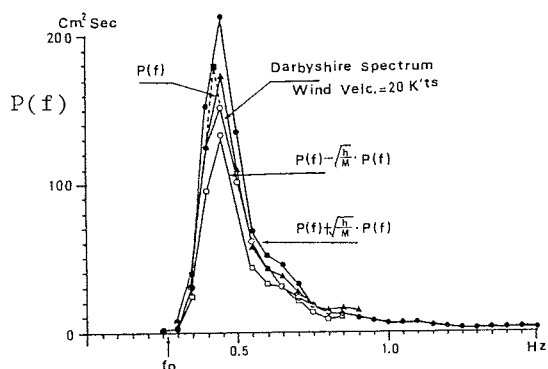


Fig.11b Comparison between Darbyshire spectrum and observed spectrum

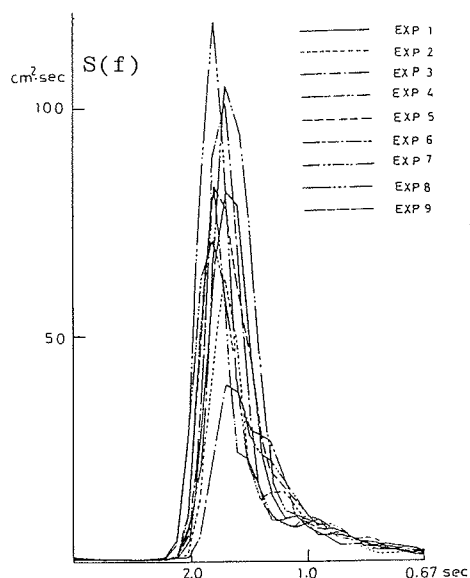


Fig.12 Wave spectra in experiment with 96 GT salmon fishing vessel

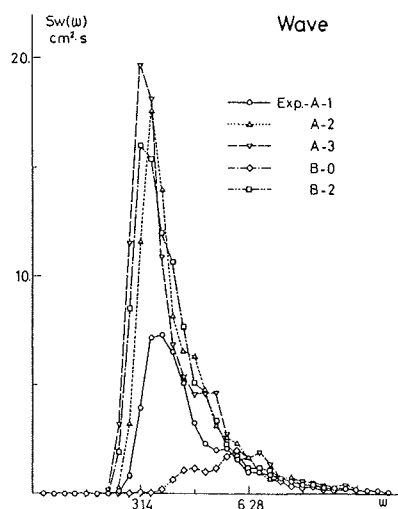


Fig.13 Wave spectra in experiment with 124 GT big offshore trawler

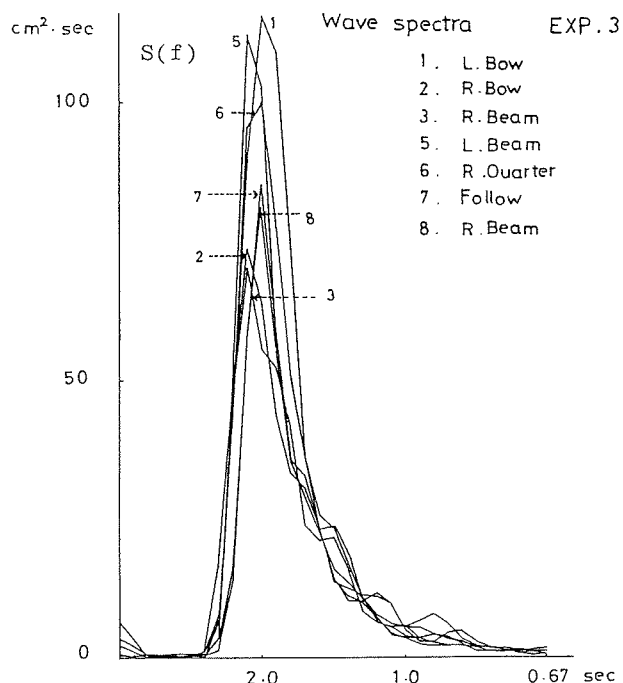
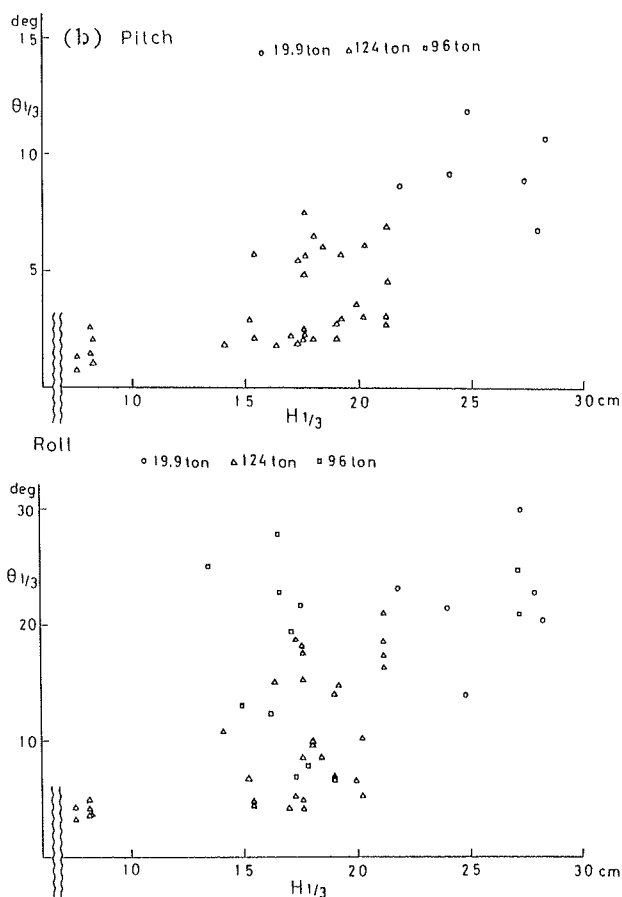


Fig.14 Wave spectra in experiment with 19.9 GT small salmon fishing vessel



(a) significant rolling angle versus  $H_{sig}$

Fig.15 Relation between ship's motions and significant wave heights ( $H_{sig}$ )

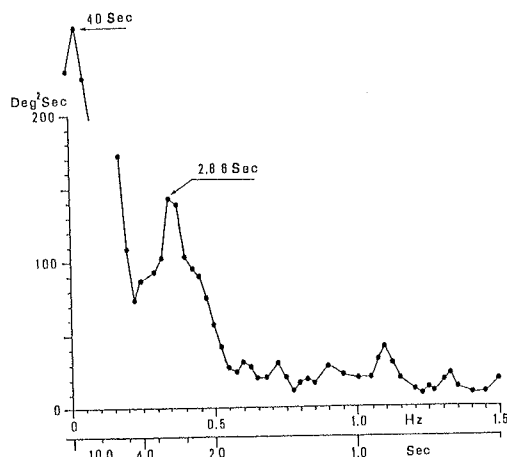


Fig.16 Spectrum for rolling angles in experiment with 96 GT salmon fishing vessel

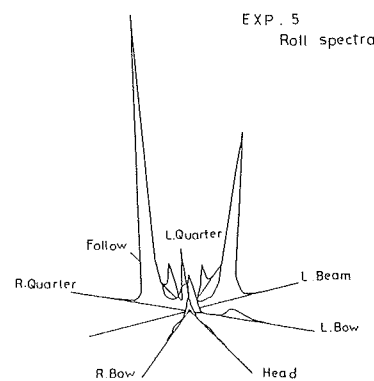


Fig.19 Spectra for rolling angles in experiment with 19.9 GT small salmon fishing vessel

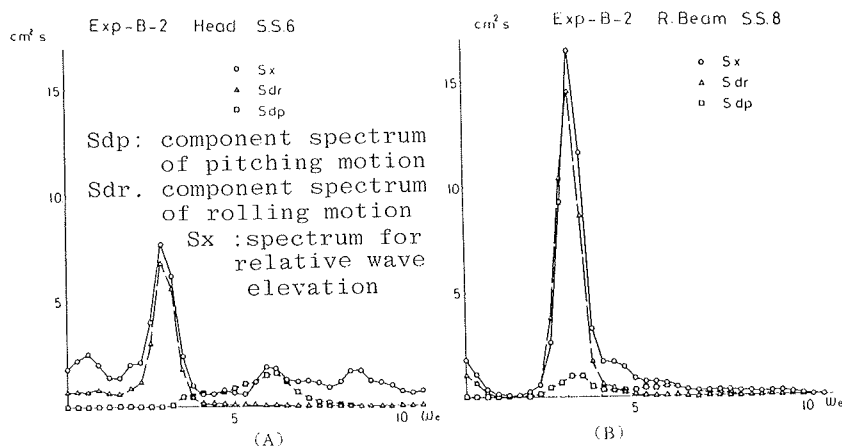


Fig.17 Power spectra for relative wave elevations in experiment with 124 GT big offshore trawler

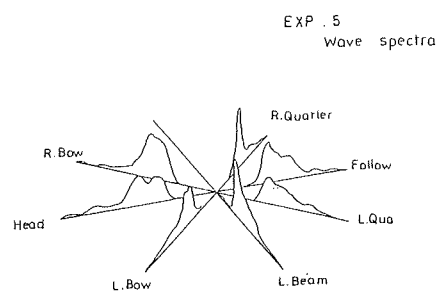


Fig.20 Wave spectra in experiment with 19.9 GT small salmon fishing vessel

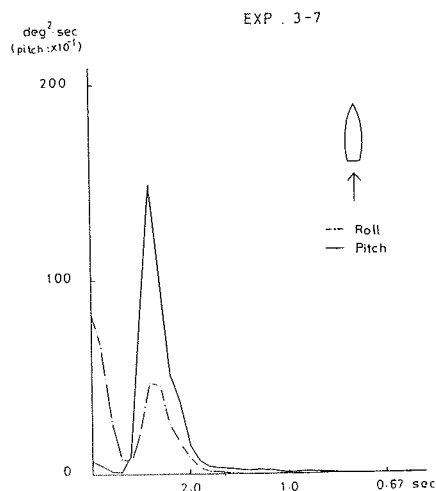


Fig.18 Spectra of ship's motions in experiment with 19.9 small salmon fishing vessel

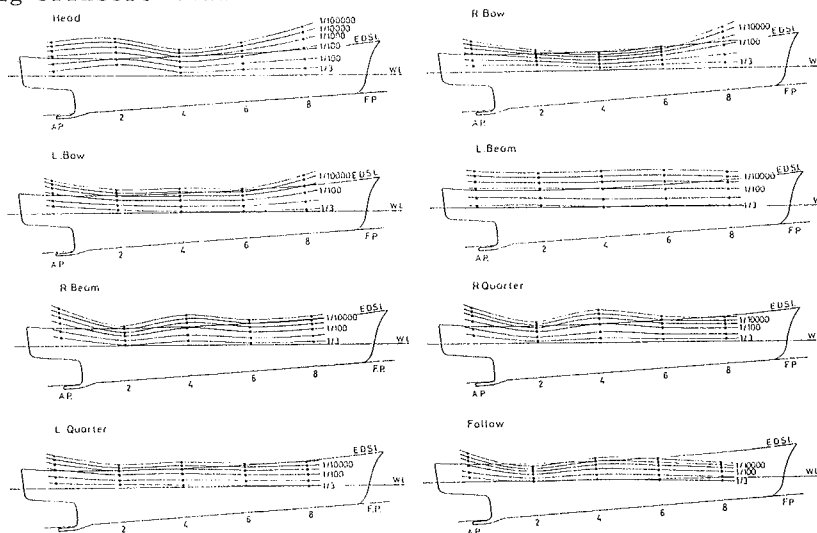


Fig.22 Equivalent probability level for relative wave elevations

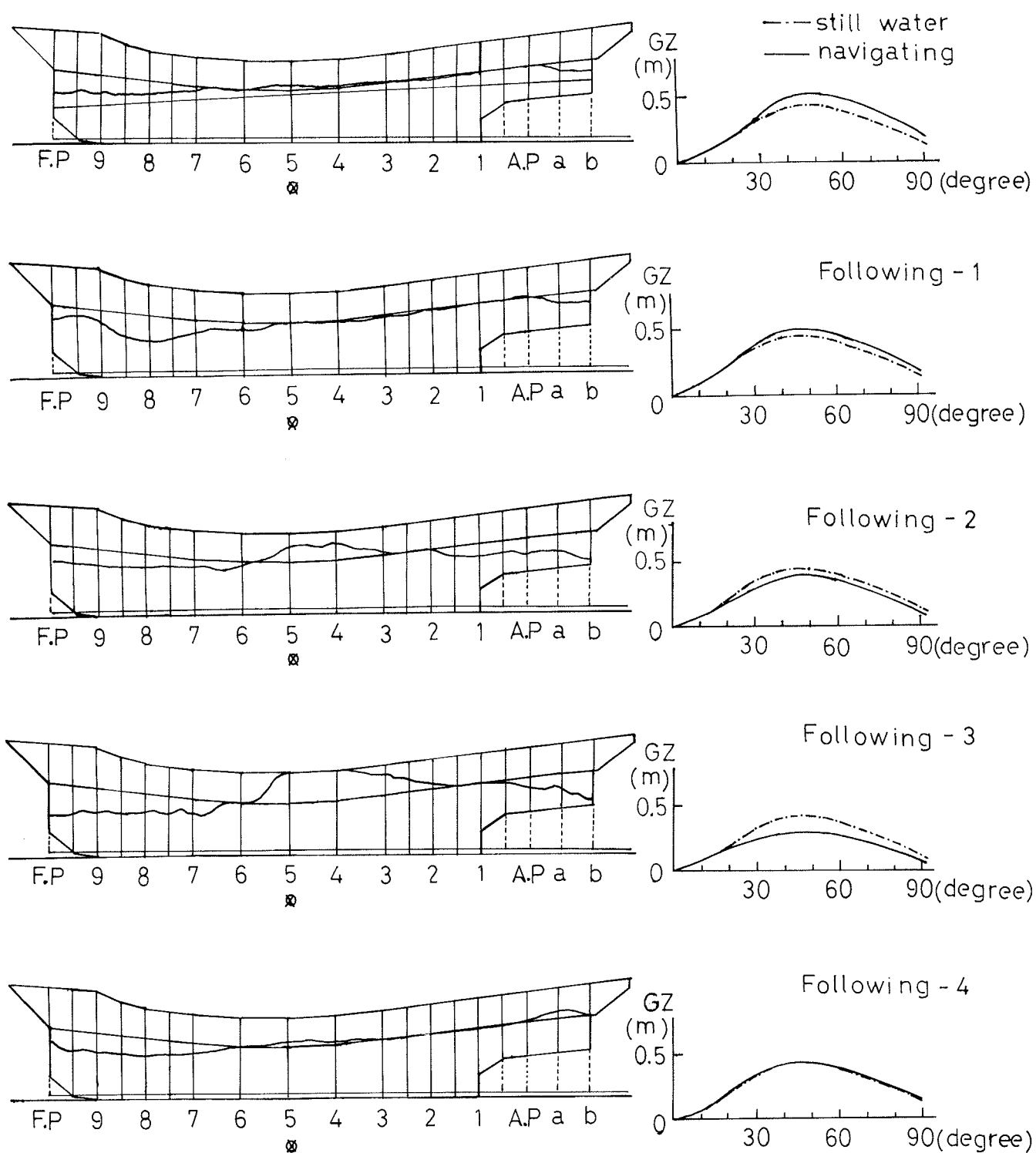


Fig.21 Relative wave profile and stability curve

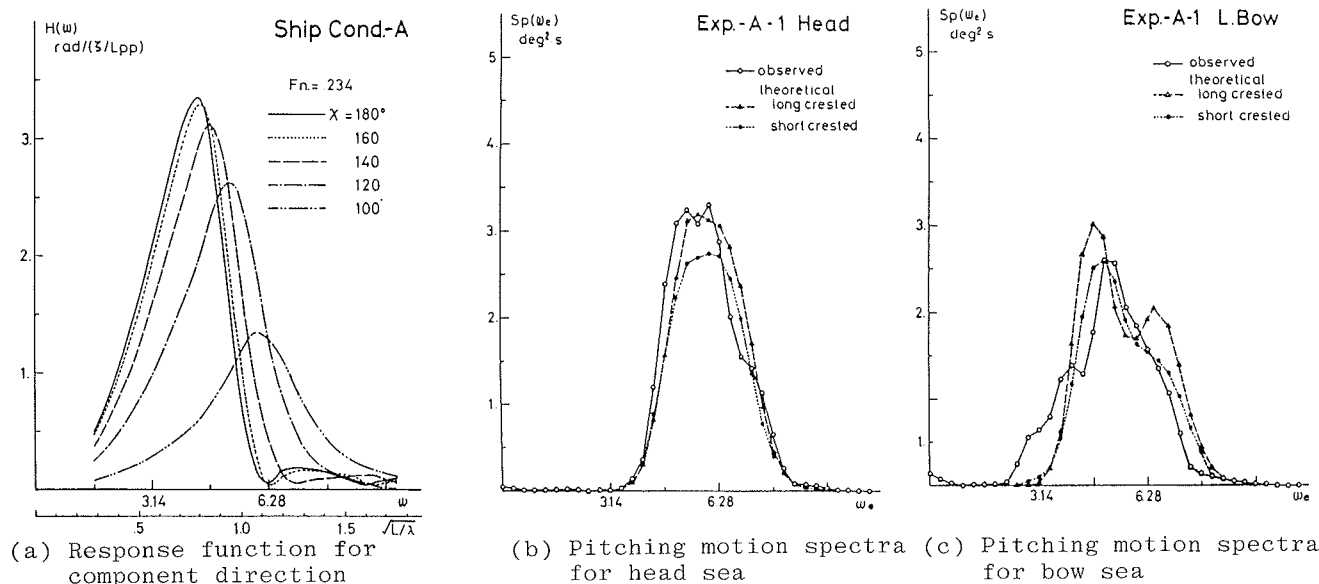


Fig.23 Comparison between calculated and observed longitudinal motion spectra in experiment with 124 GT big offshore trawler

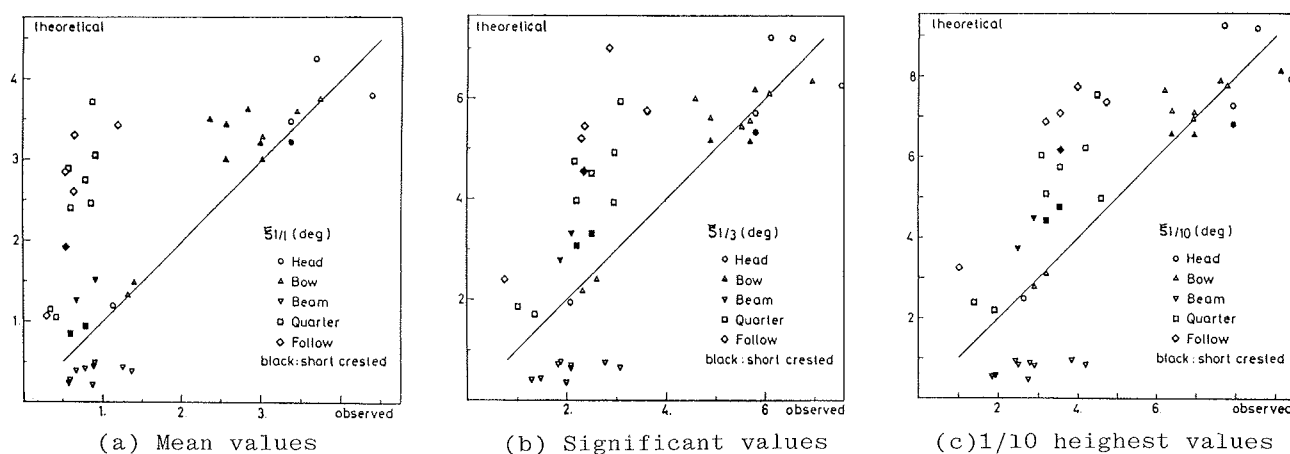


Fig.24 The highest mean values in maxima of pitching angle

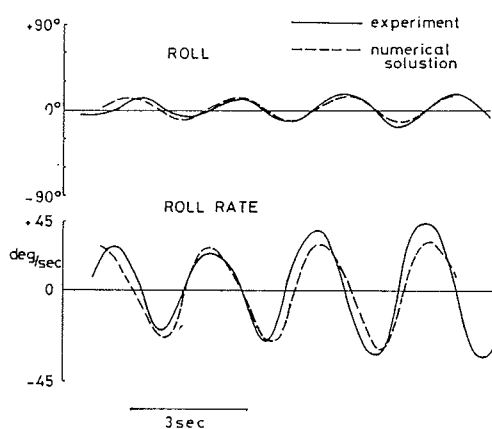


Fig.25 Comparison between calculated and measured in rolling motions

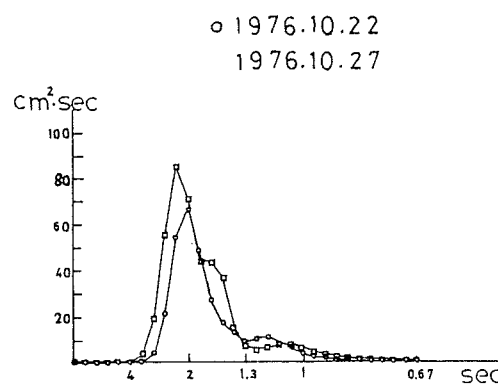


Fig.26 Wave spectra when capsizing occurred in wind waves

## Discussion

E. Dahle (University of Fisheries, Norway)

The paper can be looked upon as a progress report, as experiments will continue. Some points could be clarified, i.e.

1. More details of the various models could be given with regard to enclosed deck erections, building material for the E-maru and bilge keels.

2. In relation to 1, also GZ-curves for all vessels in still water could be given. As it stands, only GZ-curves for the H-maru is given in Fig. 21.

3. No capsizings occurred with the H- and the E-maru during the tests. This is interesting, and could be elaborated.

4. The lurching phenomena observed for the H-maru could be discussed further in light of figure 3 of the paper of Ikeda et al. of this Conference.

5. Are some of the models in paper SIV-1b (vessel 6) and in this paper (M-maru) identical? If so, a comment is appreciated.

### Author's Reply

Thank you Prof. Dahle for your discussions

1) Concerning the experiments of these 3 types of models, the concrete object of each experiment is not same for each experiment, then, the model did not have the same structure of actual fishing vessel. For example, in the case of E-maru, we did not consider that the whole structure of model, that is, under deck, upperdeck erection etc., should be similar with the actual fishing vessel, in detail. Strictly speaking, there must be some difference between the model and actual vessel.

2) We did not present the GZ curve of E-maru and M-maru for this proceedings, but we could show the characteristics of the stability for E-maru.

$d_f=1.751$ ,  $d_a=3.593$   $d_m=2.672$  Trim=1.842,  
KG=3.123, KM=3.559, GM=0.436 GZ<sub>max</sub>=0.408m  
 $\theta_{GZ_{max}}=42^\circ$ ,  $\theta_{range}=73^\circ 6'$

3) Concerning no capsizing in the case of H- and M-maru, we consider that the reason of this fact is quite different for each case. For H-maru, wave heights were not so high and the over-hanged deck was effective. In the case of E-maru, the model had the sheltered deck structure. Then, this structure was effective for stability of model as to have high free board in the experiment.

4) About lurching phenomena, as you pointed out, we would like to continue investigating further, connecting with the paper of Ikeda et al.

5) H-maru is identical with G-maru in the paper of Yamakoshi et al. concerning this problem, we had been studied with cooperation of 3 organizations, that is, SRI Japan, NRIF in Japan and Hokkaido Univ.

We are considering in stability problem especially, from the view point of capsize of fishing vessel. That we should study in closely connection with actual fishing vessel in commercial fishing, model experiment in wind wave and model experiment in basin.

According to these considerations, we have been investigated with same model using in each organization's facilities.

T. Tsuchiya (National Research Institute of Fisheries Engineering, Japan)

Stability Criteria for the following sea are expected more severe than that of the beam sea condition, and it will affect to the economic activity of fishing vessels strongly.

I would like to ask your opinion whether the stability criteria in the following sea condition for the boat builders are necessary, or they should only be reserved as the stability information to the skippers, because the skippers can easily change the course or ship speed even in the rough sea, if they think it is necessary.

### Author's Reply

We believe that in the case of fishing vessel, the navigator and the naval architect would share the responsibility for the safety of a fishing vessel on the seaway.

Also, for the navigator, it is quite necessary to have an ability to judge how to insure the safety of fishing vessel and to have knowledges concerning the stability of vessel.

The above mentioned is my fundamental point of view for the stability problem in fishing vessel, as being navigator.

Also, I consider that the stability criteria should be one of the index of the safety of vessel, but to keep the safety of the vessel, the navigator should have responsibility on the seaway, also the naval architect should make navigator to understand of situation for the stability problem.

A.R. Macnaughton (Dept. of Trade, UK)

May I compliment the authors of the thoroughness of their experimental technique and the impressive amount of interesting data produced in their paper.

My question is directed to the mutual moment of observed capsize in the models. What proportional significance would the authors attribute to the effects of parametric resonance effects compared to pure loss of stability due to heeling caused by wind and wave and water on deck?

### Author's Reply

The point of your discussion is being very difficult to answer, such as this



kind of experiments, because the phenomena observed would occur probabilistically, and also we consider that the capsize phenomenon would occur when the several causes would be superimposed.

In this paper, concerning parametric resonance phenomenon, we show that the parametric resonance were occurred in following sea on quite usual rolling motion record. Also, in Section 3-2 in this paper, we show that the capsize occurring in beam sea were observed in high wind obviously.

In this stage, it is quite difficult to speak deterministically that, which factor, as you show, that is, the effect of parametric resonance or pure loss of stability due to wind, wave and water on deck, would attribute in the capsize of fishing vessel.

But, this point is very important matter to clarify it, we would like to investigate about this problem. Thank you.

B.H. Adee (Univ. of Washington, USA)

Experience at the University of Washington in Seattle, Washington indicates that

rudder motion is very important in the loss of stability in a following sea. In your experiments how have you controlled model heading? It would also be very interesting to show rudder angle, roll motion and yaw angle as a capsize develops.

#### Author's Reply

We have controlled the model heading by the use of radio control system. We also consider that the rudder motion is very important, and we had used auto-pilot system for control heading automatically, at least to be more objective way.

But, we could not ascertain yet how effective this method is. According to rolling trajectory on phase plane, we could not consider when model are quartering and following sea, the rolling motions are to be pure rolling motion (one dimensional motion).

Further, we would like to continue studying the relation between rudder and rolling motions, as you pointed out.

*Session IVa*

## Stability of Container Ships and Environment

*Chairmen*

Mr. Joachim Jens  
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Japan

# AN INVESTIGATION ON INTACT STABILITY OF FAST CARGO LINERS

PETER BLUME AND HANS-GÜTHER HATTENDORFF

Hamburg Ship Model Basin

Federal Republic of Germany

## ABSTRACT

First results of an investigation on stability with two ships are presented. This investigation consists of tests in regular following waves extended by numerical simulations and tests in irregular following seas. The limiting values between safe and unsafe of usual stability parameters show a clear dependance from parameters describing the hull form. Therefore modified stability criteria are suggested including the influence of the hull form.

## NOMENCLATURE

$A, B, C, D_1$ $D_2, E, F$	Coefficients of equation of motion
$A(x)$	Cross section area
$A_w$	Waterplane area
$B$	Beam
$C_B$	Block coefficient
$C_w$	Waterline coefficient
$c_w$	Wave speed
$E$	Area under the rigthing lever curve
$E_R$	Remaining area under the rigthing lever curve beyond $\phi_{max}$
$F_n$	Froude number
$g$	Acceleration due to gravity
$GM_0$	Metacentric height
$GZ$	Rigthing lever
$H$	Depth of the hull
$i$	radius of gyration
$KG$	Height of the center of gravity above base line
$L_{pp}$	Length between perpendiculars

$m$	Ship's mass including hydrodynamic mass
$m_0$	Variance of the seaway
$R$	Resistance
$s$	Variance of the area $E_R$
$t$	Time
$T$	Draught
$T_0$	Draught to CWL
$T_e$	Period of encounter
$T_m$	Modal period
$T_\phi$	Natural period of rolling
$T_v$	Thrust minus thrust deduction
$V$	Volume of displacement
$v$	Speed
$W$	Faktor for the hull form
$X$	Coordinate
$Z$	Vertical acceleration
$\epsilon$	Phase angle
$\zeta_A$	Wave amplitude
$\zeta_w$	Wave height
$\eta$	Surging motion
$\lambda$	Wave length
$\rho$	Density of water
$\phi$	Rolling or heeling angle
$\phi_{St}$	Heeling angle due to a static moment

## 1. INTRODUCTION

At this time a comprehensive investigation on ship stability is underway in the Hamburg Model Basin. Within this investigation four large scale ship models shall be tested extensively during several years with the support of the German Ministry of Transport. This contribution deals with the tests and the results for the first two ships.

The test program for each ship consists of two parts. In the first part the models are tested in regular following waves at one displacement and one metacentric height only but with different static heeling moments. Up to now these tests are performed for the first ship only. Additional theoretical calculations in the time domain are done for the first model at the same conditions. The other part consists of tests in an irregular seaway. Here the static stability of the ship is changed at different displacements to find limiting values for safety against capsizing.

## 2. DESCRIPTION OF THE MODELS AND THE TEST TECHNIQUE

A semi-container ship (ship A) and a container ship (ship B) were chosen as the first test objects. The main dimensions are given in Table 1. Fig. 1 shows the cross sections of the hulls.

Table 1. Main dimensions of the ships :

		Ship A	Ship B
Length between p.p.	$L_{pp}$ [m]	135.00	202.40
Beam	$B$ [m]	23.00	32.20
Depth	$H$ [m]	10.70	18.85
Draught to CWL	$T_{CWL}$ [m]	8.20	9.00
Vol. of displ.(CWL)	$V$ [m <sup>3</sup> ]	17 190	34 585
Block coefficient	$C_B$ [-]	0.675	0.590
	$L_{pp}/B$ [-]	5.87	6.29
	$B/H$ [-]	2.15	1.71
	$H/T_O$ [-]	1.31	2.09
	$B/T_O$ [-]	2.81	3.58

Model scale

The models are built of GRP with deck and deckhouse of plywood. They were fitted with gyros for measuring the rolling, pitching and course angle and a transducer for the rudder angle.

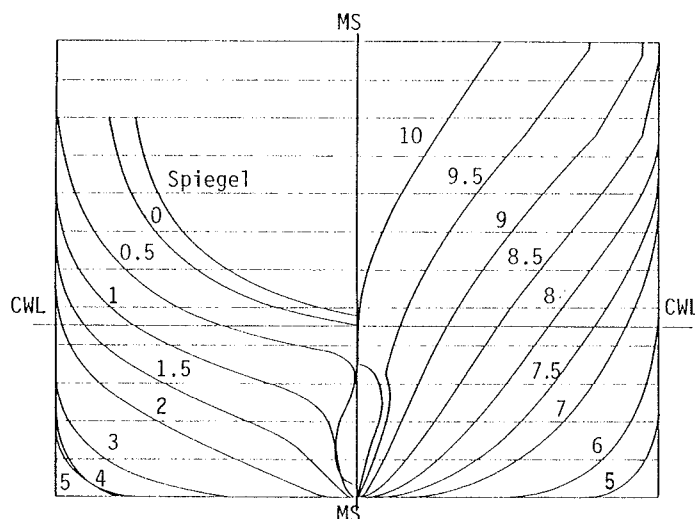
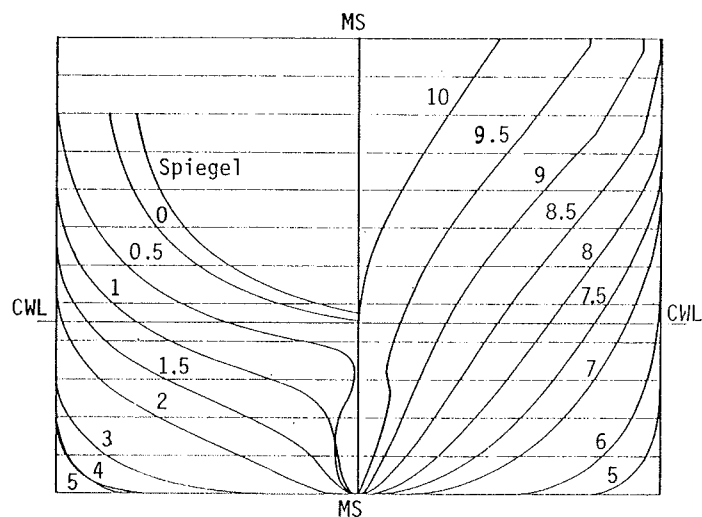


Fig. 1 Cross sections of Ship A and B

In the test series in irregular seas there was a movable weight installed within the model to simulate a heeling moment due to a steady wind force or a gust.

The self propelled model was hand operated by a helmsman being accomodated on a subcarriage, which can move across the basin in front of the main carriage.

The model was connected to the control platform by flexible cables to supply the electric current to the propelling motor and to the steering engine and, in addition, to transmit the measured values to the recorders carried on the subcarriage. During the tests carriage and subcarriage were following the model. So the influence of the cables on the motions of the model is negligible.

### 3. BEHAVIOUR IN REGULAR WAVES

### 3.1. Model Tests

The model of ship A was tested in one regular wave with a length  $\lambda$  equal to ship length  $L_{pp}$  and with a wave height  $\zeta_w = \lambda/15$ . The same wave data were used for calculating the righting lever curves in waves. All tests were performed at one load condition and one metacentric height, but at different ship speeds and 4 static heelings, which were adjusted by shifting a weight on deck. All relevant data are given in Table 2. Fig. 2 shows the righting lever curves in smooth water and in waves calculated in usual hydrostatic manner for the untrimmed ship.

Table 2. Data for the tests in regular waves with ship A

Wave	$\lambda$	[m]	135.00
	$\zeta_w$	[m]	9.00
Load condition	T	[m]	8.20
	V	[m <sup>3</sup> ]	17 190
Stability values	GM <sub>O</sub>	[m]	1.50
(smooth water)	GZ <sub>30</sub>	[m]	0.66
	E <sub>30</sub>	[m rad]	0.193
	E <sub>40</sub>	[m rad]	0.313
	E <sub>O</sub>	[m rad]	0.461
	T <sub>Φ</sub>	[sec]	14.2
Heeling angles due to static moment	Φ <sub>St</sub>	[°]	5.0; 6.9; 7.8; 10.0

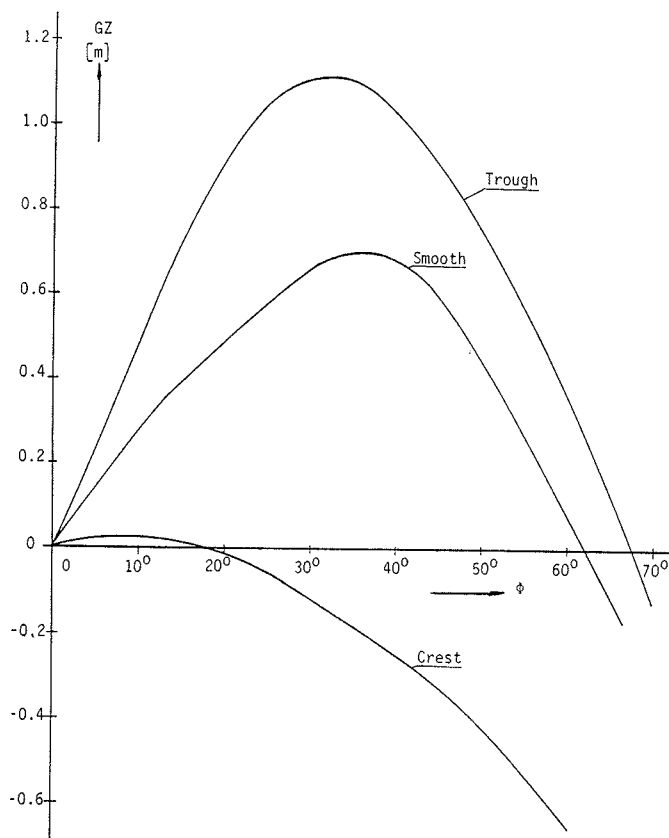


Fig. 2 Righting levers in smooth water and in waves for ship A at  $T = 8.2$  m,  $GM_0 = 1.5$  m

Although the waves were regular and the helmsman tried to keep a straight course in the direction of wave propagation, there are some deviations from the ideal course and changes in the mean speed. Therefore for analysing only those parts of the recordings were chosen, where the motions seemed to be rather stable and the circumstances were near to

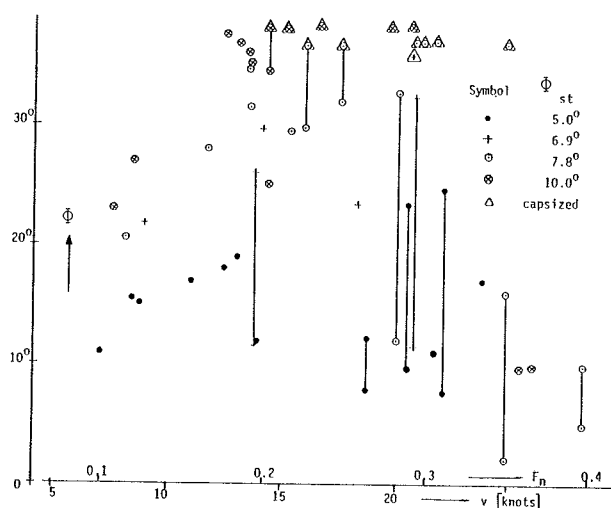


Fig. 3 Maximum roll angles in regular following waves as function of the mean speed for ship A

the ideal ones. The mean speed of the free running model had to be calculated from the measured period of encounter which could be determined from the recorded pitch angle. But small errors due to irregularities in the period of encounter can cause some differences in the calculated speed mostly at the higher mean speeds. Figure 3 shows the measured maximum roll angles as function of the mean speed. The results for the 4 static heeling moments are distinguished by different symbols. If the ship capsized after a relative short time the symbols are combined with a triangle. Capsizing after a longer nearly stable motion is marked by a symbol connected to a triangle by a line. The symbol indicates the maximum roll angle during the stable motion. Two connected symbols mean that there was a stable rolling motion with alternating higher and lower amplitudes.

### 3.2. Numerical Simulation Of Rolling Motions

For comparison rolling motions were calculated for the same conditions using a relative simple mathematical model. It takes into account the surging motion, nonlinearities in roll damping and the restoring moment and coupling effects with heaving. Effects due to course instability are not included. So the sum of forces in the longitudinal direction and of heeling and righting moments has to be zero. The balance of forces is given by the equation

$$R - T_v + \Delta R + m\ddot{\eta} = 0 \quad (1)$$

with  $R$  resistance of the ship as function of the speed ( $v_0 + \dot{\eta}$ )

$v_0$  speed in smooth water

$\eta$  periodical surging motion which is superposed to the motion  $v_0 \cdot t$

$T_v$  thrust minus thrust deduction

$\Delta R$  resistance oscillation due to the wave depending on phase

$m$  ship's mass including hydrodynamic mass

For the different parts of this equation the following estimations are used :

- the change of the resistance in smooth water is proportional to the squared speed

$$R = R_0(v_0) \cdot \frac{(v_0 + \dot{\eta})^2}{v_0^2} \quad (2)$$

- for constant torque at the propeller shaft the thrust is nearly constant

$$T_v = T_v(v_0) = \text{const} = R(v_0)$$

in combination with the resistance

$$R - T_v = R(v_0) \cdot (2/v_0 \dot{\eta} + 1/v_0^2 \dot{\eta}^2) \quad (3)$$

- the resistance oscillation is calculated using the pressure field in the undisturbed wave (Froude - Kriloff - Hypothesis). That means that the local buoyancy results in a force acting normally to planes of equal pressure. The integration of the horizontal component of these forces gives a usable estimation of the resistance oscillation

$$\Delta R = \rho g \int_L A(x) \cdot \zeta'(x) \exp(-\frac{2\pi}{\lambda} \cdot \frac{T}{2}) dx \quad (4)$$

Hereby the derivation  $\zeta'$  is given by :

$$\zeta' = \frac{2\pi}{\lambda} \zeta_A \sin \left( \frac{2\pi}{\lambda} (x_{Cr} - x) \right)$$

with  $x_{Cr}$  as the location of the wave crest relative to the midship section depending on time

$$x_{Cr} = x_0 + (c_w - v_0) \cdot t - \eta$$

The coordinate system moves with the speed  $v_0$ ,  $x_0$  is the crest-location at  $t = 0$ . After some modifications the equation (4) can be written

$$\Delta R = \Delta R_A \cdot \sin \left( \frac{2\pi}{\lambda} x_{Cr} - \epsilon \right) \quad (5)$$

with

$$\Delta R_A = \rho g \zeta_A \frac{2\pi}{\lambda} \exp \left( - \frac{2\pi}{\lambda} \frac{T}{2} \right) \cdot$$

$$\sqrt{\left[ \int_L A(x) \sin \left( \frac{2\pi}{\lambda} x \right) dx \right]^2 + \left[ \int_L A(x) \cos \left( \frac{2\pi}{\lambda} x \right) dx \right]^2}$$

and

$$\epsilon = \arctan \left[ \frac{\int_L A(x) \sin \left( \frac{2\pi}{\lambda} x \right) dx}{\int_L A(x) \cos \left( \frac{2\pi}{\lambda} x \right) dx} \right]$$

Using equation (3) and (5) the equation (1) gives after dividing by  $m$

$$A \ddot{\eta} + B \dot{\eta}^2 + C \sin \left( \frac{2\pi}{\lambda} x_{Cr} - \epsilon \right) + \ddot{\eta} = 0 \quad (6)$$

with the coefficients

$$A = \frac{R(v_0)}{m} \cdot \frac{2}{v_0}; \quad B = \frac{R(v_0)}{m} \cdot \frac{1}{v_0^2}; \quad C = \frac{R_A}{m}$$

This equation is solved after setting of starting values from  $\eta$  and  $\dot{\eta}$  by step by step integration in the time domain. Coupling effects with the roll motion (for example a resistance component depending on the roll angle) are not taken into account.

In the same manner the equation of the rolling motion given by the balance of moments is solved

$$\ddot{\phi} + D_1 \cdot \dot{\phi} + D_2 |\dot{\phi}| \cdot \dot{\phi} + E \cdot GZ(\phi, x_{Cr}, v) - F \cos \phi = 0 \quad (7)$$

The meaning of the coefficients is

$$- D_1 = \frac{B_L}{mi^2}; \quad D_2 = \frac{B_Q}{mi^2};$$

Coefficients for the linear and quadratic term of the roll damping,  $i$  means the radius of gyration for rolling including hydrodynamic effects

$$- E = \frac{g + \ddot{z}(x_{Cr})}{i^2}$$

Coefficient of the restoring moment, which includes the coupling with the heaving motion by the vertical acceleration  $\ddot{z}(x_{Cr})$  of the centre of gravity

$$- GZ = GZ(\phi, x_{Cr}, v_0 + \dot{\eta})$$

righting arm of the restoring moment depending on heeling, relative location to the wave and speed.

These values are calculated in a hydrostatic manner neglecting and including an influence of speed, which is discussed later on together with the results.

$$F = \frac{1}{i^2} \cdot g GM_0 \sin \phi_{st}$$

coefficient for the static heeling moment, which causes a static heeling angle  $\phi_{st}$ .

The connection to the surging motion is given by the location  $x_{Cr}$  of the wave crest. In practice the integration of equation (6) starts first. When the surging motion is stable the integration of equation (7) starts. At this start the static heeling angle is increased linear with time over 60 sec (model time) from zero up to the desired value. The simulation is stopped if the rolling motion becomes stable or the model capsizes.

These calculations were only done for ship A in model scale. For determination of the coefficients  $A$ ,  $B$ ,  $D_1$  and  $D_2$  the curves in Fig. 4 are used. For the model resistance  $R$  the results of older resistance tests could be used which had to be extrapolated to higher speeds corresponding to serial data. The coefficients  $B_L$  and  $B_Q$  are based on the results with similar models [1] and include the influence of bilge keels with length of 25 % Lpp and 14 mm height. Strictly speaking these values are a good approximation for roll angles up to about 20 to 25 degree only. But because of the absence of reliable information for larger roll angles the tests results had to be extrapolated.

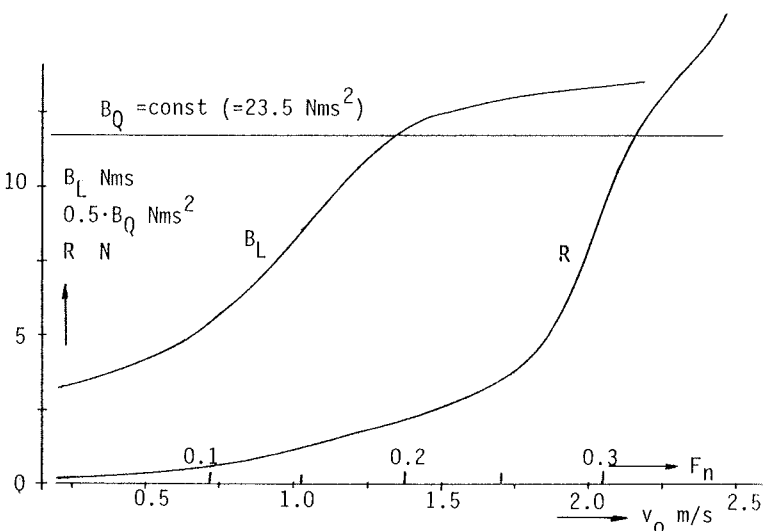
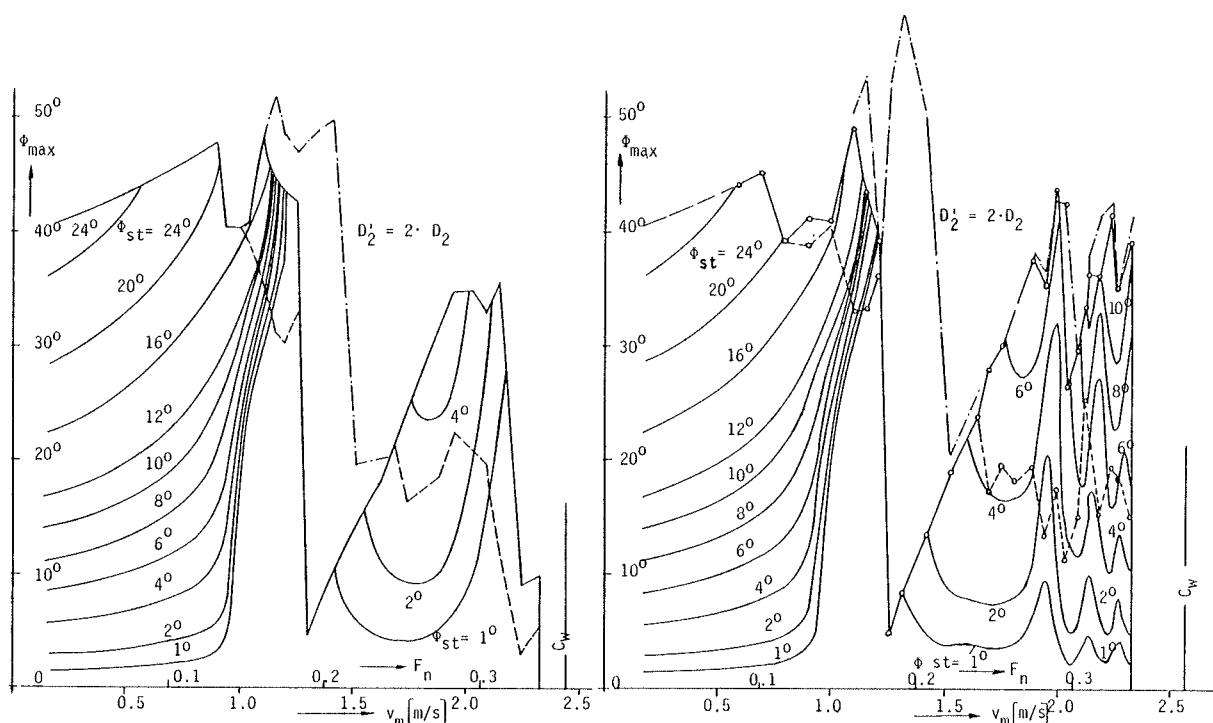


Fig. 4 Resistance and coefficients of linear and quadratic roll damping of model A.

At first these calculations were done for a range of speeds with different static heeling angles  $\phi_{st}$  up to capsizing the model using righting levers, which were calculated for an undisturbed wave. For eight locations of the ship relative to the wave these levers were calculated in the usual hydrostatic manner using trim angles of the ship obtained by dynamic calculations with strip theory. The value for the actual location and heeling angles was found by interpolation.



a) righting levers independent from Froude number

b) righting levers dependent from Froude number

Fig. 5 Results of numerical simulation - maximum roll angle as function of mean speed and static heeling moment

Fig. 5 a shows the results of the calculations. There the maximum roll angle for constant hydrostatic heeling angle  $\phi_{st}$  is drawn as function of the mean speed as in Fig. 3 which shows the test results. These curves are limited by a line beyond that the model capsized. Two resonance peaks are recognizable connected with a sharp decrease of the tolerable roll angle due to the phase shift of the motion. The dotted lines remark the lower maximum roll angle in ranges where the rolling motion shows different extreme values. In the first range near to the resonance peak the time series became somewhat unstable whereas in the range at higher speeds and near to  $T_e/T_\phi = 2$  there were alternating higher and lower amplitudes which were stable. This behaviour we could observe in our test too.

Mean speeds between 2.33 m/s and the wave speed  $C_w$  are not possible for the model. In that region the model is accelerated up to the wave speed and runs with the wave in a position with the midship section a small distance behind the trough. Here the righting levers are large. The ship isn't endangered, if it can keep speed and course stationary. But in reality this is not possible for longer times and the passage of a wave with a long period of encounter can be dangerous as shown by the results at a mean speed close to 2.33 m/s.

Because of the uncertain roll damping the effect of damping was studied by calculations with a larger coefficient  $D_2$  for the quadratic term. It was increased by a factor two.

Remarkable changes occur near to the first resonance peak only. The tolerable roll angles are increased to the dash-dotted line.

One can recognize that the tolerable static heeling angle  $\phi_{st}$  are very large at smaller speeds (up to about  $F_n = 0.15$ ), very small in the regions of resonance and small in the high speed range between the two regions of resonance. Here they are smaller than the values we found from the tests.

In the high speed range (Froude number higher than 0.2) the ship has its own wave system with a higher water level at bow and stern and a lower level midships. This must improve the low righting levers at ship on wave crest, the levers are a function of speed.

As a better approximation now righting levers were calculated using a wave profile which results from the superposition of the external wave and a profile of the wave system of the ship. For the ship's wave profile a formula was derived by regression of results in [2] and scaled by wave photos of the actual ship:

$$\zeta = 0.442 \cdot F_n \cos \left( \frac{1 - 2F_n^2 - 2 \frac{x}{L}}{2F_n^2} \right) - (0.082 + 0.025 \frac{x}{L}) F_n^2 \cos \left( \frac{4\pi x^2}{L^2} \right) \quad (8)$$

In Fig. 6 the used profiles are drawn for the situation ship on wave crest. The profile for  $F_n = 0$  is the one used for the first simulation; it is the undisturbed external wave. With increasing Froude number the crest and the trough become at first something, at  $F_n > 0.3$  much more smaller. The effect on the righting levers is shown in Fig. 7. Especially the low righting levers at the situation ship on wave crest become much higher at higher Froude numbers.

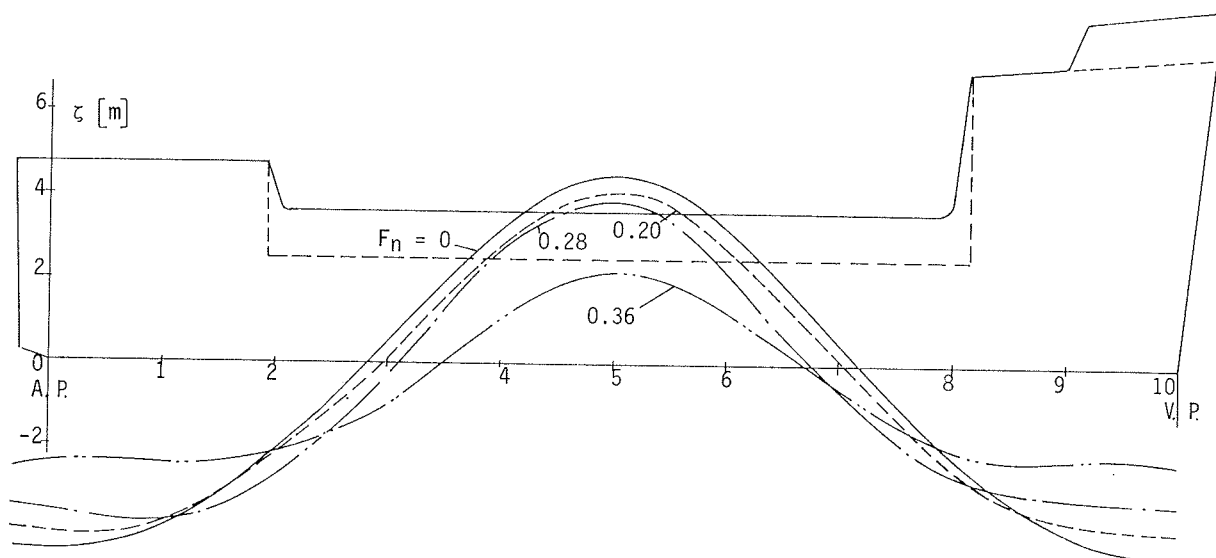
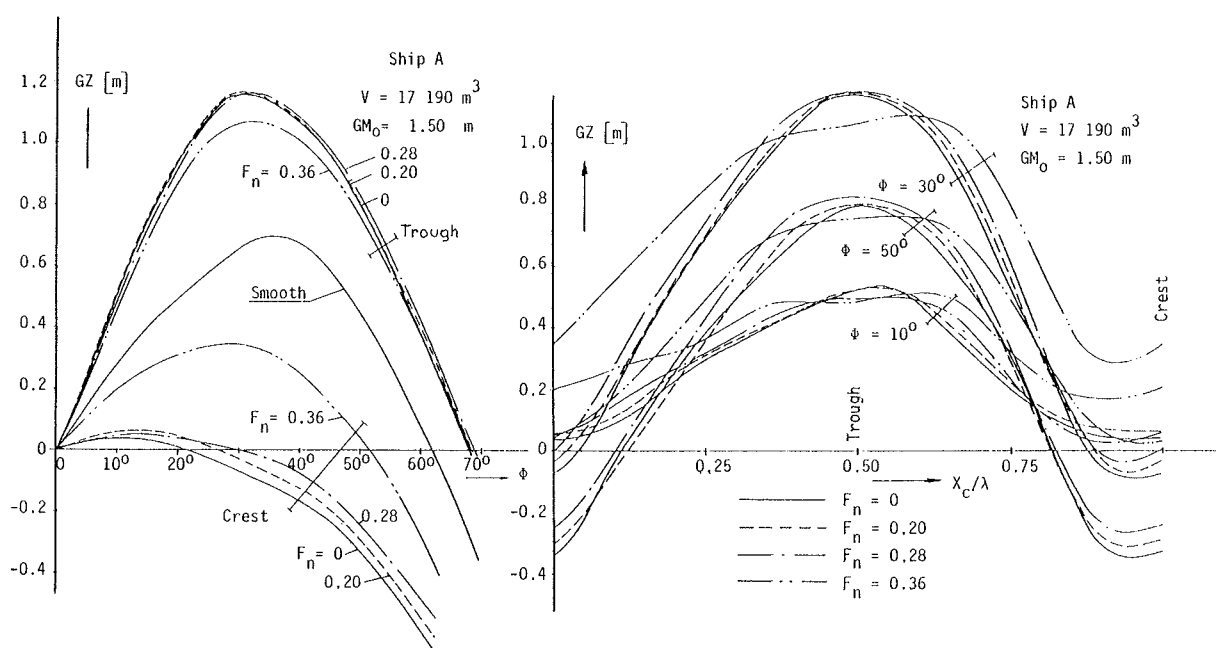


Fig. 6 Wave profiles at different speeds, ship on wave crest



a) as function of heeling angle

b) as function of phase relative to the wave

Fig. 7 Righting lever curves in smooth water and in waves at different Froude numbers



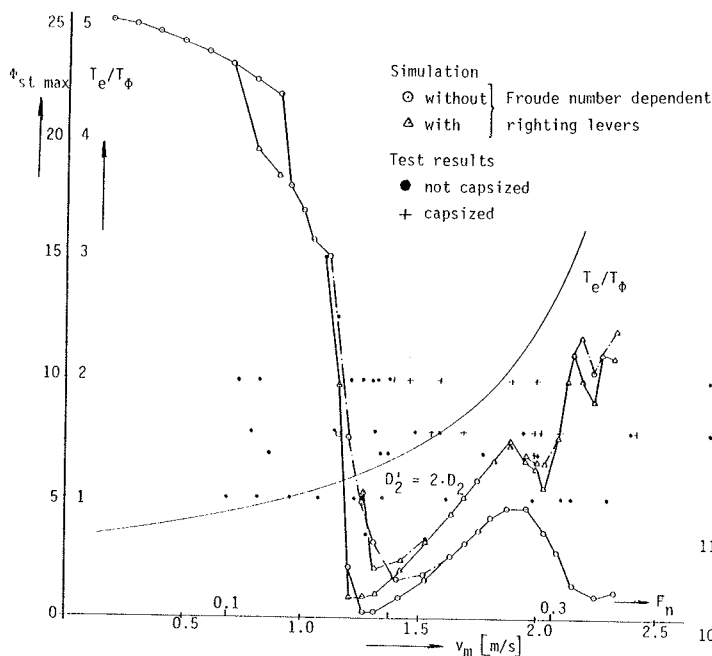


Fig. 8 Tolerable static heeling angles as function of mean speed.

The results of the simulation with Froude number dependent righting levers can be studied in Fig. 5 b. Remarkable differences can be expected only at higher Froude numbers. Because of the larger righting levers the effective natural roll period is shorter and more resonance peaks occur. The tolerable static heeling moments become larger in the high speed range and are now similar to the angles observed in the tests.

Fig. 8 shows the tolerable angles  $\phi_{st max}$  for both simulations in comparison to the test results. The line valid for the simulation with Froude number dependent righting levers separates quite satisfactorily the tests with and without capsizing heaving in mind the possible scatter in the mean speed at the test results.

The used mathematical model seems to describe these tests under idealized conditions very well. The results show the well known fact, that a ship in following sea is mostly endangered near to resonance at  $T_e/T_\phi = 1$ . The doubled quadratic roll damping doesn't alter this statement considerably. But also in the whole higher speed range the danger of capsizing is not negligible.

#### 4. MODEL TESTS IN IRREGULAR WAVES

The main part of the test program consists of tests in following and quartering irregular waves. After very few initial tests with different seaways the control signal for generating the waves was kept constant for most of all further tests. The hitherto investigated ships have nearly the same length in model scale but different scale. That means that the same seaways relative to the ship were used which makes it easier to study the influence of the hull form. Fig. 9 shows the results of the wave analysis in comparison to the JONSWAP-Spectrum with the same modal period in full scale values for ship A. The  $m_0$  - values of the different runs vary between 13 and 16  $m^2$ . The values for ship B have to be scaled

using the ratio of the model scales. Only in some tests with ship B at one draught a smaller seaway with the same full scale period and wave height as for ship A was used. The mean values of periods and significant wave heights as well as the measured extreme wave heights are given in the following table for both ships :

Table 3 : Data of the irregular seaways

	Ship A and B	Ship B
$T_m$	13.1 s	15.6 s
$\zeta_W 1/3$	14.6 m	20.8 m
$\zeta_W max$	about 22 m	about 31 m

For the judgement of the seaway with respect to safety against capsizing the extreme values should be used. The ratio of  $\zeta_W max/\zeta_W 1/3$  is smaller for the model seaways than in the nature due to the absence of the small waves in the high frequency range and the short measuring time.

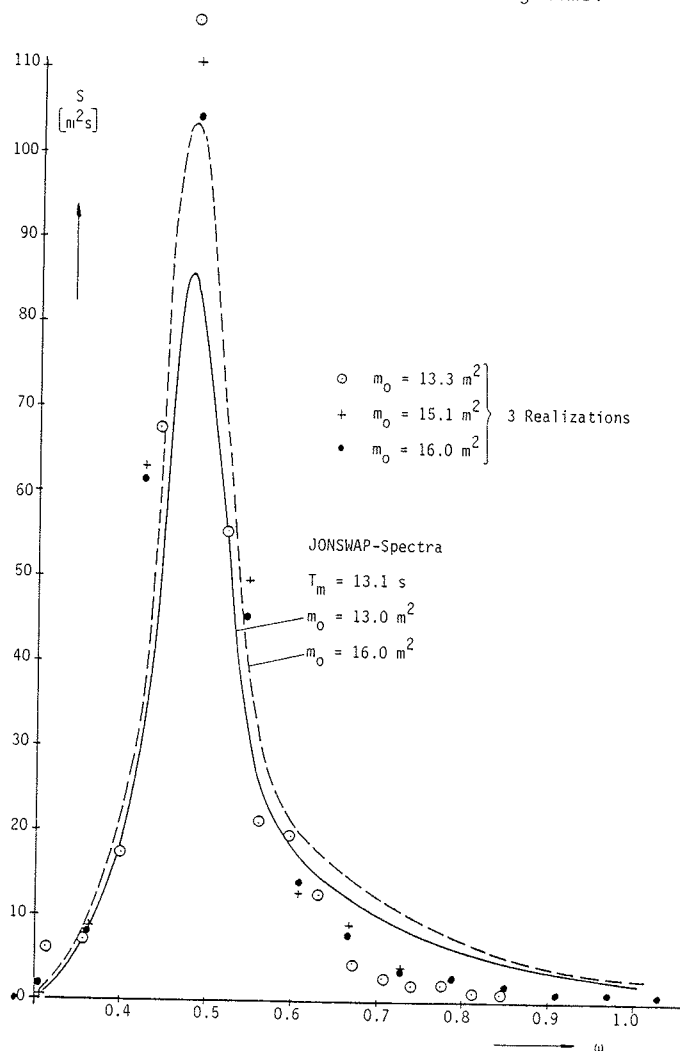


Fig. 9 Wave spectrum for the tests with ship A

Each model was tested at three displacements. At each displacement different metacentric heights were adjusted in order to find the limit between the safe and the unsafe condition. Besides some tests in beam seas at  $v = 0$  the tests were performed in following seas and at courses about 30 degree to the

direction of wave propagation. In most of the tests a constant heeling moment of 8960 kNm for ship A and of 30130 kNm for ship B (e.g. simulating a wind moment) was acting on the model; additionally a momentum simulating a gust was used in some cases during testing of ship A and not at all during testing ship B.

Tables 4 and 5 give the relevant data for the different loading and stability conditions as well as the number of runs and capsizing.  $\phi_w$  means the static heeling angle due to the heeling moment. There are less capsizings in the table for ship B than for ship A. This follows from the variations of the metacentric height. At testing ship A in the absence of any information at first to small metacentric heights were adjusted, whereas the tests with ship B were started with the higher values.

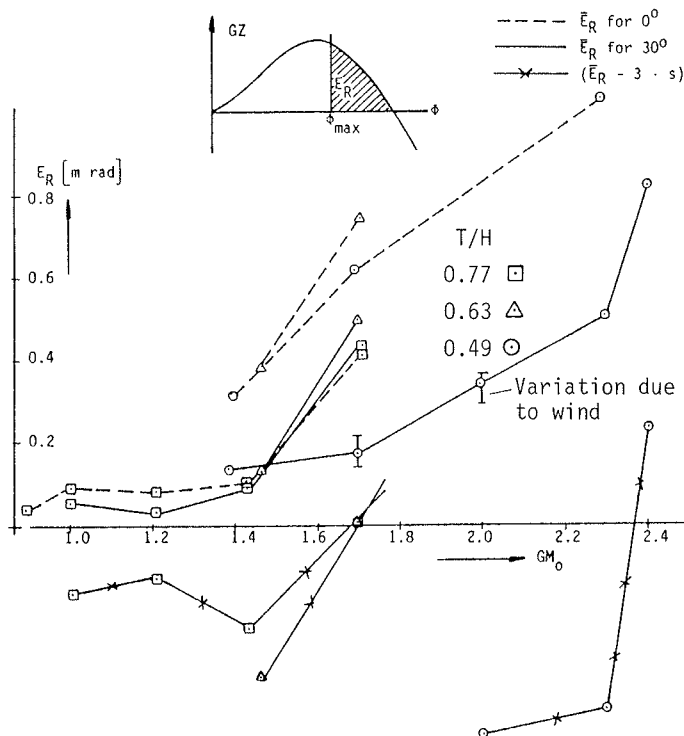


Fig. 10 Remaining area under the smooth water righting lever curve above the measured maximum roll angle for ship A

At earlier capsizing tests we found that the visual impression of the danger of capsizing corresponds very well with the remaining area  $E_R$  under the smooth water righting lever curve beyond the measured maximum heeling angle  $\phi_{max}$ . Therefore this area is used as a measure of the safety for runs where the ship doesn't capsize. The areas  $E_R$  were determined for all runs using the maximum roll angle of each run.  $E_R$  was set zero if the model capsized or if  $\phi_{max}$  was greater than the angle  $\phi_0$  where the righting levers become zero (this happened very seldom). Then the mean values  $E_R$  and the variance  $s$  over all runs at the same conditions were calculated. Because of the small number of tests  $E_R$  and  $s$  are subject to statistical errors.

This analysis shows that the differences in safety due to changes in the natural roll period in irregular seas seem to be very small. This is contrary to the impression one might get from Table 4

( $T = 8.20$  m,  $GM_0 = 1.0$  m). The mean values  $E_R$  are very small in both cases and the differences could be caused more by a different collective of mean ship's speed. The mean value over all runs with  $T\phi_0 = 16.5$  s is 15.3 knots and 13.0 knots for the runs with  $T\phi_0 = 20.5$  s. For 20.5 s there are less runs at higher speeds where most of the capsizings occur in accordance with the results of the simulation described above. Therefore the mean value  $E_R$  and  $s$  are taken over all tests at  $GM_0 = 1$  m independent from the natural roll period.

Fig. 10 shows the calculated mean values  $E_R$  for ship A as function of the metacentric height  $GM_0$  for all three draughts and both courses relative to the sea. At the large draught the danger of capsizing is about the same for both courses whereas there is a significant difference at the smaller draughts. Most of the tests were performed with a static moment acting on the ship (for example due to a side wind force). At some points the variation of  $E_R$  due to this moment is indicated; the influence is relatively small.

For finding a limit between safe and unsafe conditions curves are used, which are three times the variances  $s$  lower than the mean values  $E_R$ . For a safe condition shall be required that  $(E_R - 3 \cdot s)$  is equal or larger than zero. In Fig. 10 these curves are also drawn (lines marked with x) for the most dangerous course that means for about  $30^\circ$  at the two smaller draughts, whereas for  $T/H = 0.77$  mean values for both courses were used. In this way the limiting metacentric heights were determined with  $GM_0 = 1.70$  m for  $T = 8.2$  m and 6.7 m and  $GM_0 = 2.35$  m for  $T = 5.2$  m.

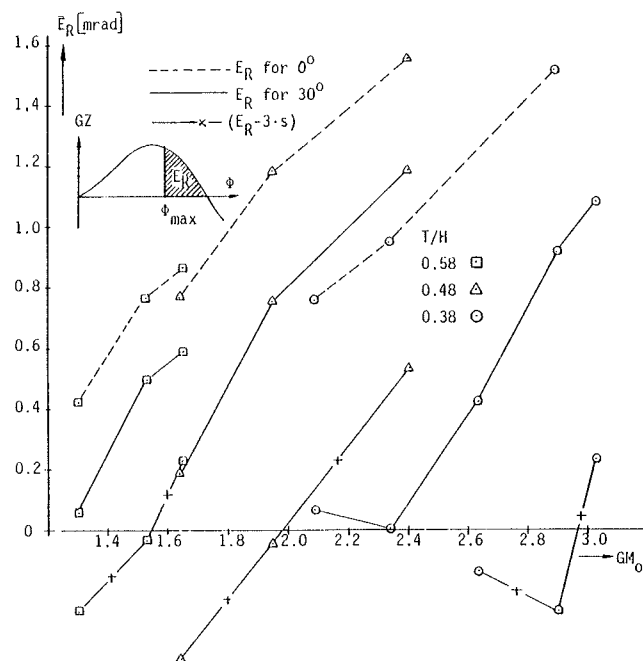


Fig. 11 Remaining area under the smooth water righting lever curve above the measured maximum roll angle for ship B,  $\zeta_{wmax} = 31$  m

Fig. 11 shows the same curves for ship B in the same seaway relative to the ship length as used for ship A. This ship is mostly endangered in quartering waves at all three draughts. The difference to the safety in following waves is increasing to smaller draughts as for ship A too. The limiting metacentric

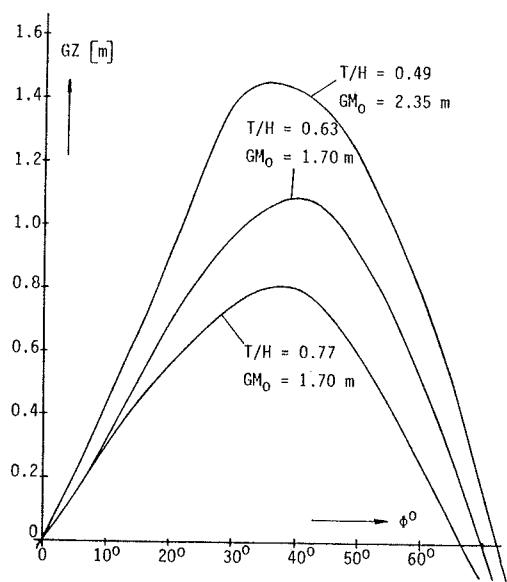


Fig. 12 Righting lever curves in smooth water for ship A at the limit safe to unsafe

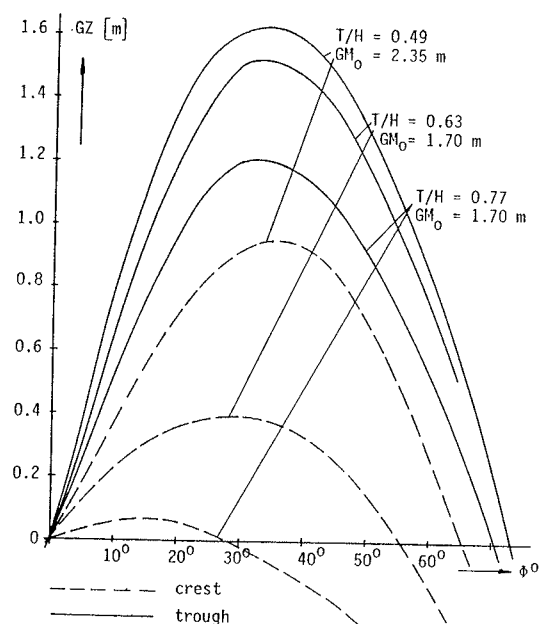


Fig. 13 Righting lever curves on wave crest and trough for ship A at the limit safe to unsafe

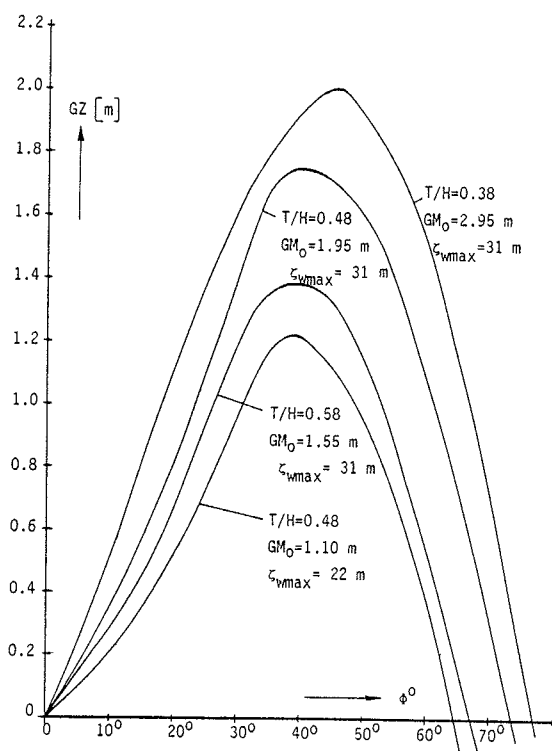


Fig. 14 Righting lever curves in smooth water for ship B at the limit safe to unsafe.

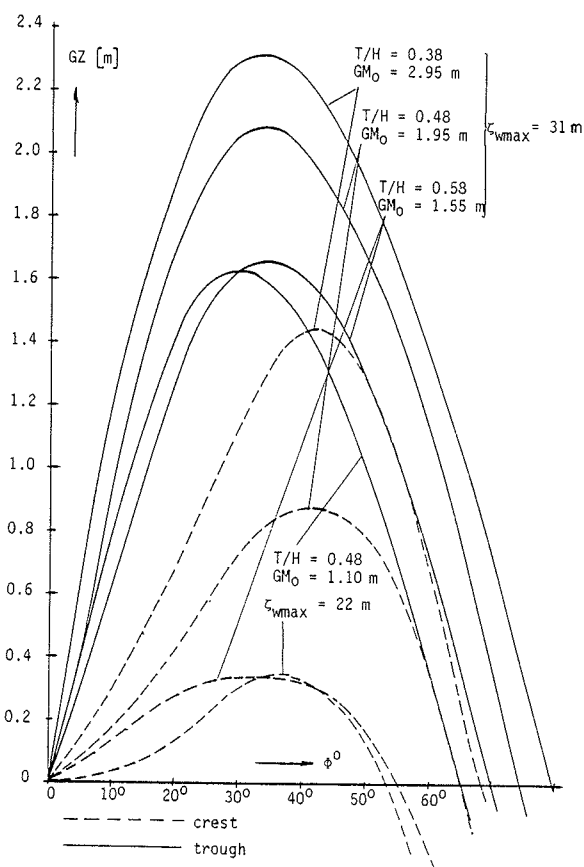


Fig. 15 Righting lever curves on wave crest and trough for ship B at the limit safe to unsafe

heights for ship B are 1.55 m, 1.95 m and 2.95 m at  $T = 11.0$  m, 9.0 m and 7.1 m. In a seaway with the same absolute wave height and period compared with ship A a metacentric height  $GM_0 = 1.1$  m at  $T = 9.0$  m would be enough.

The righting lever curves valid for the limiting metacentric heights are drawn in Fig. 12 to 15. Fig. 12 and 14 show the righting levers in smooth water and Fig. 13 and 15 the levers in waves on a crest and trough calculated for a wave length  $\lambda$  equal to the ship's length and a wave height equal to  $\lambda/15$ . Some stability parameters derived from the righting lever curves in smooth water are given in Table 6.

The values in this table show a clear trend. The righting levers  $GZ$  and the areas  $E$  under the righting lever curves increase with decreasing draught to breadth ratio whereas the angles  $\Phi_0$  and  $\Phi_m$  where the righting levers become zero or maximum seem to be influenced little. The height of the centre of gravity or  $KG/H$  is nearly constant.

Because the tests have been done at different B/T-ratios they can be considered as representative for ships with varying B/T-ratios. Therefore the influence of these ratios on the required stability can be derived from the test results. With this information it is possible to formulate improved stability criteria which assume the same values independent of the hull form characteristics. For this purpose the conventional criteria (e.g.  $GZ_{30}$ ,  $E_{30}$ ,  $E_{40}$ ,  $E_0$ ) are multiplied by a hull form factor  $W$ .

This factor takes care of three influences: the main ratios of the midship section, the hull form described by the block coefficient  $C_B$  and the water line coefficient  $C_W$ , the vertical location of the centre gravity.

Their influence is as follows:

- a large B/T and B/H ratio leads to small values for  $\Phi_m$  and  $\Phi_0$
- the hull form can be characterized by the ratio of the mean water plane area  $V/T$  and the water plane area  $A_w$ . A high value indicates steep sections with smaller variations of the righting levers in waves

$$\frac{V}{T \cdot A_w} = \frac{C_B \cdot L_{pp} \cdot B \cdot T}{T \cdot C_W \cdot L_{pp} \cdot B} = \frac{C_B}{C_W}$$

- the test results show a smaller safety at courses about  $30^\circ$  to the sea than at  $0^\circ$  especially for the small draughts. This effect may be explained with an additional heeling moment due to the lateral acceleration if the centre of gravity is much higher than the water plane. Therefore the ratio  $KG/T$  comes into consideration.

For the hull form factor the following formula is proposed:

$$W = \frac{T \cdot H}{B^2} \cdot \sqrt{\frac{T}{KG}} \cdot \frac{C_B}{C_W} \quad (9)$$

The modified stability criteria are tabulated in Table 7 for both ships. The equability of values for the different B/T-ratios at each ship is satisfactory. This is also true comparing both ships. Thereby the absolute values valid for the same absolute seaway ( $\zeta_{w \max} = 22$  m) have to be compared. Comparing the results in the same seaway relative to the ship's length ( $\zeta_{w \max} = 22$  m and 31 m respectively) the values divided by  $L/100$  have to be used.

Up to now we could not find any criterion which is based on the calculated righting lever curves for wave crest and allows to differentiate between safe and unsafe cases. Also the mean of the righting levers for crest and trough has no advantage compared with the righting levers for smooth water: The mean values lead to the same findings on numerical somewhat lower level. Therefore the simple hydrostatic righting lever curve for smooth water is still the best base to judge safety against capsizing.

## 5. CONCLUDING REMARKS

This paper presents the first results of a comprehensive investigation on ship stability which is not yet finished. The present program includes the investigation of 2 further ships. Therefore the findings are somewhat preliminary. They have to be discussed and perhaps modified later on with respect to the test results for all four ships. Nevertheless we hope to improve the methods for the judgement of the stability of ships in the indicated way using form parameters of the hull.

The absolute values of the required stability parameters in this investigation are partly remarkable higher than the actual values which such ships have in practice. As indicated by the results of ship B in smaller waves the limiting values depend strongly on the wave data. The discussion over the correct modelling of the environment or the question what is safe enough is not finished. This investigation shall give informations how stability criteria depend on hull form characteristics. Practicable and acceptable limiting values of stability criteria can only be derived by considering the corresponding values of safe ships and of ships which have capsized.

## ACKNOWLEDGEMENT

This investigation was supported by the German Ministry of Transport. The authors also owe thanks to the Germanische Lloyd, Hamburg, for the calculation of righting levers and other contributions.

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Table 4. Survey of the test program for ship A,  $\zeta_{w \max} = 22 \text{ m}$

V [m <sup>3</sup> ]	T [m] (T/H)	$\Delta T$ [m]	GM <sub>0</sub> [m]	T <sub>Φ<sub>0</sub></sub> [s]	Φ <sub>w</sub> [°]	Number of runs (N) and capsizings (N <sub>C</sub> )								
						about 0°			about 30°			all courses		
						N	N <sub>C</sub>	N <sub>C</sub> /N	N	N <sub>C</sub>	N <sub>C</sub> /N	N	N <sub>C</sub>	N <sub>C</sub> /N
17190	8.20 (0.77)	0	0.90	21.6	3.3	5	2	0.40	2	1	0.50	7	3	0.43
			1.00	20.5	3.0	11	4	0.36	5	0	0	16	4	0.25
			1.00	16.5	3.0	9	4	0.44	4	2	0.50	13	6	0.46
			1.21	15.4	2.5	5	0	0	9	4	0.44	14	4	0.29
			1.43	14.2	2.1	6	1	0.17	5	1	0.20	11	2	0.18
			1.71	12.7	1.7	14	0	0	6	0	0	20	0	0
			2.14	11.2	1.4+ 12.0	4	0	0	3	0	0	7	0	0
13540	6.70 (0.63)	0	1.46	14.9	2.6	8	0	0	6	2	0.33	14	2	0.14
			1.70	13.8	2.2	4	0	0	9	0	0	13	0	0
10184	5.20 (0.49)	1.50	1.39	16.6	3.6	5	1	0.20	22	11	0.50	27	12	0.44
			1.70	15.0	3.0	6	0	0	16	8	0.50	22	8	0.36
			2.01	13.4	2.5	-	-	-	15	5	0.33	-	-	-
			2.30	12.6	2.2	1	0	0	9	1	0.11	10	1	0.10
			2.40	12.3	2.1	-	-	-	6	0	0	-	-	-

\*) with 12° static heeling angle, simulation of a stifted grain load

Table 5 : Survey of the test program for ship B,  $\zeta_{w \max} = 22 \text{ m}$  and 31 m

V [m <sup>3</sup> ]	T [m]	ΔT	GM <sub>0</sub> [m]	T <sub>Φ<sub>0</sub></sub> [s]	Φ <sub>w</sub> [°]	Number of runs (N) and capsizing (N <sub>C</sub> )								
	(T/H)	[m]				about 0°			about 30°			all courses		
						N	N <sub>C</sub>	N <sub>C</sub> /N	N	N <sub>C</sub>	N <sub>C</sub> /N	N	N <sub>C</sub>	N <sub>C</sub> /N
ζ <sub>w max</sub> = 22 m														
34 685	9.00 (0.48)	0	0.89	29.0	5.7	7	0	0	7	1	0.14	14	1	0.07
			1.10	26.4	4.6	5	0	0	6	0	0	11	0	0
			1.30	24.3	3.9	4	0	0	7	0	0	11	0	0
ζ <sub>w max</sub> = 31 m														
45 147	11.00 (0.58)	0	1.30	22.1	3.0	10	0	0	7	4	0.57	17	4	0.24
			1.53	20.3	2.6	4	0	0	7	0	0	11	0	0
			1.65	19.6	2.4	6	0	0	6	0	0	12	0	0
34 685	9.00 (0.48)	0	1.64	20.0	3.1	4	0	0	5	2	0.40	9	2	0.22
			1.95	18.3	2.6	5	0	0	5	0	0	10	0	0
			2.40	16.8	2.1	10	0	0	8	0	0	18	0	0
25 620	7.10 (0.38)	0	2.09	20.5	3.3	6	0	0	5	3	0.60	11	5	0.45
			2.34	18.8	2.9	6	0	0	4	4	1.00	10	4	0.40
			2.63	17.6	2.6	-	-	-	8	0	0	-	-	-
			2.90	16.5	2.4	3	0	0	9	0	0	12	0	0
			3.03	16.1	2.3	-	-	-	8	0	0	-	-	-

Table 6 : Form- and stability parameters at the limit safe to unsafe

		Ship A			Ship B		
		$\zeta_{w \max} = 22 \text{ m}$			$\zeta_{w \max} = 31 \text{ m}$		$\zeta_{w \max} = 22 \text{ m}$
V	[m <sup>3</sup> ]	10 184	13 540	17 190	25 620	34 685	45 147
T	[m]	5.2	6.7	8.2	7.1	9.0	11.0
B/T		4.42	3.43	2.80	4.54	3.58	2.93
T/H		0.486	0.626	0.766	0.377	0.477	0.584
C <sub>B</sub>		0.631	0.651	0.675	0.554	0.590	0.628
C <sub>w</sub>		0.720	0.759	0.816	0.694	0.762	0.832
GM <sub>0</sub>	[m]	2.35	1.70	1.70	2.95	1.95	1.55
KG/H		0.777	0.774	0.761	0.705	0.707	0.712
KG/T		1.598	1.230	0.993	1.870	1.481	1.220
$\phi_m$	[°]	36	40	38	45	40	39
$\phi_0$	[°]	72.0	70.0	66.5	76.5	73.5	67.0
GZ <sub>m</sub>	[m]	1.45	1.09	0.81	2.00	1.75	1.38
GZ <sub>30</sub>	[m]	1.38	0.96	0.75	1.62	1.39	1.19
E <sub>30</sub>	[mrad]	0.356	0.260	0.220	0.425	0.321	0.267
E <sub>40</sub>	[mrad]	0.606	0.440	0.360	0.736	0.606	0.500
E <sub>m</sub>	[mrad]	0.505	0.450	0.335	0.907	0.606	0.475
E <sub>0</sub>	[mrad]	1.107	0.790	0.580	1.621	1.272	0.908
E <sub>40</sub> -E <sub>30</sub>	[mrad]	0.250	0.180	0.140	0.311	0.285	0.233
E <sub>0</sub> -E <sub>40</sub>	[mrad]	0.501	0.350	0.220	0.885	0.666	0.408

Table 7 Weighted stability parameters at the limit safe to unsafe

		Ship A			Ship B			
		$\zeta_{w \max} = 22 \text{ m}$			$\zeta_{w \max} = 31 \text{ m}$		$\zeta_{w \max} = 22 \text{ m}$	
T	[m]	5.2	6.7	8.2	7.1	9.0	11.0	9.0
B/T	[-]	4.42	3.43	2.80	4.54	3.58	2.93	3.58
W	-	0.0726	0.1051	0.1377	0.0753	0.1041	0.1367	0.1009
GZ <sub>30</sub> ·W	[m]	0.100	0.101	0.103	0.122	0.145	0.163	0.096
$GZ_{30} \cdot \frac{100}{L} \cdot W$	[-]	0.074	0.075	0.077	0.060	0.071	0.080	0.047
E <sub>30</sub> ·W	[mrad]	0.0258	0.0273	0.0303	0.0320	0.0334	0.0365	0.0210
$E_{30} \cdot \frac{100}{L} \cdot W$	[rad]	0.0191	0.0202	0.0224	0.0158	0.0165	0.0180	0.0104
E <sub>40</sub> ·W	[mrad]	0.0440	0.0463	0.0496	0.0555	0.0631	0.0683	0.0409
$E_{40} \cdot \frac{100}{L} \cdot W$	[rad]	0.0326	0.0343	0.0367	0.0274	0.0312	0.0338	0.0202
E <sub>0</sub> ·W	[mrad]	0.0804	0.0830	0.0799	0.1221	0.1324	0.1241	0.0732
$E_0 \cdot \frac{100}{L} \cdot W$	[rad]	0.0595	0.0615	0.0592	0.0603	0.0654	0.0613	0.0362
(E <sub>0</sub> -E <sub>40</sub> )·W	[mrad]	0.0364	0.0368	0.0303	0.0667	0.0693	0.0558	0.0323
$(E_0 - E_{40}) \cdot \frac{100}{L} \cdot W$	[rad]	0.0269	0.0272	0.0224	0.0329	0.0343	0.0275	0.0160

## Discussion

I.A. Manum (Norwegian Maritime Directorate, Norway)

The HSMB has carried out rather similar model-test for another ship type approximately 10 years ago. I am referring to the test carried out on a supply-ship model on behalf of the Norwegian Maritime Directorate. That test formed a part of the base for the additional IMO criteria for supply-ships. It is said that the results presented today is preliminary.

I would propose that the HSMB considers to summarize all their experience gained in this field of testing in a near future in order to recommend a practical model test tool for yards and owners for judgement of safety against capsizing of new design type as a supplement to the inadequate IMO A.167 for that purpose.

S. Tamiya (Tokai University, Japan)

I should like to have author's reply in the following minute questions.

- 1) What is the opposing vertical force against  $\ddot{Z}(X_{cr})$  in term E of equation(7)? Is the force guaranteed to pass through centre of buoyancy?
- 2) Is it confirmed or proved that the wave expressed by equation(8) has a pressure distribution like as the oncoming ocean wave trains?

H. Takegawa (Akishima Laboratory, Mitsui Engineering and Shipbuilding Co., Ltd., Japan)

This paper is of great interest to us who design ships and it gives me much pleasure to obtain the information of capsize problem which has a connection with a safety of ships.

The authors treat the capsize problem by testing in irregular waves. A capsize phenomenon is a transient motion itself even if it happen in regular waves and the motion in irregular waves is also a transient motion.

It is hopeful that these problems are treated at the point of statistical process. Are the tests shown in Table 4 somewhat few in order to obtain statistical samples?

Also, the authors say that the control signal for generating the waves was kept constant for most of all further tests. What does it mean?

Author's Reply

First of all I want to express my thanks to all discussers especially to Mr. Manum for his kind remarks concerning earlier capsizing tests in our model basin and their contribution to the IMO-criteria for supply-vessels. Because of the nonsystematic sample of other earlier tests we have performed in this field, it is rather difficult to get more general informations from these tests. But I can assure we try to go ahead in the indicated way and after further experience we hope to be able to give recommendations for the judgement of stability of a wider range of large ships.

Now I will reply to the more specified questions by Prof. Tamiya and Mr. Takegawa:

The vertical acceleration  $\ddot{Z}$  in equation (7) results from the dynamic vertical forces acting on the hull which are mostly but not only buoyancy forces. Nevertheless I assume that the resultant force passes through the centre of buoyancy. Together with the inertia force of same size acting opposite in the centre of gravity these forces produce a heeling moment at a heeled ship.

The equation(8) only describes the wave profile of the ship wave system, not the oncoming wave. The superposition of both profiles was used for calculation of the righting levers in the usual hydrostatic manner.

I agree it would be desirable to have a larger sample of tests for better statistic values. But we have to look for a compromise between economic and statistical requirements.

As the last point I can repeat that we use the same model seaway for most of our tests with all models which have nearly the same length of about 5 m. This is necessary because we look for the influence of the hull form. Otherwise the effects due to different hull form characteristics would be covered by effects due to the change of wave characteristics.

## THE SEAWAY MODEL AND SEAKEEPING

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### ABSTRACT

This paper presents a complete six degrees of freedom motion calculation of the S7-175 container ship using different models to describe the seaway. These include regular sinusoidal waves, long crested irregular waves and short crested irregular waves. It is found out that the inclusion of a proper spreading in the spectral representation of the seaway has the effect of smoothing the results as compared with the case of long crested waves. This may not be crucial in stipulating dimensioning criteria for example, but operational predictions of ships may be considerably in error. This is particularly true for pitching in beam seas and for rolling in following and in head seas.

The second part of the paper is concerned with the inverse problem, i.e. finding the motion responses from measurements made in natural waves. Since the motions in irregular waves are greatly affected by the directionality of the seaway, the motion responses obviously must be analysed with this in mind. The theory of irregular waves with spreading is discussed and some suggestions concerning the possibilities of analysing the measurements of ship motions in natural waves are made.

Summing up, it would be desirable to perform more measurements of sea waves including the directionality in order to obtain more reliable information as a basis for any seakeeping predictions made either theoretically or with the aid of model tests. The long crested seaway model tends to give a reasonable representation of the most severe conditions to be used in dimensioning ship's structures, but it fails in producing operational predictions for many important modes of motion.

### NOMENCLATURE

$B_{jk}$	restoring coefficients in motion equations
$C_{ij}$	co-spectra between $z_i$ and $z_j$
$F_j$	wave exciting coefficients in motion equations
$H_j$	response amplitude operator of $j$ :th mode of motion
$H_{1/3}$	significant wave height
$M_{jk}$	inertia coefficients in motion equations
$N_{jk}$	damping coefficients in motion equations
$Q_{ij}$	qudrature spectra between $z_i$ and $z_j$
$R_{ij}(\tau)$	covariance functions of $z_i$ and $z_j$
$S(\omega)$	ordinate of point spectrum
$S(\omega, \mu_w)$	ordinate of directional spectrum
$T_w$	average apparent wave period
$a_k(\omega)$	Fourier coefficients of directional spectrum
$a_n$	conformal mapping coefficients
$a_w$	wave amplitude = half the wave height
$b_k(\omega)$	Fourier coefficients of directional spectrum
$c_m(\omega)$	Fourier coefficients of motion spectra
$c_{mn}$	amplitudes of elementary waves in a directional seaway



$d_m(\omega)$	Fourier coefficients of motion spectra
$f(\omega, \mu)$	spreading function, non-dimensional
$g$	acceleration of gravity $9.81 \text{ m/s}^2$
$g_k(\omega)$	Fourier coefficients of response amplitude operators
$h_k(\omega)$	Fourier coefficients of response amplitude operators
$i$	imaginary unit, $i^2 = -1$
$k$	wave number, $k = \omega^2/g$
$k_n$	wave number of $n$ :th elementary wave
$t$	time
$u$	forward speed of ship
$x$	longitudinal coordinate
$x_j$	general coordinate of $j$ :th mode of motion
$x_{j\text{rms}}$	root mean square amplitude of $j$ :th mode of motion
$x_{j1/3}$	significant amplitude of $j$ :th mode of motion
$y$	transverse coordinate
$z$	vertical coordinate
$z_i$	general coordinates of sea surface measurement
$\alpha_k$	Fourier coefficients of spreading function
$\beta_k$	Fourier coefficients of spreading function
$\epsilon_{mn}$	random phase angles of elementary waves
$\theta$	pitch angle
$\mu$	spreading angle
$\mu_m$	spreading angles of elementary waves
$\mu_w$	course angle of the ship with respect to waves
$\tau$	time lag of covariance functions
$\varphi$	roll angle
$\psi$	yaw angle
$\omega$	circular wave frequency
$\omega_e$	frequency of encounter
$\omega_n$	frequency of $n$ :th elementary wave

## 1. INTRODUCTION

The seakeeping characteristics of ships are determined either theoretically or with the aid of model tests. The accuracy obtained is usually adequate for engineering purposes. The word adequate is, however, often misunderstood when it is claimed that a design philosophy is sound when a satisfactory correlation is found between theoretical calculations and model tests. The test may have been designed to match the theory rather than to simulate the real operating conditions of the ship.

The model of the seaway is so complicated that one seldom bothers to be critical on the results obtained by simplified modelling, e.g. regular sinusoidal waves or irregular long crested seas. The experience has taught long time ago that a considerable gap may exist between the predictions based on simple seaway models and the reality.

The problems relating to the mathematical modelling of the seaway comprise

- (1) the form of the point spectrum and
- (2) the form of the spreading function.

The shape of the point spectrum has been extensively discussed in the literature. One usually works with a spectrum having the form proposed by Bretschneider:

$$S(\omega) = a\omega^{-p}e^{-b\omega^{-q}} \quad (1)$$

where  $a$ ,  $b$ ,  $p$  and  $q$  are suitable constants. In (Ref. 1) there has been shown, that if one puts for convenience  $q=p+1$ , the spectrum of Bretschneider will match the significant wave height, the average apparent wave period and the coherency of an irregular seaway, but it fails in a seaway composed of different storms, current or swell. It is also known that the high frequency end of the spectrum may be grossly in error when compared with measurements. This is particularly true for coastal waters or in developing or in decaying seas. In coastal waters the JONSWAP spectrum has been widely used.

No matter how elegantly the point spectrum is formulated, it will be only capable of describing waves approaching from one single direction. This, however, is seldom the case in nature, since the waves are coming from many different directions making it often very difficult even to define the dominant wave direction. There may be found several dominant directions resulting from the change of the wind. If the wind is blowing constantly from one direction, an educated guess for the spreading of the waves is the cosine squared distribution. A reasonable spectral model could then be written:

$$S(\omega, \mu) = \frac{2}{\pi} \cos^2 \mu \frac{AB}{\omega^5} e^{-\frac{B}{\omega^4}} \quad (2)$$

where

$$A = \frac{H_{1/3}^2}{4}$$

$$B = \frac{1}{\pi} \left( \frac{2\pi}{T_w} \right)^4 \quad (3)$$

and the spreading angle  $\mu$  varies between  $-\frac{1}{2}\pi$  and  $\frac{1}{2}\pi$  on both sides of the dominant wave direction.

The next question then arises, what effect the seaway model might have on the predicted seakeeping characteristics of a ship. To illustrate this a six degrees of freedom ship motion calculation was performed for the S7-175 container ship using regular waves, long crested waves and short crested waves of (2) as the seaway model. Other models are not covered in this paper.

## 2. CALCULATIONS

### 2.1 Background

The calculations were performed with the six degrees of freedom ship motion program of Engineering Company M. G. Honkanen Ltd. The theory behind the program is explained in detail in (Ref.1). The hydrodynamic forces affecting the ship in regular waves are calculated with a strip theory that is slightly modified from the one of (Ref. 2) including viscous effects for the roll damping. The sectional hydrodynamic forces are computed in accordance with the conformal mapping technique of (Ref.1). The contour of the actual ship section is represented as

$$y + iz = \sum_{n=1}^N a_n e^{i(3-2n)\theta} \quad (4)$$

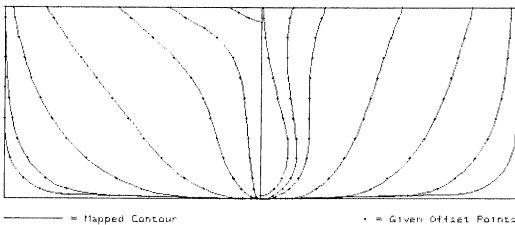
where  $i$  is the imaginary unit,  $a_n$  are the conformal mapping coefficients to be determined from the section contour by an iterative process and  $\theta$  a parameter being  $-\frac{1}{2}\pi$  at the keel and 0 at the waterline. A summary of the conformal mapping process for the S7-175 container ship is given in Fig.1, where it may be seen that an accuracy nearly equivalent to the lines drawing may be obtained with a modest number of mapping coefficients.

#### CONFORMAL MAPPING PROGRAM

Engineering Company M. G. Honkanen Ltd., 00340 Helsinki 34, Finland

Project: S7-175 Container Ship  
Date of calculation: 1982-06-19  
Summary of Conformal Mapping Process:

Nto of frame	Degree of map	Number of iterations	Res-error mm	Beam m	Draught m	Area efficient	Center of area of d
0.00	5	11	7.42	3.095	.699	.5990	-.3737
.50	12	12	5.68	9.000	9.500	.2744	-.2943
1.00	12	26	7.72	14.000	9.516	.4505	-.3517
2.00	13	24	7.64	21.252	9.502	.6591	-.4000
3.00	11	7	5.97	24.596	9.498	.8252	-.4457
4.00	7	11	5.51	25.409	9.503	.9392	-.4770
5.00	10	8	6.04	25.394	9.500	.9706	-.4864
6.00	7	11	7.50	24.510	9.509	.9146	-.4709
7.00	14	21	7.66	20.959	9.491	.8206	-.4508
8.00	16	20	7.34	14.002	9.499	.7584	-.4328
9.00	16	20	7.83	8.255	9.520	.7536	-.4547
9.50	21	34	7.83	3.089	9.510	.9342	-.5068
10.00	9	11	7.55	.177	9.355	8.3619	-.6401



S7-175 Container Ship

Hydrostatics as Integrated from the Mapping Coefficients:

Length	Lpp = 175.000 m	Center of buoyancy	LCB = -2.559 m
Beam	B = 25.400 m	Center of flotation	LCF = -7.270 m
Draught	T = 9.500 m	Height of buoyancy	KB = 5.191 m
Displacement	Vol = 24194.200 m <sup>3</sup>	Transverse metacenter	GMT = .950 m
Waterplane	Rwp = 3167.640 m <sup>2</sup>	Longitudinal metacenter	GML = 204.740 m

Fig. 1: The S7-175 Container Ship

After the hydrodynamic coefficients have been computed, the motion equations are set up as

$$\sum_{k=1}^6 M_{jk} \ddot{x}_k + N_{jk} \dot{x}_k + B_{jk} x_k = F_j a_w e^{-i\omega_e t} \quad j = 1 \dots 6 \quad (5)$$

where  $M_{jk}$  are the inertia coefficients,  $N_{jk}$  the damping coefficients,  $B_{jk}$  the restoring coefficients and  $F_j$  the wave exciting coefficients. The coordinate system is defined in Fig. 2.

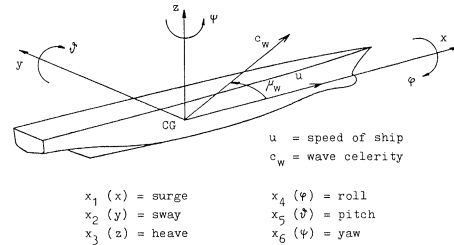


Fig. 2: Ship Motion Coordinate System

The frequency of encounter is calculated as

$$\omega_e = \left| \omega - \frac{\omega^2}{g} u \cos \mu_w \right| = \left| \omega - k u \cos \mu_w \right| \quad (6)$$

The motions in regular waves are expressed as harmonic functions

$$x_j(t) = x_j e^{-i\omega_e t} \quad (7)$$

and the response amplitude operators (RAO) are defined as the amplitude ratios

$$H_j(\omega, \mu_w) = \frac{x_j}{a_w} \quad (8)$$

with  $a_w$  being half the wave height.

The RAOs are found from (5) by grouping the terms and by separating the real and the imaginary parts of  $H_j$  and  $F_j$ :

$$\begin{aligned} P_{jk} &= -\omega_e^2 M_{jk} + B_{jk} \\ Q_{jk} &= -\omega_e N_{jk} \\ H_j &= H_{jc} + iH_{js} \\ F_j &= F_{jc} + iF_{js} \end{aligned} \quad (9)$$

After some matrix algebra there is found:

$$\begin{aligned} H_{jc} &= (RR + I)^{-1} R Q F_{jc} + (RR + I)^{-1} Q^{-1} F_{js} \\ H_{js} &= R H_{jc} - Q^{-1} F_{jc} \\ R &= Q^{-1} P, I = \text{unit matrix} \end{aligned} \quad (10)$$

The spectra in irregular short crested seas are obtained from the RAOs as

$$S_j(\omega, \mu_w) = \int_{-\pi}^{\pi} |H_j(\omega, \mu + \mu_w)|^2 S(\omega, \mu) d\mu \quad (11)$$

and the root mean square amplitudes by integrating in the frequency of encounter domain:

$$x_{j\text{rms}}^2(\mu_w) = \int_0^{\infty} \frac{d\omega_e}{d\omega} |S_j(\omega, \mu_w)|^2 d\omega \quad (12)$$

## 2.2 Extent of the Calculations

The results of the regular wave calculations are presented in Figs. 3 to 8. The magnitudes of all six motion components are plotted as functions of the wave length at a speed corresponding to a Froude number of  $F_n = 0.275$ . The motion amplitudes are non-dimensionalized with the wave amplitude or with the wave slope, and the wave lengths with the length of the ship. The course angle with respect to the waves is the parameter being 0 for following waves and 180 degrees for head seas.

The points included in the heave, pitch and roll RAOs represent model test values obtained from (Ref. 3). It is seen that a reasonable correlation is obtained in most cases.

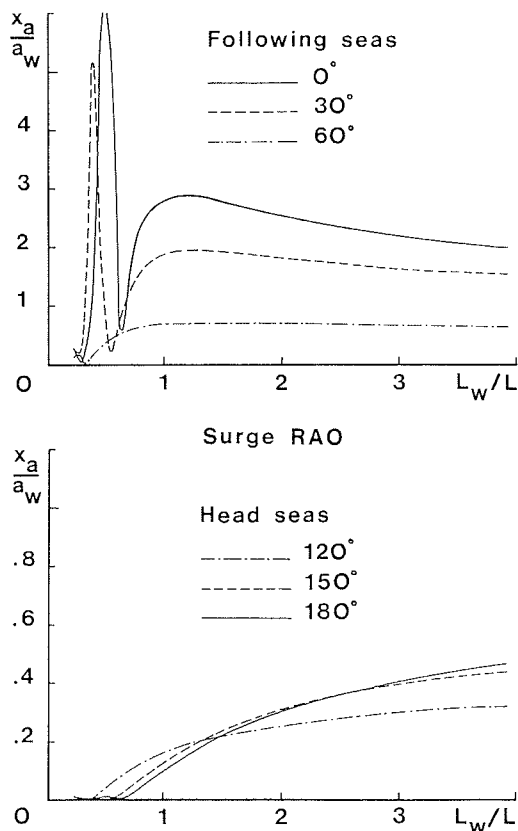


Fig. 3: Surge RAO of S7-175

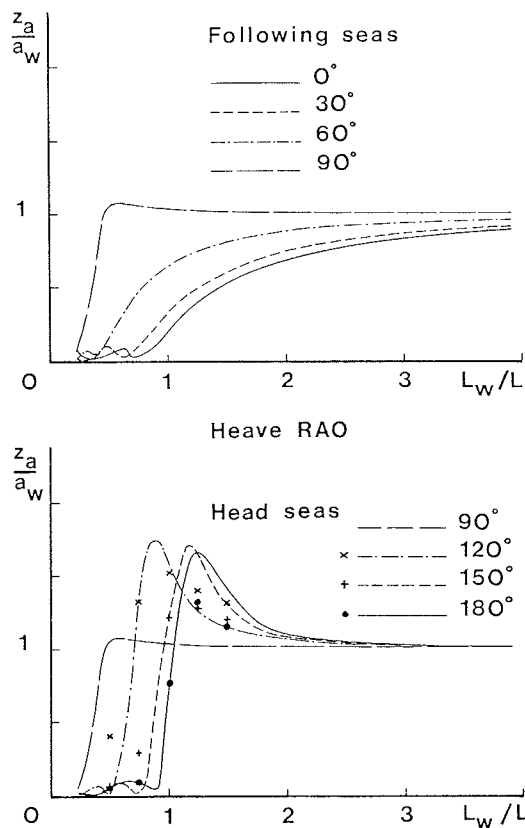


Fig. 4: Heave RAO of S7-175

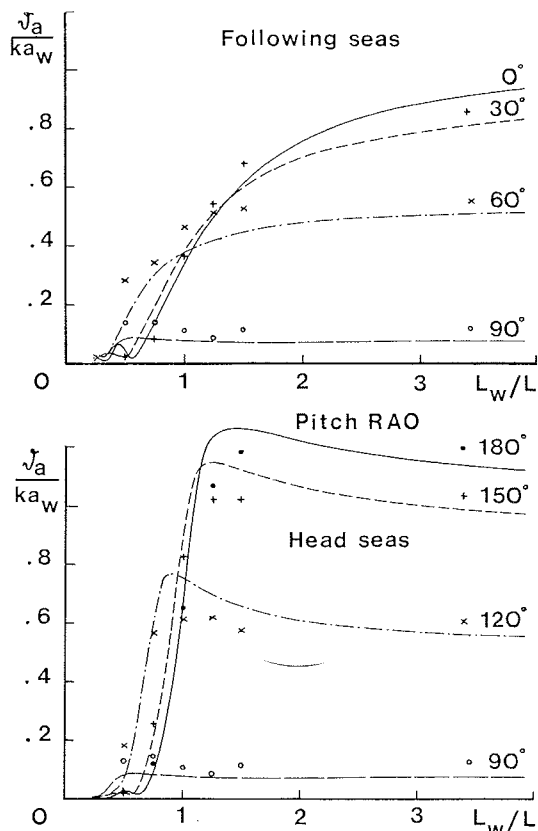


Fig. 5: Pitch RAO of S7-175

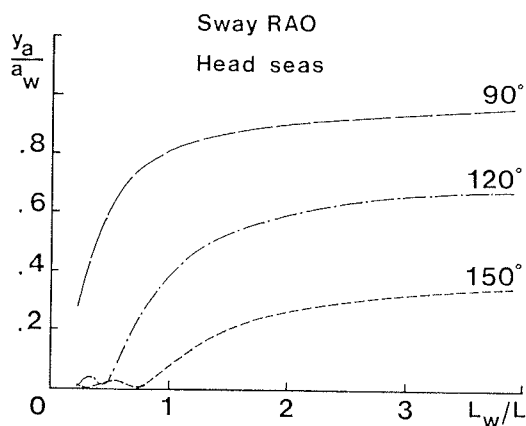
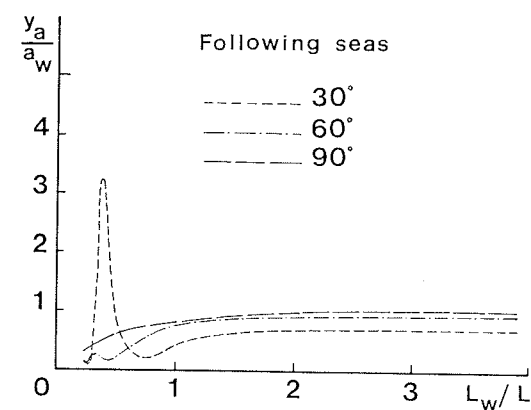


Fig. 6: Sway RAO of S7-175

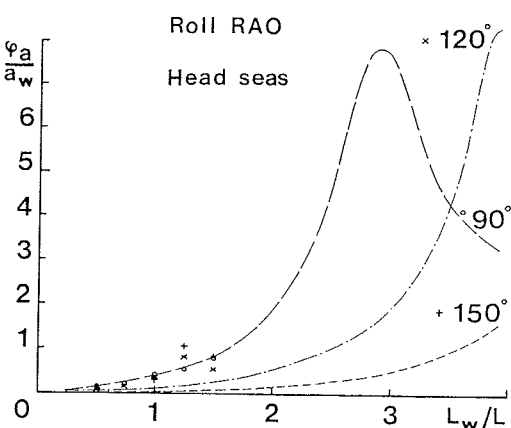
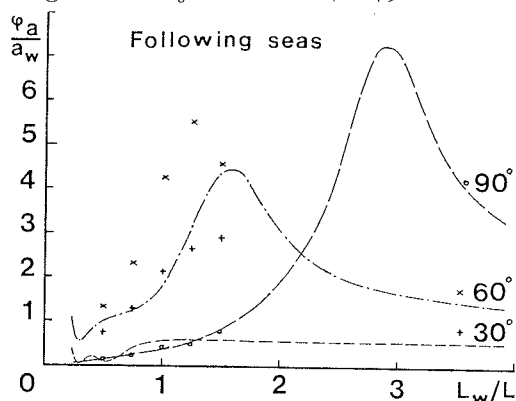


Fig. 7: Roll RAO of S7-175

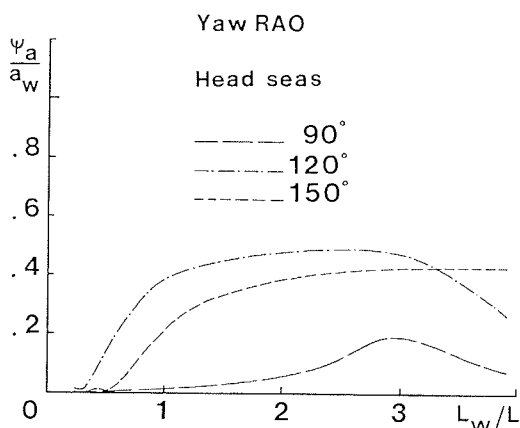
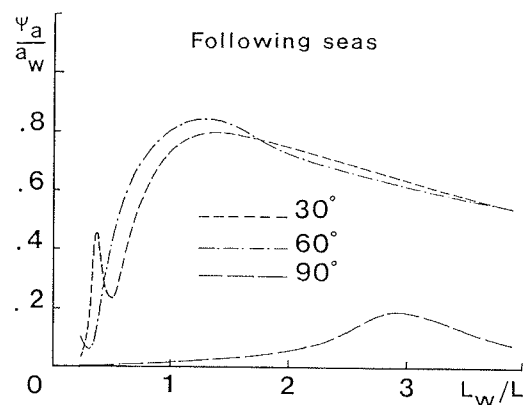


Fig 8: Yaw RAO of S7-175

Four graphs, Figs. 9 to 12, are prepared for the irregular seas. Heave, pitch, bow acceleration and roll are considered. The significant amplitudes are calculated from the root mean square values as

$$x_{j1/3}(\mu_w) = 2.0 x_{j\text{rms}}(\mu_w) \quad (13)$$

The relationship between the significant wave height and the average apparent wave period is presented in Table 1 worked out from (Ref. 7).

Period $T_w$ , s	Wave height $H_{1/3}$ , m
4	1.4
6	1.9
8	2.7
10	4.1
12	5.9
14	8.2
16	11.2

Table 1: Wave Height-Period Relation

The above relation represents the situation on an open ocean. In coastal waters one should use another relation.

Although the number of curves presented is small, the topics chosen are expected to characterize the differences of the seaway models reasonably well.

## 2.3 Results of the Calculations

The heave results in Fig. 9 show that spreading has only little effect on the computed motions. This is understandable because the heave RAOs are not very strongly affected by the course angle of the ship with respect to the waves. The situation would slightly change in coastal waters with a different wave height wave period relationship from the one presented in Table 1. This was demonstrated in (Ref. 4) where the motions of a fast patrol boat were compared in Baltic waves. Heave was still not strongly dependent on the directionality of the seaway, although larger differences than in Fig. 9 were found. It may be therefore concluded that those sea-keeping parameters having a weak dependence on the course angle may also be insensitive on the spreading function and its form.

The pitch motion already shows more differences between the long crested and the short crested seas, see Fig. 10. The responses in beam seas are particularly affected by the seaway model chosen, but the most severe conditions in head seas still remain within an accuracy acceptable for engineering purposes. If one uses the simpler long crested model as a dimensioning criterion, it will lead to a slight overprediction of the pitch motion. This would, of course, be on the safe side when dimensioning ship structures.

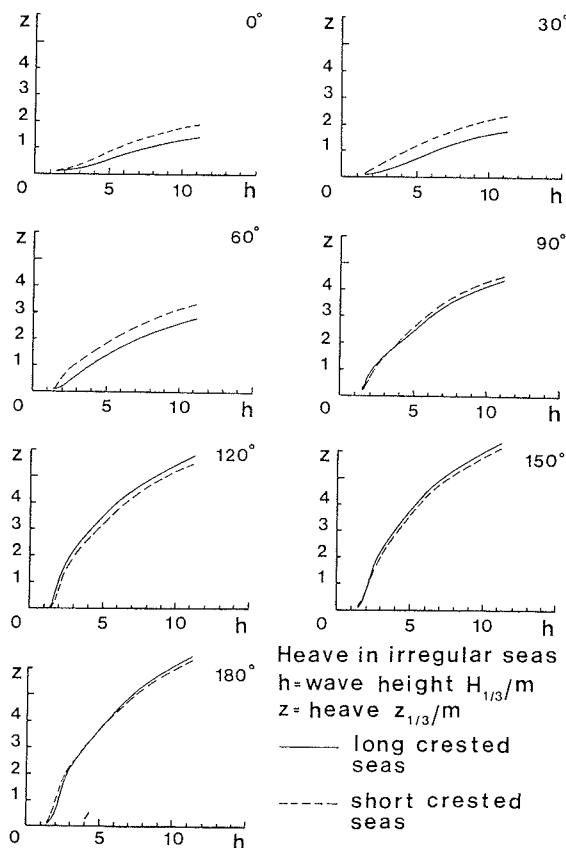


Fig. 9: Heave in Irregular Seas

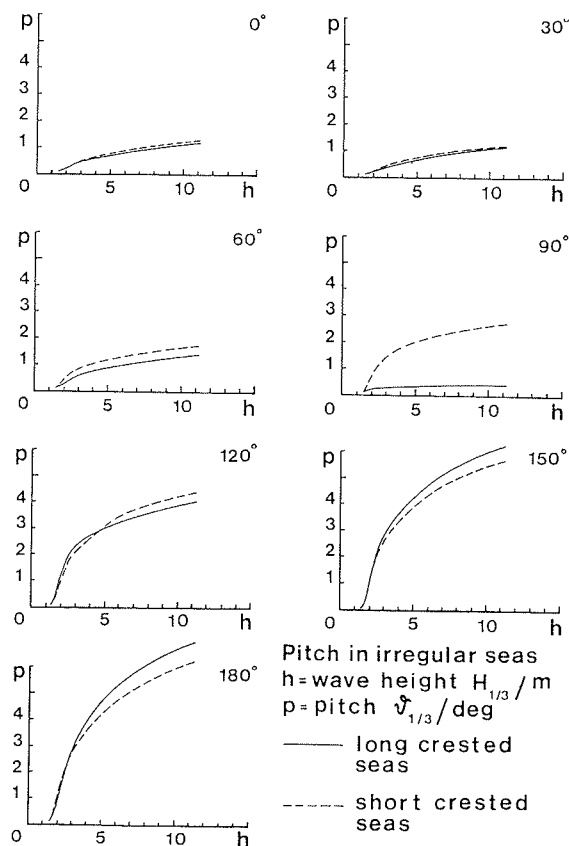


Fig. 10: Pitch in Irregular Seas

To include a derived motion in the analysis as well, the bow acceleration was determined and presented in Fig. 11. Since the acceleration is a combination of heave and pitch, one would expect to obtain curves similar to those for heave and pitch. This seems however not to be the case because the bow acceleration is considerably increased in quartering short crested seas and reduced in head seas as compared with short crested waves. The altered behaviour is explained by the different phase relations between heave and pitch in the two seaways. Thus it may be concluded, that a simple superposition of ship motions into derived seakeeping characteristics of ships may lead to considerable discrepancies depending on the seaway model used. In this case, if the bow acceleration would be a dimensioning criterion, a considerable overestimation would result from the use of a simplified seaway model. The operational performance would be underestimated at other headings than head seas. It may be noted, however, that a choice of a long crested seaway model tends to toss the designer on the safe side rather than to put him into a restless mood. Since the longitudinal motions heave and pitch with their derivatives usually determine the sea loads, this conclusion may be generalized. But it still leaves much open for operational predictions.

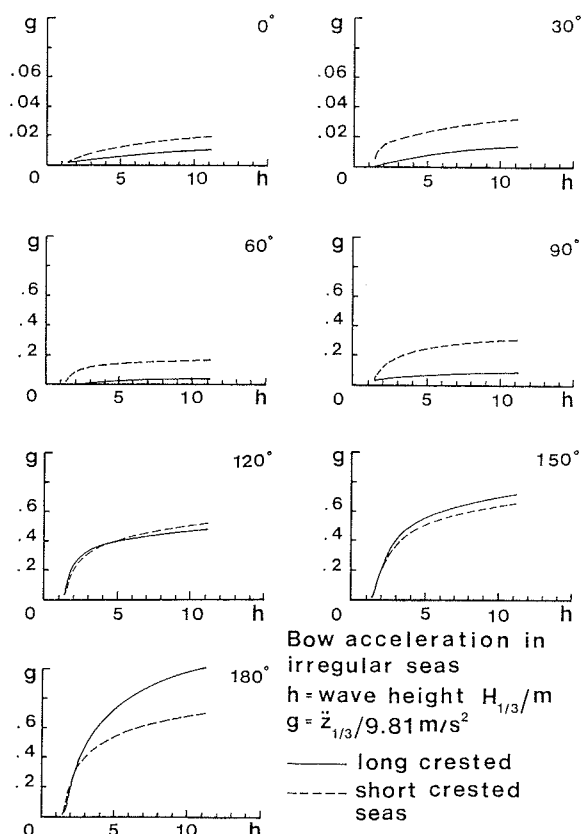


Fig. 11: Bow Acceleration in Irregular Seas

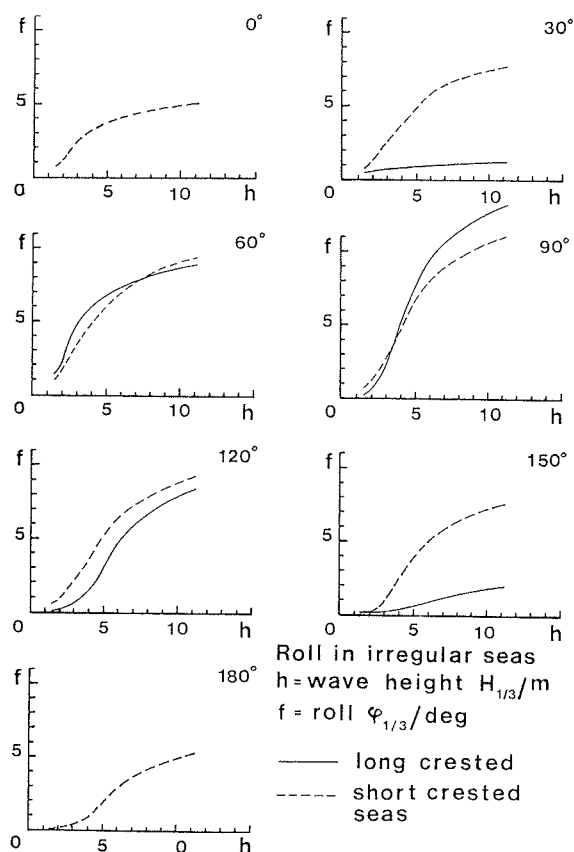


Fig. 12: Roll in Irregular Seas

The roll characteristics as shown in Fig. 12 differ considerably if determined in long crested or in short crested waves. Only in beam seas there is an agreement, but both following seas and head seas induce considerable rolling not predicted by the long crested seaway model. The worst condition is also clearly overpredicted by long crested seas. Because roll is largely determined by the resonant condition, this behaviour becomes understandable due to the higher probability of resonance in long crested waves compared with short crested ones. This is known to have lead to seriously adverse roll predictions especially with ships fitted with passive anti-roll tanks. An anti-roll tank is a device that is normally tuned to cut-off the resonant roll frequencies of the ship. Now, if the performance predictions are based on sharply tuned long crested seas, or even worse, on regular sinusoidal waves, the anti-roll tank may show a brilliant roll reduction whereas the results could be quite modest in short crested seas. The author is aware of designs that have shown a respectable 80% roll reduction in regular beam seas, but an average of only 30% in irregular short crested seas. These findings have been also confirmed by full scale measurements.

### 3. THE DIRECTIONAL SEAWAY

#### 3.1 Mathematical Formulation

As it was seen in the preceeding chapter, the directionality of the seaway has a considerable influence on certain seakeeping parameters of ships. Measurements are often made in natural waves to determine the seakeeping characteristics of ships, and one often wants to find out the RAOs from the measurements in order to be able to predict the behaviour of the ship in another seaway. The results of the previous calculations indicate, however, that the directionality has to be accounted for in order to make a meaningful analysis.

The theory of the directional seaway is outlined here for convenience following the method presented in (Ref. 4) based on the work of Longuet-Higgins, Cartwright and Smith in (Ref. 5). The sea surface is given as a function of time as

$$z(x, y, t) = \sum_m \sum_n c_{mn} \cos(k_n \xi_m - \omega_n t - \epsilon_{mn})$$

$$\xi_m = x \cos \mu_m + y \sin \mu_m, \quad k_n = \omega_n^2 / g \quad (14)$$

where  $\varepsilon_{mn}$  are random phase angles between 0 and  $2\pi$ . The directional spectrum coordinate is related to the amplitude of the elementary wave by the formula

$$\frac{1}{2}c_{mn}^2 = S(\omega_n, \mu_m) \Delta\omega \Delta\mu \quad (15)$$

If we define

$$z_1 = z \quad z_2 = \frac{\partial z}{\partial x} \quad z_3 = \frac{\partial z}{\partial y} \quad (16)$$

and further the covariance functions between any pair of these quantities as

$$R_{ij}(\tau) = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T z_i(t) z_j(t + \tau) d\tau \quad (17)$$

the following relations between the directional spectrum and the covariance functions are found:

$$\begin{aligned} R_{11}(\tau) &= \int_{-\pi}^{\pi} \int_0^{\infty} S(\omega, \mu) \cos(\omega\tau) d\omega d\mu \\ R_{22}(\tau) &= \int_{-\pi}^{\pi} \int_0^{\infty} k^2 \cos^2 \mu S(\omega, \mu) \cos(\omega\tau) d\omega d\mu \\ R_{33}(\tau) &= \int_{-\pi}^{\pi} \int_0^{\infty} k^2 \sin^2 \mu S(\omega, \mu) \cos(\omega\tau) d\omega d\mu \\ R_{12}(\tau) &= \int_{-\pi}^{\pi} \int_0^{\infty} k \cos \mu S(\omega, \mu) \sin(\omega\tau) d\omega d\mu \\ R_{13}(\tau) &= \int_{-\pi}^{\pi} \int_0^{\infty} k \sin \mu S(\omega, \mu) \sin(\omega\tau) d\omega d\mu \\ R_{23}(\tau) &= \int_{-\pi}^{\pi} \int_0^{\infty} \frac{1}{2} k^2 \sin 2\mu S(\omega, \mu) \cos(\omega\tau) d\omega d\mu \end{aligned} \quad (18)$$

By applying the appropriate Fourier transforms the following co- and quadrature spectra of the pairs of quantities  $z_i$  and  $z_j$  are formulated:

$$\begin{aligned} C_{11} &= \frac{2}{\pi} \int_0^{\infty} R_{11}(\tau) \cos(\omega\tau) d\tau = \int_{-\pi}^{\pi} S(\omega, \mu) d\mu \\ C_{22} &= \frac{2}{\pi} \int_0^{\infty} R_{22}(\tau) \cos(\omega\tau) d\tau = \\ &= \int_{-\pi}^{\pi} k^2 \cos^2 \mu S(\omega, \mu) d\mu \end{aligned}$$

$$\begin{aligned} C_{33} &= \frac{2}{\pi} \int_0^{\infty} R_{33}(\tau) \cos(\omega\tau) d\tau = \\ &= \int_{-\pi}^{\pi} k^2 \sin^2 \mu S(\omega, \mu) d\mu \end{aligned}$$

$$\begin{aligned} Q_{12} &= \frac{1}{\pi} \int_{-\infty}^{\infty} R_{12}(\tau) \sin(\omega\tau) d\tau = \\ &= \int_{-\pi}^{\pi} k \cos \mu S(\omega, \mu) d\mu \end{aligned}$$

$$\begin{aligned} Q_{13} &= \frac{1}{\pi} \int_{-\infty}^{\infty} R_{13}(\tau) \sin(\omega\tau) d\tau = \\ &= \int_{-\pi}^{\pi} k \sin \mu S(\omega, \mu) d\mu \end{aligned}$$

$$\begin{aligned} Q_{23} &= \frac{1}{\pi} \int_{-\infty}^{\infty} R_{23}(\tau) \cos(\omega\tau) d\tau = \\ &= \int_{-\pi}^{\pi} \frac{1}{2} k^2 \sin 2\mu S(\omega, \mu) d\mu \end{aligned} \quad (19)$$

The covariance functions can be calculated from the measured quantities by standard numerical methods.

We have now six integral equations defining the directional spectrum. In order to obtain a discrete solution, we assume that the directionality may be represented by a Fourier series, hence

$$\begin{aligned} S(\omega, \mu) &= \frac{1}{2} a_0(\omega) + \sum_k a_k(\omega) \cos k\mu + \\ &+ b_k(\omega) \sin k\mu \end{aligned} \quad (20)$$

A comparison with (19) reveals instantly that the Fourier coefficients must be

$$\begin{aligned} a_0(\omega) &= \frac{1}{\pi} C_{11} \\ a_1(\omega) &= \frac{1}{\pi k} Q_{12} & b_1(\omega) &= \frac{1}{\pi k} Q_{13} \\ a_2(\omega) &= \frac{1}{\pi k^2} (C_{22} - C_{33}) & b_2(\omega) &= \frac{2}{\pi k^2} C_{23} \end{aligned} \quad (21)$$

Finally, the spreading function may be separated from the spectrum by writing

$$S(\omega, \mu) = f(\omega, \mu) S(\omega) = f(\omega, \mu) C_{11} \quad (22)$$

where

$$f(\omega, \mu) = \frac{1}{2\pi} + \alpha_1 \cos \mu + \beta_1 \sin \mu + \alpha_2 \cos 2\mu + \beta_2 \sin 2\mu \quad (23)$$

with the Fourier coefficients redefined as

$$\begin{aligned} \alpha_1 &= \frac{1}{\pi k} \frac{Q_{12}}{C_{11}} & \beta_1 &= \frac{1}{\pi k} \frac{Q_{13}}{C_{11}} \\ \alpha_2 &= \frac{1}{\pi k^2} \frac{C_{22} - C_{33}}{C_{11}} & \beta_2 &= \frac{2}{\pi k^2} \frac{C_{23}}{C_{11}} \end{aligned} \quad (24)$$

In general, the spreading function depends on both the frequency and the spreading though the frequency is often omitted for the sake of simplicity. It is interesting to note that only the second harmonics can be obtained from the measurements of wave elevation and its derivatives in two directions. If a greater accuracy is desired, more information like e.g. second derivatives should be measured, but such measurements would be very uncertain if not totally impractical.

### 3.2 Tests in Natural Waves

Provided one is able to make the above measurements one is also able to compute the directional spectrum. The ship motions and their spectra may also be measured either in model scale or in full size. The next question is, how to obtain the RAOs from these measurements. Methods of analyzing such measurements are discussed in detail e.g. in (Ref. 6) where sectoral pseudo-RAOs are introduced. The half circle is divided into four sectors where the problem is treated more or less as four different long crested seas. But let us follow the way outlined above a little further. Suppose that there has been measured a set of motion spectra at different headings together with the directional spectrum of the seaway. Let us represent the motion spectrum as a Fourier series with respect to the course angle:

$$S_j(\omega, \mu_w) = \frac{1}{2} c_0(\omega) + \sum_{m=1}^2 c_m(\omega) \cos m \mu_w + d_m(\omega) \sin m \mu_w \quad (25)$$

Furthermore, let us put the unknown RAO in a similar form:

$$|H_j(\omega, \mu_w)|^2 = \frac{1}{2} h_0(\omega) + \sum_{k=1}^2 h_k(\omega) \cos k \mu_w + g_k(\omega) \sin k \mu_w \quad (26)$$

A substitution of (20) and (25) into (11) will enable after some manipulation the analytical computation of the motion spectrum in the following form:

$$\begin{aligned} S_j(\omega, \mu_w) &= \frac{1}{2} h_0 a_0 + \sum_{k=1}^2 \int_{-\pi}^{\pi} h_k a_k \cos k(\mu + \mu_w) \cdot \\ &\cdot \cos k \mu + h_k b_k \cos k(\mu_w + \mu) \sin k \mu + g_k a_k \cdot \\ &\cdot \sin k(\mu_w + \mu) \cos k \mu + g_k b_k \sin k(\mu_w + \mu) \cdot \\ &\cdot \sin k \mu \end{aligned} \quad (27)$$

The integrations in (27) result in terms proportional to  $\cos k \mu_w$  and  $\sin k \mu_w$ , after which the expression can be put equal with (25) term by term. One thus obtains a set of linear simultaneous equations for the unknown Fourier coefficients at each frequency. By solving the coefficients one finally obtains the RAO in form of (26). An interesting finding is, that the RAOs can only be obtained up to the same number of harmonics than the directional spectrum.

### 4. CONCLUSIONS

The main conclusion is that most ship motion parameters are affected by the choice of the seaway model. The long crested model both under- and overestimates the motions, but the most severe conditions are as a rule overestimated. A weak dependence of a RAO on the course angle also results in a weak dependence on the directionality of the seaway.

Another conclusion is, that if one wants to obtain RAOs from measurements made in natural waves, the spreading of the seaway should also be measured and the analysis of the results should be made with the directionality in mind.

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## Discussion

R. Barr (Hydronautics, Inc., USA)

The results presented for rolling motions in long- and short-crested irregular waves, figure 12 of the paper, indicate only small differences in motions between these waves for the wave heading angles of maximum rolling angle ( $60^\circ - 120^\circ$ ). This does not agree with the author's conclusion No.1 that the short-crestedness of the waves can have an important effect on maximum rolling motions. Does the author have calculated results for other ships which show a significant influence of short-crestedness on maximum rolling motions, that is, rolling at the wave heading angle producing maximum rolling motion.

### Author's Reply

The S7-175 container ship does not represent the best possible example with maximum roll at about  $90^\circ$  heading. I have other ships in my files where the maximum occurs very often at about  $60^\circ$  heading, and in these cases the effect of spreading is much more pronounced.

M. St. Denis (USA)

When waves are measured by a fixed wave staff, or similar instrument, which is insensitive to wave direction, the measurements are reducible to a "point" variance spectrum of the seaway. Because this is the simplest of all spectra, it was the first to be introduced. The concept was that of Pierson (1952), the formulation, that of Neumann (1952). To go from a "point" to a "directional" spectrum is not altogether a simple extension of the problem, for a complication is introduced: it now becomes necessary to eliminate ambiguous possibilities introduced by the added dimension.

The reduction of wave measurements into a wave spectrum is simple only in theory and direct only in principle. In practice, the derived spectrum is a probabilistic one colored by statistical uncertainties. These are assessable as a function of the statistical degrees of freedom of the measurement and reduction processes. Because of the limited number of statistical degrees of freedom realizable, the confidence bands about the derived spectral ordinates are widely spaced, perhaps so widely spaced as to temper with doubt any statement made about the spectrum. (Incidentally, the formula for statistical degrees of freedom usually employed, namely,  $f \approx [n-m/4]/[m/2]$ , where  $n$ =number of wave ordinates,  $m$ =number of lags, is valid only for flat spectra, which may be referred to as "white noise" or, more properly, as "white wave" spectra. The Bretschneider, Pierson-Moskowitz, indeed, all the wave spectra so far pro-

posed, are far from being flat. The applicable formula, for which see Rice (1949), yields a considerably lower number of statistical degrees of freedom.

The first suggestion for a spreading function appears to be due to Arthur (1949): it is a cosine squared variation from the dominant wave direction. For the want of a sufficiency of objective and reliable measurements, this formulation was avidly accepted and has "stuck" as one says. The Stereo Wave Observation Project (See Chase et al. 197?) produced a cosine-squared plus cosine fourth formulation for the spreading function, but this was for one sea state only and no generalization to all sea states is warranted. Longuet-Higgins, Cartwright and Smith (1961) offered an analysis of observations which indicates a spreading function which is sensitive to wave period; the results, however, exhibit considerable statistical variation so that the trend is not altogether firm. Scott (1965) has suggested a spreading function varying as the  $n$ -th power of the cosine, where  $n$  is approximately  $7/8 \cdot T$  ( $T$ =wave period). Last year, a conference was held in Paris on wave directionality; however, the proceedings appear not to have arrived as yet in Honolulu, where I make my home, or in Finland, where Dr. Honkanen resides. Of course, these proceedings warrant study.

The amount of use that has been made of a spreading function in the prediction of ship motions and in the assessment of sea-kindliness and of motion safety has not been impressive. It is to be wondered why this has been the case, for the added labor to take into account wave spreading is not conceptual but only computational. The error introduced by the assumption that there is no wave spreading is by no means trivial especially when the waves are storm driven.

### Author's Reply

I completely agree with Dr. St. Denis that the measurement of the directional spectrum is a very difficult one. I have simulated the directional seaway numerically starting from a simple spectrum of known properties in order to obtain a pure sea surface. The analysis of this surface to get the original spectrum clearly demonstrated the difficulties in obtaining a reasonable confidence even for an error-free signal not to mention measurements, where either systematic or random errors are present. I am developing in cooperation with Chalmers Institute of Technology, Gothenburg, Sweden, a wave measurement system based on the three staff principle, from which project I can only arrive, that the opinions expressed by Dr. St. Denis are rather optimistic than pessimistic.

Regarding the shape of the directionality, I also agree with Dr. St. Denis that much is open for discussion. This naturally follows from the fact, that the number of measurements is scarce and their reliability may be questionable. However, all data available at the writing of the paper indicate, that if one forces the spreading in the  $\cos^n$ -form, the power  $n$  will be frequency dependent and greater than 2. Hence, I have deliberately used the power 2 in order to make the paper provocative to the profession in producing upper limits for the seakeeping computations. And indeed, as is expressed in my paper, some seakeeping

characteristics are greatly affected by the spreading. Especially roll in following seas is much underpredicted by long crested waves, a fact that should greatly concern those being responsible for the stability of ships under the action of sea waves.

I am delighted that Dr. St. Denis shares my view that the inclusion of the wave spreading in the seakeeping computations deserves more attention than has been paid to it in the past. If we are to perform direct calculations to judge the seaworthiness of ships, a better understanding of the environmental conditions is required.

*Session IVb*

## Safety of Fishing Vessels (Part II)

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## MODEL EXPERIMENTS ON CAPSIZE OF FISHING BOATS IN WAVES

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### ABSTRACT

The model experiments of fishing vessels have been carried out in the seakeeping model basin, in order to investigate the capsizing mechanisms in rough seas. In this study, seven typical fishing vessels in Japan were used, i.e., two purse seiners, two bonito fishing boats, a pair stern trawler, a coastal small trawler and a small salmon drift net fishing boat.

In the experiments free-running radio-controlled models were used and the models' speed and the courses to relative to waves were chosen systematically in irregular or regular waves. The motions and relative wave elevations on the hull sides of the models were measured together with the records taken on the movies.

In the experiments a lot of capsizes occurred and the typical modes of capsize for fishing vessels have been found, i.e., the shipping water on deck and the dynamic effects of waves caused to capsize the models, or the model lost its stability on wave crest amidship in following seas, and so on. The modes of capsize also depend on the ships' type. For the analysis of ship motions in waves, the relative wave elevations closely concerning with the shipping water on deck have been especially investigated by both measured and calculated responses in oblique waves.

The experimental results of capsize have been compared with the GM-Freeboard diagram representing the  $C_1$ ,  $C_2$  and  $C_3$  value proposed by Tsuchiya as the stability criteria for fishing vessels, and it is shown that such values will be useful to judge the safety of fishing vessels in taking account of many factors, such as the effect of shipping water on deck or the limit of shipping water over the bulwarks

as well as dynamical stability in wind and waves.

A model experiment in the seakeeping basin seems to be effective to clarify the behaviours of a ship in severe sea state or the basic character on complicated responses occurred under the combined effects of many factors.

### 1. INTRODUCTION

In Japan, three research groups, namely the National Research Institute of Fisheries Engineering, Ship Research Institute and the Faculty of Fisheries, Hokkaido University, have been co-operatively working on the model experiments for various kinds of fishing vessels in oblique waves. For these six years, seven models have been tested in the experimental model basin.

In the first stage of investigation the essential purpose was to observe the capsizing phenomena so that the typical modes of capsize for fishing vessels could be found. The experiment results have been reported by Tsuchiya et al. [1].

According to those results, the shipping water on deck has played the most important role for capsizing of fishing vessels. The subsequent experiments have been continued for the other models concentrating the efforts to study the relative wave elevations on the hull sides of the models, because the characteristics of the shipping water on deck should be clarified. The measured responses of the models in waves have been also compared with the estimated ones based upon the strip theory and superposition theory. In addition, whether the similar mode of capsize or the other mode of capsize will be found for the other types of fishing vessels has been further problems to be

investigated.

## 2. TEST PROCEDURE (MODELS & METHODS)

### 2.1 Models

Seven different types of fishing vessels, i.e. two purse seiners (named model A and B), two bonito fishing boats (model C and D), a pair stern trawler (model E), a coastal small trawler (model F) and a small salmon drift net fishing boat (model G) were tested.

The principal dimensions of those models are given in Table 1 and the main particulars of the models tested are also given in Table 2. Fig. 1 shows body plans with the righting arm curves, GZ curves, of the each model tested.

The superstructures of the models above water were made as similar as possible to the full scale hull forms with forecastle, deckhouses and bulwark with freeing ports. The water tightness of the models were kept perfect even when the model capsized so that the measurement devices equipped on the model were protected and undesirable effects of water flooding in-board were prevented.

The models can be classified into three groups according to the differences in their hull forms especially above water. In the first group, model A and model E have wall-sided hull forms. In the second, model F and model G have the wide over-hanged deck above water as almost of small fishing boats in Japan are fitted. In the third, model B, model C and model D have a conventional hull forms.

Besides this the characteristics of the models are as follows.

- (1) Model A and model D have the improved hull forms with U-shape.
- (2) Model E is equipped with a slip-way at the stern for fishing operations.
- (3) Model G has the hard-chine hull form.

### 2.2 Test Method in the Basin

The test procedures adopted for the model experiments are as follows.

- (1) The model experiments were performed by using free-running radio-controlled models. All of the models are two meters in length with one exception which is 2.25 meters.
- (2) The tests were carried out for various load conditions or the transverse meta-centric heights, GMs.
- (3) The freeing ports were closed or opened alternatively.
- (4) The courses of the model relative to waves was chosen systematically from beam seas to following seas.
- (5) The motions and relative wave elevations of the model were measured by the gyroscopes and the capacitance type wave probes fitted on the hull side respectively.
- (6) The model speeds were also radio-controlled by changing the voltage of DC batteries supplying power to the drive motor. For most models its speeds were chosen two steps, normal (about 1.20 m/sec)

and half (about 0.8 m/sec).

(7) The courses and speeds of the model were measured by the ultra-sonic, automatic position detector equipped in the basin.

The irregular waves generated in the basin have the significant wave height of about 0.3 meters and the mean wave period of about 1.7 seconds approximately the same characters for all models. The spectrum of the waves is defined by the so-called Pierson-Moskowitz type.

The spectra measured in the basin are shown in Fig. 2.

Fig. 3 shows a photograph on the model in the experiments.

## 3. TEST RESULTS

Concerning to the investigation on mechanisms of capsizing and shipping water, the tests for each model were performed under the conditions involving with lower freeboard and poorer stability than normal.

This chapter describes the observed results on each models grouped by the differences in fishing operations or ship forms. Fig. 4 (a)-(g) show the capsized cases.

### 3.1 Purse Seiners

Though the experimental results of model A and model B have been reported already, a short description of the results here will be helpful to compare the capsizing phenomena with the other types of fishing vessels.

The model B has a conventional hull form and model A has its modified one. The tests were performed for different two load conditions, one of them was the over-loaded having the minimum allowable freeboard in the regulation. The height of center of gravity, KG, was also changed, involving the lower GM values than normal, in order to find the critical stability condition between capsizing or non-capsizing in the given sea state.

In the experiments the followings were clarified on the capsizing of those fishing vessels.

- (1) When the models ran with a high speed in heavy following or quartering seas, they were usually put into the dangerous situations and sometimes were capsized. Capsizing in head, bow and beam seas occurred in the special cases such as the poor stability and lower freeboard.
- (2) Three modes of capsize were observed, i.e.
  - a) capsizing caused by shipping water on deck,
  - b) capsizing caused by combined effects of trapped water on deck-well and dynamic forces due to waves, and
  - c) capsizing caused by a pure loss of stability on the wave crest, accompanying with the water flooding on the deck.
- (3) Capsizing caused by the rolling resonance in beam seas could not be found, and so on.

After the experiments further study were conducted for model A in regular waves. The main purpose of the experiments was to

investigate the characteristics of shipping water over the top of bulwarks. The model condition II-3 (see Tab. 2) with the freeing ports fully opened was adopted for the experiments. The experiments were held under such wave conditions that the wave height was maintained as a constant ( $H_w=25.0$  cm or  $15.0$  cm alternatively) and the wave length was varied ( $\lambda/L=0.75$  to  $2.0$ ).

The observed results showing the degree of shipping water were given in Fig. 5 (a). Each marks in the figure represent as follows:

- : a large amount of shipping water on deck over the bulwarks,
- : no shipping water on deck, and intermediate representations are indicated in proportion to the amount of shipping water over the bulwarks.

Fig. 5 (a) shows that the frequency of the shipping water on deck increased rather in steeper waves. The shipping water on deck was however observed in longer waves ( $\lambda/L = 1.5$  to  $2.0$ ) under the head and bow sea condition whereas no shipping was observed under the other heading conditions.

### 3.2 Bonito Fishing Boats

This type of fishing boats is equipped with a fishing platform all around outside the bulwark. Model C has the conventional hull form whereas model D is improved one.

The tests on model C were carried out almost in regular waves for such model conditions as shown in Tab. 2. In particular, the condition II-2 ( $GM=4.0$  cm) with freeing ports closed was adopted for the main tests. The purpose of the tests for this model was to investigate on the shipping water too.

The wave parameters were varied within the following ranges:

- wave length :  $\lambda/L = 0.75$  to  $2.0$
- wave height :  $H_w = 12.5$  cm to  $25.0$  cm

Fig. 5 (b) shows the degree of the shipping water on deck in such wave conditions that the wave height was maintained about  $12.5$  cm and the wave length was varied.

The results on model C in regular waves are summarized as follows.

- (1) The shipping water on deck depends on both model speed and wave steepness.
- (2) The more the wave is short and steep, the more the model shipped water on deck.
- (3) Capsizing only occurred in poor stability ( $GM = 2.0$  cm) under the following waves of which length are equal to the model length or  $1.5$  times of it.
- (4) No capsizing occurred in the other stability condition, nevertheless shipping water filled the deck-well and induced a large heeling of the model.

In the irregular waves the model did not capsize in all heading angles, nevertheless the model shipped much water on deck.

Model D was tested in regular or irregular waves too. The experiments in regular waves were done in such conditions that the wave-model length ratio is equal to

$1.25$  and the wave heights were varied from  $14.0$  cm to  $30.0$  cm.

The results of capsizing in regular and irregular waves are given in Fig. 4 (d) and Fig. 4 (c), respectively.

The main results on model D are as follows.

- (1) Capsizing occurred most frequently with rather lower  $GM$  ( $GM = 1.48$  cm) in both regular and irregular waves.
- (2) All of capsizes occurred in the quartering or following waves.
- (3) Total freeing port area for clearing water from the deck is insufficient to prevent the capsizing in such lower freeboard condition.

### 3.3 Pair Stern Trawler

The tests for model E were carried out under two different conditions of loading, one of them was over-loaded condition having nearly zero freeboard and the other was full-loaded condition. The center of gravity,  $KG$ , was also changed (see Tab. 2). Model E has so high protected freeboard (height between water surface and bulwark tops) that the tests were expected to produce the capsizing due to rolling resonance. But no capsizing occurred in 138 runs in any encounter angles relative to the waves, nevertheless the large amplitude of rolling was observed.

On the condition II-1 the model has so good stability due to the effects of deck-houses as the angle of vanishing stability of the model is beyond  $90$  degrees.

On the condition I-1 the angle of vanishing stability is about  $47$  degrees and the angle of the bulwark edge becoming immersed is about  $26$  degrees. The value of  $GZ$  at the angle of the bulwark edge becoming immersed is very low on this condition, so the model should be put into dangerous situation if it has a large amount of water trapped on its deck-well.

The main results on model E are shown as follows.

- (1) No capsizing occurred under all experimental conditions, since the stability of the model was improved by the effects of its deck houses.
- (2) The shipping water on deck over the bulwarks was not observed except in several cases in which the model was attacked by the severe breaking waves.
- (3) The scooping water on fore-castle deck over the top of the stem was sometimes observed in head seas, because the flare of the the model at fore parts is very small.

Besides this, a few tests were tried to observe the shipping water on deck over the slip-way on the astern condition with lower speed in following seas. In these cases the model was seriously flooded over the slip-way in spite of having the relatively high freeboard at the position (about  $16.4$  cm in condition I).

### 3.4 Small Fishing Boats

Model F and model G are typical small fishing boats in Japan. The wide over-hanged deck is one of the characteristics of this type. Model G has a hard-chine in its bilge parts throughout from fore-peak to aft end, while model F has round bilge with small circle. In the tests on those models it was expected to occur the other mode of capsize based upon these features. The attention was also paid on the effects of the over-hanged deck against shipping water in waves.

The tests on those models were held in irregular waves under following, quartering or beam sea conditions. In the experiments the initial heeling angles were given for several cases in order to investigate the effect of the corresponding moment due to wind or cargo shift.

Fig. 4 (f) shows the rate of capsizes for model F. The tests results on the model are summarized as follows.

(1) Without initial heeling capsizing occurred 6 times in 63 runs under the condition I-1 whereas capsizing occurred 3 times in 71 runs under the condition II-1.

(2) In the condition II-1 one of them occurred in head seas, whereas the other two cases in quartering seas.

(3) Capsizing in head seas occurred in such a situation that a large amount of water was shipped over the bulwark top when the model was struck by a breaking wave, the significant yawing happened immediately and it was hard to keep on course, and the model ended up capsize.

(4) the other two capsizes were happened in the state that the model shipped a large amount of water on the deck and it was much heeled, and the model capsized by the subsequent one or two steep waves. In these cases the GZ reduction due to the wave crest amidship may be occurred, because the trapped water on the deck did not capsize the model in calm water or in moderate waves.

(5) No capsizing occurred in 9 times runs in condition II-1 when the freeing ports of the model were opened.

(6) Capsizing occurred in high frequency when the model had an initial heeling. In these cases the model capsized to the initially heeled side only. When the model initially heeled to weather side, it capsized in all cases.

(7) With the freeing ports closed the model easily capsized if it once shipped a large amount of water on the deck, because the shipping water on deck gave a significant capsizing moment to the model.

(8) The shipping water happened mostly at the midship or the stern of the model in quartering seas, while the water was shipped at midship in beam seas.

(9) Capsizing increased when the freeing ports were opened in lower freeboard, because the water easily flew onto the deck through the freeing ports.

The tests on the model G were carried

out almost in the constant speed. The tests results on the model are summarized as follows.

(1) No capsizing occurred in 47 times runs under the following, quartering and beam sea conditions.

(2) Capsizing occurred one time in 8 runs, when the model was initially heeled. In this case the model was initially heeled to the weather side and resulted to capsize to the same side.

(3) Shipping water over the bulwark tops was happened only when the model encountered a breaking wave.

(4) The bulwark parts of the model had been constructed solid to guarantee the watertightness and strength of the hull. Therefore, the GZ values of the model at the angles beyond which the bulwark edge becomes immersed are higher than the other ones. The dotted line in Fig. 2 (g) represents the GZ curve corresponding to the real ship. Supposing the model having such stability represented as the dotted line, the model may capsize when it trapped a large amount of water on its deck-well.

Through the experiments on the both models the followings are clarified: i.e. the over-hanged deck is effective to increase the stability of the model and to prevent shipping water on deck except the shipping of breaking waves. However, the stability of the model, with the freeing ports closed, significantly decreases if the model once ships water on the deck.

### 4. ANALYSIS

#### 4.1 Ship Motions and Relative Wave Elevations

Today, it is commonly recognized that the strip theory offers a very useful tool for estimating ship responses in regular oblique waves. The usefulness of this theory as well as limitation of its applicability has been proved through comprehensive studies including comparison between calculations and experiments for various types of vessels. This is also true for fishing boats as already shown by one of the present authors [2]. Therefore, the ship motions as rolling and pitching as well as relative wave elevations on the hull sides were estimated by means of strip theory (OSM) for all seven models tested. In the calculations the roll damping coefficients measured on the same models as tested one were applied according to the conventional way. The relative wave elevations were estimated for several points along model sides as A.P., midship or fore part of the model where the wave probes were attached during the experiments.

The responses in irregular waves were also estimated by using those in regular waves and the spectrum of the irregular waves used for the experiments, basing on the linear superposition theory. Fig. 6 (a)-(p) show the results of calculations together with the measured values of the responses in

irregular waves for model D, model F and model G. In the figure the standard deviations are derived from the total area of the spectrum.

From these figures, the followings concerning to ship motions can be seen.

(1) The calculated values represent the same tendency as the measured ones which show some scattering.

(2) Rolling motion becomes largest in quartering seas.

(3) Responses of relative wave elevations on hull sides show rather complicated variation in accordance with the ship type and the course against waves.

By means of calculated values of relative wave elevations, the possibility of occurrence of shipping water on deck can be estimated, i.e. the shipping water may be possible to occur when the amplitude of relative wave elevation exceeds the protected freeboard.

In Fig. 7, the critical lines indicating the possibility of shipping water in regular waves are shown in comparison with the observed results on the model A. Those critical lines were derived from the calculated results on relative wave elevations at the weather sides of St. 5 (mid-ship), St. 2-1/2 and St. 7-1/2. They show that those critical lines also indicate nearly the shipping water range on the model, except the case of  $\lambda/L = 1.5$  where the shipping water in head waves can be explained by considering relative wave elevations at more fore-part of the model.

#### 4.2 Capsizing

The experiments on all models were conducted in the irregular waves having almost the same characteristics. A lot of capsizing occurred in such sea states. Table 3 shows the ratio of number of capsizes to number of tests for each conditions, without initial heeling. Total number of capsizes for seven models is 66 times in 528 runs. Capsizing in quartering seas occurred 41 times, while in following seas 18 times. However, in beam seas no capsizing occurred in 59 runs. According to these results, it is deduced that fishing vessels running in quartering or following sea condition in severe seas would be put into much dangerous situations.

If the model once ships water on the deck, capsizing depend on the ballance between the restoring moment determined by the shape of GZ curve and the capsizing moments due to the trapped water on deck-well. When the model trapped much water on deck-well, it is heeled either to starboard side or port side and results its protected freeboard reduced. In such a condition, the model more easily ships water on the deck in severe seas. The water on deck thus increases rapidly and finally causes the model to be heeled at the angle of its bulwark edge becoming immersed.

In this condition, if the GZ value at the angle is negative or nearly equal to

zero, the model will easily capsize. When the GZ at the angle is positive in such case as shown in Fig. 1 (e), capsizing possibility depends on the magnitude of the hydro-dynamic forces acting on the model due to the subsequent waves. Indeed, capsizing occurred mostly by the hydro-dynamic forces of waves combined with trapped water when the model run in quartering seas, while in following seas capsizing may occur closely relating to the GZ reduction on the wave crest amidship.

The encounter period to the waves in quartering seas is longer than the other heading angles, so the motion becomes very slow. Then in such a condition, the GZ of the model could be estimated by a hydro-static calculation. Fig. 8 gives the typical GZ curves in following waves for model B and model F. For the wave profile as shown in Fig. 9, sine waves have been assumed with wave crest or trough amidship. The GZ curves are calculated until the angle of the bulwark edge becoming immersed when the free- ing ports are closed, and the calculations were performed with trim free condition.

These figures show that the angles of the bulwark edge becoming immersed in waves are smaller than in still water. In addition the GZ values of model B becomes smaller in wave crest amidship. So the model may easily capsize by the stability loss when it catches the water onto deck in such a condition. While, model F has good stability in waves rather than in still water till the angle of the bulwark edge becoming immersed.

This feature also appears on model G under a certain condition for the stability calculation in waves. Although there are only two examples, the fishing vessels having a wide over-hanged deck won't considerably reduce its stability in wave crest amidship.

However, the angles of the bulwark edge becoming immersed reduces in waves as like as those of model B. Therefore, if the model has been heeled by the moment due to the trapped water on deck or cargo shift, the model will be endangered to capsize.

#### 4.3 Stability Criteria Indicating Safety Region Against Capsizing

The experimental results have been compared with the GM-Freeboard diagram representing the  $C_1$ ,  $C_2$  and  $C_3$  values proposed by Tsuchiya [3] as the stability criteria for fishing vessels of intact condition.

$C_1$  value is the criteria for preventing shipping water,  $C_2$  value is one of the so-called weather criteria for taking into account of rolling motion in severe wind and waves and  $C_3$  value is the criteria for withstanding to the effect of the trapped water on deck-well, respectively.

The results are shown in Fig. 10 by full scale. The  $C_1$ ,  $C_2$  and  $C_3$  value are calculated by such conditions corresponding to the experiments. The conditions for the calculation are as follows : the limit



angle needed for  $C_1$  or  $C_2$  calculation is assumed to be 40 degrees, the heeling moments due to wind are not taken into account and the angle of bulwark edge becoming immersed is adopted as the flooding angle.

All of the results shows that the safety zone on ship's stability is restricted by  $C_1$  and/or  $C_3$  value. Since fishing vessels are operated with rather low freeboard condition, therefore, such stability criteria as  $C_1$  or  $C_3$  value concerning with shipping water should be introduced for judging the safety of fishing vessels.

## 5. CONCLUSION

The capsizing experiments were carried out for seven models of fishing vessels in oblique regular and irregular waves which were controlled to have the same spectrum form so as to be able to compare capsizing modes between vessel's type.

During the experiments, capsizing phenomena were visually observed, while the model motions and relative wave elevations were measured so that the characteristics of shipping water on deck which thought as the main factor in capsizing for fishing vessels could be clarified quantitatively.

The principal results of this investigation are as follows.

(1) Capsizing modes or the behaviours observed are somewhat different for each other in accordance with ship types as described as follows.

- a) Purse seiners showed typical modes of capsizing, i.e., capsizing caused by the effect of hydro-dynamic force due to waves combined with the trapped water on deck in quartering sea condition, and the pure loss of stability in following sea condition.
- b) Bonito fishing boats have tendency easier to shipping water because of lower protected freeboard and capsized or heeled heavily when the GM value was lowest. Large heeling was observed in following seas even in the better stability condition.
- c) The pair stern trawler has a relatively high protected freeboard and large superstructure on deck. Therefore, the shipping water was prevented and no capsizing was found.
- d) The small fishing boats were protected from the shipping water by a over-hanged deck. However, they receive much effect from the shipping water, if they once shipped water on the deck.

Generally speaking, the effect of shipping water on capsize is significant.

(2) The possibility of shipping water can be estimated by the calculation of strip theory, especially the anticipation of critical line dividing shipping or no-shipping water conditions in quartering regular waves. The relative wave elevations on the hull side in irregular waves can be also estimated by the linear superposition theory applying to the responses in the regular waves.

(3) The effects of shipping water on cap-

sizing are significant for fishing vessels, in particular, for small fishing boats. The protected freeboard increases the safety of the vessel against shipping water as it increases. However, a fishing vessel could not have so high protected freeboard from the viewpoint of fishing operations. Therefore, when water is once shipped on the deck over the bulwarks and she trapped water on her deck, she will be endangered to capsize.

Thus the effects of shipping water should be taken into account for stability criteria of fishing vessels to have the ability preventing shipping water or withstanding against it.

(4) The effectiveness of freeing ports for exhausting the water from the deck reduces in the case of almost zero-freeboard.

(5) The effect of stability loss due to waves should not be ignored.

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Table 1 Principal Dimensions of the Tested Model

Model	A	B	C	D	E	F	G
Length (m)	2.00	2.00	2.00	2.00	2.00	2.25	2.00
Breadth (m)	0.455	0.458	0.427	0.398	0.417	0.477	0.050
Depth (m)	0.179	0.183	0.210	0.176	0.189	0.216	0.195
Scale Ratio	16.5	15.3	14.75	19.6	15.1	6.4	7.6

Table 2 Tested Conditions of the Models

Model	Cond'n	d (cm)	trim (cm)	$\Delta$ (kg)	$\overline{GM}$ (cm)	$\overline{KG}$ (cm)	$\overline{OG}$ (cm)	$T_R$ (sec)	FB (cm)	PFB (cm)
A	I - 1	17.9	3.56	110.8	2.92	18.8	14.1	2.13	0.03	6.70
	I - 2	18.0	3.66	110.3	3.92	17.8	14.3	1.68	-0.10	6.57
	II - 1	15.7	3.55	91.7	3.14	19.2	12.5	2.20	1.58	8.85
	II - 2	15.6	3.55	90.9	3.90	18.4	12.4	1.80	1.70	8.97
	II - 3	15.7	3.55	91.7	4.58	17.7	12.5	1.64	1.58	8.85
B	I - 1	18.24	3.26	136.7	5.13	19.5	13.2	1.67	0.07	7.25
	I - 2	18.24	3.26	136.7	6.10	17.3	13.2	1.52	0.07	7.25
	II - 1	15.88	3.37	114.3	4.06	18.9	11.8	1.87	2.42	9.61
	II - 2	15.88	3.37	114.3	4.63	18.2	11.8	1.79	2.42	9.61
C	I - 1	18.79	8.40	115.0	2.00	18.5	9.76	2.30	2.22	10.02
	I - 2	18.79	8.40	115.0	3.00	17.5	9.76	1.87	2.22	10.02
	I - 3	18.79	8.40	115.0	4.00	16.5	9.76	1.56	2.22	10.02
D	I - 1	17.96	7.00	96.9	1.47	18.1	9.60	2.75	-0.36	3.93
	I - 2	17.96	7.00	96.9	1.99	17.6	9.60	2.36	-0.36	3.93
	I - 3	17.96	7.00	96.9	2.50	17.1	9.60	2.04	-0.36	3.93
	II - 1	16.76	7.00	88.3	2.18	17.3	8.78	2.03	0.84	5.13
	II - 2	16.76	7.00	88.3	2.50	17.0	8.78	2.26	0.84	5.13
E	I - 1	18.87	2.65	124.2	1.75	18.7	11.2	2.60	0.00	10.21
	I - 2	18.94	2.65	124.7	3.18	17.2	11.3	1.84	-0.07	10.13
	II - 1	16.74	2.65	106.0	4.58	16.1	10.3	1.66	2.14	12.34
F	I - 1	18.77	16.8	164.7	5.26	23.8	26.3	1.96	2.80	14.16
	II - 1	16.55	16.8	137.7	2.17	23.9	25.0	2.44	5.00	16.38
G	I - 1	18.00	7.50	135.5	3.89	22.4	11.1	2.30	1.47	14.76

Table 3 Number of Capsized Cases in Irregular Waves

Speed	Heading Angle	Freeing Ports	Model						Total
			A	B	D	E	F	G	
Half	180 (Head)	close	12 - 4	1 - 0		5 - 0			18 - 4
		open	3 - 0	1 - 0		1 - 0			5 - 0
	135 (Bow)	close	7 - 2	1 - 0		2 - 0			10 - 2
		open		1 - 0					1 - 0
	90 (Beam)	close	7 - 0	1 - 0	2 - 0	4 - 0	2 - 0		16 - 0
		open	2 - 0	2 - 0			1 - 0		5 - 0
	45 (Quart.)	close	12 - 8	21 - 5	10 - 7	8 - 0	7 - 0		58 - 20
		open	8 - 4	8 - 4	4 - 0	2 - 0	3 - 0		25 - 8
	0 (Follow)	close	12 - 4	7 - 0	10 - 1	19 - 0	11 - 0		59 - 5
		open	4 - 1	3 - 1	4 - 0	3 - 0	3 - 0		17 - 2
Normal	180 (Head)	close		1 - 0		7 - 0	6 - 1		14 - 1
		open			1 - 0	6 - 0			7 - 0
	135 (Bow)	close		1 - 0		4 - 0	3 - 0		8 - 0
		open				5 - 0			5 - 0
	90 (Beam)	close		2 - 0	2 - 0	7 - 0	6 - 0	7 - 0	24 - 0
		open		3 - 0		7 - 0	1 - 0	3 - 0	14 - 0
	45 (Quart.)	close		8 - 0	10 - 5	23 - 0	37 - 3	12 - 0	90 - 8
		open		4 - 0	4 - 2	10 - 0	12 - 3	5 - 0	35 - 5
	0 (Follow)	close		11 - 0	9 - 6	18 - 0	28 - 0	10 - 0	76 - 8
		open		10 - 1	5 - 0	7 - 0	14 - 2	5 - 0	41 - 3

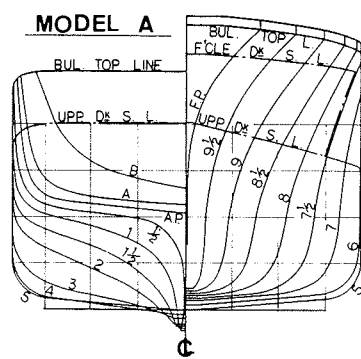


Fig. 1 (a)

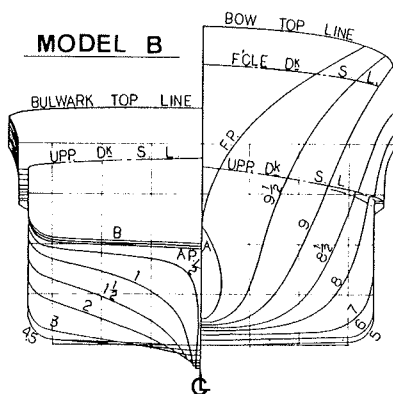
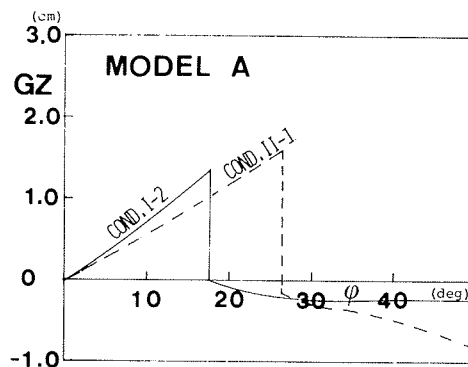


Fig. 1 (b)

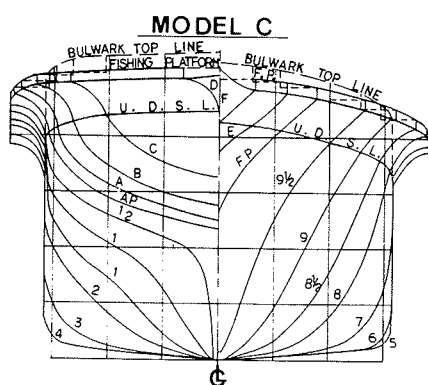
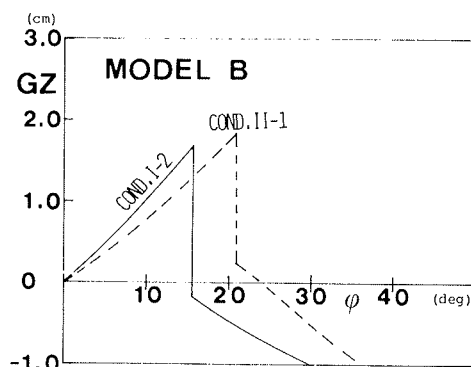


Fig. 1 (c)

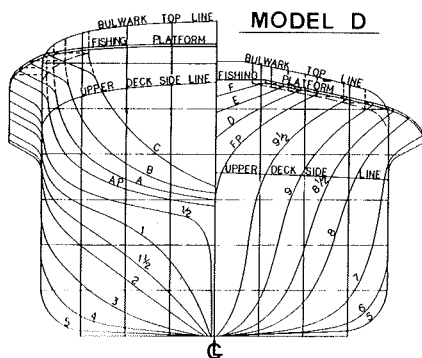
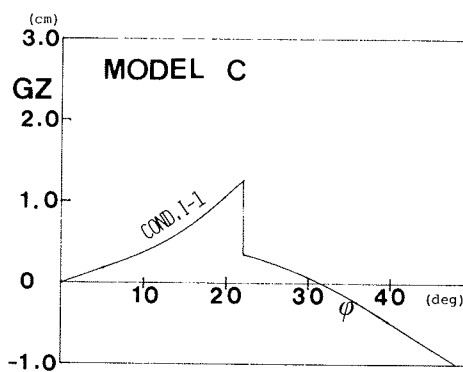


Fig. 1 (d)

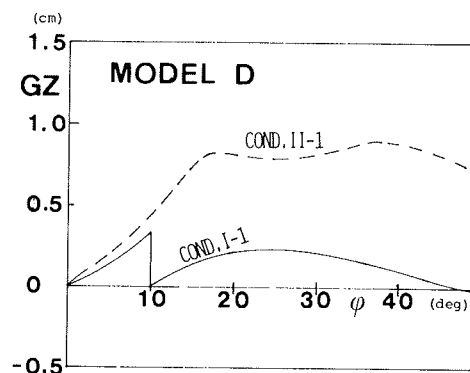


Fig. 1 (a) - (d) Body Plans and Righting Arm Curves

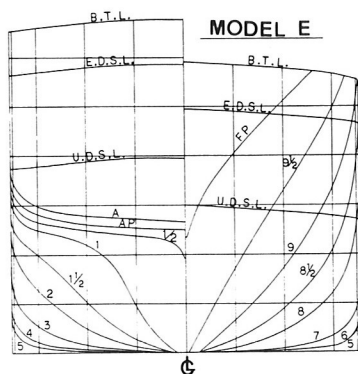


Fig. 1 (e)

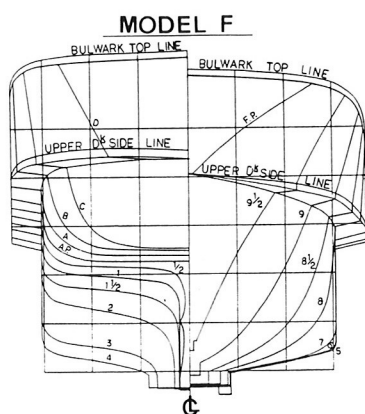


Fig. 1 (f)

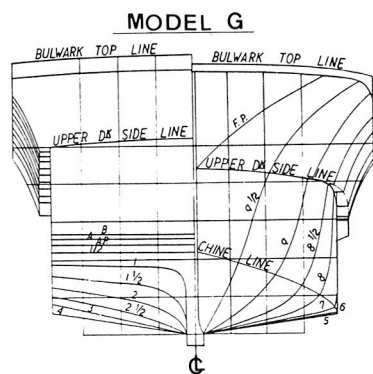


Fig. 1 (g)

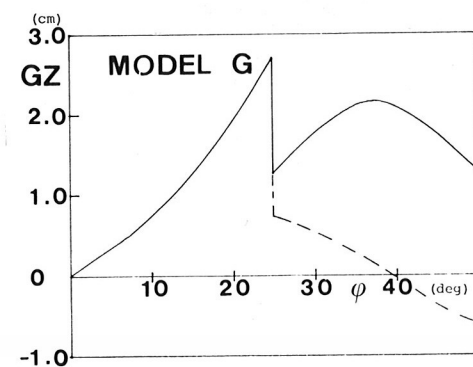
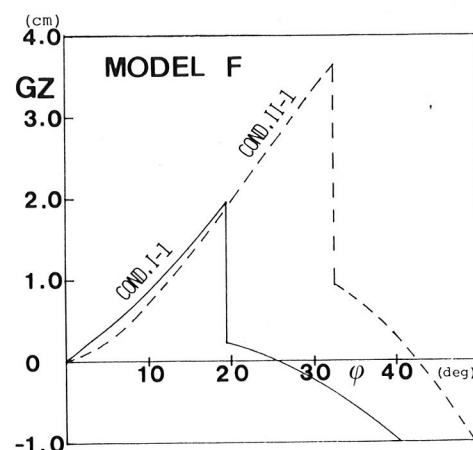
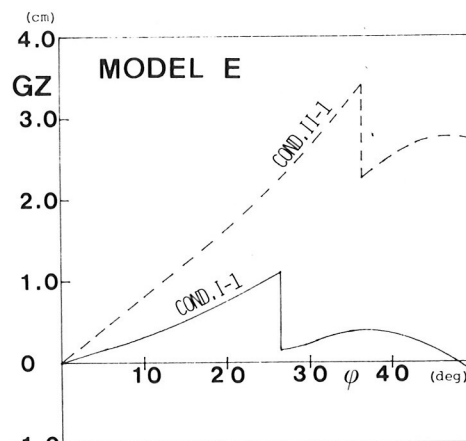


Fig. 1 (e) - (g) Body Plans and Righting Arm Curves

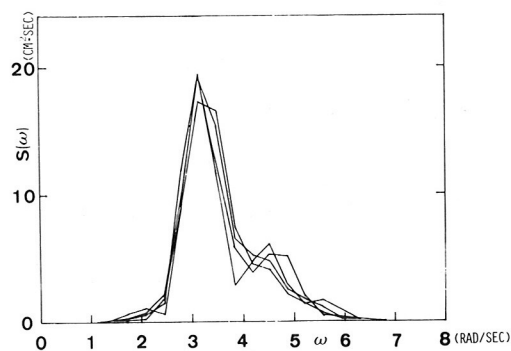


Fig. 2 Spectra of Irregular Waves Generated in the Basin



Fig. 3 Photograph of the Model in Waves

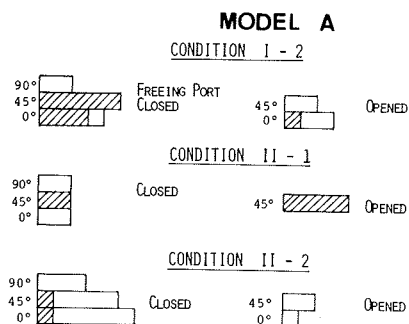


Fig. 4 (a) Rate of Capsizing in Irregular Waves for Model A

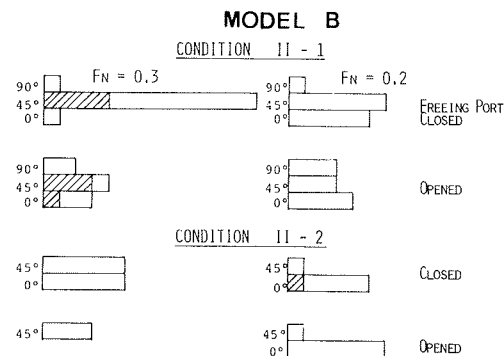


Fig. 4 (b) Rate of Capsizing in Irregular Waves for Model B

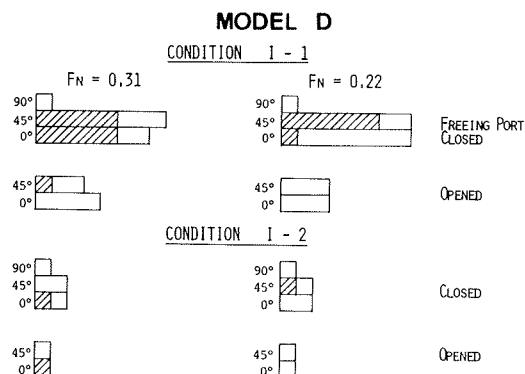


Fig. 4 (c) Rate of Capsizing in Irregular Waves for Model D

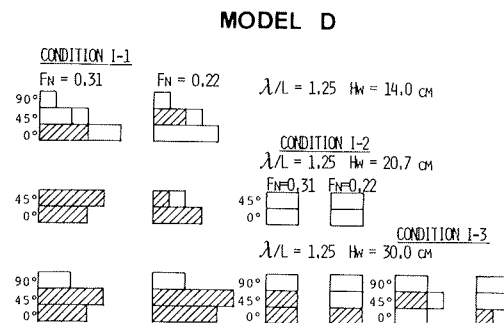


Fig. 4 (d) Rate of Capsizing in Regular Waves for Model D

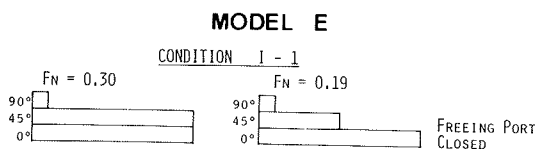


Fig. 4 (e) Rate of Capsizing in Irregular Waves for Model E

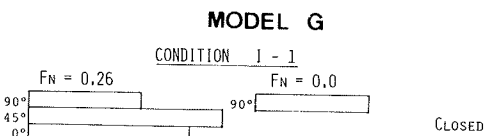


Fig. 4 (g) Rate of Capsizing in Irregular Waves for Model G

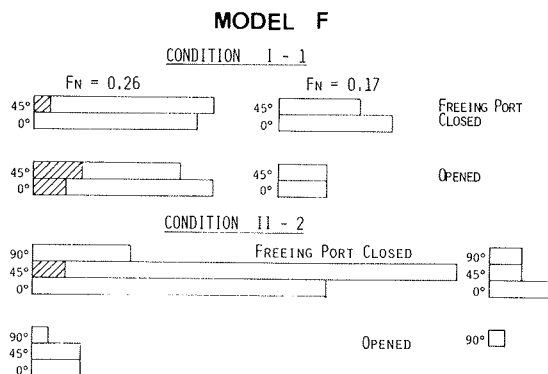


Fig. 4 (f) Rate of Capsizing in Irregular Waves for Model F

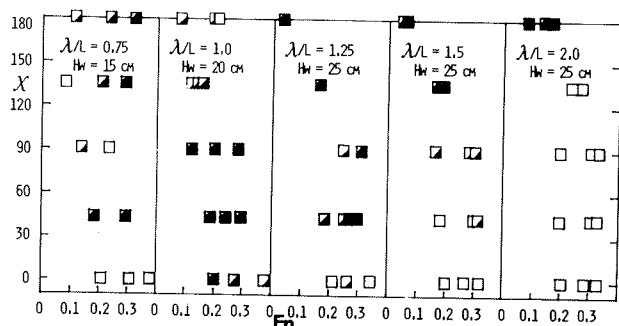


Fig. 5 (a) Degree of Shipping Water in Regular Waves for Model A

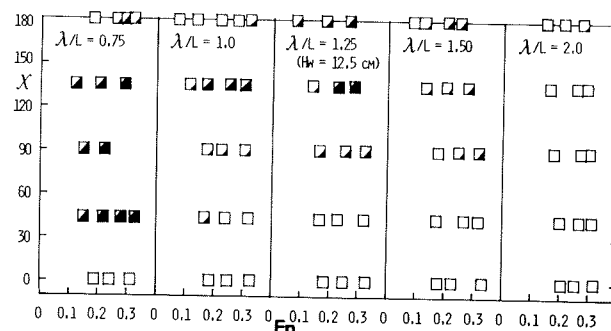


Fig. 5 (b) Degree of Shipping Water in Regular Waves for Model C

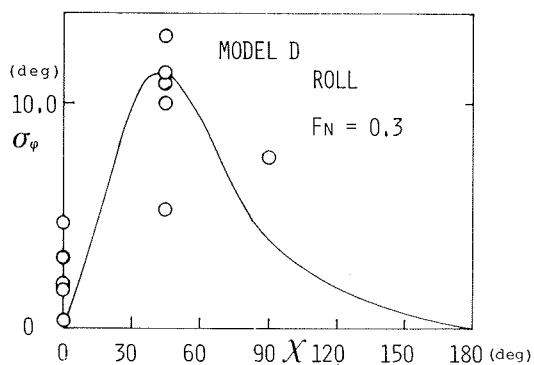


Fig. 6 (a) Rolling of Model D

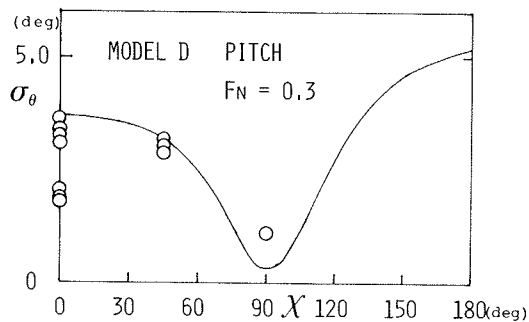


Fig. 6 (b) Pitching of Model D

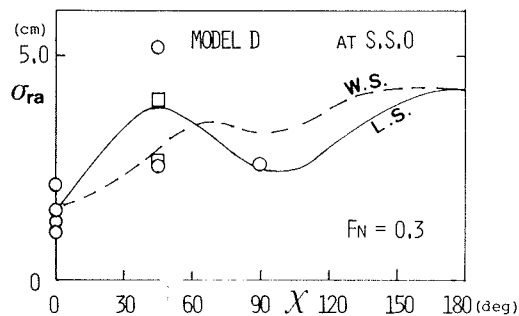


Fig. 6 (c) Relative Wave Elevations at A.P. of Model D

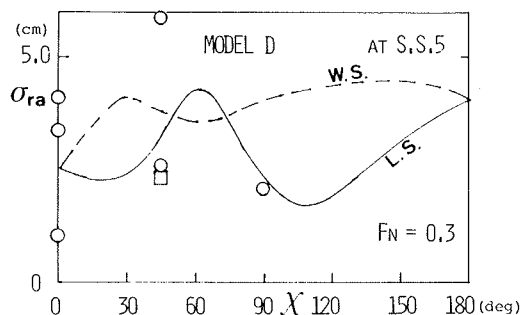


Fig. 6 (d) Relative Wave Elevations at Midship of Model D

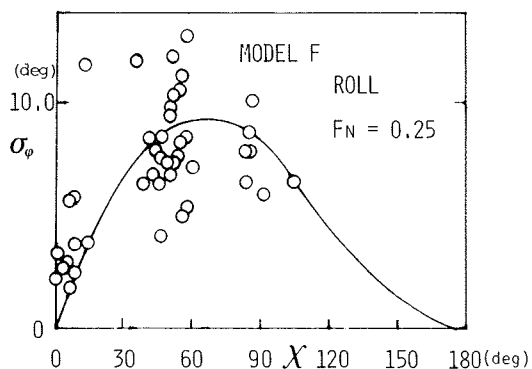


Fig. 6 (e) Rolling of Model F

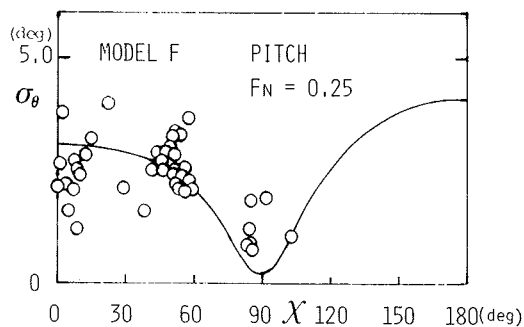


Fig. 6 (f) Pitching of Model F

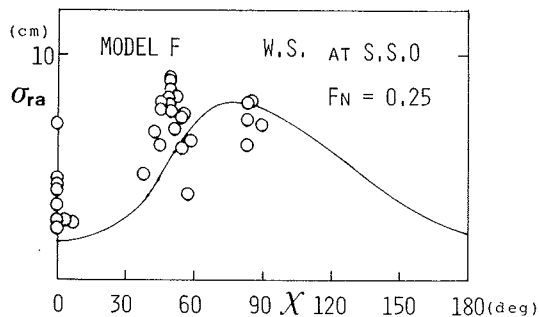


Fig. 6 (g) Relative Wave Elevations at A.P. of Model F, W.S.

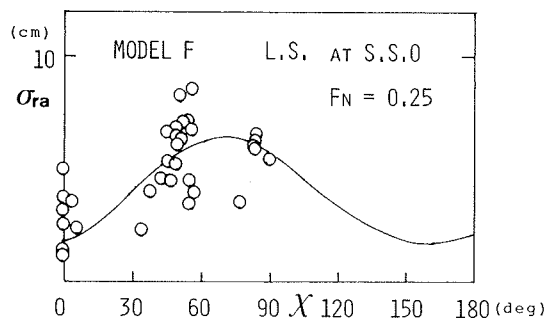


Fig. 6 (h) Relative Wave Elevations at A.P. of Model F, L.S.

Fig. 6 (a) - (h) Standard Deviations of Responses in Irregular Waves

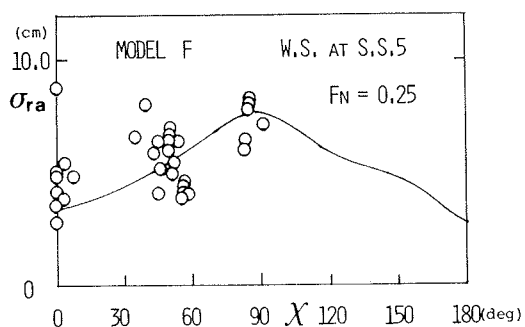


Fig. 6 (i) Relative Wave Elevations at Midship of Model F, W.S.

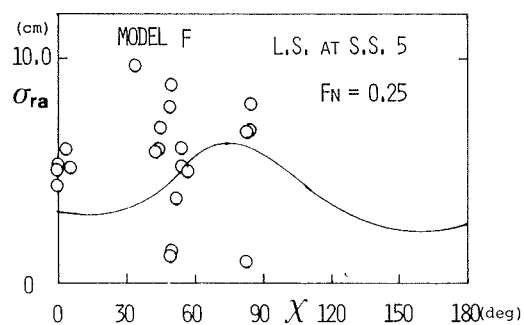


Fig. 6 (j) Relative Wave Elevations at Midship of Model F, L.S.

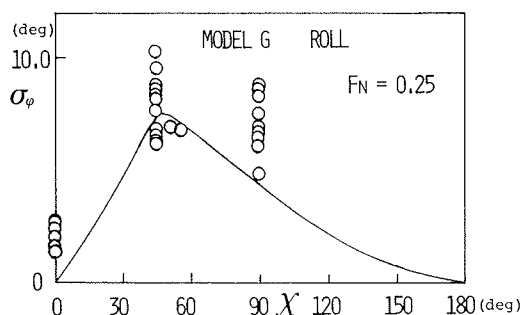


Fig. 6 (k) Rolling of Model G

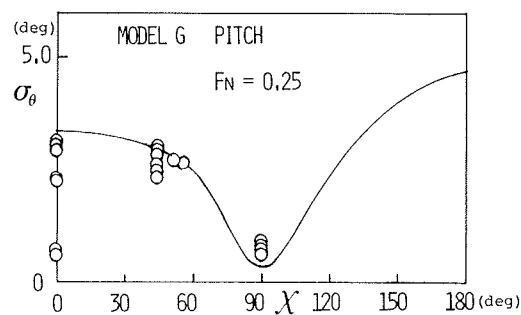


Fig. 6 (l) Pitching of Model G

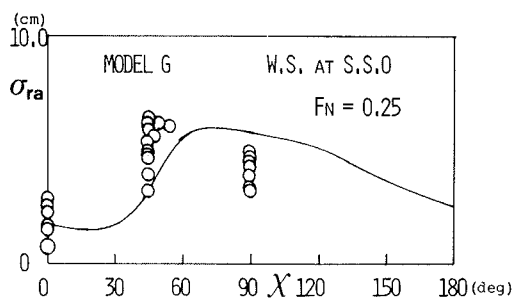


Fig. 6 (m) Relative Wave Elevations at A.P. of Model G, W.S.

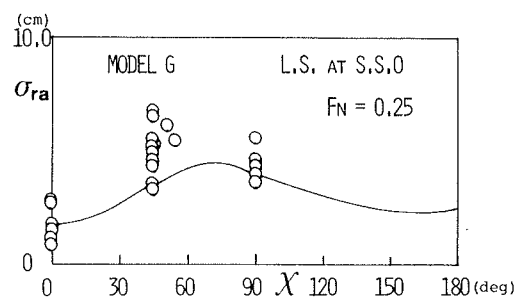


Fig. 6 (n) Relative Wave Elevations at A.P. of Model G, L.S.

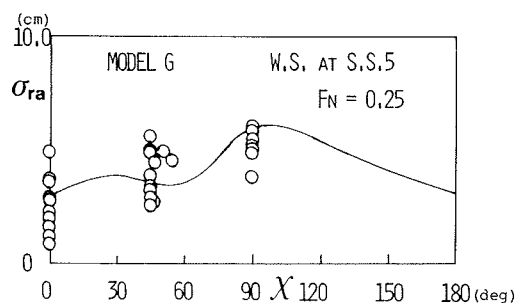


Fig. 6 (o) Relative Wave Elevations at Midship of Model G, W.S.

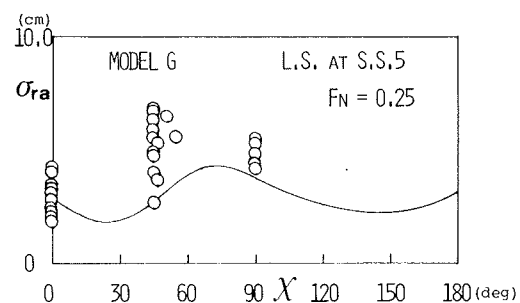


Fig. 6 (p) Relative Wave Elevations at Midship of Model G, L.S.

Fig. 6 (i) - (p) Standard Deviations of Responses in Irregular Waves

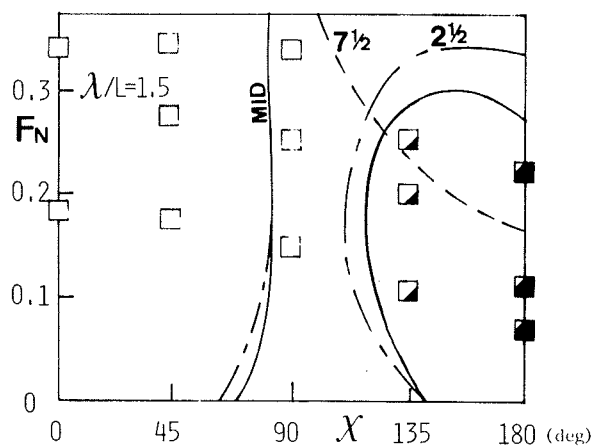


Fig. 7 (a)  $\lambda/L = 1.5$ ,  $H_W = 15$  (cm)

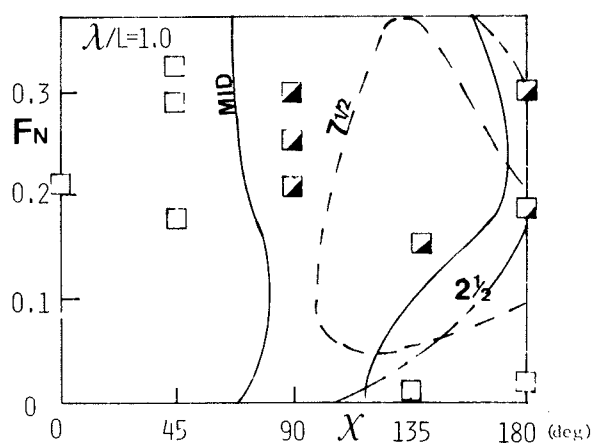


Fig. 7 (b)  $\lambda/L = 1.0$ ,  $H_W = 15$  (cm)

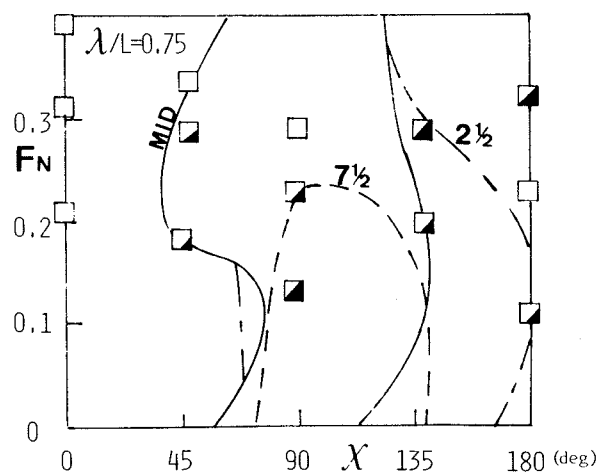


Fig. 7 (c)  $\lambda/L = 0.75$ ,  $H_W = 15$  (cm)

Fig. 7 Critical Lines Indicating Possibility of Shipping Water in Regular Waves for Model A

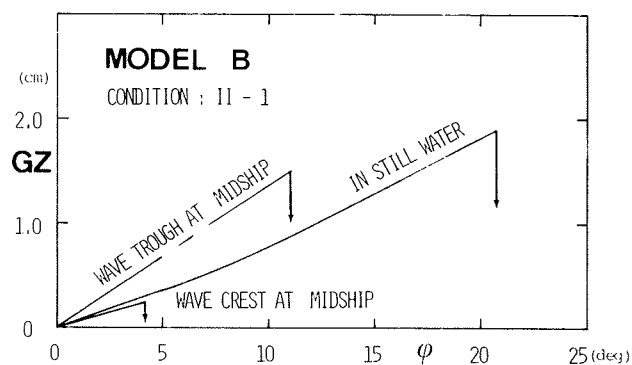


Fig. 8 (a) Righting Arm Curves of Model B in Still Water and in Wave

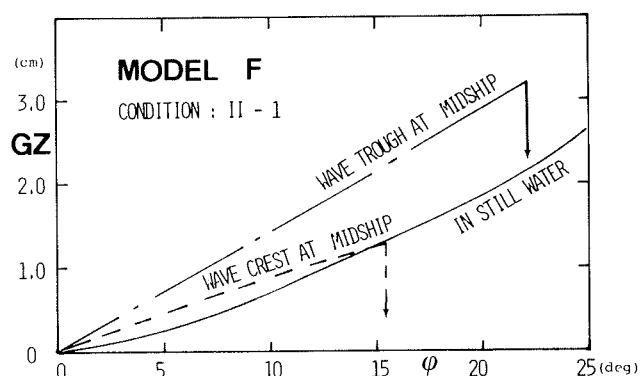


Fig. 8 (b) Righting Arm Curves of Model F in Still Water and in Wave

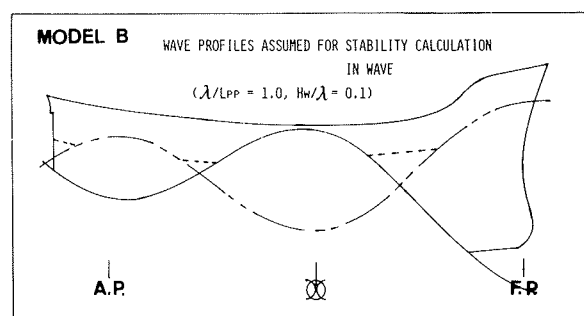


Fig. 9 Wave Profile Assumed for Stability Calculation of Model B in Wave, Corresponding to Fig. 8 (a)



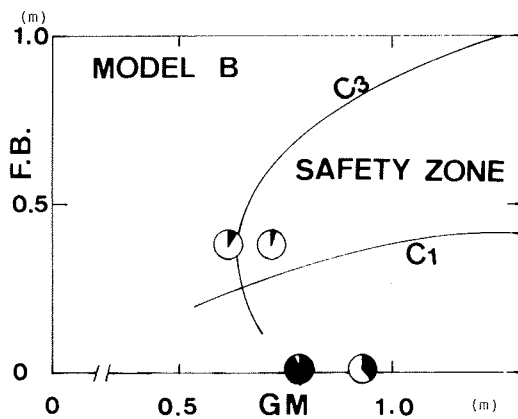


Fig.10 (a) GM - Freeboard Diagram for Model B

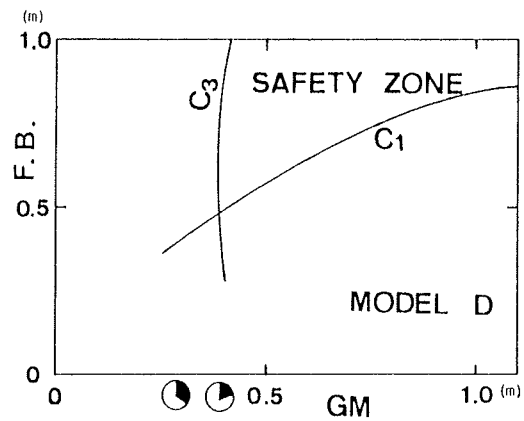


Fig.10 (b) GM - Freeboard Diagram for Model D

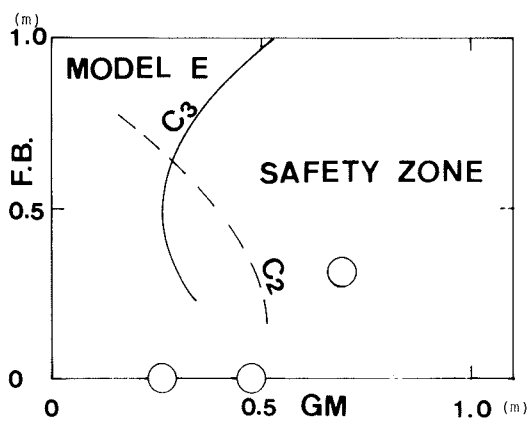


Fig.10 (c) GM - Freeboard Diagram for Model E

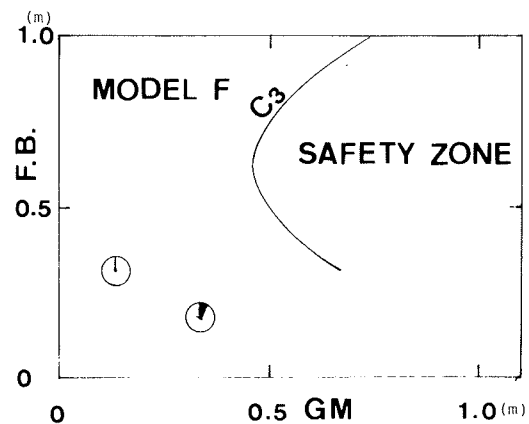


Fig.10 (d) GM - Freeboard Diagram for Model F

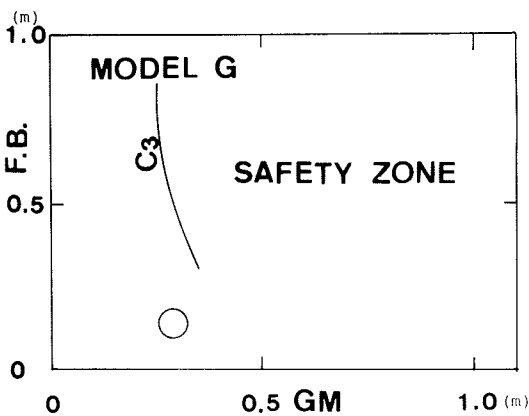


Fig.10 (e) GM - freeboard Diagram for Model G

Fig.10 (a)-(e)  
GM-freeboard Diagram for Each Model

\*the dark  
parts of each small circle  
indicate the rate of capsizing  
in irregular waves.

## Discussion

E. Dahle (University of Fisheries, Norway)

Some very interesting results are presented in this paper. To ease the reading of figures, values could be given in full scale instead of model scale. Also, more details on erections on the deck would be helpful.

These results are of special interest:

1. Model E does not capsize, although in cond I-1, the  $\theta_r=47$  degrees (printing error in paper). This is due to intact deck erections. This is important information to designers.
2. Model E has an excellent GZ-curve,  $\theta_r$  being above 60 degrees, due to the marine bulwarks on the model. Again, using voids in the bulwark is an excellent way of substantially improving GZ for small vessels. The idea has already been used by advanced designers.
3. The shape of model F shows slightly improved GZ-values for a wave crest amidships. Again, this is valuable information to designers.

I would appreciate the authors comments on these points. Finally I would like to know the relation between the GZ-curves used in the experiments and the IMCO(IMO) recommendations for fishing vessels.

### Author's Reply

The authors would like to thank Dr. Dahle for his constructive comments.

In generally speaking, it is very hard to know the exact navigation conditions for actual fishing vessels, especially when a ship capsized. Therefore, the aim of our study was firstly to find out the models of capsize for fishing vessels through the model experiments of various ship forms in the identical wave conditions, so as to make clear the effects of GZ-curves on capsize directly in model scale. Dr. Dahle pointed out the importance of the angle of vanishing stability against capsizing for fishing vessels. The authors would agree with his opinion in principle, and it's effect on capsize should be studied in more detail.

The over-hanged deck improves the GZ-curve of a ship up to the angle of bulwark edge becoming immersed. Fig.1 shows the wave profile (at  $\phi=0$ ) computed on a free-trim basis for model F. The over-hanged deck immerses even in the wave crest amidship and the GZ-values are slightly improved in such a loaded condition.

The effects of the erection on deck are very important for ship's safety, but the stability based on the erection on deck should be considered as the residual stability due to a reserve buoyancy for saving a ship against an abnormal wave condition.

Although the GZ-curves recommended by IMO for fishing vessels are not shown immediately, the GM-values required by IMO are shown in Fig.2. In these calculations, the effects of the bulwark are not taken into account.

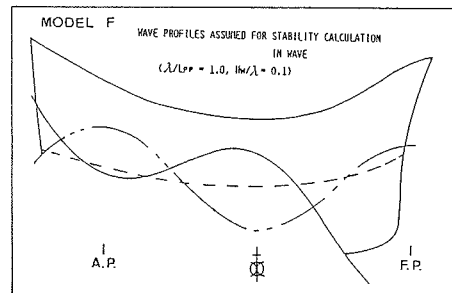


Fig.1 Wave Profile Assumed for Stability Calculation of Model F in Wave, Corresponding to Fig.8 (b)

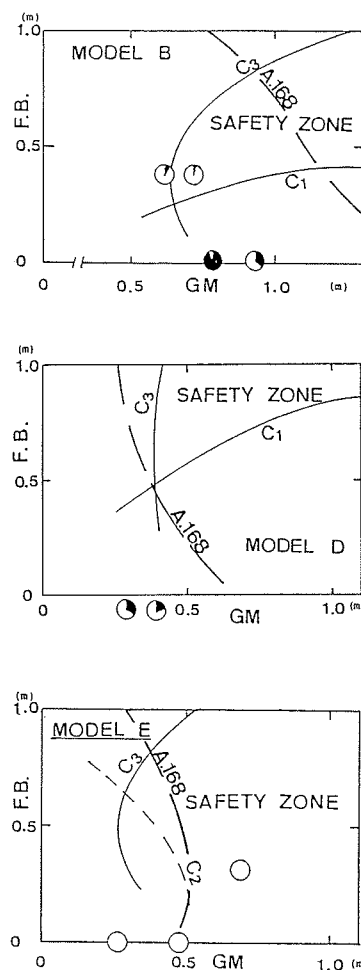


Fig.2 GM-values required by IMO

S. Kastner (College of Engineering  
Bremen, FRG)

I want to congratulate the authors on their thorough paper, in which they combine experiments with theoretical studies. The large number of results given - for seven different hull shapes in different seaways and headings - give a good statistical and experimental evidence on the extreme motion behaviour.

I have found it of particular interest, that in 59 runs in beam seas, you could not detect any capsizing, where as in following sea you had a lot of capsizes, but in quartering sea even more. I have two questions concerning the ship modelling.

1. Why did you use bulwarks in the model? We have tested without, as bulwarks do not contribute to the buoyancy of the heeled ship, and thus have not yet been included in any of the stability calculation procedures and regulations. If you wanted to measure the impact of trapped water on deck, did you check the similarity of the outflow of water through the freeing ports?
2. Did you employ an autopilot for course-keeping of the model in waves?

#### Author's Reply

The authors also would like thank Prof. Kastner for his valuable comments. The authors also have interests in the fact that any capsizing didn't occur in beam seas, except the several experiments held under initially heeled condition in which the model sometimes capsized even in beam seas. Therefore, the study on capsize in beam seas should be continued.

As to the first question, the similarity of the out-flow of water through the freeing ports was not checked yet. The model tests were carried out with bulwark in principle, because most of fishing vessels in full-size are equipped with the bulwark.

As to the second question, the models were controlled by manual steering. Of course, the auto-pilot is considered to be a better controlling method for the model tests.

#### T. Tsuchiya (Author's Reply)

The model tests of the fishing boat with bulwarks were not held. The reasons are the fishing boats in Japan have bulwarks mostly, and also the tests without bulwarks were thought unrealistic according to the

authors comparative tank tests with and without bulwarks held in regular waves. The tests result showed that the motion of a fishing boat with bulwarks was quite irregular and much different to that of the boat with bulwarks.

#### H. Hormann (Germanischer Lloyd, FRG)

1. Tests have been performed with models having different scales to the full size ship. Therefore the probability that the fullsize ship meets the seaway used is different; it decreases with increasing size.
2. The "Tsuchiya-factors" [ $C_1$ ,  $C_2$ ,  $C_3$ ] are different in their importance; I note that the factor accounting for wind is in no case considered here decisive. This points to my remarks on various occasions that one must not overestimate the effect of wind.
3. Fig. 8(b) What is the explanation for the surprisingly good wave-crest GZ-values of model F.
4. GZ-curves which take into account the bulwark may be misinterpreted; they are valid just in one direction, as soon as the bulwark edge comes to immerse, the curve as shown is not restored, when the ship is uprighting again, because the water on deck will be freed not simultaneously.

#### Author's Reply

The authors would like to thank Mr. Hormann for his useful comments.

As to the first and second comments, the authors agree with his opinions. As to the third comment, the authors have already explained in Dr. Dahle's discussion.

As to the last comment, the authors also would agree with his opinion in principle. However, the effects of the over-hanged deck and the bulwark on ship's safety should be studied in more detail, because most of the fishing vessels in Japan are low-built vessels.

Although in the  $C_1$ ,  $C_2$  and  $C_3$  calculations the effects of the over-hanged deck and the bulwark are not taken into account for GZ calculation, the breadth of the over-hanged deck and the height of bulwark are taken into account for calculating  $C_1$  or  $C_3$  value which is used to judge safety against shipping water on deck.

## TRANSVERSE STABILITY OF SHIPS IN A FOLLOWING SEA

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### ABSTRACT

In this paper, the transverse stability of small fishing boats in a following sea is investigated both theoretically and experimentally.

For the investigation, a ship model is towed at the same velocity as the phase velocity of a wave train generated by means of a wave-generating screen board attached beneath the towing carriage. The relative position of the ship to the wave is fixed at several different positions. The ship model has a given angle of heel and is free with regard to sinkage and trim. The righting moment and ship resistance acting on the model are measured with a 4-component dynamometer.

Next, calculations based on the Froude-Krylov Hypothesis are made to compare the results of the experiment. The calculations are carried out both with the effect of the wave orbital velocity taken into account and with it neglected. The Froude-Krylov calculations give consistent results for the experiment if the trim and sinkage of the ship on the wave are taken into account, thereby proving that the wave orbital velocity does have an effect.

In conclusion, a practical method of calculating the righting arm  $\overline{GZ}$  of a ship in a following sea is proposed.

### 1. INTRODUCTION

It is well known that when a ship is traveling in a heavy following wave of nearly the same length as the ship, the wave exciting force has a significant influence on the transverse stability of the ship. An additional effect of the

wave is to make a remarkable decrease of the ship resistance when the ship is positioned on the down slope of the wave. As a result, the ship will tend to travel with the wave at a position balanced on the wave, or with a velocity nearly equal to the phase velocity of the wave. Thus, the stability will vary with respect to the relative position of the ship to the wave, rather than to the encounter frequency of the ship to the wave. Such a condition seems to be a typical situation of a ship traveling in a following sea.

Thus, as an ideal theoretical example of this situation, we will herein consider the simplest condition where the ship is traveling at exactly the same velocity as the wave, because all dynamic quantities related to acceleration and frequency are irrelevant in this condition. We may be able to consider the stability of a ship in a following sea by using the traditional concept of the righting arm of a ship in still water.

In still water, when a ship is heeling at an angle  $\phi$ , the righting arm  $\overline{GZ}$  can be obtained by the following equation:

$$\overline{GZ} = \overline{BR} - \overline{BG} \sin \phi \quad (1)$$

where B is the center of buoyancy at the upright condition, G is the center of gravity, and R is as defined in Fig. 1. In a following wave, if the ship is traveling with the wave, the center of buoyancy will vary a bit from the original position. Due to this variation  $\overline{BB^*}$ , the righting arm will have to change as:

$$\overline{GZ} + \Delta \overline{GZ} = (\overline{BR} + \Delta \overline{BR}) - (\overline{BG} + \overline{BB^*}) \sin \phi \quad (2)$$

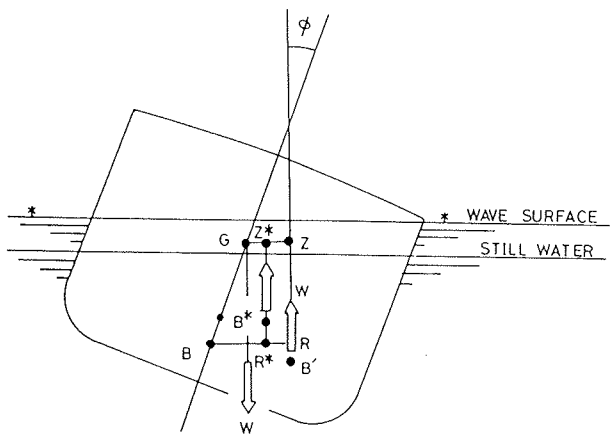


Fig. 1 Righting arm of ship model

and the change  $\Delta \overline{GZ}$  of the righting arm  $\overline{GZ}$  is obtained by the relation:

$$\Delta \overline{GZ} = \Delta \overline{BR} - \overline{BB^*} \sin \phi \quad (3)$$

where  $\Delta \overline{BR}$  and  $\overline{BB^*}$  are the changes in the form stability and in the gravitational stability, respectively.

In addition, the derivative of  $\Delta \overline{GZ}$  at a heel angle equal to zero will render the following change  $\Delta \overline{GM}$  in the metacentric height:

$$\Delta \overline{GM} = \Delta \overline{BM} - \overline{BB^*} \quad (4)$$

where  $\Delta \overline{BM}$  is the change in the metacentric radius BM which is equal to  $I/V$ . Since the inertia of the water plane  $I$  and the volume of the ship displacement  $V$  in still water will vary in a following wave as  $I + \Delta I$  and  $V + \Delta V$ ,  $\Delta \overline{BM}$  is approximately:

$$\Delta \overline{BM} = \overline{BM} (\Delta I/I - \Delta V/V) \quad (5)$$

The metacentric height  $\overline{GM}$  changes in a wave and the small change  $\Delta \overline{GM}$  of GM can be written as:

$$\Delta \overline{GM} = \overline{BM} (\Delta I/I - \Delta V/V) - \overline{BB^*} \quad (6)$$

where  $\Delta I$  is the small change of the inertia of the water plane.  $\Delta V$  is the small change in the displacement volume and  $\overline{BB^*}$  is the small change of the center of buoyancy at the upright condition. As is clear from equation (6),  $\Delta \overline{GM}$  increases with an increase of  $\Delta I$ ; however, increases of both  $\Delta V$  and  $\overline{BB^*}$  result in decreasing GM.

In this paper, based on such a consideration, an investigation of the stability of a ship model in a wave is carried out both experimentally and theoretically.

## 2. CAPTIVE MODEL TEST IN A FOLLOWING WAVE

In order to evaluate the change in  $\overline{GZ}$  quantitatively, we carried out a captive model test to measure the righting moment and ship resistance acting on a heeled model towed with the same velocity as the phase velocity of the wave. Fig. 2 shows a sketch of the set up for this experiment. The set up consists of a dynamometer to measure the righting moment and ship resistance, an apparatus to which

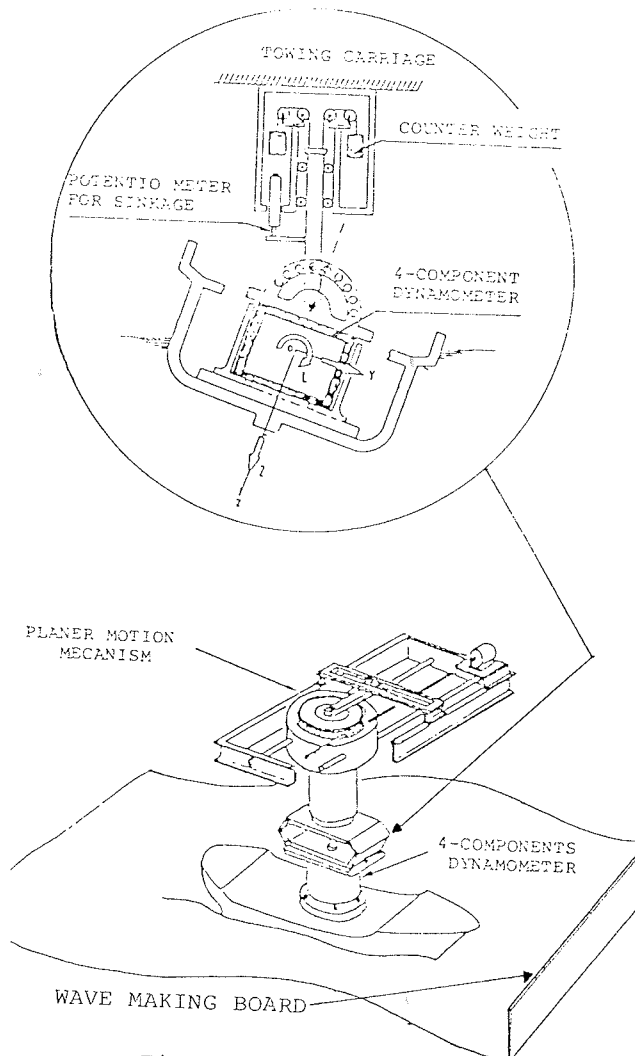


Fig. 2 Set up for experiment

to fasten the model so that it is free with regard only to sinkage and trim, and a wave generating screen board (Ref. 1) traveling with the towing carriage to make a wave train. The board is attached beneath the towing carriage with the lower edge about level with the surface of the water in still water, and it is tapered at an angle of 45 degrees. Towing the board at a constant velocity  $U$ , a wave train is generated on the surface of the still water disturbed by the edge, and the wave

train develops very uniformly. The crests and troughs lie at right angles to the direction of advance. In order to measure

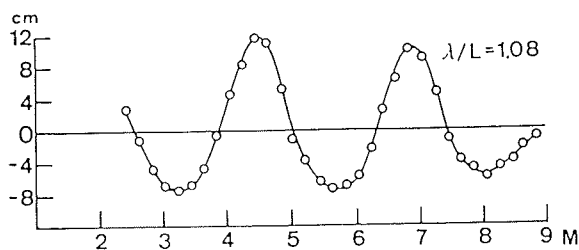


Fig. 3 Wave profile generated by wave generating board.

the waves exactly, a wave probe is attached to the carriage behind the board. The wave profiles shown in Fig. 3 were detected. According to the linearized

$L_{pp}$	2.2500 <sup>m</sup>
$B$	0.4766 <sup>m</sup>
$D$	0.2156 <sup>m</sup>
$d$	0.1360 <sup>m</sup>
$KM$	0.272 <sup>m</sup>
$GM$	2.38 <sup>cm</sup>

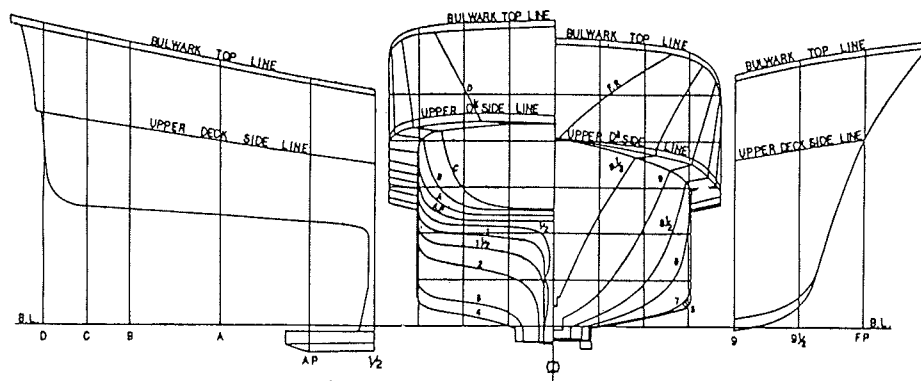


Fig. 4 Lines of ship model

theory of surface waves,

$$\lambda/U^2 = 2\pi/g \quad (7)$$

However, instead of the theoretical value, the observed value of  $\lambda/U^2$  was 0.57 where  $\lambda$  denotes the wave length and  $U$  denotes the phase velocity of the wave. And the ratio  $H/\lambda$  of the wave height and length shows that the waves generated by using this method were somewhat steeper than the  $H/\lambda = 1/20$  generally anticipated in the ship building industry, and the profile was similar to a trochoidal wave rather than a sinusoidal one. The model tests for a fishing boat as shown in Fig. 4 were conducted so as to tow the model heeled in the wave generated behind the board.

When the model is heeled up to an angle of  $\phi$  from the upright condition, the righting moment  $-K$  (which is detected with the dynamometer mounted to the model) is given as:

$$-K = W(\overline{BR} - \overline{BG}\sin\phi) \quad (8)$$

in still water, and that can be written as:

$$-(K + \Delta K) = W[(\overline{BR} + \Delta\overline{BR}) - (\overline{BG} + \Delta\overline{BG})\sin\phi] \quad (9)$$

in a generated wave. Because the change  $\Delta K$  can be obtained by the relation:

$$-\Delta K = W(\Delta\overline{BR} - \Delta\overline{BG}\sin\phi) \quad (10)$$

we can experimentally detect the  $\Delta K$  by towing the model heeled in a generated wave where  $W$  is the displacement of the model and  $-\Delta K$  is equal to  $W\Delta GZ$ .

The experimental results of  $\overline{GZ^*}/\overline{GZ}$  versus  $\xi_0/\lambda$  are plotted in Fig. 5. Where  $\overline{GZ^*}$  is the righting arm of the model in the wave and  $\xi_0$  is the distance to the midship of the model measured from the wave trough behind the model.  $\lambda$  is the wave length and the ratio  $\xi_0/\lambda$  represents the relative position of the model to the wave. As mentioned previously, the  $\overline{GZ^*}$  obtained from our model tests is greater

than the still water value with the wave trough amidships, and less than the still water value with the wave crest amidships.

Next, let us consider how the moment of inertia of the water plane and the displacement volume of the model affect the righting arm of the model in the wave. The sinkage  $(\zeta^* - \zeta)/a$  and the trim  $(\theta^* - \theta)/2\pi a/\lambda$  of a dimensionless form as a function of the relative position of the model to the wave,  $\xi_0/\lambda$ , are shown in Fig. 6, where  $(\zeta^* - \zeta)$  and  $(\theta^* - \theta)$  are the differences of the sinkage and trim of the model towed with the same velocity in still water and in a generated wave.  $a$  is the wave amplitude and  $2\pi a/\lambda$  is the maximum wave slope.

This model has large flared sections at the ends and wallsided sections amidships as shown in Fig. 4, and it can be seen that not only the sinkage and trim, but also the inertia of the water plane are different for each position of the model in relation to the wave. For

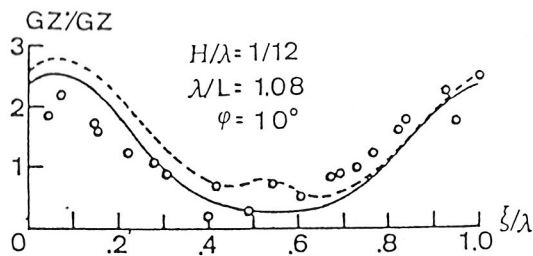


Fig. 5 Change in righting arm with relative position of model to wave

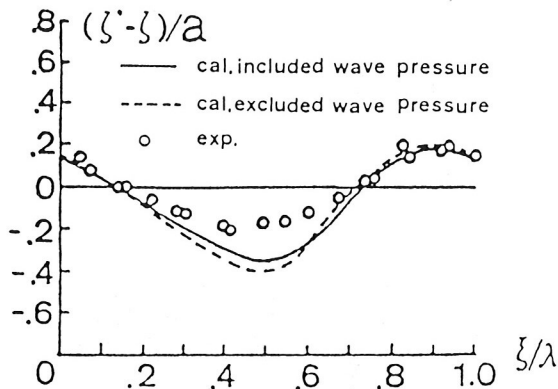


Fig. 6 Change in sinkage with relative position of model to wave

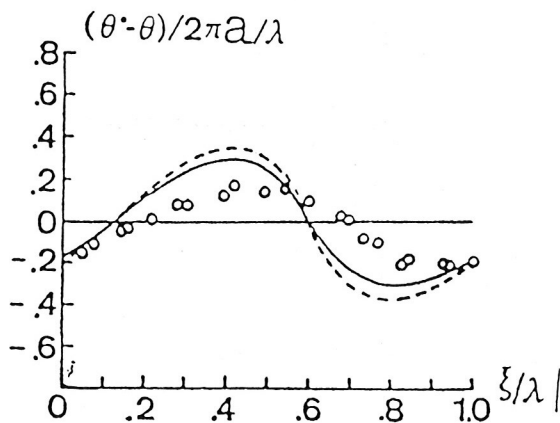


Fig. 7 Change in trim with relative position of model to wave

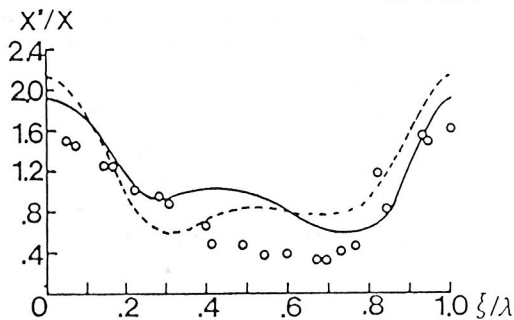


Fig. 9 Change in ship resistance with relative position of model to wave

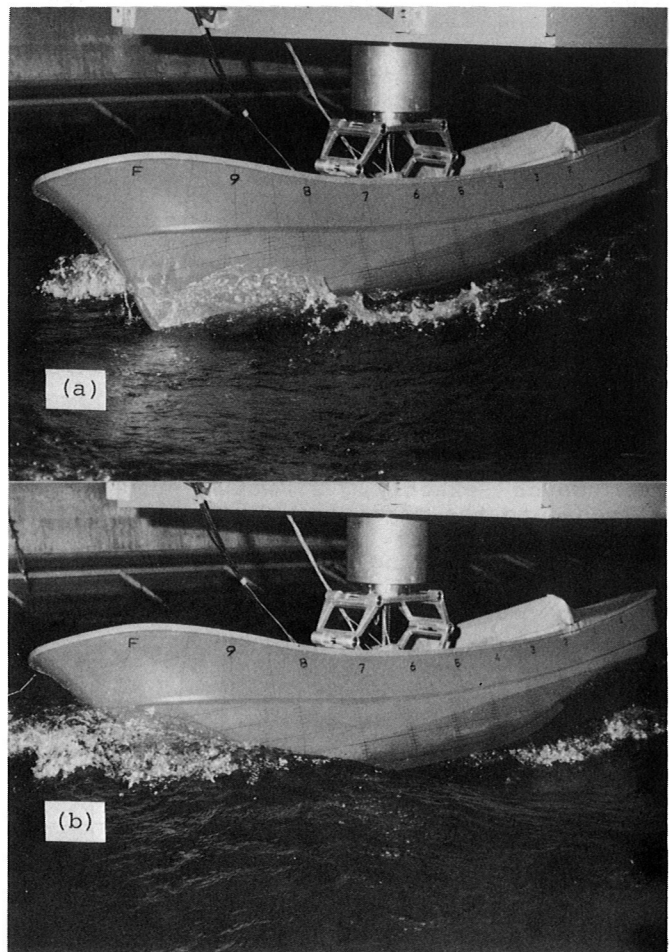


Fig. 8 Photographs of model in wave crest (a) and in wave trough (b)

example, (a) and (b) in Fig. 8 are photographs of the model heeled in a wave crest and in a wave trough, respectively.

In addition, how the ship resistance varied with respect to the relative position of the model to the wave is shown in Fig. 9, where  $X^*/X$  is the ratio of the ship resistance of the model towed in a wave and in still water, at the same velocity. From this diagram, it can be seen that the ship resistance remarkably decreases in the down slope toward the wave trough.

### 3. THEORETICAL ANALYSIS BASED ON THE FROUDE-KRYLOV HYPOTHESIS

In discussing the stability of a ship in waves, how to estimate the water pressure caused by the waves and how to obtain the forces and moments acting upon the ship are the most important problems.

In general, because of the orbital motion of the water particles, the water pressure at the crests is slightly less than that of still water, and is larger at the troughs. For this phenomenon we already have a solution which was obtained through a small wave theory. Now, suppose

a wave which has an amplitude  $a$  and a wave number  $k$  is moving steadily at a phase velocity  $c$ , as is seen in Fig. 10. By fixing the co-ordinates  $o \sim \xi, \eta, \zeta$ , to the still water level and the wave profile,  $\zeta_w$ , at a certain time  $t$ , can be represented by the following equation:

$$\zeta_w = a \cdot \cos k(\xi - ct) \quad (11)$$

Wave pressure  $p$  can then be calculated as:

$$p = \rho g \zeta - \rho g a e^{-k\zeta} \cos k(\xi - ct) \quad (12)$$

where  $\rho$  is the water density and  $g$  is the gravity acceleration. Accordingly, the vertical pressure gradient at any point  $\zeta$ ,  $dp/d\zeta$  then becomes:

$$dp/d\zeta = \rho g [1 + a k e^{-k\zeta} \cos k(\xi - ct)] \quad (13)$$

Recalling  $k = 2\pi/\lambda$  then  $k\zeta = 2\pi\zeta/\lambda$ ,  $e^{-k\zeta}$  does not much change at the water surface ( $\zeta=0$ ) through the ship's bottom ( $\zeta=d$ ). It is handy and sensible then, to use a certain constant value of  $e^{-k\zeta}$ , i.e.,  $e^{-k\zeta_s}$  from the surface to the bottom.  $\zeta_s$  is thus a representative or effective draft of a cross section of a ship, defined as the sectional area divided by the sectional breadth on the surface. The vertical pressure gradient at a cross section of a ship is then

$$(dp/d\zeta)_{A(x)} = \rho g [1 + a k e^{-k\zeta_s} \cos k(\xi_0 + x)] \quad (14)$$

where  $\xi_0$  denotes the distance of the midship section in front of the nearest trough and  $x$ , the distance of the cross section in front of the midship.  $\zeta_s$  is the representative draft of the section.

Incidentally Formula (14) implies that the effect of the orbital motion on the vertical pressure gradient can virtually be expressed by changing the water density; apparent high density at troughs and low density at crests. Such an apparent density at a cross section  $A(x)$ , we write it  $\rho^*$ , is:

$$\rho^* = \rho [1 + a k e^{-k\zeta_s} \cos k(\xi_0 + x)] \quad (15)$$

On the other hand, the righting moment  $WGZ$  for a heeled ship in a longitudinal wave can be written as:

$$WGZ = - \int_{x_A}^{x_F} (dK/dx) dx - W \overline{OG} \sin \phi \quad (16)$$

In equation (16), the second term on the right hand side indicates the righting moment caused by the ship's weight. The  $K$  in the first term is another righting moment caused by the water pressure around the ship's hull and, therefore, this term might be called the righting moment caused by the hull form.

Fig. 10 is a schematic explanation of the relations mentioned above. In the figure,  $b(z)$  is the breadth of the water plane as a function of the variable  $z$ .  $d$  is the draft at the upright condition in still water.  $Z_p$  and  $Z_s$  indicate port and starboard side immersions in a wave, respectively, as shown in Fig. 10. Now we can calculate the  $dK/dx$  by using the

following equation:

$$(dK/dx) = -\rho g [1 + a k e^{-k\zeta_s} \cos k(\xi_0 + x)]$$

$$\begin{aligned} & \cdot \left\{ \cos \phi \int_{Z_p}^{Z_s} \frac{1}{2} b^2(z) dz \right. \\ & - \sin \phi \left[ \int_{Z_p}^{Z_s} 2zb(z) dz + \int_{Z_p}^0 zb(z) dz \right. \\ & + \left. \int_0^d zb(z) dz \right] \\ & - \frac{(1 - \tan^2 \phi) \sin \phi}{6} \\ & \cdot [b^3(Z_p) + b^3(Z_s)] \\ & - \frac{\tan \phi \sin \phi}{2} \\ & \cdot [Z_p b^2(Z_p) - Z_s b^2(Z_s)] \} \quad (17) \end{aligned}$$

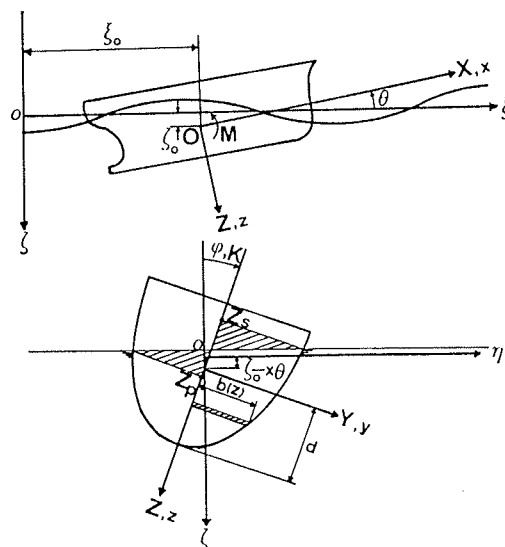


Fig. 10 Coordinate system

The initial righting moment  $WGM$  can also be obtained through a relation,  $GM = (dGZ/d\phi)_{\phi=0}$ , and practical calculations can be performed by the following formula:

$$\begin{aligned} WGM = & \rho g \int_{x_A}^{x_F} \frac{2}{3} b^3(Z_c) dx - \rho g \int_{x_A}^{x_F} dx \int_{Z_c}^0 2zb(z) dz \\ & - WBG + \rho g a k e^{-k\zeta_s} \int_{x_A}^{x_F} \frac{2}{3} b^3(x) \\ & \cdot \cos k(\xi_0 + x) dx - \rho g a k e^{-k\zeta_s} \int_{x_A}^{x_F} dx \\ & \cdot \int_0^d 2zb(z) \cos k(\xi_0 + x) dx \quad (18) \end{aligned}$$

In formula (18),  $Z_c$  is the immersion in a



wave at the upright condition. When all  $Z_p$  and  $Z_s$  values for a heeled ship in longitudinal waves are found, the WGZ values can easily be calculated by formula (16).

The  $Z_p$  and  $Z_s$  are at this stage still unknown quantities. In order to define them, the equilibriums in regard to trim and sinkage are used. Those are:

$$\begin{aligned} & -\rho g \int_{x_A}^{x_F} [1 + a k e^{-k \zeta_s} \cos k(\xi_0 + x)] \\ & \cdot A(x) dx + W = 0 \\ & \int_{x_A}^{x_F} [1 + a k e^{-k \zeta_s} \cos k(\xi_0 + x)] \\ & \cdot x A(x) dx = 0 \end{aligned} \quad (19)$$

where  $A(x)$  denotes the sectional area of the ship under water. Referring to Fig. 10,  $A(x)$  can be represented as:

$$\begin{aligned} A(x) = & \int_0^d 2b(z) dz + \int_{Z_p}^0 b(z) dz + \int_{Z_s}^0 b(z) dz \\ & + \frac{1}{2} [b^2(Z_p) - b^2(Z_s)] \tan \phi \end{aligned} \quad (20)$$

Now, if we let  $A(x)$  equal the sectional area of the ship at the upright condition and  $A_0(x) = \int_0^d 2b(x) dz$ ,  $A(x)$  then becomes:

$$A(x) = A_0(x) + A^*(x) \quad (21)$$

The second term of the right side of equation (21) is given:

$$\begin{aligned} A^*(x) = & \int_{Z_p}^0 b(z) dz + \int_{Z_s}^0 b(z) dz \\ & + \frac{1}{2} [b^2(Z_p) - b^2(Z_s)] \tan \phi \end{aligned} \quad (22)$$

By putting formula (22) into equation (19) and by neglecting small quantities of the second order, two simultaneous equations are obtained as follows:

$$\begin{aligned} & a k e^{-k \zeta_s} \int_{x_A}^{x_F} A_0(x) \cos k(\xi_0 + x) dx \\ & + \int_{x_A}^{x_F} A^*(x) dx = 0 \end{aligned} \quad (23)$$

$$\begin{aligned} & a k e^{-k \zeta_s} \int_{x_A}^{x_F} x A_0(x) \cos k(\xi_0 + x) dx \\ & + \int_{x_A}^{x_F} x A^*(x) dx = 0 \end{aligned} \quad (24)$$

For solving formulas (21) and (22), a perturbation method with a small heel angle  $\Delta\phi$  is best. When a ship is intended to heel  $\Delta\phi$  more from the heel angle  $\phi$ , the underwater sectional area  $A^*(x) + \Delta A^*(x)$  is expressed:

$$\begin{aligned} A^*(x) + \Delta A^*(x) = & \int_0^d b(z) dz + \int_{Z_p}^{Z_p + \Delta Z_p} b(z) dz \\ & + \int_{Z_s}^0 b(z) dz + \int_{Z_s}^{Z_s + \Delta Z_s} b(z) dz \\ & + \frac{1}{2} [b^2(Z_p + \Delta Z_p) - b^2(Z_s + \Delta Z_s)] \\ & \cdot \tan(\phi + \Delta\phi) \end{aligned} \quad (25)$$

In this case, there is a special relationship among  $x$ ,  $y$ ,  $z$ ,  $\zeta_0$ ,  $\phi$ , and  $\theta$  as follows:

$$\zeta = \zeta_0 - x\theta + y \cdot \sin \phi + z \cdot \cos \phi \quad (26)$$

Through the above relationship,  $Z_p$  and  $Z_s$  can be found by the following formulas:

$$Z_p = - \frac{\zeta_0 - x\theta - b(x, Z_p) \sin \phi - a \cdot \cos k(\xi_0 + x)}{\cos \phi} \quad (27)$$

$$Z_s = - \frac{\zeta_0 - x\theta + b(x, Z_s) \sin \phi - a \cdot \cos k(\xi_0 + x)}{\cos \phi} \quad (28)$$

Small changes in  $Z_p$ , and  $Z_s$ , namely  $\Delta Z_p$  and  $\Delta Z_s$ , caused by small heel angle are:

$$\Delta Z_p = - \frac{\Delta \zeta_0 - x \Delta \theta - \Delta \phi [b(Z_p) \cos \phi + Z_p \cdot \sin \phi]}{\cos \phi - b'(Z_p) \sin \phi} \quad (29)$$

$$\Delta Z_s = - \frac{\Delta \zeta_0 - x \Delta \theta + \Delta \phi [b(Z_s) \cos \phi - Z_s \sin \phi]}{\cos \phi + b'(Z_s) \sin \phi} \quad (30)$$

Let us again neglect second order small quantities, and  $A^*(x)$  becomes:

$$\begin{aligned} A^*(x) = & \frac{(\Delta \zeta_0 - x \Delta \theta) [b(Z_p) + b(Z_s)]}{\cos \phi} \\ & - \Delta \phi \frac{[b^2(Z_p) - b^2(Z_s)] (1 - \tan^2 \phi)}{2} \\ & - \Delta \phi \frac{2[Z_p b(Z_p) + Z_s b(Z_s)] \tan \phi}{2} \end{aligned} \quad (31)$$

An equation for satisfying the relationship between  $\Delta\phi$  and  $A^*(x)$  is obtained:

$$\int_{x_A}^{x_F} \Delta A^*(x) dx = 0, \quad \int_{x_A}^{x_F} x \Delta A^*(x) dx = 0 \quad (32)$$

Substituting equation (31) into equation (32) results in simultaneous equations obtained as follows:

$$\begin{aligned}
& \Delta \zeta_0 \int_{x_A}^{x_F} [b(Z_p) + b(Z_s)] dx \\
& - \Delta \theta \int_{x_A}^{x_F} x [b(Z_p) + b(Z_s)] dx = \Delta \phi \quad F(x) dx \\
& \Delta \zeta_0 \int_{x_A}^{x_F} x [b(Z_p) + b(Z_s)] dx \\
& - \Delta \theta \int_{x_A}^{x_F} x^2 [b(Z_p) + b(Z_s)] dx = \Delta \phi \quad x F(x) dx
\end{aligned} \quad (33)$$

$$\begin{aligned}
2F(x) = [b^2(Z_p) - b^2(Z_s)](1 - \tan^2 \phi) \cos \phi \\
+ 2[Z_p b(Z_p) + Z_s b(Z_s)] \sin \phi
\end{aligned} \quad (34)$$

In practical calculations of equation (33), we have to start from the upright condition before advancing to the next heeled condition; and by starting from the heeled condition, we can reach another condition heeled twice as much,  $2x\Delta\phi$ . By repeating these operations several times, we can reach the desired condition defined by  $\zeta_0$  and  $\theta$ .

At the beginning of these calculations it is necessary to fix the  $\zeta_0$  and  $\theta$  values at the upright condition  $\phi = 0$  in waves. In the upright condition, the immersions at port and starboard sides,  $Z_p$  and  $Z_s$ , are equal to each other; thus, instead of  $Z_p$  and  $Z_s$ , if we put  $Z_c$  into equation (27), the following formula is obtained:

$$Z_c = -\zeta_0 + x\theta + a \cdot \cos k(\xi_0 + x) \quad (35)$$

Through equations (33), (34), and (35), the changing of the sectional area,  $A^*(x) + \Delta A^*(x)$ , can be expressed:

$$\begin{aligned}
A^*(x) + \Delta A^*(x) = \int_{Z_c}^{Z_c'} 2b(z) dz \\
+ \int_{Z_c + 2Z_c}^{Z_c} 2b(z) dz
\end{aligned} \quad (36)$$

Equations (23) and (24) can now be rewritten in solvable form as follows:

$$\begin{aligned}
& \Delta \zeta_0 \int_{x_A}^{x_F} 2b(Z_c) dx - \Delta \theta \int_{x_A}^{x_F} 2xb(Z_c) dx \\
& = \Delta a \int_{x_A}^{x_F} G(x) dx \\
& \Delta \zeta_0 \int_{x_A}^{x_F} 2xb(Z_c) dx - \Delta \theta \int_{x_A}^{x_F} 2x^2 b(Z_c) dx \\
& = \Delta a \int_{x_A}^{x_F} xG(x) dx
\end{aligned} \quad (37)$$

$$G(x) = [2b(Z_c) - ke^{-k\zeta_{SA_0}(x)}] \cos k(\xi_0 + x) \quad (38)$$

In this case, the above equations might be solved from a basic step of  $a = 0$ , and by adding a small amount of  $a$ , namely  $a$ , we can reach a new step of  $a + \Delta a$ . By adding the  $\Delta a$  several times, the desired values of  $\zeta_0$  and  $\theta$  for a ship in waves can be

obtained. Body plan forms of a ship at each square station are usually defined by half breadths and heights listed in an off-set table. However, for performing the calculations mentioned above, it is better to express the information by using a continuous function of  $b(z)$  rather than an off-set table. For this purpose, a half breadth  $b(z)$  is expressed:

$$b(z) = b_0 + b_1 z + b_2 z^2 \quad (39)$$

In formula (39),  $b_0$ ,  $b_1$ , and  $b_2$  are all constants which are defined by  $x$  and  $z$ . These constants can easily be obtained by applying the formula for three half breadths two layers of water line apart at a certain position of the body plan. For the next two layers from the above ones, other constants are obtained in the same way. By repeating the operation, the body plan form from the keel to the bulwark top can be expressed by a series of formula (39). In Fig. 11, curves show the correct body plan forms and small black points show the results obtained by the formulas. Looking at this figure, we may say that the two body plan forms resemble each other quite closely.

The diagrams in Figs. 5, 6, and 7 show the righting arm  $GZ^*$ , the sinkage  $\zeta^*$ , and the trim  $\theta^*$  versus the

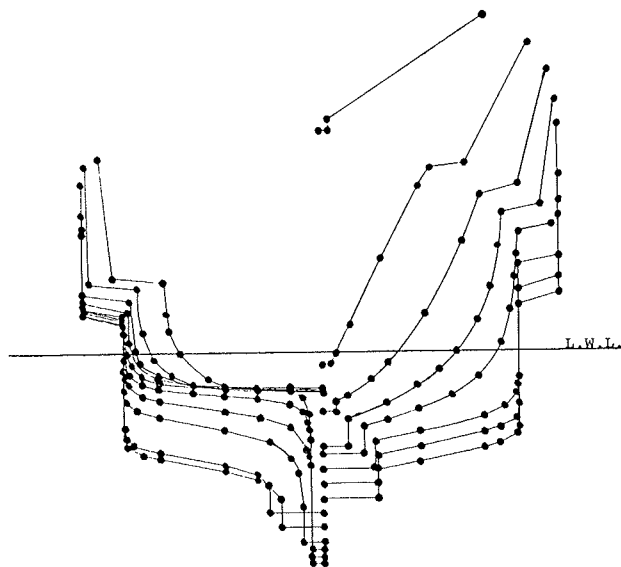


Fig. 11 Mapping of body plan

relative position of the model to the wave. These items have been calculated in this manner in order to compare the experimental results for which the solid lines and dotted lines denote cases in which the effect of wave orbital motion was considered and in which it was neglected, respectively. Fig. 12 shows the  $GZ$  curves of the model in still water, in the wave crest, and in the wave trough

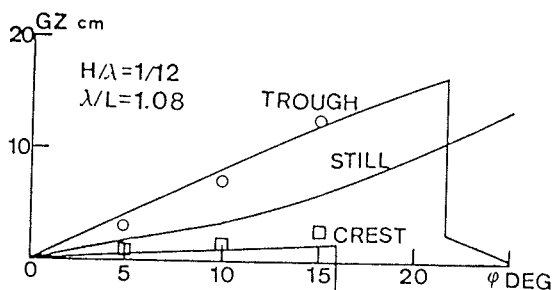


Fig. 12 Righting arm of ship model at wave crest and trough

amidships. Fig. 13 shows the  $\overline{GZ}$  curves of three ships different in body plan in order to consider the effects dependent on the ship hull geometry. From the results of calculations, it has been discovered that the change in the righting arm produced by a wave is remarkably dependent on the ship hull geometry but is not so much affected by the wave orbital motion.

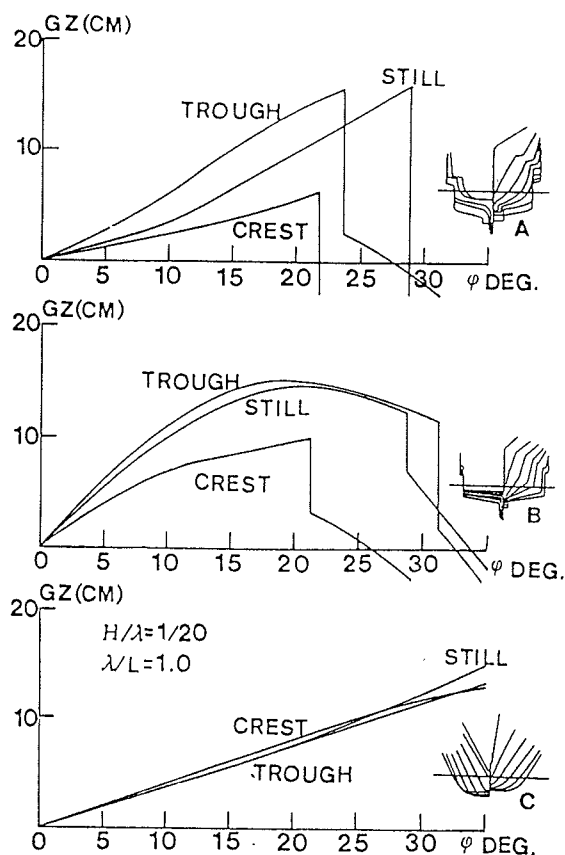


Fig. 13 Righting arms of several ships at wave crest and trough

In order to consider the causes analytically, it is very informative to

formularize  $\Delta I$ ,  $\Delta V$ , and  $\overline{BB^*}$ , as was carried out in the previous section. If the half breadth of the water line  $b(x, z)$  is approximated as:

$$b(x, z) = b(x, 0) + (\partial b / \partial z) z \quad (40)$$

where the terms higher than the second order are omitted,  $\Delta I$ ,  $\Delta V$  and  $\overline{BB^*}$  can be approximated by substituting equation (40) into equation (18) as follows:

$$\Delta I = - \int_{z_A}^{z_F} 2b^2(x) (\partial b / \partial z)_{z=0} z dz \quad (41)$$

$$-\overline{BM}\Delta V = ake^{-k\zeta_s}$$

$$\int_{z_A}^{z_F} \frac{2b^3(x)}{3} \cos k(\xi_0 + x) dx \quad (42)$$

$$\begin{aligned} \overline{VBB^*} = & \int_{z_A}^{z_F} b(x) z dz \\ & + ake^{-k\zeta_s} \int_{z_A}^{z_F} Zb(x) A_0(x) \cos k(\xi_0 + x) dx \end{aligned} \quad (43)$$

where  $Zc$  is obtained from equation (35),  $A(x)$  is equal to the first term of equation (11) and  $Zb(x)$  is the center of buoyancy measured from the origin of the coordinate system. As shown in these equations, the change  $I$  in the inertia of the water plane is caused by the flare of sections  $(\partial b / \partial z)_{z=0}$ , the change  $\Delta V$  in the volume of ship is induced by the wave orbital motion, and the change in the center of buoyancy  $BB^*$  is affected by both the wave orbital motion and the  $Zc$  related to the sinkage and trim of a ship in a wave. Therefore, the things to be emphasized are the effects of  $\Delta I$  and  $\overline{BB^*}$ .

#### 4. CONCLUDING REMARKS

For measuring the righting moment of a ship model heeling and traveling with a wave, an experimental approach was made under the simplest conditions to tow the model in a wave generated by the means of a wave-generating screen board. The experimental results show that the righting moment produced by a wave varies with respect to the relative position of the model to the wave. For example, it can be seen that the righting moment decreases in a wave crest amidships and increases in a wave trough amidships as mentioned in sistent with those of the experiment. For small fishing boats, it is necessary to take into account the sinkage and trim but the effect of wave orbital motion seems to be very small.

It was confirmed experimentally and previous investigations (Ref. 2, 3, 5, and 6).

To analyze the righting moment produced by a wave theoretically, a

theoretical calculation was carried out on the basis of the Froude-Krylov hypothesis. In this calculation, the sinkage and trim of the ship model are balanced in the wave and the effect of wave orbital motion is taken into account.

The calculation rendered results consistent with those of the experiment. For small fishing boats, it is necessary to take into account the sinkage and trim but the effect of wave orbital motion seems to be very small.

It was confirmed experimentally and theoretically that the righting arm of a ship in a following sea can be calculated practically by the method of calculation derived in this paper.

#### ACKNOWLEDGEMENTS

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numerical calculations were done by the aid of the ACOS77/NEAC/800 computer at the Computer Center of Osaka University. The authors would also like to express their sincere thanks to all colleagues who cooperated in this study.

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## Discussion

M.R. Renilson (Univ. of Glasgow, UK)

The authors have developed a very useful technique for studying the effect of a wave on a model in a following sea condition. The wavedozer (as we in the UK call their wavemaking flap) is a very useful facility and, in fact, we have used one in the large CWC at the National Maritime Institute. I have one or two questions to ask about the experimental technique used. Firstly, what was the angle between the water surface and the wavedozer flap. We found after some preliminary experiments that about 15° was best. Too high an angle caused too much turbulence. Secondly, we found that the steepness of the wave train depended on the depth of immersion of the flap. Would Professor Nomoto agree with this? Thirdly, how far from the wavedozer was the model? Obviously the closer to the wavedozer the higher the disturbances in the wave.

For simulation of a ship in a following sea it is important to know the longitudinal force and it is interesting to note that the authors have quite good agreement between theory and experiment. Could Professor Nomoto explain how this was calculated?

Were the side force and yawing moment due to heel measured? If they were I look forward to seeing the results when they become available.

Finally, I would like to ask whether the model was self-propelled and at what

speed were the results quoted in the paper obtained? Perhaps I overlooked it but I could not find any reference to model speed at all.

#### Author's Reply

Our wave generating board is slanted by 45°, just clear of the surface on still water, say, by 1 m/m. The steepness of the wave employed in the present study is about 0.07,  $(h/\lambda)$ . The model is located 4 - 6 m aft of the board. (measured at midship).

In order to obtain the longitudinal force acting on the hull in a following wave, we integrate the longitudinal component of the water pressure on the hull, with and without Smith correction.

We did measure the lateral force and yaw moment. We carried out also PMM test on a wave in the same manner though not included in the present paper.

Speed at the test is Froude No.=0.44 in this case. In general, the speed at the wave-dozer experiment is defined by the wave length to ship length ratio. In our case it is 
$$Fn = \sqrt{\frac{\lambda/L}{0.57 g}}$$

A.Y. Odabasi (The British Ship Research Association, UK)

I would like to ask a few questions to the authors about both their calculation method and their experiments.

(1) The authors state that when the ship is travelling at exactly the same velocity as the wave, all dynamic quantities related to acceleration and frequency are irrelevant in this condition. I would like to know what is the justification of this statement as there is ample experimental evidence to indicate otherwise and computations performed in the following paper indicate the significance of including roll damping effects.

(2) The authors express the change in the metacentric radius by:

$$\Delta BM = BM (\Delta I/I + \Delta V/V)$$

To the knowledge of the present discussor the standard metacentric radius relationship of ship statics, i.e.  $BM = I/V$ , assumes that the intersection between the free surface and ship hull lies on a plane. Since, however, in the following waves this intersection is no longer a planar curve, I would like to know what is actually implied by  $I + \Delta I$ . Does this inertia belong to the projection of the intersection surface?

(3) The authors conclude that the effect of the wave orbital motion is small. The authors also note the differences between the laboratory-generated waves and the infinitesimal wave theory, i.e.  $\lambda/U^2=0.57$  instead of 0.64 with a profile similar to a trochoidal wave. However, since their computations are based on the results of the infinitesimal wave theory, i.e. closed circular orbits with simple exponential relationship, how have the authors reached the above conclusion? Did they carry out any observations on particle trajectories in waves?

(4) From their experiments the authors measure a moment which they call "righting moment". I am, however, not too sure whether this is actually a righting moment since what they measured will be the total moment acting on the model, including steady wave-induced and drag components. Did the authors carry out a special analysis to separate different components?

#### Author's Reply

"Running-with-a-wave" experiment is an idealization of a ship's situation in follow sea. In reality a ship does have a non-zero frequency of encounter, but by this idealization we can make the analysis as well as experiment practicable. We believe this is one of good approaches to the problem. The effect of roll damping is important as Dr. Odabasi mentioned.

The computation of the water-plane inertia should be made along the wave surface, i.e., curved water plane, but the suggestion of the discussor to use the projection will work similarly as the direction cosine is very close to unity.

We did flow survey under water on the dozer-generated wave to know that the wave is not exactly the deep-water sinusoidal

wave, but closely described by the second approximation of the Stokes wave. The difference of flow structures of these two wave systems is, however, not significant from the practical point of view.

There is certainly an appreciable amount of Smith effect as we presented in the paper. Considering the accuracy of the present calculation of the force on the hull in a wave, however, Smith effect may be discarded for the sake of simplicity. One may take account of it if desirable, however.

The last point Dr. Odabasi raised is a question of definition. We define the righting moment to be all the hydrostatic and hydrodynamic moment acting upon the hull about the centre of gravity, but as a function of angle of heel, excluding the damping moment and added mass moment, if any exist.

E. Dahle (University of Fisheries, Norway)

The results of the measurements in the paper confirms that the loss of GZ with a wave crest amidships can be calculated with a reasonable degree of accuracy.

I would, however, like to ask the authors about the GZ-curves of Fig.13, Model A. When comparing with the theoretically calculated curves of Mr. Yamakoshi's paper for this Conference (Model F, Fig.8(b), cross-section on Fig.1(f)),  $H/\lambda=0.1$  the results are quite different, although  $H/\lambda=1/12$  and the cross-section look rather identical.

Is the discrepancy only caused by different wave-profiles, or are there other reasons?

#### Author's Reply

In fact, Model F of Mr. Yamakoshi's paper, Fig.4 of our paper and Fig.10 of Prof. Nakato's paper are all of an identical lines. For some reasons, however, displacement, draught and metacentric height taken at different institutes were more or less different from each other.

The test conditions in our paper and Prof. Nakato's paper are nearly the same except the height of centre of gravity. Its correction is simple, and so we did. The result showed a practically good agreement.

In the case of Mr. Yamakoshi's paper, considerably deeper draught was chosen, which resulted in much slighter loss of transverse stability at the wave crest. This depends much upon the "out-rigged" bulwark that is a particular design feature of small fishing boats of Japan.

Incidentally the corresponding actual ship did capsize and the draught and metacentric height at the accident correspond to that taken in our paper.

*Session V*

## Stability Assessment

*Chairmen*

Dr. Anthony Morrall  
National Maritime Institute  
U.K.

Dr. Tsutomu Tsuchiya  
National Research Institute of Fisheries Engineering  
Japan

# SHIP STABILITY CRITERIA BASED ON TIME-VARYING ROLL RESTORING MOMENTS

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## ABSTRACT

This paper is concerned with the formulation of rational stability criteria for small ships moving in waves. The approach is deterministic and employs ordinary differential equations of motion. Applications of mathematical stability theory proposed in the ship stability literature are discussed. It is argued that such theories should be pursued to the final stage of formulating stability criteria which can be assessed in practice, even if the underlying assumptions and equations of motion are thought to be over simple. In practice, theories at all levels of sophistication will always be needed, since the level employed will be governed by the extent of the data available for a given ship. In this spirit, the paper proposes equations of motion incorporating the rolling moment which acts on the ship by virtue of its instantaneous position and attitude in an incident wave. The effects usually described as direct excitation and roll restoring (including parametric excitation) are thus included. A simple calculation based on the Froude-Kriloff hypothesis is reported. The results are made the basis for applications of three possible stability theories namely, the Mathieu parametric resonance theory, Lyapunov's direct method, and a work/energy balancing method based on existing wind criteria. In each case, the emphasis is on clarifying the form in which stability criteria emerge from the theory, and the external inputs which are required. Numerical results are presented for two ships.

## NOMENCLATURE

a	parameter in Mathieu equation
$b(\phi, t)$	scaled GZ
B	beam
C	damping constant
d	scaled damping constant
$d^{\min}(\phi)$	minimum d for safe rolling to angle $\phi$
$\Delta$	ship displacement in $m^3$
$f_1(t), f_3(t)$	terms in restoring
$F(\phi)$	work done by applied forces during trial half roll
GM	metacentric height
$\overline{GM}$	mean GM
$\delta GM$	amplitude of GM variation
GZ	righting arm
h	waveheight
I	moment of inertia
I	domain of initial conditions
L	ship length
$\lambda$	wave length
N	domains in definition of Lyapunov function
$\phi, \phi_1, \phi_2, \Phi$	roll angles
$\psi$	roll angular velocity
$\phi_v(t)$	vanishing angle at time t
$\phi_d(t)$	restriction on $\phi$ from Lyapunov method
$\rho$	density of water
s	parameter in Mathieu equation
S	safe domain
t	time
$t_1$	time at start of half roll
$\tau$	scaled time
V	Lyapunov function
$\omega$	$\pi/\omega$ = time for half roll
$\omega_e$	encounter (angular) frequency
$W(\phi)$	wind moment

## 1. INTRODUCTION

### The Underlying Approach to Ship Stability

The continued losses of small ships emphasise the urgent, short-term need for improved stability criteria, but recent experience shows how readily dynamical stability work becomes long-term research. Long-term research is needed, but it may be possible for the work to evolve in a way to maximise the interim benefits. Interaction with the practical application is essential for the theoretical development, since only this can ultimately validate theoretical criteria. Therefore, it is valuable at an early stage to consider the whole application process from initial measurements to the formulation of criteria, even if the theory employed is thought to be an oversimplification.

More emphasis needs to be placed on:

- (a) assessment of the accuracy with which one can expect to measure the coefficients in proposed equations of motion, and the implications of accuracy for the consequent stability theories,
- (b) for any proposed application of mathematical stability theory, a careful statement of the form which resulting stability criteria might take,
- (c) the manner in which such criteria could be assessed and validated.

This paper is concerned mainly with (b).

There is an outstanding need to agree on appropriate equation(s) of motion for use in stability work. However, there is no virtue in seeking too exact a description in view of the severe limitations of accuracy in measuring even simple things like the position of the centre of mass [1]. No one seriously suggests that any of the equations can accurately predict a particular ship motion record right up to the moment of capsize. The object of a theoretical model is more modest; it is to identify some "characteristic property" of the ship and environment to which tests can be applied to reveal a predisposition to unacceptably large motions. At present, the "characteristic property" is the statical GZ curve. The tests currently applied have been frequently questioned, following losses of ships which appear to satisfy them (see [2] for example). Many new tests have been proposed, as for instance [3], and several papers within [4]. However, improvements of the tests are limited by the nature of the "characteristic property". We seek a "characteristic property" which carries more information about the dynamical behaviour of a ship in waves.

At the various stages of design and operation, it is necessary to make assessment of stability from very modest data for the ship. This will always be necessary even if a very detailed dynamical theory should become available. We propose the concept of "levels of stability guidance", linked to the nature of available data. This paper aims at a level in which the hull geometry and position of the centre of mass are known. It employs equations of motion with some empirical terms and chosen reference wave conditions.

### Brief Review of Mathematical Stability Theories

The fundamental difficulty in using mathematical stability theory is that the definition of stability is quite different from the notions of the ship case. The theory is mainly concerned with infinitesimal stability of a particular motion record to some kind of perturbation. Given a measure of the tolerable change in the motion (no matter how small), is it possible to find an upper bound on the perturbation which ensures the perturbed motion stays within this tolerance? The upper bound on the perturbation may be very small indeed and this is perfectly acceptable in the theoretical work. Gross simplifications are often made, losing valuable information, because the aim is merely to show that some suitable bound on the perturbation can be found. In contrast, ship stability requires only a kind of large scale boundedness of the motion (and possibly motion derivatives), and criteria must inevitably allow large motions, but not too large. This calls for a very careful application of the theory, without rough simplifications which would build in undue safety factors. The applications of the theory are in two main divisions: linear or linearised stability theory and direct methods (Lyapunov).

Linear stability theory takes as "characteristic property" the set of growth/decay rates of solutions of a linear equation (or system of equations) without excitation. Such equations may arise directly as equations of motion or as the first variational equation for a special solution (e.g. periodic) of a non-linear equation of motion. One takes the appearance of a growing solution (parasitic motion) as the test of instability. It is necessary to believe that the infinitesimal stability of the basic solution considered is crucial for ship stability, or that the sudden destabilizing of a solution as a parameter is/



is slowly varied is a signal of danger (e.g. the influence of the static bias parameter in [5]). These matters can only be resolved by some kind of practical testing; it is easy to construct theoretical examples where linearised theory signals instability but the actual motion is limited by non-linear effects; conversely linear theory fails to predict certain other kinds of instability.

Direct methods avoid computing any particular solution of the equations of motion, but seek properties of a family of solutions simultaneously. This is in closer accord with ship stability notions which regard stability as a property of the ship and environment (i.e. the parameters in the equation of motion), rather than of any one motion record. Lyapunov's direct method, used initially for the study of infinitesimal stability, immediately provides some global bounds on motion and so seems well adapted to the ship case. However, to avoid over pessimistic results, it is necessary to provide efficient Lyapunov functions (see §3), and it has not yet been made clear, in the ship stability literature, exactly how this may be done. The method is presented and developed by Odabasi [6], [7], [8], Kuo and Odabasi [9], and Özkan [10]. Criteria based on this approach are partly in the form of inequalities to be satisfied by ship/environment parameters, and partly expressed in terms of properties of the motion in the phase plane.

## 2. TIME-VARYING ROLL RESTORING MOMENTS

### The Basic Idea

When a ship moves in waves, the shape of the displaced volume of water varies with the position of the wave and the attitude of the ship. That the ordinary GZ calculation should be influenced by this, was first proposed by Grim [11] who suggested that periodic waves should lead to periodic GZ variations. The idea has been taken up by many authors, for instance Paulling [12] and Abicht [13]. The qualitative effects of periodic variations in GZ are indicated by the Mathieu equation (see [14]):

$$\frac{d^2\phi}{d\tau^2} + (a - 2s \cos 2\tau)\phi = 0 \quad (1)$$

where  $\phi$  is "the roll angle",  $\tau$  is a non-dimensional time variable, and the constants  $a, s$  are related to the static GM value and its time-variations respectively. The periodic term is called parametric excitation. The physical effects represented by (1) are described by Paulling [15]. Solutions oscillate with/

with an amplitude which grows or decays according to the values of  $a, s$  in a manner represented by the Mathieu stability diagram (see fig. 5). Possible danger points are represented by the parametric resonances, especially the subharmonic, in which a roll oscillation at the natural roll period builds up in response to arbitrarily small parametric excitation with half this period. The parametric excitation may arise through encounter with following or head seas (suggesting dangerous encounter periods and hence speeds) or via motion in another mode. Although subharmonic build up of roll has been observed in experiments by Paulling [15], the true importance of this instability for ships has not been established. Reservations are that the rate of build up of roll is quite small, so that a master could take action (by slowing down or changing heading) to prevent further build up; irregularity of the waves could prevent the tuning of periods for the time necessary to produce significant build up; and equation (1) excludes the non-linearities of the GZ curve which could also limit the rolling. Variations of restoring in waves may also cause sudden large roll excursions known as "lurching" [12], or in extreme cases "pure loss of stability" (when GM is drastically reduced for long periods as in following seas with nearly the same speed as the ship).

Most attempts to describe rolling with righting arm variations lead to equation (1) by a sequence of simplifying, but not completely consistent assumptions. A proper assessment of the importance of time-varying roll restoring moments requires their derivation as leading terms in equations of motion under a rational approximation scheme. Care is needed to specify precisely which moment is to be computed; it is not sufficient to use a displacement calculation by analogy with the statical case unless it is established that hydrodynamic pressure changes in the incident wave and due to diffraction and body motion are negligible. Our work in this area is not complete and so the presentation is restricted to a careful statement of the arguments and assumptions currently employed, leading to a non-linear variant of (1), to which the various stability theories are applied.

### The Quasi-Static Approximation

In principle, the forces and moments on a ship in waves can be computed only by solving the complete dynamical problem for the fluid motion as well as the ship. Even if viscous effects are ignored (which is thought to be an oversimplification, especially/

especially in roll), the resulting problem is non-linear via its boundary conditions, and involves partial differential equations for the fluid motion coupled to ordinary differential equations for the ship motion. The only known approximation which leads to equations of motion of the ship expressed in terms of ship parameters and the incident wave alone, is the small amplitude linearised theory (also requiring slenderness if there is significant mean speed). This theory is not appropriate here. Since all boundary conditions are 'transferred' to the mean positions of the boundaries, the theory is insensitive to the very factors such as flare and free-board variations which are thought to enhance GZ variations in waves. Furthermore, the only quantities which oscillate are the first order variables, and so the parametric excitation term would be second order. There are many other second order effects and it is inconsistent, without justification, to include only one. A full second order treatment is difficult and, possibly, wrong in principle, for if second order terms are as important as first order this contradicts the basis of the approximation scheme. It seems we must reject the assumption that wave amplitude is small compared to all ship length scales, though it may be reasonable to retain small wave steepness in the incident wave, giving the usual sinusoidal profile and approximately circular particle paths.

The natural order of magnitude assumptions for small ships in waves are that the wave is comparable in length with the ship length and in height with the ship transverse dimensions. This suggests choosing slenderness as a small parameter without making the small amplitude assumption. Of course, the use of inviscid calculations is far more questionable in this case. A rough order of magnitude calculation then suggests that the leading order effects correspond closely with those of the small amplitude, long wave, slender body theory of Newman [16]. In modes with restoring, the leading terms represent a quasi-static balance of excitation and restoring, both calculated by the Froude-Kriloff hypothesis (F-K). Even inertia effects are of lower order. For small ships, the conclusions make good sense in the case of heaving and pitching in following or quartering seas, when the natural frequencies are high compared to encounter frequency and the response is well on the in-phase side of the resonance. For rolling, the conclusion is less satisfactory since near-resonant rolling is common. This suggests that a slenderness calculation should/

should also involve the slenderness parameter in another way which reduces the order of magnitude of both excitation and restoring in roll to agree with the order of magnitude of the inertia. If we had been considering rolling about a fixed axis, this would require a 'near-roundness' assumption. These indications are in accord with the assumptions made by Paulling [12] and suggest the following procedure based on F-K.

Let  $Oxyz$  denote ship-fixed axes with  $Ox$  pointing forward and  $Oz$  vertically upwards when the ship is lying in still water. At an arbitrary attitude in a wave, the roll angle  $\phi$  is defined to be the angle swept out by  $Oz$  in order to bring this axis into a vertical plane by rotation about  $Ox$ . If the roll angle  $\phi$  is specified, the leading order pitch and heave equations amount to the free trim condition. Keeping  $\phi$  fixed, the heave and pitch position is adjusted so that the vertical forces are in balance and the centre of pressure is in the same transverse plane as the centre of gravity. This is carried out by an iteration process involving repeated calculation of the F-K forces and moments as the heave and pitch position is adjusted. Finally, the component about  $Ox$  of the F-K moment is computed. By specifying the encounter frequency and the initial position of the wave, the result can be written as a function of  $\phi$  and time  $t$ , which we term the time-varying roll restoring moment, and denote by  $\rho g \Delta GZ(\phi, t)$ . This includes the F-K approximation to both excitation and restoring.

The F-K approximation involves integrating, over the wetted hull surface, expressions which depend on the hydrodynamic pressure in the incident wave. By the divergence theorem, these surface integrals can be expressed as volume integrals over the displaced volume, defined to be the volume inside the ship and below what would have been the free surface of the incident wave had the ship not been present. The resulting volume integral is the sum of a purely geometric contribution (direct analogy of the ordinary statical calculation) and a contribution from the hydrodynamic pressure changes. Paulling [12] is concerned to estimate the relative importance of these terms since the former are much easier to calculate by hand than the latter. Such considerations are not of much concern in our computer program, but some comparative calculations were carried out for the two terms. We find that the hydrodynamic contribution is negligible as regards both GZ and GM, (the latter in disagreement with Paulling).

The function  $GZ(\phi, t)$  is used in subsequent sections through a roll equation suggested by the above, but which in some contexts will be modified by the addition of other empirical terms. The most complicated form considered is:

$$I\ddot{\phi} + C\dot{\phi} + \rho g \Delta GZ(\phi, t) - W(\phi) = 0 \quad (2)$$

Here  $\dot{\phi} = d\phi/dt$ ,  $\ddot{\phi} = d^2\phi/dt^2$ . In the absence of reliable data, and since added mass and damping are somewhat imprecise concepts in the circumstances of high waves, we take the inertia (which can be thought of as virtual) to be  $I = \rho \Delta (0.4 B)^2$ , where  $B$  = beam, and compute the damping constant  $C$  from the Japanese damping program (see [17]); conditions on  $C$  also emerge from the theories. The wind heeling moment is calculated according to the U.K. recommendation to IMCO [18] giving  $W(0) = 480 AH/\rho g \Delta$ , where  $A$  = lateral area of vessel above water line and  $H$  = wind moment arm between centre of lateral area and half the mean draught. The variation with  $\phi$  is chosen as  $W(\phi) = W(0) \cos^2 \phi$ . The angle  $\phi$  is treated as if it were measured in an inertial frame. This is not exactly so, but the resulting inertial forces are omitted. The equation includes position dependent coupling from heave and pitch through the  $GZ$  calculation, but no velocity or acceleration dependent coupling.

### 3. LYAPUNOV'S DIRECT METHOD

#### A Definition of Practical Ship Stability

Consider a second order differential equation

$$\ddot{\phi} + f(t, \phi, \dot{\phi}) = 0 \quad (3)$$

which could be regarded as a roll equation such as (2) or, by interpreting  $\phi$  as a vector, could represent coupled motion in several modes. Equation (3) is replaced by an equivalent first order system by setting

$$\begin{aligned} \dot{\phi} &= \psi \\ \text{so that } \dot{\psi} &= -f(t, \phi, \psi) \end{aligned} \quad (4)$$

For any particular motion record, the graph of  $(\phi(t), \psi(t))$  is called a trajectory in phase space (or phase plane for a single roll equation). As well as having theoretical attractions, the phase plane is a natural setting for the description of practical stability requirements. The Lyapunov approach presupposes that we have a set of conditions on  $\phi$  and  $\psi$  which, if violated, we take to imply a dangerous condition; these/

these conditions do not come from the Lyapunov theory and must be supplied through other considerations. Examples are:

- (a)  $|\phi| <$  statical downflooding angle,
- (b)  $|\phi| <$  statical vanishing angle.
- (c) a restriction on roll velocity  $\psi$ ,
- (d) a restriction on roll acceleration  $\ddot{\phi}$  which, via the equation (3), can be expressed as conditions on  $t, \phi, \psi$ .

Many refinements are possible:

- (e) replace (a) by a time-dependent condition which, if violated, indicates shipping of green water from wave crests,
- (f) replace (b) by  $|\phi| <$  vanishing angle at time  $t$  of  $GZ(\phi, t)$ .

However, since we do not expect the differential equation to be accurate all the way to capsizes, there is probably little virtue in trying to be very accurate in applying these conditions. Together the conditions define  $S_t$  the safe domain at time  $t$ .

Having chosen  $S_t$  for a given ship/environment, we clearly have a safe condition if all trajectories starting in  $S_t$ , at any time, remain in  $S_t$  for all later times. In application, this condition is very restrictive. If it is relaxed, then safety becomes a matter of initial conditions. We think of the differential equation (3) holding during distinct intervals separated by special events which it does not describe. The question is whether the ship can recover from a special event during the interval when (3) holds. The special events are represented by a domain  $I$  of initial conditions  $(\phi, \psi)$  into which the ship may be thrown from time to time. If we can show that no trajectory starting in  $I$  ever passes out of  $S_t$ , we interpret this as safety. This is akin to the notion of practical stability defined by La Salle and Lefschetz [19]. The domain  $I$  must also be supplied by considerations outside the Lyapunov theory. In [7] Odabasi attempts to estimate  $I$  by a separate theory, also employing Lyapunov functions. He finds a bound on the trajectories of the differential equation which pass through the origin of phase space (there appears to be some inconsistency in this work as regards the inclusion of direct excitation in one part of the calculation but not in another). The estimation or choice of  $I$  and  $S_t$  is somewhat analogous to the choice of windward and leeward extreme roll angles in the wind criteria.

#### Summary of the Method

To outline Lyapunov's direct method, it is convenient first to consider an equation having the null solution,  $\phi \equiv 0$ , so/

so that  $f(t, 0, 0) = 0$  for all  $t$ . For rolling this means following seas. The method is used initially to study the infinitesimal stability of the null solution. The key theorem states that the null solution is infinitesimally stable if there exists a continuously differentiable function  $V(t, \phi, \psi)$ , defined in some neighbourhood  $N_t^*$  of  $(0, 0)$ , such that

$$V(t, \phi, \psi) > 0 \text{ on } N_t^* \text{ except that } V(t, 0, 0) = 0 \quad (5)$$

$$\text{and } \frac{DV}{Dt}(t, \phi, \psi) \leq 0 \text{ on } N_t^* \quad (6)$$

Here  $\frac{DV}{Dt} = \frac{\partial V}{\partial t} + \frac{\partial V}{\partial \phi} \dot{\phi} + \frac{\partial V}{\partial \psi} \dot{\psi}$  which is the total rate of change of  $V$  following a trajectory, and can be evaluated using (4). Some technical uniformity conditions have been omitted from the statement of the theorem. An intuitive understanding of the theorem is quite straightforward. Condition (5) ensures that near  $(0, 0)$ , typically in a neighbourhood  $N_t$  smaller than  $N_t^*$ , the curves given by  $V(t, \phi, \psi) = k$ , constant, are closed loops, nested so that  $k$  increases as  $(\phi, \psi)$  moves out across the loops from  $(0, 0)$ . Condition (6) then ensures that  $V$  cannot increase on any trajectory in  $N_t$  and so a trajectory starting inside a loop labelled by  $k$  remains inside the loop labelled by  $k$  at all later times (the loop can vary in time - typically pulsate in some manner).

Many generalisations are possible. For  $n$  coupled equations the loops are replaced by hypersurfaces in  $2n$  dimensions. If equation (3) does not have the null solution (due to direct excitation), the above theorem is not applicable, but it may be possible to find a function satisfying (5) and (6) in an annular region which may be adequate for the practical purposes. However, many results in the mathematical literature treat the direct excitation as a small perturbation. This may lead to excessive restrictions in the ship case, where direct excitation is a major factor.

#### Lyapunov Functions and Practical Ship Stability

If we have found a Lyapunov function  $V(t, \phi, \psi)$  and associated neighbourhood  $N_t$ , it may be possible to draw immediate conclusions about practical stability. If we can find a value of  $k$  so that the corresponding loop lies in both  $N_t$  and  $S_t$  and encloses  $I$  for all times, then practical stability follows. The Lyapunov theory may indicate conditions on ship/environment parameters in order to/

to find such a loop. An efficient Lyapunov function is one for which  $N_t$  is a large neighbourhood (so that  $S_t$  places the major restrictions on  $k$ ) and does not place undue restrictions on the ship/environment parameters.

#### An Application to Rolling in Following Seas

The section employs the Lyapunov function proposed by Odabasi [7] though not with the generality claimed in that reference. Consider the differential equation

$$\ddot{\phi} + d\dot{\phi} + b(\phi, t) = 0 \quad (7)$$

obtained from equation (2) by setting  $d = C/I$ ,  $b = \rho g \Delta GZ(\phi, t)/I$ , and omitting the wind moment. We consider only following seas, and further represent the restoring term by

$$b(\phi, t) = f_1(t)\phi - f_3(t)\phi^3 \quad (8)$$

where  $f_1$  and  $f_3$  are chosen so that (8) agrees with the numerical computations in respect of the vanishing angle and the area under the curve up to vanishing angle, at each time  $t$ . The Lyapunov function proposed in [7] is

$$\begin{aligned} V(t, \phi, \psi) &= \frac{1}{2}(\psi + d\phi)^2 + \int_0^\phi b(q, t)dq = \\ &= \frac{1}{2}(\psi + d\phi)^2 + \frac{1}{2}f_1(t)\phi^2 - \\ &\quad - \frac{1}{4}f_3(t)\phi^4. \end{aligned} \quad (9)$$

By considering the behaviour of  $V$  along the line  $\psi + d\phi = 0$ , it is seen that the loops can be nested in the required manner only if

$$|\phi| < \phi_v(t) = \{f_1(t)/f_3(t)\}^{\frac{1}{2}} \quad (10)$$

which is the vanishing angle at time  $t$ . This is the first step of finding  $N_t$  but we do not proceed with this now. To keep the discussion as simple as possible, we shall specify  $S_t$  by the same condition (10). Although quite a natural choice, it omits restrictions on velocity and acceleration and pays no attention to downflooding. The second Lyapunov condition (6) places further restrictions on  $N_t$  and on the damping constant  $d$ :

$$d\{f_1(t)\phi^2 - f_3(t)\phi^4\} \geq \frac{1}{2}f_1'(t)\phi^2 - \frac{1}{4}f_3'(t)\phi^4 \quad (11)$$

This condition can be interpreted in two ways. We make the natural assumption that  $f_1$  and  $f_3$  are positive, and consider  $|\phi| < \phi_v(t)$  so that

f/

$f_1(t)\phi^2 - f_3(t)\phi^4 > 0$ , but  $f_1'$  and  $f_3'$  can have either sign.

(i) A condition on damping, necessary if  $N_t$  is to extend up to angle  $\phi$ :

$$d > d^{\min}(\phi) = \frac{1}{4} \max_t \left\{ \frac{2f_1'(t) - f_3'(t)\phi^2}{f_1(t) - f_3(t)\phi^2} \right\} \quad (12)$$

(ii) If  $d$  is specified for a given ship, then we have a restriction on  $\phi$  and hence on  $N_t$ :

$$|\phi| < \phi_d(t) = \left\{ \frac{4df_1(t) - 2f_1'(t)}{4df_3(t) - f_3'(t)} \right\}^{\frac{1}{2}} \quad (13)$$

which must be imposed at all times of the cycle when  $\phi_d(t) < \phi_v(t)$  and the quantity in the brace is positive.

These conditions give an indication of the kinds of restrictions on ship/environment parameters and on initial conditions which the full theory will require.

#### 4. A WORK/ENERGY BALANCE METHOD

Since all the results in this paper are based on the time varying roll restoring arm  $GZ(\phi, t)$ , it is natural to question whether this function, or some particular features of it, can be used directly as the "characteristic property" as discussed in §1. In this section, we propose a way in which this might be done, by analogy with the wind criteria based on Moseley's dynamical stability theory. The argument is plausible but non-rigorous. Consequently, a Rahola type, statistical approach is likely to be necessary as validation. The attraction of the approach, especially as an interim measure before full dynamical criteria are established, is its similarity to the current wind criteria. It is, therefore, easy to understand and apply.

##### Moseley's Dynamical Stability Theory

Moseley's [20] theory, involving the balance of work done by exciting and restoring moments, can be established rigorously for a simple class of roll equations, namely:

$$I\ddot{\phi} + R(\phi) - W(\phi) = 0, \quad (14)$$

where  $R$  and  $W$  denote restoring and wind moments respectively, both depending only on the roll angle  $\phi$ . Such an equation possesses a first integral (the energy equation), and if we consider a half roll beginning with  $\dot{\phi} = 0$  at  $\phi = \phi_1 < 0$ , this takes the form

$$\frac{1}{2}I\dot{\phi}^2 + \int_{\phi_1}^{\phi} R(q) dq - \int_{\phi_1}^{\phi} W(q) dq = 0. \quad (15)$$

The/

The end of this half roll is given by  $\dot{\phi} = 0$  again. This occurs at the angle  $\phi$  satisfying:

$$\int_{\phi_1}^{\phi} R(q) dq = \int_{\phi_1}^{\phi} W(q) dq, \quad (16)$$

(work done by wind balanced by work done by restoring). The wind criterion requires that this angle must be less than another specified angle  $\phi_2$  (such as downflooding), and achieves this by demanding:

$$\int_{\phi_1}^{\phi_2} R(q) dq > \int_{\phi_1}^{\phi_2} W(q) dq, \quad (17)$$

which is the familiar area condition.

##### Generalisation

Suppose, instead of (14), we wish to, consider moments which depend on  $t$  and  $\phi$  as well as  $\dot{\phi}$ . The general equation of this type may be written as

$$\ddot{\phi} + f(t, \phi, \dot{\phi}) = 0. \quad (18)$$

The step analogous to (15) gives formally:

$$\frac{1}{2}\dot{\phi}^2 + \int_{\phi_1}^{\phi} f(t, q, \dot{q}) dq = 0.$$

In this integral, it is implicit that  $t$  and  $\dot{q}$  are related to  $q$  via the actual motion record determined by the differential equation and initial conditions, (we need to specify also  $t_1$ , the time at which the half roll begins). This would normally require numerical evaluation which is precisely what stability theories seek to avoid. As for (16), we require that the half roll ends at  $\phi < \phi_2$ , where  $\phi$  satisfies

$$\int_{\phi_1}^{\phi} f(t, q, \dot{q}) dq = 0, \quad (19)$$

but this statement has the same difficulty. We proceed from here in a way which is plausible but non-rigorous. We replace the unknown motion record by a plausible class of trial functions. Measured half rolls have the appearance of fig. 1 in which the initial conditions  $\phi = 0$ ,  $\dot{\phi} = \dot{\phi}_1$  at  $t = t_1$  are specified, but all other characteristics are then determined by the dynamics. We consider a family of trial curves, parameterised by  $\phi$  and other parameters, including  $t_1$  and  $\omega$  (see fig. 1). A natural choice is

$$\phi = Q_{\phi}(t) \equiv \frac{1}{2}(\phi_1 + \Phi) + \frac{1}{2}(\phi_1 - \Phi)\cos\omega(t-t_1), \quad t_1 < t < t_1 + \pi/\omega. \quad (20)$$

For these trial functions, the integral in (19) can be evaluated to give, say:

$$F(\Phi) = \int_{t_1}^{t_1 + \pi/\omega} f(t, Q_{\phi}(t), \dot{Q}_{\phi}(t)) \dot{Q}_{\phi}(t) dt. \quad (21)$$

The/

The value of  $\Phi$  which makes  $F(\Phi) = 0$  is some kind of approximation to the actual maximum roll angle, and this might be improved by appropriate adjustment of the free parameters in  $Q_\Phi$ . Typically  $F(\Phi)$  increases with  $\Phi$  as the value  $\Phi_2$  is approached and the condition  $F(\Phi) = 0$  for  $\Phi < \Phi_2$  is usually ensured if  $F(\Phi_2) > 0$ . This can be investigated numerically in each case.

The final form of the criterion is as follows. We select a family of trial functions  $Q_\Phi(t)$ , of sufficient generality to include a good approximation of the actual roll record in the family. For each trial function, we evaluate the integral (21) and demand  $F(\Phi_2) > 0$ . This condition is very similar to the familiar wind criterion area condition. Instead of the statical GZ curve, it employs a curve obtained from  $GZ(\Phi, t)$  by relating  $t$  to  $\Phi$  via the trial function (see fig. 2). All other moments (whether depending on  $t$  or  $\Phi$ ) can be included by expressing them as functions of  $\Phi$  via the trial function. Having represented all the moments in this way, the criterion is precisely the familiar area condition (see fig. 3).

## 5. RESULTS AND DISCUSSION

Results in this paper are for two ships (I, II, particulars in table 1). Most are for Ship I which is a 56 m stern trawler, the sister ship of a casualty. The wavelengths considered are approximately  $\lambda = 0.75 L, 1.0 L, 1.5 L$ , where  $L$  = ship length, and the waveheight in each case is taken to be  $h = \lambda / (10 + 0.05\lambda)$  as proposed by Kastner [21]. Fully loaded and half loaded conditions are considered. The paper is mainly concerned with following seas but makes brief reference to  $45^\circ$  quartering sea calculations.

A representation of the function  $GZ(\Phi, t)$  as a surface in three dimensions forms part of fig. 2, for one of the following sea cases. Fig. 4 shows the maximum and minimum envelopes of  $GZ(\Phi, t)$  over a complete wave cycle in a number of following and quartering sea cases. It is to be noted that the minimum GZ envelope would often fail to comply with existing regulations. We feel this cannot be of crucial importance, since the minimum GZ envelope is relevant only for short intervals during the wave cycle; a more careful interpretation of  $GZ(\Phi, t)$  is required.

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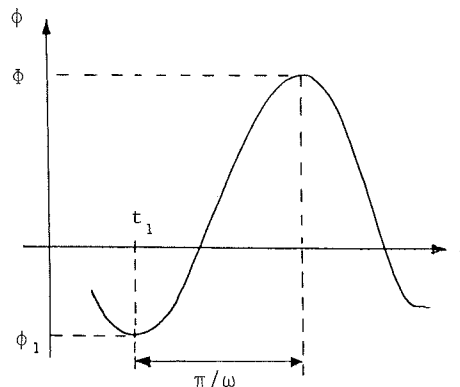


Figure 1: Typical half-roll.

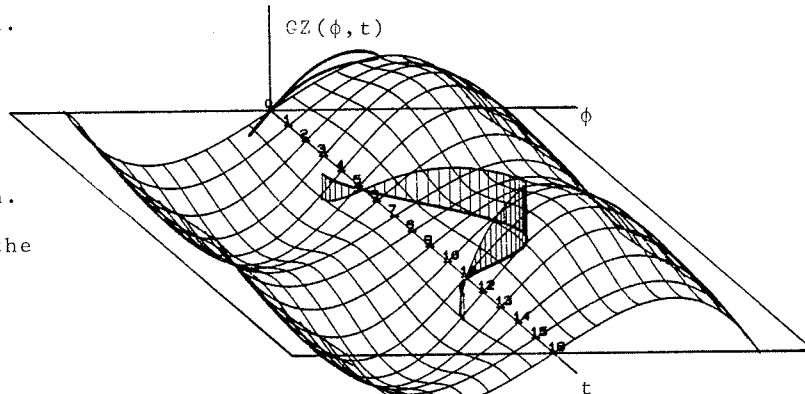


Figure 2: Representation of  $GZ(\Phi, t)$  as a surface. For a trial half roll, a section of the surface is constructed and projected onto the  $(GZ, \Phi)$  plane. The projection is used in the wind criterion area condition.

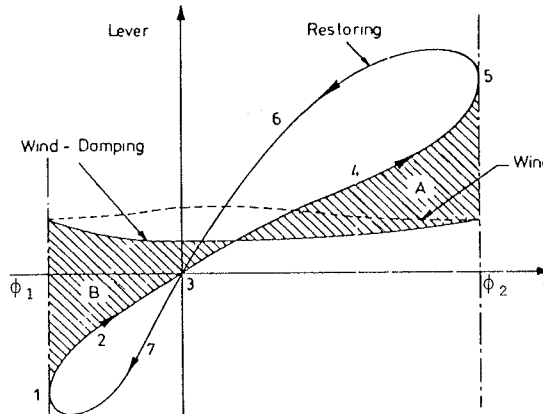


Figure 3: The wind criterion area condition as applied to time-varying GZ via a trial half roll:

$$\text{Net area } A - B > 0.$$

Table 1. Particulars of ships I and II.

Ship Parameter	I		II	
	Fully Loaded	Half Loaded	Fully Loaded	Half Loaded
Length (LBP), m	56.85	56.85	22.09	22.09
(LOA), m	66.07	66.07	25.91	25.91
Breadth, m	12.19	12.19	6.86	6.86
Depth, m	7.77	7.77	3.35	3.35
Mean Draft, m	4.8	4.18	2.56	2.32
Displacement, Tonnes	1850.6	1500.1	189.7	159.3
Block Co-efficient	0.542	0.505	0.476	0.445
Prismatic Coefficient	0.626	0.596	0.635	0.610
Waterplane Area Coefficient	0.818	0.768	0.824	0.775
KG, m	5.42	5.57	3.092	3.216
GM, m	0.7	0.457	0.804	0.725
BM, m	3.23	3.51	2.17	2.27
LCG, m	- 1.42	- 1.11	- 0.70	- 0.753
KB, m	2.89	2.52	1.64	1.49

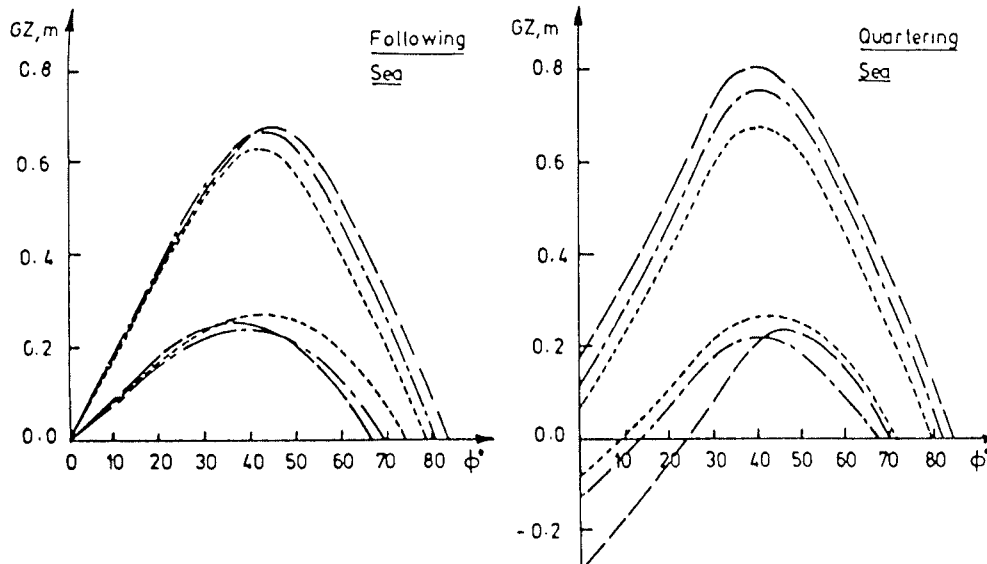


Figure 4: Maximum and minimum GZ envelopes for ship I, fully loaded, in following sea of wavelength  $\lambda = 45$  m (-----), 60 m (-----), and 90 m (———).

To apply the theory of §2 based on the Mathieu equation, the computed  $GZ(\phi, t)$  is approximated by  $(GM - \delta GM \cos \omega_e t)\phi$  where  $\omega_e$  = encounter frequency,  $GM$  is the mean value of  $GM$  and  $2\delta GM$  is the difference of maximum and minimum  $GM$ . By neglecting the damping and wind moment terms in equation (2) and suitably scaling the time variable, we obtain equation (1) with

$$a = \frac{4\rho g \Delta \overline{GM}}{I\omega_e^2} \quad \text{and} \quad s = \frac{2\rho g \Delta \delta GM}{I\omega_e^2}.$$

The standard Mathieu stability chart is shown in fig. 5 with lines superimposed to represent the calculated points  $(a, s)$  for a number of following sea cases and speeds ahead ranging from 0 to 10 knots. The corresponding dangerous ranges of speed are given in table 2. The dangerous ranges of speed are wider in the light loading condition and this is due to the lower value of  $GM$ . The effect of damping is to reduce the unstable regions in the neighbourhood of the  $a$ -axis and so reduce or remove the width of the dangerous speed ranges. The effect of increasing wave height is to shift the plotted lines to the right, so increasing the dangerous speed ranges, though this is a very mild effect; the growth rates of the instability will also be increased.

Since these conclusions are linked so closely to wave conditions and ship handling, the main outcome is advice on dangerous combinations of wavelength, heading, speed and loading condition which could be made available in stability booklets. If this type of instability were subsequently to be considered significant then regulations would have to place minimum restrictions on  $GM$  and maximum restrictions on  $\delta GM$ , in reference wave conditions.

Table 2. Dangerous speed ranges, according to the Mathieu parametric resonance theory, for ship I in following seas.

Loading Condition	Ranges of Speeds, Knots			The combined effect of the three waves
	$\lambda = 45 \text{ m}$	$\lambda = 60 \text{ m}$	$\lambda = 90 \text{ m}$	
Fully Loaded	0.0 - 2.77 & 8.06-8.82	7.47-8.51	6.68-7.72	0.0 - 2.77 & 6.68-8.82
Half Loaded	0.0 - 5.15 & 8.96-10.12	0.0 - 2.86 & 8.3 - 9.97	8.03-9.87	0.0 - 5.15 & 8.03-10.12

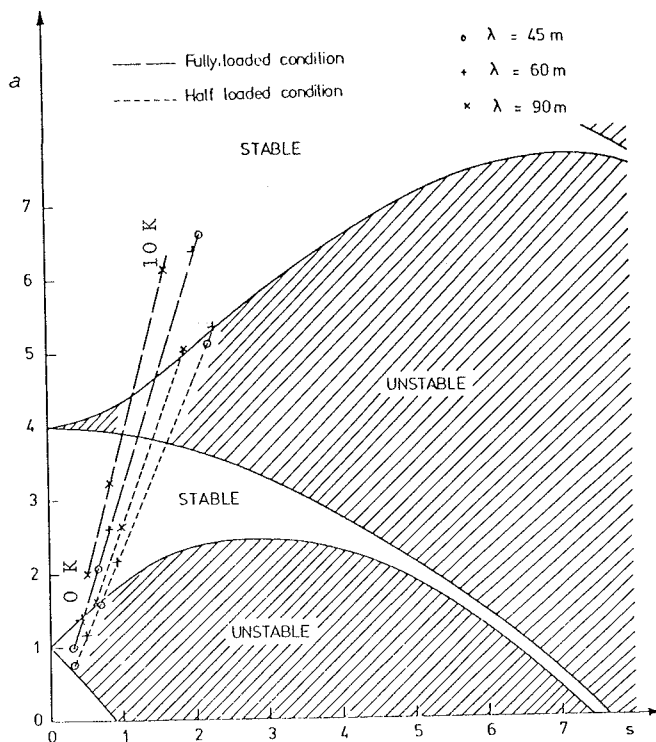


Figure 5: The Mathieu stability diagram. Parametric resonances occur for  $a = 1, 4, 9, \dots$  and  $s$  arbitrarily small;  $a = 1$  gives the subharmonic. Plotted lines show  $(a, s)$  combinations for the cases indicated.



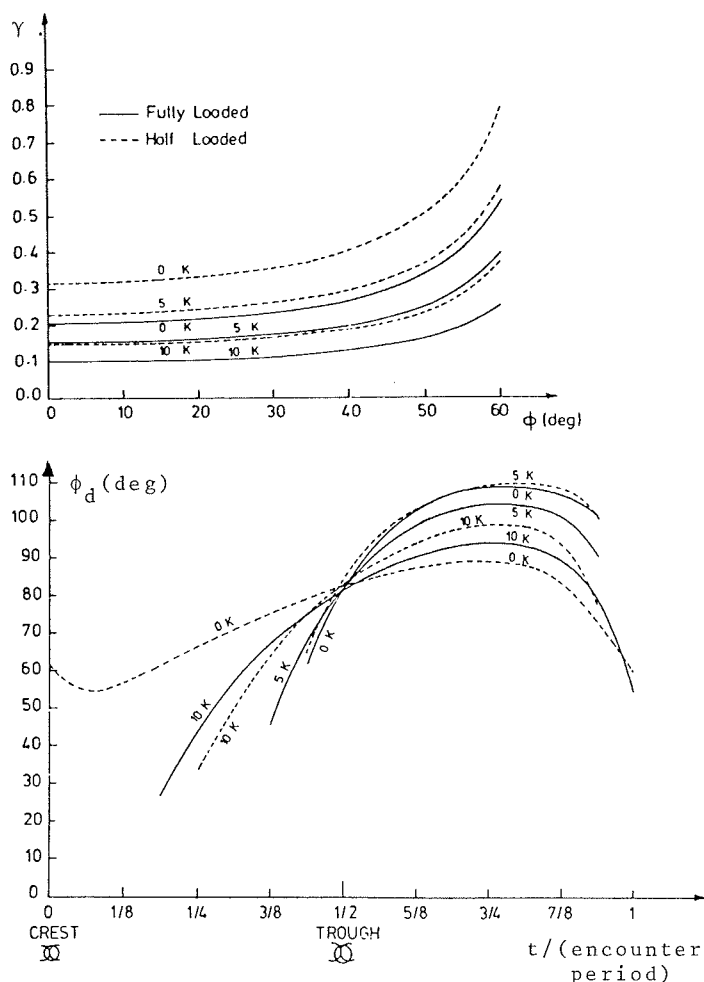


Figure 6: Stability conditions derived from the Lyapunov method applied to ship I, in following seas with  $\lambda = 60$  m.

- (i)  $\gamma = d^{\min}(\phi)/(2 \times \text{natural roll angular frequency})$
- (ii)  $\phi_d(t)$ , (breaks in curve indicate vanishing of  $\phi_d$  at next computation).

To apply the theory of §3 based on Lyapunov's direct method, conditions (12) and (13) were examined. Graphs of  $d^{\min}(\phi)$  and  $\phi_d(t)$  are presented in fig. 6 (ship I) and fig. 7 (ship II). for ship II, the requirement on  $d$  rapidly becomes severe if we wish to allow roll angles of order  $30^\circ$ , in contrast to ship I where the figure is  $60^\circ$ . This feature, which appears to distinguish between the two ships, is attributable to the small vanishing angles for ship II. The graphs of  $\phi_d(t)$  illustrate that, for both ships, the value of  $\phi_d(t)$  can become zero. Put another way, the values of  $d$  derived from the Japanese damping program do not satisfy (12) in some cases. The violation occurs/

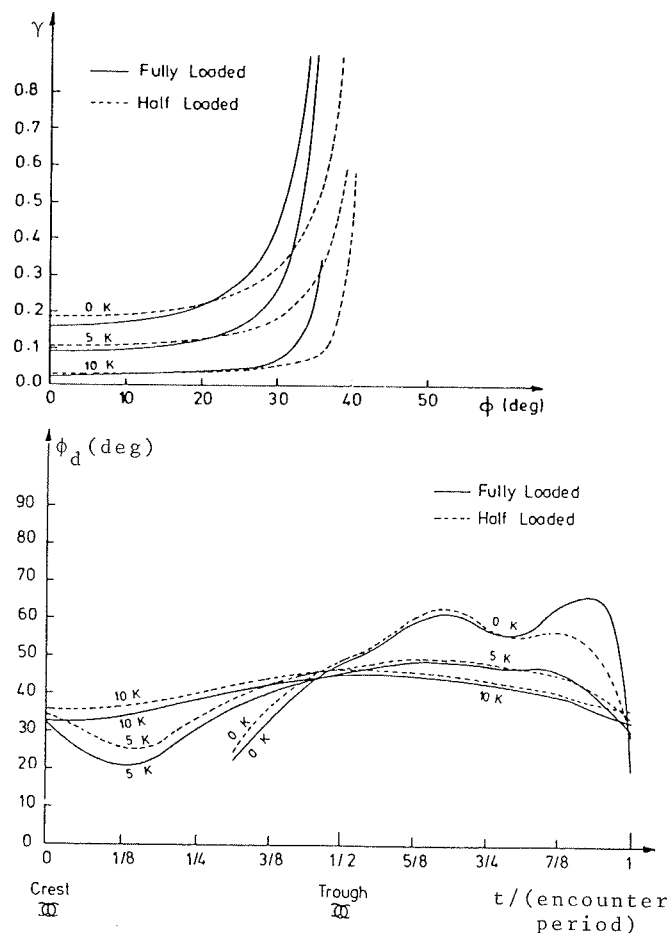


Figure 7: Caption as figure 6, for Ship II with  $\lambda = 24$  m.

occurs when the crest is a short distance forward of the midship section. Both ships are, therefore, sure to fail a criterion based on this method. It follows that either both ships require greater damping for safety or our implementation of the damping program gives an underestimate, or the chosen roll equation is too unrealistic, or the proposed Lyapunov function is inefficient.

The results show that the Lyapunov theory based on the function (9) places very severe requirements of roll damping if we wish to allow rolling of an order usually considered acceptable. The method does draw distinctions between ships. The distinctions are explicable in more elementary terms, but the method has the virtue of quantifying them.

To illustrate the method of §4 based on work/energy balance, we consider the roll equation (2) and the class of trial functions (20).

We suppose that large rolling is most likely to occur at about the encounter frequency (principal resonance) or half that frequency (subharmonic resonance) when one of these coincides with the natural roll frequency (linear theory) and so we choose as test cases  $\omega$  equal to the encounter frequency or half this value; the corresponding ship speeds are easily computed. In the results to be presented, we select  $t_1$  to give the most unfavourable application of the area condition. This appears to be satisfactory in following seas, but for quartering seas is probably unrealistic because the worst cases represent rolling in anti-phase with long waves. The extreme roll angles  $\phi_1$  and  $\phi_2$  have received considerable attention in relation to the ordinary wind criteria. A natural choice for  $\phi_2$  is the statical downflooding angle. There can be no hard and fast rule about  $\phi_1$  which is merely a trial initial windward heel; relevant factors are summarised in [3], and empirical formulae are given in [22] and [23]. We have chosen  $\phi_1 = -10^\circ$ .

We first consider ship I and omit damping altogether. The results for the net area (as defined by fig. 3) are given in table 3, with  $\phi_2 = 40^\circ$ . We observe that in every case the half loaded condition is worse than the fully loaded condition (actually all the half loaded conditions fail). For each loading condition and frequency ratio, the worst wavelength is that closest to the ship length, in agreement with Kastner[22]. We next consider ships I and II and introduce the damping term. For ship I we retain  $\phi_2 = 40^\circ$ , but for ship II, which has a small downflooding angle, we choose  $\phi_2 = 25^\circ$ . Results for net area are given in table 4. The net area is now positive in all cases, indicating that damping has made the difference in some cases between pass and fail, although it is less helpful in the more dangerous half loaded condition. In most of our computations, the principal resonance gives the stronger condition (as for ship I), though the reverse is true of ship II. These trends are in accord with many experimental observations and support the suggestion that significant information on the dynamics of a ship in waves is contained in the function  $GZ(\phi, t)$  employed in the manner suggested.

We /

Table 3. Net area (m.rad) results (as defined in fig. 3), for ship I in following sea, without damping.

Loading	$\omega/\omega_e$		
	$\lambda$	0.5	1.0
Fully Loaded	45	0.0115	-0.0189
	60	0.0193	-0.0083
	90	0.0347	0.0148
Half Loaded	45	-0.0296	-0.0538
	60	-0.0235	-0.0432
	90	-0.0146	-0.0266

Table 4. Net area (m.rad) results (as defined in fig. 3), for ships I and II in following seas, with damping included.

Ship/loading	$\omega/\omega_e$		
	$\lambda$	0.5	1.0
I, Full	45	0.0688	0.0657
	60	0.0653	0.0594
	90	0.0725	0.0677
I, Half	45	0.0028	0.0021
	60	0.0017	0.0013
	90	0.0083	0.0099
II, Full	18	0.0193	0.0276
	24	0.0151	0.0213
	36	0.0144	0.0159
II, Half	18	0.0099	0.0220
	24	0.0079	0.0190
	36	0.0086	0.0157

We have presented results for three distinct stability theories, all employing an equation of motion in which time-varying restoring moments play an important role. It is evident that conditions for stability emerge from the theories in qualitatively different ways. Consequently comparison is difficult although similar trends can be seen, namely that parametric excitation must be limited and sufficient damping ensured. As discussed in §1, the theories do not yield rigorous and precise safety/capsize conditions, but instead produce a /

a 'characteristic property' of the ship/environment to which 'tests' for stability can be applied to reveal a predisposition to large motions. Certain tests are suggested by the theories but in each case some form of experimental or statistical study would be required to formulate and validate the tests. Precisely how this should be done remains as a difficult problem.

The Mathieu theory predicts certain dangerous ranges of speed in following seas. This is an example of linear stability theory in which the 'characteristic property' is the set of growth/decay rates of parasitic motion, and the 'test' for instability is the appearance of a growing parasite. It remains to be determined whether this is truly dangerous, or limited to safe motion by non-linearity or randomness. The Lyapunov theory places conditions on ship/environment parameters (such as damping) in order that motions starting with a prescribed range of initial conditions should respect certain prescribed bounds on roll angle and angular velocity. The 'characteristic property' can be chosen in a number of ways; for instance the set of all initial conditions for which the Lyapunov theory can guarantee that the bounds are respected. Conditions derived from the Lyapunov theory are sufficient but not necessary and can be over-restrictive (as we find in the above application which uses a Lyapunov function proposed in the recent ship literature). If the method is to be used in practice, more efficient Lyapunov functions must be found. The work/energy balance method selects as 'characteristic property' a modified GZ curve which takes into account the variations of  $GZ(\phi, t)$  over a trial half roll. The 'test' proposed is the analogue of the familiar wind criterion area condition.

## 6. CONCLUSIONS

- (i) Applications of mathematical stability theory, proposed in the ship stability literature, should be pursued to the stage of formulating criteria which can be assessed in practice, even if the underlying equations of motion are considered to be oversimple. There will always be a need for simple methods based on very modest data.
- (ii) The quasi-static approximation leading to  $GZ(\phi, t)$  is proposed as a readily computable property containing some information on the dynamic behaviour of a ship in waves.
- (iii) The Mathieu parametric resonance theory using computed  $GZ(\phi, t)$  provides estimates of dangerous speed ranges for a ship in following seas.
- (iv) An application of Lyapunov's direct method produces criteria which are too restrictive to be useful. If this method is to be used in the ship application, it is necessary to specify efficient Lyapunov functions.
- (v) A modification of the wind criterion, to include time variations in  $GZ(\phi, t)$  over a trial half roll, produces trends in accord with experience and could be used as an interim criterion which is readily understood and applied.

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## Discussion

A.Y. Odabasi (The British Ship Research Association, UK)

I have thoroughly enjoyed reading this paper and I express my appreciation to the authors for the clarity of presentation. To a very large extent I fully agree with the opinions expressed by the authors. In particular, I would also like to emphasise the need for a more representative mathematical model which in the view of the present discussor poses the most serious problem.

In the discussion of Lyapunov's direct method, the authors point out a possible inconsistency in the results of Ref. [7] due to the inclusion of direct excitation in one part of the calculation but not in another. To clarify this point I would like to offer the following explanation.

In system stability assessment one has to verify two distinct properties, stability limit, i.e. domain of attraction, and boundedness. By definition, stability limit is a property of the system irrespective of the

purely time-dependent excitation term, and therefore in the determination of the domain of attraction this term does not play any role. The bound of motion on the other hand depends both on the system and on the excitation and hence both have to be present in its determination.

I shall tend to agree with the authors' comment in relation to the application of Lyapunov's direct method, i.e.

(1) The damping estimates based on Ref.[17] are quite likely to be inaccurate especially for large angles and for the ship scale. In fact, in a recent review and modelling study [D1] it has been discovered that due to both the relatively small size of the models employed in the derivation of formulae and the assumed functional relationships for viscous contributions, their success for full-scale prediction is quite doubtful.

(2) As the authors also point out, the equation of motion is also suspect, not only from the hydrodynamics point of view but also from kinematical considerations. For

example, in a recent mathematical modelling study [D2] it has been shown that roll equation contains a gyroscopic term:

$$M r_F \frac{dr_F}{d\phi} \dot{\phi}^2 \text{sign}|\phi|$$

where  $M$  is the ship mass and  $r_F$  is the distance between the instantaneous centre of roll and the co-ordinate origin. It can be seen that this term acts like a damping term, depending on the sign of  $dr_F/d\phi$ .

(3) It is also quite probable that Lyapunov function (9) is inefficient. In fact there is no universality claim attached to this particular expression. Provided there is an agreement on the form of the equation of motion, derivation of a more efficient Lyapunov function does not pose a serious problem since one may ultimately employ numerical means to derive it once and for all.

In relation to the application of Lyapunov's direct method, I would like to ask the following questions to the authors:-

(1) When the method of Ref.[17] is used to obtain a linearised damping coefficient, one needs to assume a roll amplitude. What value (or values) of roll amplitude(s) have been used in the computations?

(2) Since the restoring moment is represented by:

$$f_1(t)\phi - f_3(t)\phi^3$$

how well does this representation agree with the actual computation?

I would also like to raise several points in relation to the work/energy balance method. In principle, the method outlined in the paper differs only slightly from a numerical simulation for a half period since the simulated peak response value, i.e.  $\phi_2$  would satisfy the authors' condition (19) in a numerical approximation. There still remains, however, the question of the choice of  $\phi_1$  from where the integration is started and this choice determines the value of  $\phi_2$ . Assuming that  $\phi_1$  is chosen successfully, the method predicts a bound for the motion, just like Moseley's integral does. The question of stability, however, still remains, since the boundedness, with the exception of uniform asymptotic boundedness, does not imply stability in non-linear systems.

I am sure the authors will also agree that one can always find environmental conditions sufficient to capsize a ship and hence this cannot be raised as a serious objection. From the characteristics given in Table 1, one may conclude that Ship I was lost under some unknown circumstances, and experiments carried out by Beukelman and Versluis[D3] showed that ships similar to Ship II capsized in regular following seas when the maximum roll angle reached the range 24 to 30 degrees. In view of these facts, the results from the application of Lyapunov's direct method may not appear too restrictive, in spite of the

possible deficiencies expressed by the authors and agreed by the present discussor.

Finally, I would like to point out a difference in the basis of comparison of criteria presented in the paper. As a result of the form of equation(7) it is assumed that wave passage is a steady state event leading to possible resonant build-ups, whereas in the work/energy balance method the ship is implicitly assumed to have a wave passage of limited duration, for otherwise  $\phi_1$  should be equal to the negative peak response. If one rewrites equations(7) and (8) for this case, they become:-

$$\dot{\phi} + d\phi + a_1\phi - a_3\phi^3 = [-p_1(t)\phi + p_3(t)\phi^3] [U(t) - U(T^*)] \quad (22)$$

where  $U(t)$  is the Heaviside function,  $a_1$  and  $a_3$  correspond to still water conditions and:

$$p_1(t) = f_1(t) - a_1, \quad p_3(t) = f_3(t) - a_1$$

and  $T^*$  is the time interval,  $T/2$ , in accordance with the work/energy method.

If  $T^*$  is of the order of a period, the right-hand side of (22) may be integrated with respect to time as:-

$$\begin{aligned} & \int_0^\infty [-p_1(t)\phi + p_3(t)\phi^3] [U(t) - U(T^*)] dt \\ &= \int_0^{T^*} (-p_1(t)\phi + p_3(t)\phi^3) dt \\ &\leq \int_0^{T^*} (-p_1(t)\phi_* + p_3(t)\phi_*^3) dt \\ &= -P_1(T^*)\phi_* + P_3(T^*)\phi_*^3 = E_1 \end{aligned} \quad (23)$$

where  $P_1(T^*) = \int_0^{T^*} p_1(t) dt$ ,  $P_3(t) = \int_0^{T^*} p_3(t) dt$ ,

and  $\phi_*$  is the limiting angle to be determined from the stability analysis to be carried out below.

Rewriting the equations in quasi-canonical form as:-

$$\begin{aligned} \dot{\phi} &= \psi - d\phi - E_1 \\ \dot{\psi} &= -a_1\phi + a_3\phi^3 \end{aligned} \quad (24)$$

and defining the Lyapunov function of the unperturbed system (i.e.  $E_1=0$ ) as [D4]:-

$$V = \frac{1}{2} \dot{\phi}^2 + \frac{a_1}{2} \phi^2 - \frac{a_3}{4} \phi^4 + d\phi\dot{\phi} + \frac{1}{2} d\phi^2 \quad (25)$$

substantial derivative of the Lyapunov function along the unperturbed motion (i.e.  $E_1 \neq 0$ ) becomes:

$$\frac{DV}{Dt} = (-a_1\phi + a_3\phi^3) (d\phi - E_1). \quad (26)$$

Since the stability domain of the unperturbed motion is limited by the smaller of the vanishing angle or any other practical limit, for a given system, i.e. mass, damping, etc., the limit of stability is

determined from the solution of:-

$$(-a_1\phi + a_3\phi^3) (d\phi - E_1) = 0 \quad (27)$$

or, since the solution of the first bracket, (other than origin) is,  $\phi = \phi_v$ ,

$$d_1\phi_* + P_1(T^*)\phi_* - P_3(T^*)\phi_*^3 = 0$$

$$\phi_* = \sqrt{[d_1 + P_1(T^*)]/P_3(T^*)} \quad (28)$$

If,  $\phi_*$  is less than vanishing or down-flowing value, the motion is stable.

If  $T^*$  is longer than a period the corresponding state space equations can be written as:

$$\dot{\phi} = \psi - d\phi \quad (29)$$

$$\dot{\psi} = -a_1\phi + a_3\phi^3 + e(t)$$

In accordance with the practical stability definition:

$$\frac{DV}{Dt} = d\phi[e(t) - a_1\phi + a_3\phi^3] + \dot{\phi}e(t) \quad (30)$$

where

$$e(t) = [-p_1(t)\phi + p_3(t)\phi^3][U(t) - U(T^*)]$$

$$\leq -P_1\phi + P_3\phi^3 \quad (31)$$

where  $P_1$  and  $P_3$  correspond to maximum values of  $p_1(t)$  and  $p_3(t)$  in the interval  $(0, T^*)$ .

Substituting (31) in place of  $e(t)$  in (30) one obtains the following equation for the bound of motion (in the positive quadrant of  $\phi$ - $\dot{\phi}$  plane):

for  $\phi > 0$  :  $d\phi[-(a_1+P_1)\phi + a_3+P_3]\phi^3 + \dot{\phi}(-P_1\phi + P_3\phi^3) = 0 \quad (32)$

By setting  $\dot{\phi} = 0$  the maximum amplitude is obtained as:

$$\phi_* = [(a_3+P_3)/(a_1+P_1)]^{1/2} \quad (33)$$

If  $\phi_*$  is less than  $\phi_v$  then the motion is practically stable.

From our computational experience, the result to be obtained by (28) and (33) will be more realistic and yet they are derived without any inconsistency.

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#### Author's Reply

The authors would like to thank Dr. Odabasi for his constructive remarks and hope the following notes will answer most of his questions.

Regarding the value of roll amplitude, one need to assume, to evaluate the damping coefficient by the method of Ref. [17], the amplitude was assumed to be 25° for Ship I and 20° for Ship II. In Section 3, the GZ curve was fitted using a cubic form. We found that this representation gives satisfactory agreement with the actual computations.

As far as the value of  $\phi_1$  is concerned, a value of -10° was assumed in the present calculations. However, we would like to point out to the fact that we have advanced our work since the preparation of this paper and now in the process of evaluating  $\phi_1$  according to the results of simulated rolling motion.

B. Johnson (U.S. Naval Academy, USA)

My question concerns the parameter "b" in equation (7) (for which the typist has used the symbol  $\Delta$  rather than  $\nabla$  to indicate volume of displacement). In a sea-way, both  $I$  and  $\nabla$  vary with time as a result of heaving and the variation of the waterplane area. Is this variation taken into account in the time-varying roll restoring moment method? If so, how is this accomplished and if not so, do you have plans to do this in the future?

#### Author's Reply

Thank you for the question.

The variation in the moment of inertia ( $I$ ), and volume ( $\nabla$ ) are not taken into consideration in the present calculations. For  $I$  and  $\nabla$  to vary with time, a full dynamic approach to the ship stability problems must be adopted where as our present approach is a quasi-static treatment that indirectly incorporates dynamic information. The basis of the solution ensures that the volume is constant while trimming the ship to achieve longitudinal balance. For applications, such as stability of a ship in the following seas, it is justified because heaving and pitching forces are small. The full applicability of this approach is being established.

As regards future plans, it is our intention to incorporate as many of the dynamic effects in a step by step manner while bearing in mind the need, at all times, to provide the users with methods which can be employed in design and/or for regulatory purposes. We hope to report further advances at a future date.

## CONSIDERATION ON THE DANGEROUS SITUATIONS LEADING TO CAPSIZE OF SHIPS IN WAVES

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### ABSTRACT

In this paper, the author describes firstly the results of statistical analysis of casualty records about flooding and capsizing accidents of 448 cases. The situation these accidents occurred are classified into ten modes corresponding to ship conditions and navigation conditions as well as weather conditions so that the typical patterns and the main causes of capsizing accidents can be clarified.

Secondly, these typical patterns of capsizing accidents in rough weather are compared with the model test results carried out in the seakeeping model basin. The model experiments confirm that the main causes of capsizing for small vessels be cargo shift and shipping water on deck accompanied with the poor stability of the ship herself. Casualty records and model experiments show that the successive attack of large waves will strongly endanger the ship to capsize.

Such dangerous situations can be explained by considering the encounter waves in quartering or following seas and ship motions in such sea conditions. The author shows an example of encounter waves measured in following seas and deduces that the ship traveling with an irregular wave could be accompanied with the highest waves in the irregular waves in certain probability and be attacked successively by largest waves for a long time which would be sufficient to result a large amplitude of rolling motion or to catch a large amount of water shipped onto the deck. Considering special characteristics of energy spectra of encounter waves in following or quartering seas, a diagram indicating such dangerous situations for ships is derived, which show the relationship between ship speed, encounter angle and wave period.

### 1. INTRODUCTION

The true picture of the capsizing phenomena of ships can be recognized by careful analysis of casualty records and their statistics as well as by means of observation of capsizing model experiments in rough seas, as Kobilinsky stated [1]. The present author had both opportunities to analyse the casualty records and to participate the model experiments of small fishing boats so that the typical patterns of capsizing phenomena have been classified and identified with the model results.

The most dangerous situations which may lead a ship to capsize are the following three conditions, i.e. beam sea, following sea and quartering sea conditions as same as Kobilinsky's opinion [1]. The model experiments carried out by various researchers can be also classified into these three conditions in which the capsizing patterns are different for each other. According to the author's analysis, the most dangerous situation is to navigate in quartering sea condition when the ship travels with relatively higher speed. By considering the ship motion and shipping water characteristics in waves, such dangerous situation of quartering sea condition will be explained. Ship motion theory in regular waves as well as the superposition theory in irregular waves are efficient to understand and indicate such dangerous situations.

To prevent to encounter such situations, the proposed diagram will be useful for both designer and operator of ships. Model experiments in regular waves can be related to those in irregular waves at such special situations which is thought the most dangerous condition in waves.

## 2. CLASSIFICATION OF CASUALTIES OF SHIPS ON FLOODING AND CAPSIZING

The documents describing the decisions by Japanese Maritime Disaster Inquiry Agency for five years from 1973 to 1977 include 3,212 sea disasters. Among them, 448 cases are flooding or capsizing accidents. The other cases are 717 for collision, 285 for wrecking, 264 for fire, 112 for person accidents due to fall into sea, and 1,026 for machinery or shaft system troubles [2]. By the analysis of processes of each accident, the patterns or causes of flooding and capsizing accidents have been clarified.

### 2.1 Classification of Flooding and Capsizing Accidents

The patterns or causes of flooding and capsizing accidents can be classified into ten modes or situations corresponding to the ship condition and weather condition as well as navigation or operation conditions as shown in Table 1; namely such accidents occurred:

- (1) during navigation in quartering or following seas,
- (2) during navigation in head, bow and beam seas,
- (3) during navigation in calm water,
- (4) during fishing or towing ship at work,
- (5) due to hull break down,
- (6) due to mishandling of piping or valve systems,
- (7) during anchoring in harbours under the storm or typhoon,
- (8) due to misloading of cargo,
- (9) due to icing or drift ice, and
- (10) due to reasons other than (1) - (9).

The numbers of each patterns are shown in Table 1, while the number in parenthesis indicates the cases of capsizing accidents. Considering these numbers in Table 1, the following tendencies can be found in general.

(a) The number of casualties occurred on fishing boats is about one third of the total number (153/448).

(b) The number of casualties occurred in rough weather is almost as many as those in calm weather (225/223).

(c) Casualties in quartering or following seas amount to one fourth of total cases which occurred during navigation in rough weather (51/201). That means the accidents of this kind in rough weathers occurred irrespective to the encounter angle between ship course and wave direction, while the casualties in quartering and following seas occurred in relatively higher percentage on fishing boats than the other kinds of vessels.

(d) Casualties during navigation in head, bow and beam seas occupied one third of all cases (150/448).

(e) Casualties in still water or at work for fishing or towing took place more frequently on cargo vessels than fishing boats.

(f) Casualties due to hull break down occurred at the same rate both on fishing boats and the other vessels.

(g) Casualties due to mishandling of pipe lines or valve systems occurred equally both for fishing boats and the other vessels (24/22).

(h) Casualties caused by icing or drift ice occurred only for fishing boats.

(i) Casualties in rough weather occurred in the most cases when the ships were being under navigation, while casualties in calm weather occurred equally both under navigation and anchoring.

(j) Capsizing casualties occupied about 60 percent of the total cases.

### 2.2 Patterns of Flooding and Capsizing

- (1) Case 1: Flooding and capsizing during navigation in quartering or following seas.

When a ship having poor stability, i.e. of top heavy or over loaded condition, was navigating with relatively higher speed in quartering or following seas of moderate or rough sea states, she was exposed to heavy rolling which sometimes induced cargo shift if lashing was loose, and sea water shipped on deck over the bulwark top at the midship or stern parts. She caught so much water that it flew into the holds or some the other spaces through openings or doors if the tightness or closing was defective. The heeling angle increased to force crews to get off the ship, and then the ship was foundered or capsized. This pattern of casualties can be found equally both for fishing boats and the other vessels, and capsizing accidents occupied about 75 percent.

- (2) Case 2: Flooding and capsizing during navigation in head, bow or beam seas.

The ship was forced to navigate in high waves caused by misjudgement of weather or by unexpected change of weather. The high waves attacked the ship and water shipped on the deck frequently which sometimes broke her worn-out hatch covers and flew into the holds. This led the ship to flooding or cargo shift, finally to capsizing. There are some cases that the fishing nets on deck closed the freeing ports which induced much water on deck, or the liquid-like cargo shifted to one side. This pattern of casualties occurred more frequently on cargo ships than fishing boats. Capsizing accidents are two third of this case.

- (3) Case 3: Flooding and capsizing during navigation in calm water.

This kind of casualty occurred on the ships having poor stability or over-loaded cargo when they navigated in calm water with high speed and occasionally steered too hard. The influence of strong current can be sometimes seen. This happened mostly on cargo vessels and all cases are capsizing accident.

- (4) Case 4: Flooding and capsizing during fishing or towing ship at work.

Almost all cases of this casualty happened on the towing ships when the communication between tug boat and the towed vessel was insufficient so that the tension of towing lines acted transversely on the towing ship.



- (5) Case 5: Flooding and capsizing due to hull break down.

Looseness of aged hull or collision to sea bottom or the other under-water objects produced crack of hull. The flooding and capsizing occurred when watch on bilge water was failed. Almost all cases are flooding accidents.

- (6) Case 6: Flooding and capsizing due to mishandling of piping or valve systems.

Insufficient closing of valves or forgetting to close them, rust or dust, mishandling of handles of valves, etc, were main causes of this kind of casualty. Such cases occurred mostly during anchoring in harbours if watch on bilge water was neglected. This kind of casualty happened equally on fishing boats and cargo vessels, and are mostly flooding accidents.

- (7) Case 7: Flooding and capsizing during anchoring in harbours under storms or typhoons.

Flooding by shipped sea water by high waves developed in the harbour, and collision or contact with the quay wall due to breaking chains caused this casualty. This case occurred equally on fishing boats and cargo vessels, mostly by flooding.

- (8) Case 8: Flooding and capsizing due to misloading of cargo.

One side loading of cargo due to insufficient communication between loader and crew or shift of liquid-like cargo were the main causes of this kind of casualty. This case happened mostly on cargo vessels by capsizing.

- (9) Case 9: Flooding and capsizing by icing or drift ice.

This case happened only by fishing boats. Casualties by icing and by drift ice happened equally. The latter case was a special accident that a group of fishing boats was surrounded by packed ice and forced to capsize. All cases are capsizing accidents.

- (10) Case 10: Flooding and capsizing due to reasons other than cases (1)-(9).

This case happened equally on fishing boats and cargo vessels. Accidents due to design failures and intentional accidents were found. Flooding and capsizing occurred equally.

The casualties in rough weathers and in calm water occurred half and half of the total cases. The causes of casualties in rough weathers are in general very complicated because a lot of factors of ship conditions, weather conditions as well as human factors is combined, while causes of casualties in calm water are rather simple.

### 2.3 Classification of Factors Caused Capsizing Accidents in Rough Weather

The main factors caused capsizing accidents in rough weather have been picked out from the descriptions of each accident which were summarized in Table 2. The following items have been chosen as the main factors of capsize so that the number of each items have been summarized both for fishing boats and cargo vessels, respectively.

The main causes picked out are as follows;

- (1) For loading condition of ships:
  - a) Over-load, b) Top-heavy,
- (2) For the state of cargo:
  - a) Insufficient lashing
  - b) Inferior loading
- (3) For ship construction:
  - a) Open door, b) Inferior hatch cover,
- (4) For ship responses in waves:
  - a) Cargo shift, b) Breaking hull,
  - c) Shipping water, d) Broaching.

From Table 2, it can be found that shipping water on deck is the most superior factor for capsize of fishing boats while cargo shift is for cargo vessels.

### 3. MECHANISMS OF CAPSIZING OBSERVED THROUGH MODEL EXPERIMENTS IN THE BASIN

At the first International Conference on Stability of Ships and Ocean Vehicles, Glasgow, 1975, many papers describing capsizing model experiments carried out at various laboratories were read [3]. After this conference, capsizing model experiments have been carried out successively in various countries, but in larger scaled and more systematical ways [4].

Though the ship types and the methods as well as the purposes of these researches were different for each other, a systematical consideration of the methods and the results leads us to a thorough understanding on the capsizing phenomena of ships in heavy seas, because a visual observation of capsizing phenomena in various situations is possible only through model experiments.

The capsizing experiments could be classified into three categories according to the experimental methods, i.e.

- (1) Experiments in beam sea,
- (2) Experiments in following sea, and
- (3) Experiments in quartering seas.

The present author describes in this chapter the patterns and main causes of capsizing of ship in rough seas found through the experiments of fishing boats carried out in the model basin of SRI [5],[6].

#### 3.1 Capsizing Experiments in Beam Sea

Capsizing in beam sea has been thought possibly to occur when a ship loses her controllability of engine or steering gear by means of some trouble or accidents so that the ship becomes to be tossed about by wind and waves. It has been thought that this situation is the worst one to be checked for keeping safety of ship or stability criteria. The so-called weather criteria used in several countries as Japan are based on the dynamic consideration of rolling ship in beam sea condition. Therefore, a lot of researches in this situation have been carried out; see Tamiya et al. [7], Miller [8], Dudziak [9], Morrall [10],[14], Wright and Marshfield [11], Dahle et al. [12], and Sadakane [13].

In the experiments at SRI, it has been observed on the models running in beam sea

condition that the shipping water occurred at first by attack of steep waves which sometimes seemed as like as breaking waves, then the ship inclined to the weather side so that the shipping water became easier to happen thereafter. The dynamic forces of waves striking the ship beyond her bulwark gave immediate capsizing moment to the ship. When the weight shift occurred on the model, capsizing was significantly accelerated. Therefore, it can be said that the successive attacks of steep waves after the first heavy strike of breaking waves, steady heeling induced by shipping water on deck and the weight shift are the most significant factors for capsizing phenomena in beam sea condition. Weight shift can occur to both sides equivalently, depending on the occasional ship responses such as lateral accelerations acting on the weight. Initial heeling of ship increased the danger of capsizing. Capsizing by means of the synchronous rolling motions happened scarcely while the weight shift could happen easier in this situation.

The examples to represent the response characteristics of lateral accelerations and relative motions of a small fishing boat in beam sea and in quartering sea are shown in Fig.1 and 2 [6]. In the Fig.2 it is observed that the shipping water may occur easier at the weather side than the lee side, provided the weight shift does not occur.

### 3.2 Capsizing Experiment in Following Sea

Three kinds of capsizing phenomena were observed by the experiments in the following sea condition; i.e.

- (a) Capsizing caused by the pure loss of stability when the ship rides on a wave crest: In this case, the crest of wave reached to the bulwark top so that the water flew onto the deck remarkably when the ship speed was almost the same as the wave propagation velocity.
- (b) Capsizing caused by the parametric excitations of rolling motion when the encounter frequency of ship to waves coincided with the natural frequency of rolling motion itself: The restoring moment on the ship fluctuated periodically so that the so-called Mathew's equation of motion can be applied to consider this unstable motion. The ship repeated large angle of rolling successively exerted on the wave crest and up-righted in the wave trough. The capsizing did not occur in such cases that the stability of ship was good enough.
- (c) Capsizing caused by the so-called broaching tendency occurred when the ship was accelerated on the wave slope if the ship speed was approximately the same as the wave propagation speed. In this case, the course of ship was forced to deviate abeam side so that the ship occasionally inclined heavily. This case happened in the last stage of the experiment in the following waves when the model ship was steered to change her course. It seemed that the rudder effect excited the heeling moment on the model.

Miller reported the experiments in the following waves which showed capsizing caused by the pure loss of stability [8]. The stability fluctuations in the following waves were measured by various researchers, for example, Upahl [15].

### 3.3 Capsizing Experiments in Quartering Sea

Through the experiments in quartering waves it has been observed that capsizing happened under combined action of many factors concerning ship responses in waves. In this situation the model was overtaken by one or two specially steep waves and then caught much water on deck which resulted a large heeling of ship. After then the model was capsized when the successive waves were large enough. It was estimated that the large resonant rolling motions and large relative motions of wave surface on the side of ship hull occurred in the quartering waves as shown in Fig.1 and 2. The weight shift occurred occasionally by the action of lateral accelerations on the weight placed on deck [6].

For the ship having poorer stability, i.e. the smaller GM value, the resonant rolling motion occurred in the quartering waves when the waves were steep and short, because the encounter period will approach to the natural rolling period of such vessels. Besides, the relative wave elevations on the ship sides became large in the quartering waves at the midship part where the protected freeboard, i.e. the freeboard plus bulwark, is lowest so that the water can flow onto the deck easily.

Therefore, it can be said that the worst factors both in beam and following sea conditions occur in quartering sea simultaneously when the ship navigates in steep and short waves with high speed.

As described in chapter 2, the main reasons of capsizing of small vessels are weight shift and shipping water on deck when the stability of the ship itself is insufficient. The same tendencies have been observed through the model experiment as mentioned above. Namely, capsizing occurred in higher percentage when both weight shift and shipping water on deck happened simultaneously. If the weight shift occurred after the enough accumulation of shipped water on deck, capsizing occurred immediately, while the capsizing was advanced gradually if the weight shift occurred at first and the shipping water increased thereafter by means of successive attacks of waves on the heeled ship [6].

## 4. CONSIDERATION ON DANGEROUS SITUATIONS IN QUARTERING SEAS

Through the model experiments in oblique waves, the danger of navigation in quartering seas was clearly recognized. This situation will be explained by considering the encounter wave characteristics of quartering or following seas.

It is well known that a spike-like peak

will appear on the spectrum when an irregular wave spectrum,  $S_W(\omega_0)$ , is transformed into the encounter wave spectrum,  $S_W(\omega_e)$ , by means of the following relation, i.e.

$$S_W(\omega_e) = S_W(\omega_0) \cdot \frac{d\omega_0}{d\omega_e} \quad (1)$$

$$= S_W(\omega_0) \cdot \frac{1}{|1 - 2\omega_0 V \cos \chi / g|} \quad (2)$$

$$\text{where } \omega_e = \omega_0 (1 - \omega_0 V \cos \chi / g) \quad (3)$$

$$\text{and } \frac{d\omega_e}{d\omega_0} = 1 - 2\omega_0 V \cos \chi / g \quad (4)$$

In these equations,  $V$  is the ship speed,  $\chi$  is the encounter angle of ship course to wave direction ( $\chi=0$  is the following wave) and  $g$  is the gravity acceleration.

If  $\omega_0$ ,  $V$  and  $\chi$  satisfy the following relation, i.e.

$$1 - 2\omega_0 V \cos \chi / g = 0 \quad (5)$$

$$\text{or } \omega_0 V \cos \chi / g = \frac{1}{2},$$

then the encounter wave spectrum  $S_W(\omega_e)$  will tend to infinity at the critical encounter frequency  $\omega_{ec}$ .

The critical encounter frequency,  $\omega_{ec}$ , at which the spike-like form of the transformed spectrum appears is related to the critical wave frequency,  $\omega_{0c}$ , that is  $\omega_0$  satisfying the equation (5) by the following relation, i.e.

$$\omega_{ec} = \omega_{0c} / 2 \quad (6)$$

$$\text{or } \omega_{ec} V \cos \chi / g = \frac{1}{4} \quad (7)$$

These relationship between both spectra  $S_W(\omega_0)$  and  $S_W(\omega_e)$  is represented schematically by Fig.3. When  $\omega_{0c}$  is nearly equal to the dominant wave frequency or the frequency of the peak of the spectrum, the transformed spectrum of encounter waves will have a narrow and sharp form with the high value. In such a case, the encounter waves will become similar form as a regular wave train having a constant encounter period while the wave height can be either high or low occasionally.

To examine this fact, the wave elevations were measured by means of travelling wave probe attached on the carriage in the model towing tank in which an irregular wave train was generated. Fig.4 shows the record of wave elevations measured by the fixed wave probe at the midway of the tank, and Fig.5 shows its energy spectrum analysed. The encounter wave spectrum estimated by the equation (1) from the spectrum shown in Fig.5 takes such form as that shown in Fig.6 when the wave probe travels with the critical velocity,  $V_c$ , which has the relationship with the frequency of the peak of the spectrum,  $\omega_p$ , as shown by the equation (5), i.e.

$$\omega_p V_c \cos \chi / g = \frac{1}{2} \quad (8)$$

$$\text{or } V_c \cos \chi = \frac{1}{2} \cdot \frac{g T_p}{2\pi} = \frac{1}{2} c^* = c_g^* \quad (9)$$

where  $c_g^*$  is the group velocity of the wave having its frequency of  $\omega_p$ .

The record of wave elevations measured by the travelling wave probe is shown in Fig.7, that is quite similar as a regular wave train. Of course, the other records of such waves have been obtained through repeated runs with the same velocity but different phases of start of carriage in the same irregular wave. The wave heights of encounter waves seem to depend on the phase. Fig.7 shows the examples of highest wave among measured records. By comparing this wave with that shown in Fig.4, it is easily recognized that the wave height of this encounter wave train is larger than the highest wave elevation in the irregular waves measured at the fixed point and this high wave elevation appears repeatedly like as a regular wave train.

This fact means that the ship travelling with the irregular wave, i.e. running in following or quartering seas, may be accompanied with the highest wave expected in this irregular wave, occasionally for a long time according to the probability of occurrence of wave heights in the irregular wave so that the ship will be attacked by such high waves successively which is sufficient to result a large amplitude of rolling motions or to catch a large amount of shipped water. The most dangerous situation is that when the natural period of rolling or the other period of responses dangerous for capsizing, for example, the period at which the rolling motion becomes unstable, coincides with the critical encounter period  $T_{ec} = 2\pi / \omega_{ec}$  of the equation (7).

Fig.8 shows the relationship between  $\omega_{0c}$ ,  $\omega_{ec}$ ,  $V_c$  and  $\chi$  of the equations (6), (7) and (8). By means of this figure, the dangerous situations of a ship can be related to the navigation and wave conditions. For example, if the modal period of the sea is 4 seconds and the course of ship is 60 degrees quartering sea, the dangerous ship speed is 12 knots as shown by the circle mark in this figure. The encounter period of waves which coincides with the critical period is 8 seconds. Therefore, the danger increases very much if the natural rolling period of the ship is 8 seconds.

Recently the danger induced by a large wave group attracts attention in the viewpoint of preventing sea disasters in waves. In the author's opinion, the ship running in following or quartering seas can encounter to a large wave group virtually even if the irregular wave has no remarkable groupiness. It means the groupiness should be considered in the phase of encounter wave elevations.

Kastner [16] has made a comprehensive study on capsizing of ships by considering fluctuations of stability curve in following seas. He treated with the transformation of spectra of stability fluctuations in irregular waves.

The spectrum of rolling in such a special condition that the encounter waves are similar as regular waves and the natural rolling period is nearly equal to the encoun-

ter wave period has such form as shown in Fig.9. In this case, the rolling motion resonant to the highest waves in the irregular waves should be considered for judging the stability.

There is an example of ship motion records measured on the fishing boat model running in quartering or following wave condition on the lake [5], see Fig.10. In this figure, the rolling and pitching motions are shown till model capsizing. Pitching motion shows responses like as that in regular wave at the midpoint of the record, while the heeling angle of the model increased very much which could be considered almost as capsizing in spite of moderate rolling amplitudes. This means that the model was attacked successively by a large wave elevation in following or quartering waves so that the pitching motion became large as well the shipping water on deck. This should be the main cause of capsizing of the model though some length of remainder run continued for several seconds thereafter.

## 5. CONCLUSION

The results of casualty record analysis and model experiments have been summarized, in order to compare the reasons of capsizing occurred for actual ships and models in wave, especially in the viewpoint of the effects of encounter wave directions.

(1) According to the casualty records, capsizing or flooding of ships occurred even in moderate sea states by a successive attack of high waves when the ship was navigating in quartering sea.

(2) The main reasons of capsizing are weight shift and shipping water on deck. These facts could be observed clearly also in the model experiments in oblique waves, i.e. the model was occasionally attacked by large waves. If the wave elevations at the model position repeated large amplitudes, the model was exposed to large responses which related to shipping water or capsizing.

(3) Such phenomena can be explained by considering encounter wave characteristics of the ship in quartering waves which has a singular feature at a certain wave and navigation condition.

(4) In this singular condition, the encounter waves act virtually as regular waves which have the largest wave height in the sea state concerned. In such cases, the ship responses in irregular waves can be closely related to those in regular waves.

(5) The diagram to find this singular condition has been derived by considering the encounter wave spectrum transformed from the wave spectrum itself. This should be useful for ship operators to find the dangerous situations according to ship condition, sea condition as well as navigation condition. In considering stability criteria, the ship responses in quartering waves which sometimes is the worst condition for capsizing should be taken into account.

(6) Initial heeling and sudden steering in waves endanger the ship remarkably to cap-

sizing. The possibility of weight shift and the effects of shipped water could not be ignored as well the stability reduction due to wave.

## ACKNOWLEDGEMENTS

The statistical analysis of casualty records was done by the research committee organized in the Japan Association for Preventing Marine Accidents, the purpose of which was to make a comprehensive guidance or manual for prevention of sea disasters on small vessels. The investigation on capsizing and flooding accidents were done by the No.3 sub-committee, the chairman of which was Prof. S. Tamiya, during 1978-1979 [2]. The experimental results in this paper have been referred from mainly [6] and partly [5]. The measurement and analysis of encounter waves was done by co-operation of Mr. Saruta, SRI.

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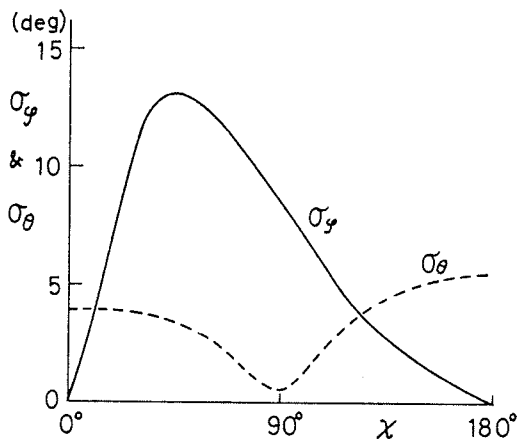
Table 1 Classification of Flooding and Capsizing Accidents of Ships

Case	Causes or Conditions of Casualties	Number of Casualties		
		Fishing Boats	Cargo Vessels	Total
1	Navigating in Quartering or Following Seas	25 (19)*	26 (19)	51 (38)
2	Navigating in Head, Bow and Beam Seas	49 (37)	101 (64)	150 (101)
3	Navigating in Calm Water	5 ( 5)	15 (15)	20 (20)
4	Working as Fishing or Towing Ship	5 ( 5)	25 (20)	30 (25)
5	Hull Break Down	17 ( 1)	37 ( 5)	54 ( 6)
6	Mishandling of Piping or Valve System	24 ( 3)	22 ( 5)	46 ( 8)
7	Anchoring in Harbour When Storm or Typhoon	12 ( 3)	12 ( 4)	24 ( 7)
8	Misloading of Cargo	3 ( 3)	47 (45)	50 (48)
9	Icing or Drift Ice	8 ( 8)	0 ( 0)	8 ( 8)
10	Reasons Other Than 1-10	5 ( 2)	10 ( 6)	15 ( 8)
	TOTAL	153 (86)	295 (183)	448 (269)

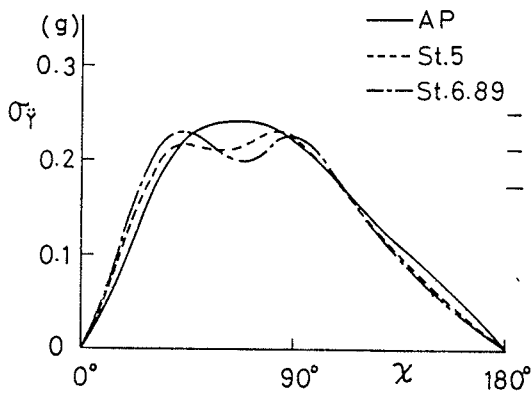
\*Note: Number in parenthesis indicates the number of capsizing accidents.

Table 2 Classification of Main Factors Caused on Capsize

Factors Caused on Capsize	Fishing Boat	Cargo Vessel	Total
Over Loaded	9	8	17
Top-Heavy	13	11	24
Insufficient Lashing	1	20	21
Inferior Loading	7	17	24
Cargo Shift	13	41	54
Open Door	9	22	31
Inferior Hatch Cover	2	18	20
Hull Break Down	2	1	3
Shipping Water on Deck	29	26	55
Broaching	1	3	4
TOTAL	52	87	139

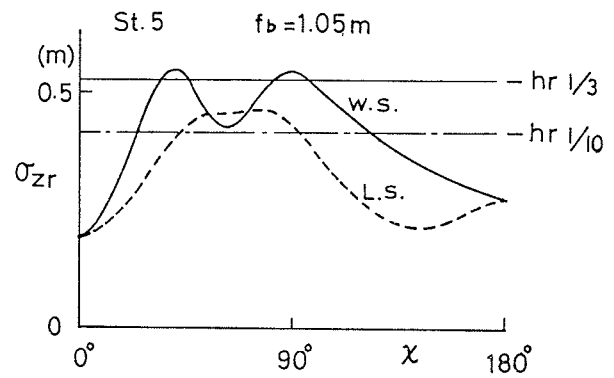


a) Rolling and Pitching

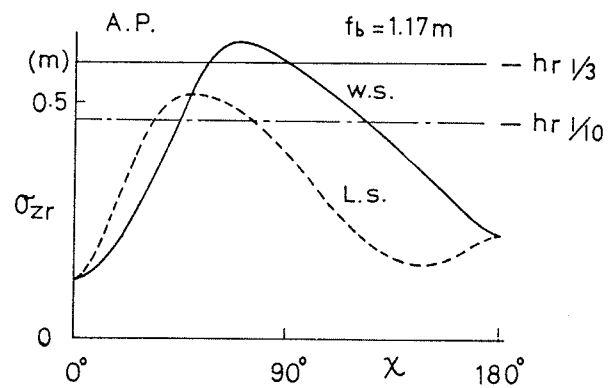


b) Lateral Accelerations

Fig. 1 Ship Responses in Irregular Waves



a) Midship



b) Stern (A.P.)

Fig. 2 Relative Wave Elevations in Irregular Waves

Fig. 3 Transformation of Irregular Wave Spectrum into the Encounter Wave Frequency,  $\omega_e$

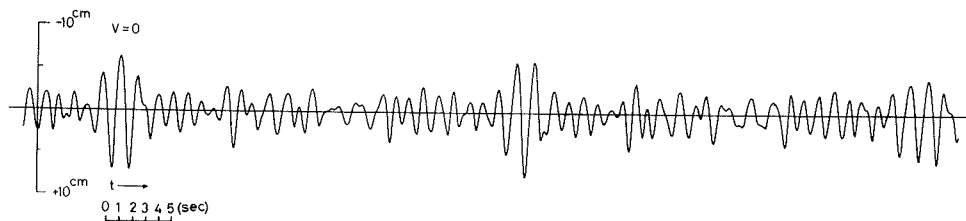
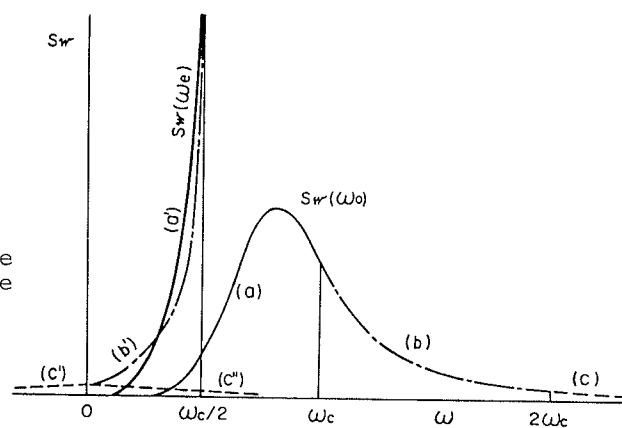


Fig. 4 Record of Irregular Wave Elevations Measured at the Fixed Point in Basin

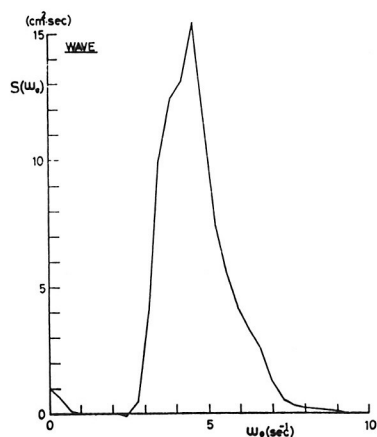


Fig. 5 Spectrum of Irregular Wave Measured at the Fixed Point

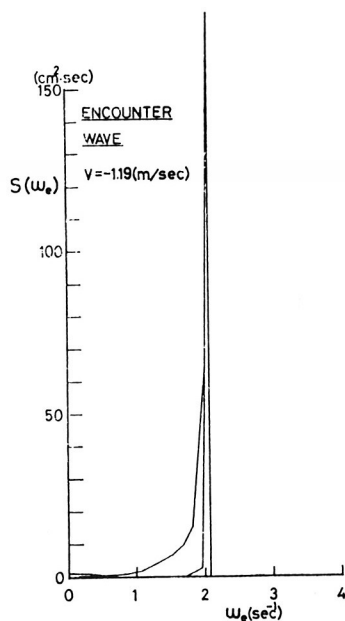


Fig. 6 Spectrum of Encounter Waves Transformed From Fig. 5

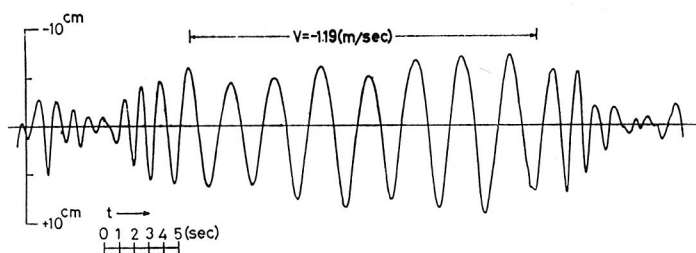


Fig. 7 Record of Encounter Wave Elevations at Critical Velocity

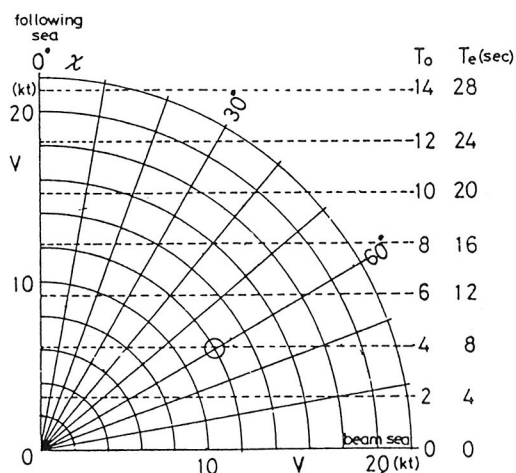


Fig. 8 Diagram Indicating Dangerous Situations in Quartering Sea

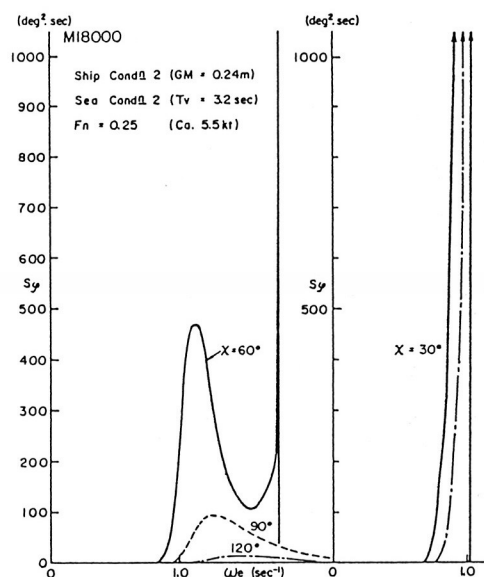


Fig. 9 Spectra of Rolling Motion in Irregular Waves

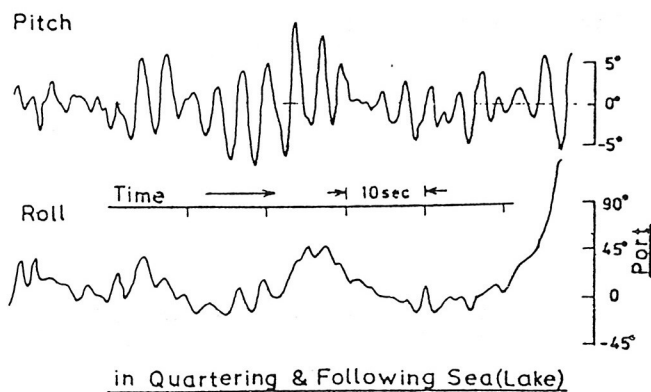


Fig. 10 Capsizing Time History of Model in the Lake [5]

## Discussion

N. Toki (Nagasaki Experimental Tank, MHI, Japan)

The authors should be congratulated for the presentation of this interesting study. This paper reminds me of our experiments for SWATH design. In the planning stage, I expected it would be most dangerous case when encounter period to following wave became close to the long natural period of rolling. On the contrary, however, ship motion and relative motion amplitudes became largest when encounter frequency became close to zero. Actually it was the only case when we felt the risk of deck wetness during the experiments. This phenomenon is closely related to the "broaching to" phenomenon, I suppose.

As being discussed by a few papers in this symposium, "broaching to" is observed in following seas with the wave length twice as long as the ship length, and when encounter frequency is close to zero. Having this fact in mind, I made a short calculation result of which is shown in Fig. A. Using the formulae (1) through (4) of this paper and ISSC spectrum as an incident wave spectrum, the energy spectrum of encounter wave is calculated in the case of  $V_s = 25$  knots, encounter angle = 15 degrees and  $T_v = 5.4$  seconds. As shown in Fig. A, rather sharp peak of encounter wave spectrum is observed where encounter frequency is equal to zero. If the length of dominant component wave must be twice as long as ship length of cause "broaching to" phenomenon, ship length must be of the order of 40 meters. Then, 25 knots correspond to  $Fn = 0.6494$ . This operating condition is considered quite usual in the cases of fishing boats and high speed craft. Have the author observed any examples of casualty corresponding to this condition? Any comments by the author would be highly appreciated.

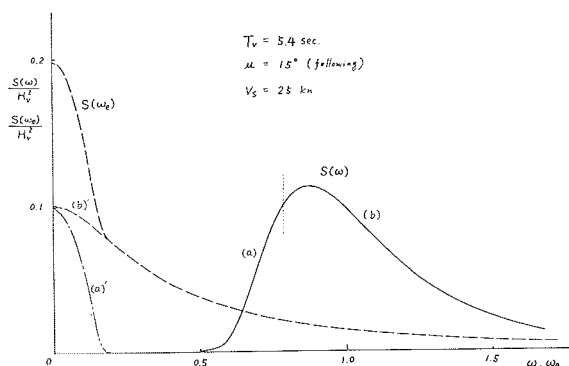


Fig. A An Example of Calculated Encounter Wave Spectrum

S. Kastner (College of Engineering Bremen, FRG)

Dr. Takaishi gives us an account on results of classifying casualty statistics for the 5-year period 1973/1979. Is the data base worldwide? I wonder if the updated statistics for the next five year period will give similar baffling results.

Based on free running model experiments, Dr. Takaishi gives us a thorough description of the capsizing mechanism. I have been particularly interested in his treatment of the different headings ship to waves. I agree with Dr. Takaishi, that many ships will find the worst conditions in quartering seas. This is the heading with the most narrow encounter wave spectrum, i.e. the most regular parametric excitation, but superimposed by external forces. Let me ask you: 1. Can you give some indication, if your models, apt to capsize in following and in quartering seas, could survive in beam seas at the otherwise same ship conditions? Changing the heading is the only quick way for ship operators. 2. Did you test with a higher freeboard? Obviously, the capsizing is closely related with wave crests leaping over the deck side.

### Author's Reply

1. I agree with Mr. Toki's opinion that the ship is endangered to broaching-to when the encounter frequency approaches zero. My paper deals with the situation where the ship speed is nearly equal to the group velocity of wave. Of my opinion, there are two pit-falls for the ship navigating in following or quartering seas, one is phase velocity at which the ship will be accelerated by the wave front to broaching-to, the other is group velocity at which the ship will be attacked by a large wave group and endangered to large rolling motions or shipping water. Fig.8 in this paper is useful to find the latter condition. In our experiment, we also observed broaching-to tendency on the model running on the wave front.

2. I thank Prof. Kastner for his detailed discussions.

1) The statistics is only for Japanese vessels, the disaster of which are described in the documents in these five years from 1973 to 1977. Of course, the statistics of casualty records for the successive years will be very much meaningful but we have not performed yet.

2) As to the dangerous situation of quartering or following seas, I agree with Prof. Kastner's opinion which is presented in his paper for this Conference, at the last paragraph 12.



My opinion is that the danger in quartering seas may occur rather in the moderate sea state whereas the danger in beam sea condition will arise mostly when the sea state is severe, as I explained in my paper.

3) In changing the heading from quartering sea to beam sea at our experiment in the basin the model(s) survived always, while

the changing of heading from following sea to beam sea without decreasing ship speed caused sometimes capsizing supposedly by the effect of rudder actions.

4) We tested only low freeboard cases because the small fishing vessels have almost always low freeboard. Therefore, the shipping water occurred easier in steep waves.

## SIMULATION AND ASSESSMENT OF ROLL MOTION STABILITY

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### ABSTRACT

Stability assessment reaches beyond the common static considerations so far only for ships, which experienced stability problems, such as fishing vessels or coasters, i. e. usually small ships, or for some particular design for the sake of research.

However, calculation of expected large roll motion and capsizing are advisable for all ships which show some peculiarities after screening standard static stability characteristics, and for any newly designed ship, on which little knowledge and experience exists, such as barge carriers or wide shallow vessels.

Stages on a procedure of judging ship stability and for developing capsizing criteria based on roll motion studies are discussed, to be applied to any sea-going ship.

A number of steps for assessing roll motion stability is suggested, with its centerpart being the modelling of the roll motion dynamics by a computer simulation, where a step by step procedure allows for the solution of a set of nonlinear differential equations.

It is pointed at different modelling problems such as number of degrees of freedom, roll damping, initial conditions, ship speed, and heading ship towards waves.

An example of simulation studies is demonstrated at a Wide Shallow Vessel as compared to a Conventional Ship type. This unconventional design has been developed at the Special Research Pool SFB98 in Hamburg and Hannover for sea-going vessels at limited draught in tidal harbours.

### NOMENCLATURE

$x$	displacement of motion
$\{x\}$	column vector of displacements
$t$	independent time variable
$a$	inertia coefficient
$b$	damping coefficient
$c$	restoring coefficient
$d$	excitation
$y_i$	variable substituted for velocity $\dot{x}_i$ and displacement $x_i$
$i$	order number of degree of freedom
$j$	order number of coupled d.o.f.
$ndf$	number of degrees of freedom
$idp$	identifier for program subroutine
$m_{ij} \cdot \ddot{x}_j$	d'Alembert's dynamic inertia term
$F(i)$	sum of all other forces
$m$	inertia (mass)
$\Delta$	displacement (mass) of ship
$\alpha_{eff}$	effective wave slope
$V_s$	Ship speed
$f_E$	encounter frequency with regular wave
$X_s$	position of crest before midships
$L_w$	wave length
$H_w$	wave height
$L_s$	Ship length
$\chi$	heading, degrees (zero aft)
$GMO$	still water metacentric height
$GM_m$	mean metacentric height in seaway
$GZ$	hydrostatic righting lever
$GZ_m$	mean righting lever in seaway
$GZ_v$	time variation of righting lever
$HKR$	heeling lever
$CXX$	coupling portion of restoring term
$CXI$	restoring term accounted for all degrees of freedom
$S_n$	normalized spectrum
$Stoch$	time function derived from $S_n$
$p$	order of discrete spectral components
$*$	index for transformed to running ship
$\sigma$	standard deviation of normal distribution $N$

## 1. ON DEVELOPING MOTION RELATED CAPSIZE CRITERIA

Existing stability criteria have been reviewed extensively, see for example Proceedings of the Stability Conference in Glasgow, 1975 / 1 / .

Basically, the goal to be reached by stability criteria is to improve or establish safety of ships from capsizing in normally encountered operational conditions in a sea environment. It is therefore certain that stability considerations for the upright floating position of the ship in still water cannot be sufficient.

Naval Architects in recent years have agreed upon the fact that new and better stability criteria ought to be based on some kind of ship motion criteria in extreme situations, but there still remains the problem of adequately modelling and predicting of extreme ship motions.

Capsizing is a particular event of a ship motion, and we might describe it as the ship assuming another stable position at large heel, due to excessive roll motion. This large heel results in inoperability of the ship and it might subsequently lead to a total loss of the ship due to flooding or otherwise.

A ship should not capsize, even under dynamic environmental loads. This is all we want to assure, and is what a Naval Architect means when he considers ship stability. Thus we have to look at the motions, and try to avoid excessive roll and capsizing by proper ship design. In order to accomplish this goal, we need guidance in the form of criteria.

Over decades Naval Architects have only looked into the static stability of the ship and the magnitude of the restoring moment of the ship body when heeled, and have tried to derive sufficient criteria from that. Therefore, the terms "stability" and "stability criteria" have been used traditionally in this sense.

However, as we have pointed out, this is not the proper term, when we extend our considerations from stability of the upright position to the safety from capsizing under environmental loads. Hence, if we stick to the term "stability", while extending to motions (including capsizing), we might be tempted to look at the "ship motion stability". But how would this term be defined?

We can think of "motion stability" as the property of a ship in particular environmental conditions which forces her always to return to the initial upright position without capsizing, i.e. without assuming another (generally topside down) statically stable condition. This is the kind of definition we would like to follow.

However, the "stability of motion" has already been defined in mathematical terms for quite a different property of the motion. This "stability of motion" could be described as follows: Any motion is mathe-

matically stable, if sufficiently small disturbances do not result in another motion, but in always the same motion with only a very small deviation / 2 / .

This "mathematical stability of motion" can be used in order to check, if any estimated motion solution is stable. In other words, to check if we really get this solution, even if we have small disturbances along the time history of the roll motion, say due to inevitable variations in the assumed physical properties for modelling our physical system of ship and environment.

Say, for example, a ship assumes a large roll amplitude around the vanishing point of the righting arm curve,  $x_{40}$ . This is a point of equilibrium, but unstable. Any further small disturbance will suffice to upright or capsize the ship, and certainly, since small disturbances are of a random nature, we cannot predict the outcome in a single event. Once the ship comes to this position she might upright again, or capsize. Thus, even a very small disturbance will very much decide the final outcome of the motion, and mathematical stability analysis of the motion might be applied. However, certainly Naval Architects are more concerned with the time history, i.e. how the ship could come to this dangerous situation at all, rolling to angles where the righting lever vanishes.

On the other hand, we might go even further and assume that variations of physical properties of the ship due to the environment, such as for example, the variation of the uprighting moment of a ship in a following seaway, can be tackled by assuming small disturbances, and follow the mathematical stability approach, generally by the Lyapunov method / 3 / . If we succeed, we might have found mathematical stability criteria for the mathematical motion stability, based on parameters of the ship motion equation. The small disturbance assumption held reasonably well for linear ship motion theory. It does not apply to non-linear roll motion, and it most probably will not apply to ship motion stability. Furthermore, this approach has one major deficiency: If the disturbed motion solution deviates from the undisturbed solution (i.e. is unstable), this does not necessarily mean that the disturbed solution includes capsizing, and vice versa (even a solution that leads to capsizing might turn out to be mathematically stable). Furthermore, the frequency of the small disturbances certainly needs attention, since we know about the importance of the encounter frequency of ship and waves.

In conclusion, the use of the same term "stability" for different approaches, for assessing different properties of physical systems, has led to considerable confusion, but for us Naval Architects, stability has traditionally and due to the nature of ships, a very clear cut meaning. When we talk about criteria for stability

assessment, we do not mean criteria for a "mathematical motion stability", but we mean criteria for ensuring that the ship will not capsize, i.e. that the ship when in motion due to large variations in up-righting moments and of heeling moments, will always return to the upright statically stable condition. In order to avoid any further confusion with mathematical motion stability, only the term "capsize criteria" might better be used for motion stability, if we refer to a motion where no capsizing ought to occur.

## 2. STAGES OF STABILITY ASSESSMENT

From the practical point of view, we may not only look for new and supposedly better criteria, but still apply all the experience and judgement gathered and applied so far. There are certainly ships, after common static evaluation, where no doubts arise on their abilities to survive stable. Others will need more specific attention and further studies. Thus we might think of different levels of sophistication in the treatment of ship stability, as Kuo suggested in 1979 /4/. Following this idea of increasing levels or stages on stability assessment, we might think of classing the needed and realized effort for safety from capsizing for any ship. We then can look at the stages of stability assessment as follows:

### 2.1 Pure ship hydrostatics-

KG, GM, and up-righting levers in still water are considered.

### 2.2 Application of any of the so-called "weather criteria".

### 2.3 Moment balancing, according to Wendel's method /5,6/.

A set of possible dynamic moments is represented by static levers. Hence, the influence of the seaway is accounted for by the mean and the maximal reduction of the righting levers in a longitudinal single wave of ship length. Heeling moments according to expected situations are included.

### 2.4 Resonance considerations

Encounter spectra of ship and seaway versus ship heading, ship speed, and GM, and righting arm data with variations, for the areas of ship operation, are investigated. Overlapping of the probability distributions of natural roll frequencies-and of double the natural roll frequencies- for linear systems and depending on the roll amplitude for accounting for the nonlinearity, against the encounter seaway spectra are being looked at /7,15/.

### 2.5 Dynamic motion parameters

Criteria on motion parameters may be developed from theory and experiments. We can refer again to required righting levers, e.g. to the positive extent of

righting lever curves, but include also damping parameters, natural frequencies, radius of gyration.

### 2.6 Roll motion simulation with computer programs

- (i) in regular waves
- (ii) in an irregular wave pattern
- (iii) evaluation by probability means say from the occurrence of severe motion build-up, capsizes, relative frequency of capsizes, or the capsizing time distribution /1,7/.

### 2.7 Model experiments

- (i) tank tests in regular waves
- (ii) tank tests in irregular waves
- (iii) open water model tests in natural waves, e.g. /8, 9, 1/, already carried out in some research institutions in different countries.

I think it would be worthwhile to set up experiments on ship stability not only after completing stage 2.1, pure ship hydrostatics, but also the five intercepting stages. However, many ships will be judged adequately with stage 2.3, i.e. Wendel's moment balancing, for the safety from capsizing, as long as theoretical and experimental results from the upper stages 2.4, 2.5, 2.6, and 2.7 as well as gathered experience have been included.

## 3. MODELLING OF THE MOTION EQUATIONS

The motion of a free floating rigid ship body, say in a seaway, may be described by a set of six-degree-of motion equations such as, in matrix notation

$$(a)(\ddot{x}) + (b)(\dot{x}) + (c)(x) = (d) \quad (1)$$

Since for capsizing we are concerned with the resulting roll motion, for convenience, often just a single-degree-of-freedom system is being considered. However, it is well-known that the roll motion will be affected by the other modes of motion as well, such as heave, pitch, sway, surge, yaw. Paulling and Rosenberg/10/ first treated the non-linear coupling of motions. In /1/ we find several demonstrations on the impact of the coupling effect, most spectacular by Kure and Bang.

One can try to look at a full degree-of-freedom system. However, it might be harder to deduct results and criteria.

Once we have set up motion equations, will a solution be necessary in order to derive capsizing criteria?

Criteria based solely on the coefficients of the motion equations can be sought for. Since such methods save tedious computation and end up with some criteria or formulas on stability from the coefficients of the roll motion equations, it seems to be a comfortable method.

However, derived criteria have to be

looked at carefully, if they really can account for all possible extreme motions and influence factors.

Most coefficient - based criteria (of the motion equation) rely on looking at just the hydrostatic terms, based on the righting levers in the restoring term or in extension, together with static equivalents of the exciting moments (d), equ. (1).

With the aid of high speed digital computers, brute force solving of non-linear motion equations in a time-step integration procedure is possible. However, this method needs extensive calculations, and the problem to evaluate a number of time series arises. Furthermore, we must make sure the time series simulated are representative.

Extreme roll motion, and capsizing, have always been considered a non-linear phenomenon, because of the extreme non-linearity in the uprighting moment at large heel in the first place. Thus, hydrostatic calculations of the uprighting moments at large heel are part of the established stability calculation procedure. Furthermore, the roll damping at large roll amplitudes turns out to be highly non-linear too. This non-linear damping term has been considered to be of significant influence for the roll motion at resonance by damping excessive roll motion. However, how exactly the non-linear roll damping can be calculated and taken into account has not been clear so far. There is some work published by Grim / 11 /, which might be applied for incorporating the non-linear damping calculations into ship stability criteria as well, see also /1,12,13/. For new capsize criteria based on motion dynamics, roll damping will need to be examined more closely.

#### 4. CAPSIZING AND IRREGULAR SEAWAY

The main advantage of linear theory besides linearisation of the Euler equation and subsequent solving for the hydrodynamic pressures around the ship hull may be seen as a means of transferring results in the frequency domain for discrete wave frequencies onto an irregular (random) seaway, described by spectra. Therefore, from the seaway spectrum we get the motion response spectrum via the transfer function  $Y_{\xi}$  calculated with linear hydrodynamic theory for regular waves, as can be seen in the known relation

$$S_x = Y_{\xi}^2 \cdot S_{\xi} \quad (2)$$

From the motion spectrum  $S_x$ , any statistical parameter, such as significant motion, may be calculated.

However, this procedure is not valid for a non-linear problem. Non-linear motion results for regular wave input cannot be superimposed. Thus the problem arises of how to handle irregular ship motion in the non-linear domain.

There have been two approaches -

- (i) Approximation of the irregular seaway by one single frequency, e.g. Grim/14/. This method seems to be reasonable for a wave group, which builds up in an irregular seaway and constitutes a dangerous situation for capsizing.

Furthermore, the most dangerous situations for capsizing appear in following and aft quartering seas. In these conditions and for a moving ship the encounter frequencies transform the irregular seaway into a narrow band spectrum, thus bringing about closer resemblance to a discrete harmonic excitation. Thus, with some caution about this approximation, useful results from non-linear motions in harmonic waves might be applied to ships in a random sea / 7,15 /. Time domain numerical simulation of the roll motion can be applied, as described in Chapter 10 of this paper.

- (ii) Direct simulation of irregular motion in the time domain. The solution of a set of non-linear motion equations for an irregular seaway is possible by applying a step-by-step integration procedure, where irregular variations of equation coefficients are introduced.

For a stationary seaway (in a statistical sense) certain ship conditions and headings have to be chosen, and motion time series will result for each particular condition.

The resulting irregular output  $x_4(t)$  might then be sampled and analysed with respect to severe motions or even capsizing. A statistical analysis of calculated time series and capsizings was introduced in 1969 / 16 /. Besides the amount of calculation work included in the approach, which might be overcome in a reasonable way, the problem of incorporating the correct hydrodynamic parameters in the non-linear motion range has not yet been solved. So far, however, the motion mechanism, and the different modes of capsizing in an irregular seaway, could be studied by this approach, and compared with experiments. In order to study all possible outcomes, a wide parameter range must be covered, which leads to a large amount of conditions to be simulated. For a statistical evaluation, this method therefore leads to numerous calculations, and the requirements on the statistical accuracy had to be looked at / 1 /.

The above mentioned simulation method might be worthwhile for further basic research on extreme motion behaviour for comparative studies with new criteria, and for comparison with model tests in regular and random seas.

## 5. SIMULATION PROGRAMS

At the present time there already exist numerous programs based on the simulation approach, and we cite only a few. In 1969 / 16 / the simulation and statistical evaluation of a one-degree-of-freedom non-linear system with a random parametric excitation was introduced, i. e. seaway excitation was due to the variation of the uprighting moments of the ship in a seaway. A Runge-Kutta-Nystroem integration procedure was used. For regular linear parametric simulation, the corresponding motion differential equation is known as the Mathieu equation. Based on this approach, the here described program RSIM has been developed and extended to coupled motions.

Paulling and Wood / 17 / in 1974 set up a six-degree-of-freedom motion simulation program. They were using Frank's close-fit-method in order to incorporate the linear hydrodynamic parameters correctly. For large motion amplitudes, they rely on the assumption that during the time of the capsizing, the roll motion itself tends to have an extremely large period, or conversely, a very small frequency and slow motion. This is at the point in the righting lever curve, where the ship at large heel has her unstable point, i. e. where the righting arm curve vanishes (is crossing zero). A very small disturbance in this condition decides the further outcome, capsizing or uprighting again. From this assumption, one may introduce the hydrostatic non-linearities solely, which is of course more readily done.

However, application is not widely spread in ship stability but more in the Ocean Engineering field on Offshore Structures /18/. In particular, simulation studies have been widely applied on anchored offshore structures.

Kaplan / 19 / in 1975, at the PNSS Conference presented a six-degree-of-freedom simulation system as well. He applied his program in particular to non-linear motions of offshore structures, where all forces such as forces due to currents, seaway, and anchoring, might easily be included.

For slamming loads and the resulting ship behaviour, motion simulation was set up by Meyerhoff and Schlachter / 20 /.

There are still some disadvantages of the numerical solution procedure. Even after fitting the correct coefficients into the scheme, and after overcoming problems related with numerical stability and the computer time required, analysts still have their reservations, mainly stemming from the fact that it is hard to know which parameter contributed most to the outcome. In other words, there is no readily available formula which allows us to look at the influence of any of the many parameters involved. Only by a sensitivity numerical analysis, i. e. through variation of the parameters, one will be able to judge their influence on the outcome.

## 6. PROBABILISTIC SAFETY CONCEPT OF CAPSIZE CRITERIA

It would be an almost impossible task to solve for the real actual probability of capsizing for any ship during her lifetime, because of the many parameters involved such as ship characteristics, environment, service routes of ship etc. Furthermore, any probabilistic safety measure should be related to some sort of analysis for the savings which improved safety will bring about versus the efforts to be made. This is also true even for the saving of lives, since there certainly exists some limit beyond which any more improvement of safety is not worthwhile any more, because of a decreasing safety-effort gradient versus effort. This means that in a probabilistic sense, we must be aware of the fact that there is no total safety, but there always is the very rare possibility of a failure, even for a ship considered "reasonably" safe. However, this does not prevent us from determining and improving capsizing safety criteria, but it shows that the best way to proceed would be to develop sound criteria in order to reach a comparable safety level for all ships rather than try to reach total safety.

As Kobylinski in Glasgow 1975 / 1 / pointed out, the probability of capsizing concept means that we allow a ship to capsize. He says "This can not be accepted". Obviously, one must agree on his example, that one out of a hundred ships prone to capsize is too high a rate, i. e. with a safety probability of 0.99. A discussion could start about the level of safety required, but not with the probability concept as such, since it is a picture of reality. However, at the present time, it is certainly not appropriate to discuss just the probability level of safety required, since we have not yet established any satisfying capsizing criteria which would allow us to calculate with reasonable accuracy the level of safety from capsizing. At this stage, it seems to be more important to introduce a motion based capsizing criteria as such, in order to make ship designs comparable and reach an equal standard of safety.

Therefore, we need to establish a set of physical models for real situations, even with some simplifications. According to a suggestion by Krappinger in Glasgow, 1975 / 1 /, citation: "because the optimum degree of safety (i. e. the conditional probability that a ship will not capsize under the assumed set of situations) is not known explicitly, it seems good enough to assume such values of wind forces, wave heights, etc. or combinations thereof, which do not capsize ships which are considered sufficiently safe", end of citation. From the practical point of view, it might be "good enough" to establish criteria without explicitly calculating any capsizing probability. This does not mean that we do not

always bear in mind the random nature of events, and will not take that into account.

## 7. CAPSIZE CRITERIA BASED ON MOTION DYNAMICS

New capsizing criteria must be based on the up-to-date knowledge of motions, hydrodynamics, and insight into the capsizing phenomena of ships. Although they are to be applied to any ship, experience on the impact on different ship types of these criteria will be gathered and certainly help with a further improvement or necessary adjustment of these criteria.

This will allow us to incorporate further developments in motion hydrodynamics, since for some of the dynamics coefficients in the nonlinear range at large roll, advances are to be expected in the near future. However, it is felt that the theoretical knowledge so far available can be used successfully for new motion based criteria on safety from ship capsizing.

The amount of knowledge and insight into the problem may be seen at different levels. The least insight would be experienced with the so-called black-box-method, where one knows the input, i.e. the excitation of the system, and the output, i.e. the motions, but nothing about the way this comes about within the system itself. This is, of course, a much too simple approach, and will mean for example, reliance on experiments solely. Since we have more insight into the capsizing problem, efforts are still very much devoted towards setting up expressions and calculation procedures in order to account for the capsizing phenomena.

In order to tackle the problem adequately, all three procedures such as experiment, simulation, and analytical derivation must be pursued in order to get closer to solutions and draw correct conclusions for safety criteria from capsizing. Numerical simulation, together with experiments, will allow comparison of specific cases under consideration and can be used for backing suggested capsize criteria.

Therefore, a practical choice of ship and environment conditions must be made, the results of motion calculations and subsequent criteria being a representative approximation of the real world. In other words, because of the limitations of existing stability criteria with respect to capsizing, it is more important to introduce motion based and probabilistically oriented criteria rather than try to solve the complete description of the motion and obtain a total overall safety with the exact figure of the safety probability. Therefore, capsizing criteria should be based on a choice of some particularly dangerous situations to be expected, and by a reasonably accurate modelling of the equations of motion, so that the important parameters for extreme roll motion are included. The limits of any such assumption must be considered.

In the following, we will try to outline some steps for a procedure on how to apply motion based capsize criteria.

## 8. CHOICE OF DYNAMIC SITUATIONS

For every ship under consideration, a table of the dynamic situations to be calculated must be set up. This table should include ship parameters to be varied versus relations of the ship to the environment to be looked at, see Fig. 1.

(I) Ship parameters or conditions.	Ballast conditions.
	Fully loaded conditions.
(II) Relations with the environment:	Choice of seaway spectra according to the operational area (North Sea, North Atlantic, etc.) Define spectra shapes.
	Operational ship speed and reduced speed in heavy seas.
	Following seaway.
	Aft quartering seaway (45 degrees);
	Include wind heeling moment.

Fig. 1 Table of Situation Parameters

## 9. WAY TO PROCEED IN CALCULATING MOTION BASED CAPSIZING CRITERIA

9.1 Calculate encounter spectrum due to ship speed for following and quartering sea. A Program TRASP was written in FORTRAN, which also approximates the encounter spectrum by a sum of discrete regular components, see example in Fig. 2.

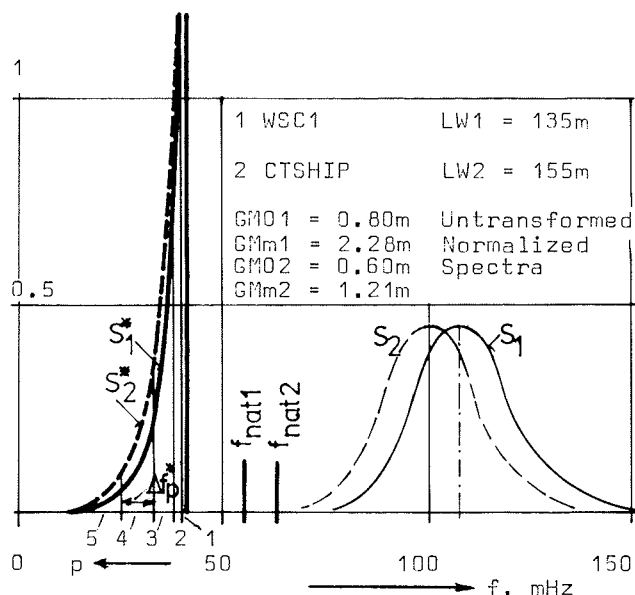


Fig. 2 Spectra of Righting Arm Variations Normalized, with Approximation by five Discrete Harmonics of Transformed Spectra  $S^*$  in Following Sea at Speed  $V_s = 18.5$  kn

9.2 Approximate irregular encounter spectrum by mean frequency and effective bandwidth.

9.3 Calculate surface of righting lever variation in a hydrostatic way, in following and quartering sea, with either 4 or at least two equidistant positions in a regular wave for wave-ship length ratio of 0.75, 1.0, 1.5 (ship length equal to the length in the respective waterline).

Height according to the formula (from Wendel's moment balancing method, /5, 6/ )

$$H_w = \frac{L_w}{10.0 + 0.05 L_w} \quad (3)$$

with  $L_w$  = wave length in metres.  
For extreme single waves, as in this case, trochoidal wave shape is a common choice.

9.4 Search for the largest variation of righting arms and for the largest decrease of righting arm, for following and quartering wave, within the results of 9.3.

9.5 Introduce results of 9.4, i.e. the most extremely different cases, for loaded and unloaded condition each, into one d.o.f. nonlinear motion equation.

9.6 Combine 9.5 with corresponding mean encounter frequency from 9.2.

9.7 Calculate nonlinear roll damping and added mass for motion amplitudes of up to 30 degrees.

9.8 Check, if there is Mathieu resonance to be expected from comparison of 9.3 and 9.2.

Include nonlinear effect on Mathieu resonance.

9.9 Perform simulation of roll motion, first with 1 d.o.f. equation. In extension, study selected coupling effects from yaw (broaching-to), sway and heave (phase relations), pitch, surge, all motion directions combined (six-degree-of-freedom system).

Simulation Program RSIM will be described in the next Chapter 10.

9.10 Evaluate results with respect of capsizing to be expected. No one single capsizing may be allowed for any of the above calculation modes. However, no defined limits are set for judging the ship to be safe.

Excessive roll in only one of the criteria modes requires more extensive calculations or model experiments.

Excessive roll will be considered any roll angle beyond  $0.75 \times 40$ , where  $40$  is the angle of vanishing righting arm of

the minimum curve according to 9.4, usually crest C.

9.11 Add hydrostatic heeling moment criteria to account for free surfaces, passengers on one deck side (if applicable), shifting loads, wind, icing (if applicable) by static balancing of uprighting and heeling levers according to Wendel's moment balancing /5, 6/.

Check for dynamic effects such as sloshing and motion acceleration on structural safety and lashing, and incorporate into simulation.

9.12 Decide on safety class of ship being reached, and if design was to be changed, calculations to be repeated or extended, or specific experiments to be set up.

The capsizing criteria steps outlined above must be carried out according to calculation procedures with programs tested for this purpose, in order to allow authorities to judge on the same basis. However, these steps cannot serve as a final procedure, but need a feedback from developing and gathering of experience in practical application.

## 10. DESCRIPTION OF PROGRAM RSIM

### 10.1 Equations of Motion

For a rigid body in motion, all acting forces and moments must be in equilibrium at any time, i.e. the sum of all forces and moments must be zero:

$$\sum_{j=1}^6 m_{ij} \ddot{x}_j + F_i = 0; \quad i = 1, \dots, 6 \quad (4)$$

Thus, the equations of motions result from the sum of all forces and moments acting in any of the six possible directions for a rigid body, i.e. for three lateral directions (forces for  $i=1, 2, 3$ ) and for three angular directions (moments for  $i=4, 5, 6$ ).

The motion equations are coupled, if forces or moments result from motion directions  $j$  other than  $i$ .

From evaluation the forces  $F_i$  in Eq. (4), we get terms proportional to motion velocity ( $\dot{x}$ ), to motion displacement ( $x$ ), and an external exciting term ( $d$ ) independent of any motion parameter, and we write down the wellknown system of second order ordinary differential equation for describing rigid body ship motions, in matrix notation

$$\ddot{\vec{x}} = [ (d) - (c)(x) - (b)(\dot{x}) ] / (a) \quad (5)$$

The set of second order differential equations can be transformed into a double number system of first order differential equations by substitution:

$$\dot{\vec{x}} = \vec{y} \quad (6)$$

$$\dot{\vec{y}} = [ (d) - (c)(x) - (b)(y) ] / (a) \quad (7)$$



Or rewritten with  $y_1 = \dot{x}$ ,  $y_2 = \ddot{x}$ ,

$$\dot{y}_1 = y_2 \quad (8)$$

$$\dot{y}_2 = [d - (c)(y_1) - (b)(y_2)] / (a) \quad (9)$$

If we solve for the particular motion acceleration  $\ddot{x}(i)$ , we get

$$\ddot{y}_2 i = \left( d i - \sum_{j=1}^{ndf} b_{ij} y_2 j - \sum_{j=1}^{ndf} c_{ij} y_1 j - \sum_{j \neq i} a_{ij} y_2 j \right) / a_i \quad (10)$$

with  $y_i = \dot{x}_i$  and  $y_{2i} = \ddot{x}_i$ ,  $i = 1, 2, 3$ .

As soon as any term of these equations, as e.g. the roll restoring term  $C44 \cdot x_4$ , becomes non-linear, a time domain stepwise solution can be used for calculating the motion  $(x)$ .

## 10.2 Reference System

The coordinate system has its origin in the centre of gravity G of the ship. Motions are given with reference to the gravity system, i.e. positive heave is directed opposite to gravitational acceleration, and zero roll means no heel of the ship with respect to the gravity axis. This reference system can only be applied for a purely longitudinal seaway.

However, simulation in quartering seas is of even greater importance, since GZ variations and external excitation moments are superimposed, and the encounter spectrum has a very narrow bandwidth which is close to a regular excitation. The effective wave slope  $\alpha_{eff}$  as shown in Fig. 3 can be used as a reference.

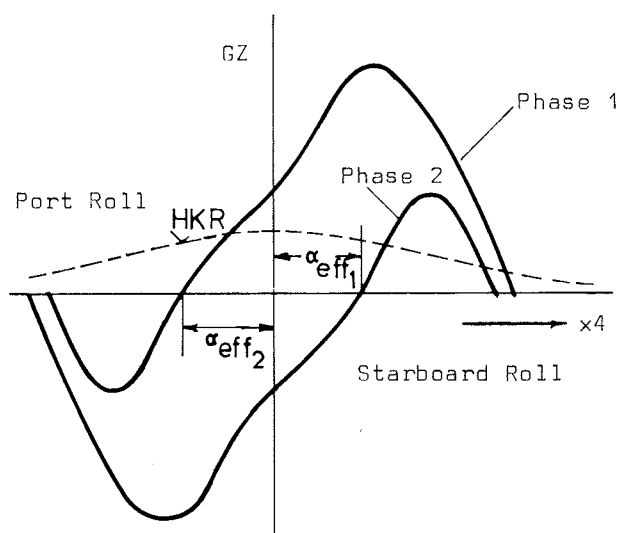


Fig. 3 Variation of GZ - Curves in Quartering Sea at Different Phase Conditions Wave and Ship

The Fourier-coefficients for the time-varying effective wave slope in phase with the passing wave and the corresponding righting lever variations in that wave can be calculated from hydrostatics programs and read in with program RSIM. Then the "apparent" roll angle  $\tilde{x}_4$ , as we might call it, is the roll with reference to the hydrostatic initial heel  $\alpha_{eff}$  depending on the underwater hull in quartering or beam waves, is given by

$$\tilde{x}_4 = x_4 - \alpha_{eff} \quad (11)$$

It is important to introduce this apparent roll angle  $\tilde{x}_4$ , because the motion dependent parameters in the motion equation such as roll damping, must be related to the actual motion.

## 10.3 Program Structure and Numerical Integration Procedure

There has been a rapid development in numerical procedures in recent years, and we refer to the work by Gear / 21 / and by Lambert / 22 /.

There are currently available reliable, well tested and documented software packages for a system of first order equations at most computer centers, which can be applied easily.

Fig. 4 shows the main block diagram of the program RSIM. Centerpart is the inclusion of the function  $f = \text{SUMidp}$  (equations (2) and (4) in Fig. 4) for our particular motion problem considered.

The program has been set up by means of structural programming. This implies that the program has been hierarchically structured in modules.

Thus we are able to use the program already developed to a certain stage for application, and the program is still open for further development and inclusion of more sophisticated submodules with corresponding subroutines or functions.

## 10.4 Righting Levers in Waves

In order to account for the non-linearity of the righting levers, the restoring roll motion term is given by

$$c44 \cdot x_4 = GZ(x_4, t) \cdot \Delta \cdot g \quad (12)$$

The righting lever function GZ must be calculated in the well-known hydrostatic way, usually, for the sake of simplicity, in a regular wave.

At the current level of development of the program RSIM, for GZ there are two subroutines incorporated, either GZREG in regular waves, the time dependence being defined by a Fourier sum of the coefficients of the 5-th order approximation of the righting lever curve versus heel, or by GZIR for irregular waves via a spectral representation of righting lever variations, according to the assumptions introduced earlier by this author / 16 /.

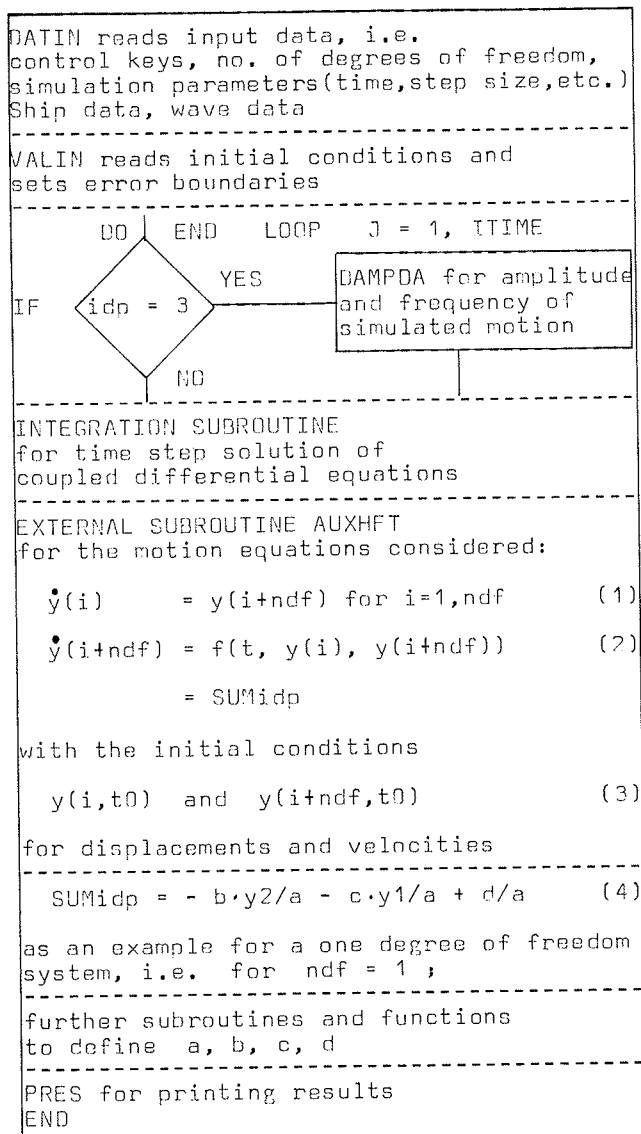


Fig. 4 Main Flow Chart of Simulation Program RSIM

#### 10.4.1 GZREG

The non-linear righting lever function GZ depending on the roll angle (resp. heel) is represented by

$$GZ(x_4, t) = F_0(t) + F_3(t)x_4^3 + F_5(t)x_4^5 \quad (13)$$

$$F_i(t) = 0.5 a(i, 1) + \sum_{j=2}^5 (a(i, j) \cdot \cos \beta_j + b(i, j) \cdot \sin \beta_j) \quad (14)$$

where

$$\beta_j = 2\pi(j-1)(t \cdot f_E + X_s/L_w) \quad (15)$$

$$f_E = f_W \left( 1 - \frac{2\pi}{g} f_W \cdot V_s \cdot \cos X \right) \quad (16)$$

$$f_W = \sqrt{g/(2\pi L_w)} \quad (17)$$

#### 10.4.2 GZIR

In function GZIR, Equ.(12) is replaced by spline functions in  $x_4$ -intervals (heel) such as

$$GZ(x_4) = ((D \cdot x_4 + C) \cdot x_4 + B) \cdot x_4 + A \quad (18)$$

The time varying righting lever can be split up into a mean righting lever curve in a seaway  $GZ_m$ , independent of time, and into the variation of the righting lever  $GZ_v$ , such as

$$GZ(x_4, t) = GZ_m(x_4) + GZ_v(x_4, t) \quad (19)$$

$GZ_m$  is determined by the mean of the righting lever curves at crest and trough, (as a first but reasonable approximation) or with more effort, from the mean of four equidistant phase conditions of the ship in a wave of about ship length (see Chapter 2). Furthermore,

$$GZ_v = \sqrt{m_0(GZ)} \cdot \text{Stoch}(x_4, t) \quad (20)$$

$$\text{where } \sqrt{m_0(GZ)} = t_R \cdot DGZ \quad (21)$$

$$\text{and } DGZ(x_4) = 0.5(T - C) \quad (22)$$

$DGZ$  is set to half the maximal righting lever variation from crest  $C$  to trough  $T$ .  $\text{Stoch}$  is the name of the irregular time function, which determines the actual size of the righting lever with respect to time.

This function  $\text{Stoch}$  can again be approximated by a mean time varying function valid for all heels, developed from the mean normalized spectrum  $S_n$  for all heels such as

$$S_n(f) = 1/(\hat{x}_4^4) \int_0^{\hat{x}_4} S_n(f, x_4) dx_4 \quad (23)$$

After comparative calculations on a ship form,  $S_n$  can be assumed to follow a log-normal shape,

$$S_n(f) = \frac{0.43}{\sigma \cdot f} N(0.43(\log f - \log f_{med})/\sigma) \quad (24)$$

Thus, we get the time varying function  $\text{Stoch}$  in (20) for all heels  $x_4$  from the normalized spectrum  $S_n$  as follows:

$$\text{Stoch}(t) = \sum_{p=1}^n \sqrt{S_{np} \cdot \Delta f_p} \cdot \cos(2\pi f_{pt} + \epsilon_p) \quad (25)$$

The total sum of the normalized spectrum area is equal to one, i.e. for all heels. The heel dependence comes in by multiplying  $\text{Stoch}$  with the square root of the actual area of the heel dependent spectrum of righting arm variations  $m_0 GZ$ .

We relate these  $m_0$ -values versus heel  $x_4$  in an irregular seaway to the maximum righting lever variation  $DGZ$  between crest  $C$  and trough  $T$  in a regular wave (according to Equ. 22) by the ratio  $t_R$ ,

$$t_R = DGZ(x_4) / \sqrt{m_0 GZ(x_4)} \quad (26)$$

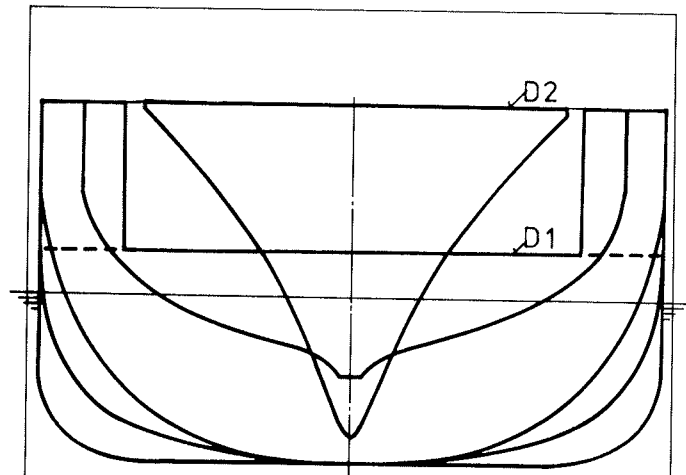


Fig. 5 Tschebischeff Cross Sections  
(1) Wide Shallow Ship Design C1

$L_{pp}=L_w=135m$        $D1$  9.50m  
 $B$  29.80m       $D2$  17.10m  
 $T$  8.00m       $H_w$  8.10m

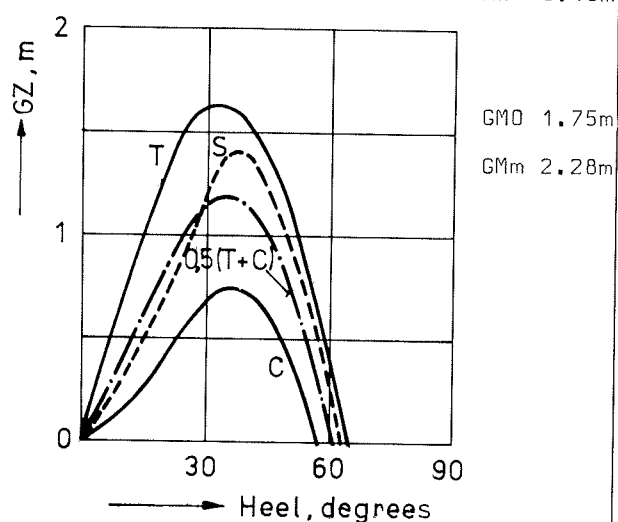


Fig. 6 Righting Levers in Longitudinal  
C Crest      T Trough  
(1) Wide Shallow Ship Design C1

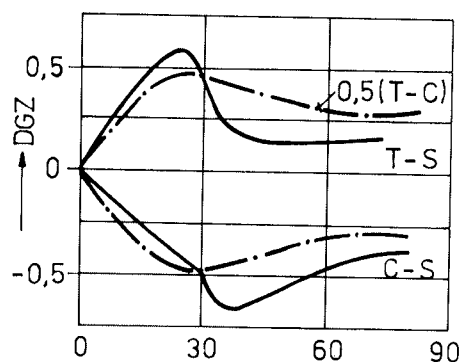
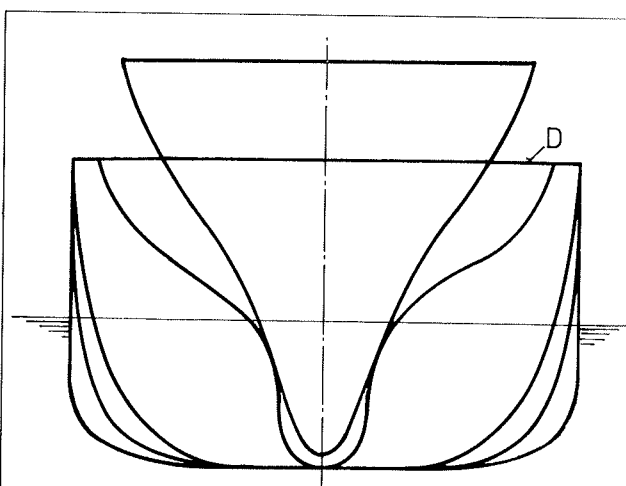
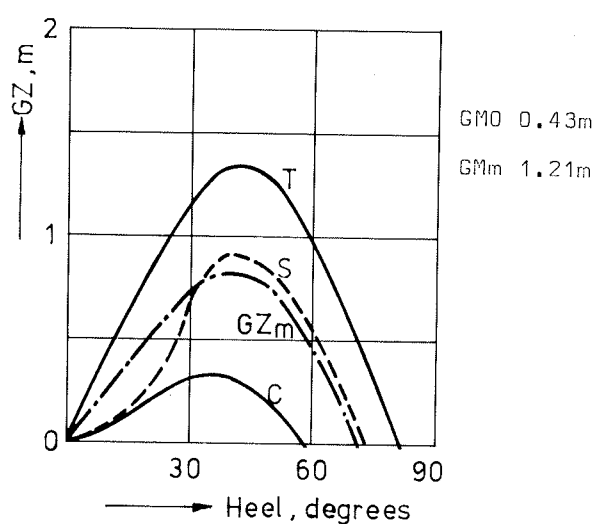


Fig. 7 Differences of Righting Levers

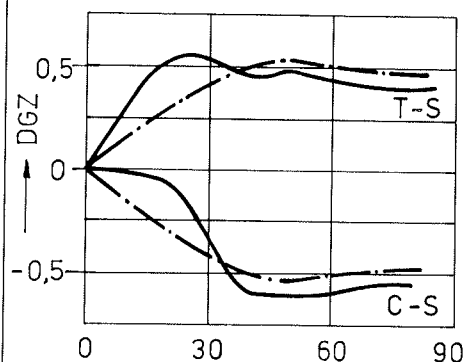


(2) Conventional Container Ship

$L_{pp}=L_w=155m$        $D$  14.60m  
 $B$  24.50m  
 $T$  7.00m       $H_w$  8.75m



Regular Waves of Ship Length  
Mean  $GZ_m = 0.5 (C + T)$ , S Stillwater  
(2) Conventional Container Ship



according to Fig. 5;  $DGZ=0.5(T-C)$

i.e. the ratio of the maximum of variation versus the standard deviation of the righting levers in a seaway. This  $t_R$  is equal to  $1/3$  for the Gaussian distribution, with only 0.27 p.c. exceeding, and  $1/2$  for 5 p.c. of all levers exceeding C and T. Thus we derive finally

$$GZ(x_4, t) = GZ_m(x_4) + DGZ(x_4) \cdot \frac{1}{R} \cdot \text{Stoch}(t) \quad (27)$$

The values of  $S_{np}^*$  and  $A_{fp}^*$  are calculated from the encounter spectrum with program TRASP, example graphically shown in Fig. 2.

In Program RSIM, the functions

$$GZ_m(x_4) \quad \text{and} \quad DGZ(x_4) = 0.5(C-T) \quad (28)$$

as shown in Figs. 6 and 7, are approximated by spline functions.

Naturally, the assumed mean function  $\text{Stoch}$  can be extended to represent the actual shape of the righting arm variations at any heel  $x_4$ , as done for a regular wave in GZREG. This numerical effort is rarely necessary.

A heeling lever  $HKR(x_4)$  can be read in and is again replaced by a spline function within RSIM. In the motion equation (1) the restoring term comes in with the residual righting lever from

$$CXX(4,4) \cdot y(4) = GZC = GZ(x_4, t) - HKR(x_4) \quad (29)$$

Finally, if coupling is to be considered, all coupling terms are added up

$$CXI = \sum_{j=1}^{ndf} CXX(i, j) \cdot Y(j); \quad i=4 \text{ roll} \quad (30)$$

All different righting lever contributions are printed at chosen time steps in order to allow insight into the origin of the simulated motion behaviour, see Fig. 8. The resulting lever  $CXI$  is printplotted (symbol L against I) in exchange with the line print of the motion data, in order to allow easy check of results, together with the roll angle  $x_4$ , symbol R against Symbol 4 as the reference, i.e. zero heel at 4.

## 11. EXAMPLE OF WIDE SHALLOW SHIP DESIGN

Ships featuring wide shallow hull forms for better economy in trades with tidal harbours have been designed within the Special Research Pool SFB98 /23/.

It was shown, that for bulk cargoes (tankers and ore carriers), the required freight rate can be kept at the same level, even if the costs of deepening waterways and harbours for growing ship size are taken into account. The same hull form design was applied to carry containers, but this project was not that successful with respect to the required freight rate calculated. Fig. 5 shows the cross sections of a Wide Shallow Ship Design (C1) compared to the lines of a Conventional Container Ship.

The Wide Shallow Container Vessel C1 needs upper walls in order to extend the

range of positive GZ-values in still water. Since for that reason the still water  $GMO$  is considered to be necessarily higher than for the conventional Containership, the mean natural roll frequencies in a seaway are large as well, larger than the encounter frequency range between wave and ship at the reduced speed of 18.5 kn, see Figs. 2 and 9, where 20 kn is the regular speed in operation.

Fig. 6 depicts the righting levers calculated in a longitudinal, regular trochoidal wave of ship length. Fig. 7 shows the differences of the righting levers accordingly.

The heel dependent functions  $GZ_m$  from Fig. 6 and  $DGZ$  from Fig. 7 were read into the program RSIM.

Fig. 8 shows the printout of a large roll from motion simulation at one degree of freedom with the Wide Shallow Design C1, but at a significantly increased height  $KG$  of the Center of Gravity, resulting in a still water  $GMO = 0.27m$ , chosen from the resonance condition with the encounter frequency of maximum spectral energy, which is the case at the mean natural roll frequency (linearly) for  $G_{mm}$  equal to 0.80 m.

However, with non-linear damping, this large roll reduces in the simulation. No capsizing was found in the first few runs.

Thus, in aid to the stages of 2.3 moment balancing and of 2.4 resonance considerations, see Chapter 2, this clearly shows the importance of 2.5 and 2.6 on including dynamic motion parameters such as damping and hydrodynamic mass (or the roll radius of gyration) in motion based capsizing criteria.

Calculations should now be extended to study the impact of coupling effects at resonance, and of large accelerations for higher  $GM$ -values.

For the Wide Shallow Ship Design, model experiments would be the next step, rather than a wait and see attitude on the operational experience, even more so, since we are concerned with a new development.

## 12. HEAD AND BEAM SEAS

In the above treatment of the capsizing problem, only following and aft quartering seaway have been considered to be a threat to the safety from capsizing. How about the other headings of ship and waves, or in other words, what course should we Naval Architects recommend to the Shipmaster?

Head sea cannot be put off completely, and certainly the magnitude of the righting lever variations may be as large as in following waves. However, since the encounter frequencies at head seas shift towards greater values, roll resonance is only to be expected at larger restoring moments, which may lead to severe motion accelerations, but not to capsizing.

# START OF MOTION SIMULATION IN THE TIME DOMAIN

J	X	DIS1	DIS2	DIS3	VELO1	VELO2	VELO3	H	IFAIL	XREL,-	HEEL	GZRES	+	TR	D6Z	•	ST(=GZVAR)=GZC	CXI,M	ACC/G
-	SEC			RADS			RAD/SEC	SEC	-		DEGRS.							-	
1	0.0	0.00	0.00	-0.17	0.00	0.00	0.00	0.10	0	0.00	-10.00	-0.187	0.50	0.244	0.00	0.000	-0.187	-0.187	0.020
6	1.0	0.00	0.00	-0.17	0.00	0.00	0.01	0.10	0	0.04	-9.61	-0.179	0.50	0.235	-0.38	-0.745	-0.224	-0.224	0.022
11	2.0	0.00	0.00	-0.15	0.00	0.00	0.03	0.10	0	0.07	-9.39	-0.155	0.50	0.206	-0.74	-0.076	-0.231	-0.231	0.022
16	3.0	0.00	0.00	-0.11	0.00	0.00	0.04	0.10	0	0.11	-6.35	-0.116	0.50	0.156	-1.06	-0.083	-0.199	-0.199	0.017
21	4.0	0.00	0.00	-0.06	0.00	0.00	0.05	0.10	0	0.15	-3.67	-0.066	0.50	0.091	-1.33	-0.050	-0.126	-0.126	0.009
26	5.0	0.00	0.00	-0.01	0.00	0.00	0.05	0.10	0	0.19	-0.65	-0.012	0.50	0.016	-1.53	-0.012	-0.024	-0.024	0.002
UPWARD ZERO CROSSING AT JNEW= 28 0.563 DEG HEEL																			
AMP:ITEST,/RMX,DEG/,FMOT,S-1,EQU,LIN,DPC,S-1 3431 10.0 0.582 0.000 D,- = 0.0000																			
31	6.0	0.00	0.00	0.04	0.00	0.00	0.05	0.10	0	0.22	2.35	0.018	0.50	0.058	-1.64	-0.048	-0.029	-0.029	0.001
36	7.0	0.00	0.00	0.09	0.00	0.00	0.05	0.10	0	0.26	5.29	0.042	0.50	0.130	-1.68	-0.109	-0.067	-0.067	0.003
41	8.0	0.00	0.00	0.15	0.00	0.00	0.05	0.10	0	0.30	8.33	0.070	0.50	0.204	-1.63	-0.166	-0.096	-0.096	0.005
46	9.0	0.00	0.00	0.20	0.00	0.00	0.06	0.10	0	0.33	11.56	0.104	0.50	0.280	-1.49	-0.210	-0.105	-0.105	0.006
ELDAM1: LIN. EQUIV. DAMPING COE. DELTA,1/SEC																			
51	10.0	0.00	0.00	0.26	0.00	0.00	0.06	0.10	0	0.37	15.00	0.146	0.50	0.355	-1.29	-0.229	-0.082	-0.082	0.003
56	11.0	0.00	0.00	0.32	0.00	0.00	0.06	0.10	0	0.41	18.56	0.195	0.50	0.416	-1.02	-0.212	-0.017	-0.017	0.004
61	12.0	0.00	0.00	0.38	0.00	0.00	0.06	0.10	0	0.44	21.97	0.244	0.50	0.455	-0.71	-0.161	0.083	0.083	0.014
66	13.0	0.00	0.00	0.43	0.00	0.00	0.04	0.10	0	0.48	24.87	0.275	0.50	0.472	-0.37	-0.087	0.168	0.168	0.024
71	14.0	0.00	0.00	0.47	0.00	0.00	0.03	0.10	0	0.52	26.89	0.283	0.50	0.475	-0.02	-0.004	0.276	0.276	0.031
76	15.0	0.00	0.00	0.48	0.00	0.00	0.00	0.10	0	0.56	27.73	0.280	0.50	0.475	0.32	0.076	0.357	0.357	0.038
81	16.0	0.00	0.00	0.47	0.00	0.00	-0.02	0.10	0	0.59	27.16	0.282	0.50	0.475	0.63	0.150	0.432	0.432	0.043
86	17.0	0.00	0.00	0.44	0.00	0.00	-0.05	0.10	0	0.63	24.95	0.276	0.50	0.473	0.90	0.212	0.488	0.488	0.047
91	18.0	0.00	0.00	0.37	0.00	0.00	-0.08	0.10	0	0.67	21.02	0.230	0.50	0.446	1.11	0.246	0.477	0.477	0.043
ELDAM1: LIN. EQUIV. DAMPING COE. DELTA,1/SEC																			
96	19.0	0.00	0.00	0.27	0.00	0.00	-0.11	0.10	0	0.70	15.50	0.153	0.50	0.365	1.24	0.227	0.380	0.380	0.030

FIG. 8 Print from R S I M - Simulation with Wide Shallow Vessel C1 in Astern Irregular Seas, Speed 18.5 kn, GM 0.27 m in Still Water, Mean GM 0.80 m in Following Sea, L-I Residual Lever, R-4 Roll angle

Only with one out of several ship models, that were tested in natural open water waves, severe motions and capsizes developed in head seas. Roden /25/ explained that head sea capsizing by resonance at the encounter frequency of the wave groups.

Beam seas in their effect appear as an external exciting term in the equation of motion, whereas in longitudinal seaway we find a Mathieu type excitation, from variations in the restoring term.

Shipmasters sometimes suggest running the ship in beam seas in case of danger of the total loss of the ship. Others would object, because the large roll build-up in beam seas might severely damage ship and load, even when no capsizing will occur.

The solution to these apparently contradictory recommendations is, that there is no such general rule at any operational condition as to the forward or beam seas heading. It all depends on the particular seaway in combination with the ship properties, defined by characteristics such as GM, righting lever curve GZ, natural roll frequency, damping, and others.

In case of large GM-values, the ship in beam seas will roll excessively, thereby causing trouble by damaging hatches, deck load, superstructures, from splashing green water, inducing large roll accelerations, shift of cargo etc. This may not only cause damage, but eventually loss of the ship by local structural failure and flooding.

However, in the case of danger from capsizing, i.e. at low GM and GZ-values, a ship in beam seas will be safer than in astern seas, since the ship then does not suffer from righting lever reduction and the resonance conditions of aft or quartering seas, see Fig.9.

Large motion amplitudes might show up in beam seas as well, but not as dangerous as in astern sea conditions, with respect to the danger of the total loss of the ship.

From experiments with free running models in astern natural waves, often extreme roll with capsizing builds up quickly, with no warning at all, after a short time motion at obviously small roll amplitudes, caused by a wave group together with phase conditions between wave crests and roll motion, but can be explained either by Mathieu resonance or total loss of righting lever.

Thus, it is hard to judge say by "feeling" from the severe motions of the ship, if she is to be in danger.

There are reports by experienced Ship Masters, who relate their survival at sea only to their turning the ship beam seas. Their experience fully complies with our notions.

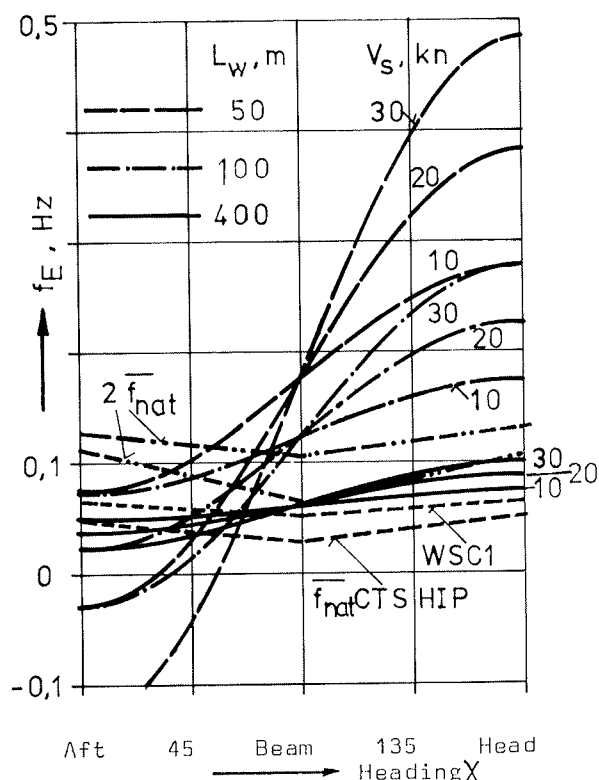


Fig. 9 Possible Resonance due to Coincidence of Natural Roll Frequency  $f_{nat}$  with Encounter Frequency  $f_E$  of Single Waves

#### ACKNOWLEDGEMENT

For this paper, use has been made of the work during my stay with Prof. Kuo at the University of Strathclyde, Glasgow, and of my work within the Special Research Pool (SFB98) in Hannover and Hamburg, sponsored by Deutsche Forschungsgemeinschaft (DFG). The shown righting levers were computed at the University of Hanover with ARCHIMEDES by N.A. Student Mr Muxfeldt.

With respect to the Computer program RSIM, I want to acknowledge the Computer Centers where I had the opportunity to do the programming and calculations, i.e. the RRZN Hannover, the Center at Strathclyde University, and the Center at the Hochschule fuer Technik Bremen.

The printing of this paper was accomplished using a Computer file at HfT Bremen. I wish to thank Ms Klaiber and Ms Wagenfeld for typing this manuscript onto the file, and N.A. Student Mr Moteka for his help with the drawings.

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## Discussion

A.Y. Odabasi (The British Ship Research Association, UK)

Professor Kastner proposes that the simulation may be used as a means of stability assessment and describes a simulation program developed to achieve this purpose. He also makes some sweeping comments on the definition and usefulness of "stability of motion" derived directly from the equations of motion.

I am afraid Professor Kastner is misinformed on the use of direct methods. The

mathematical stability theory furnishes a large number of stability definitions, each appropriate to specific conditions, cf. La Salle and Lefshetz [D1], Yoshizawa [D2]. The choice of an appropriate definition certainly poses a problem and was presented to be so by Kuo and Odabasi [D3]. In fact, all three utilisations of this method by Odabasi [D4, D5] and Ozkan [D6] contradicts the author's claim as they are based on the concept of stability of an equilibrium condition in the large under the action of persistent perturbations. To verify this

statement Fig. 10 may be useful, where two different stability definitions are illustrated, and the author's definition applies only for orbital stability assessment by the use of toroidal phase space.

As to the application of numerical simulation as a means of intact stability assessment, the following comments apply:-

- (1) Frequently, capsizing takes place through the phase-locking mechanism and to observe this effect a large number of runs of long duration need to be made with differing starting conditions.
- (2) Since the parameters in the governing equations cannot be predicted with required accuracy, a large number of runs need to be made to cover the band of variation of these parameters.

Since the above steps need to be performed for all the possible loading conditions, the impracticality of the proposed stability assessment method becomes apparent. One should, however, note that simulation has a very important role in the development of a rational stability criterion, viz:

- (1) Helping the development of a more representative mathematical model.
- (2) Providing qualitative and quantitative information on the possible mechanisms leading to capsize or dangerous rolling.
- (3) As a means of checking the efficiency of direct stability assessment methods.

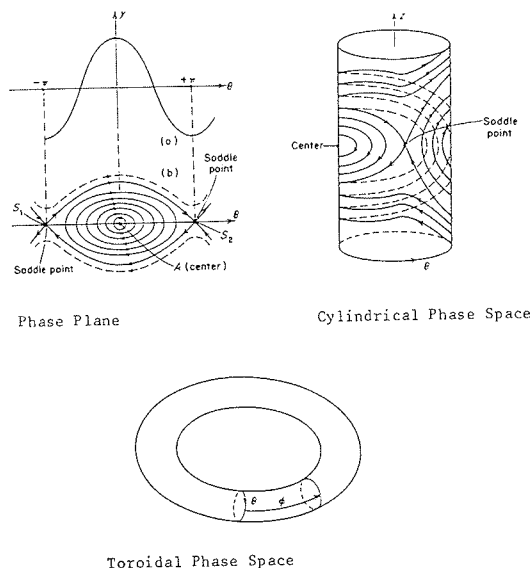


Figure. Use of Different Conceptions in the Stability Assessment

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C. Kuo (Univ. of Strathclyde, UK)

Thank you for your support to our levels approach. May I ask your views on two points.

Firstly, what are the changes that the application of simulation techniques will lead to effective practical stability criteria.

Secondly, if such criteria are developed what form would they take and whether they can be regarded as "simple".

C.M. Lee (Office of Naval Research, USA)

As the author pointed out in his paper, when we treat a nonlinear roll equation, we cannot utilize the conventional frequency-domain analysis to irregular waves. However, a serious drawback in using time-domain analysis to a nonlinear ship motion in waves is that the coefficients in the equations motion should be time dependent. It is extremely difficult to evaluate these coefficients either analytically or experimentally. I would like to know how the author has evaluated the hydrodynamic coefficients in his roll equation which was used for the time domain simulation. Another question is how one should choose the time history of waves which would represent a desired wave spectrum. As the author would agree, there could be quite a few ways of generating time series of waves. Main question is if the chosen time series could realistically represent the type of wave conditions which would be critical for roll assessments.

E. Dahle (The Norwegian Maritime Directorate Norway)

The paper gives an good overview of the present state of the West German research.

Probably the future approach to time domain simulation is to use advanced potential theory.

Finally, the statement of the author in Chapter 12 is restricted to larger vessels as smaller vessels may be endangered in beam seas by breaking waves or a series of steep



waves as described by other authors in this Conference.

#### Author's Reply

I am glad Dr. Odabasi could clarify, that the Lyapunov method stability is not at all limited to small perturbations. This is to be the case for an "orbital stability" definition only.

Simulation is like a physical experiment, but the parameters in the motion equations have to be set corresponding to nature. However, I do not think it is worthwhile to set up philosophical schools as of rational stability criteria against the application of numerical simulation. Both approaches should interact. By applying probability laws, one can limit the number of runs in simulation. We then only need to calculate short-term extreme situations of the ship in operation.

Professor Kuo asks about deriving practical and "simple" stability criteria from simulation. I have tried to put the simulation in the right perspective as a means of stability assessment in its own by the different stages (2.1 through 2.7 in my paper). For assessing motion stability, I would always start at the first stage, i.e. pure hydrostatics, only to proceed to stage 6, simulation, after levels of lower order have not yet satisfied us, and we need more information on the extreme behaviour to be expected.

Of course, after I have done simulation, I might eventually try to deduce more general results, i.e. criteria, from evaluation of the samples. But I would not consider to set up simulation for the mere purpose of deriving criteria, at least for the moment being, since we still need to know more about all the hydrodynamic parameters in-

involved. I feel we should not aim too much for that "simple" criterion, although practical it must be, but rather look for procedures which allow us at reasonable accuracy to cover the safety aspects for any of the wide variety of different ship types and ocean structures. Simulation can be such a useful procedure for stability assessment.

Dr. Lee is quite right in saying that once we simulate in the time domain, the question arises how we can tie frequency domain coefficients to it. In Program RSIM, I estimate the motion frequency from the latest simulated time history in order to pick the corresponding coefficients. However, I think we should develop new methods of integrating the pressure distribution around the shiphull in the time domain in a step-by-step procedure, i.e. simulate the flow as well. First attempts have been made by Soeding, as cited in Chapter 3 of my paper [13]. I use the common Fourier series representation of the time function. For irregular waves, I have simplified by calculating righting moment variation in an extreme regular wave, taken according to a formula based on weathership statistics. The irregular time function comes in by assuming a narrow band normalized spectrum. We might certainly change regular wave or change the spectral shape, to define other sea conditions.

I must agree with Prof. Dahle, that the statement in Chapter 12 of my paper on a ship in beam seas to be safer than in a stern seas applies only to ships with sufficient freeboard, which is usually the case for large ships. This may not apply for small ships, where beam seas wave breaking and water on deck can pose a severe danger to the vessel.

Let me finally thank the discussers for their questions, which allowed me to clarify some of the points I made in the paper.

*Session VIa*

## Damage Stability

*Chairmen*

Prof. Jong Heul Hwang  
Seoul National University  
Korea

Prof. Takeo Koyama  
University of Tokyo  
Japan

## ONBOARD CALCULATION OF DAMAGE STABILITY FOR ACTUAL LOADING CONDITIONS

TORBEN MUNK AND PETER FLERON

Danish Ship Research Laboratory

Denmark

### ABSTRACT

Damage stability calculation programs for computers are generally too large and complicated to be used on an onboard computer. Therefore, the Danish Ship Research Laboratory has developed a computer program, which does not require a large computer, and which is easy to use.

The main features of the new program are:

- (1) Hulls and departments are described by a limited number of data.
- (2) The iteration for the determination of the equilibrium position of the ship is rapidly convergent, even for large trim and heel angles.
- (3) The effect of fluids in tanks is taken correctly into account.

The program is based on a curve representation by means of specially developed spline functions.

The calculation of the restoring forces for change of the position of the ship in the water gives rise to problems at large trim and heel angles. The definition of trim and heel is therefore discussed, and a method for the calculation of the forces and moments acting on the ship together with their first derivatives with respect to trim, heel and draft is given.

### 1. INTRODUCTION

It is a requirement that certain types of ships must remain floating in a nearly upright position, if they are damaged in a specific way, and that they must even in this condition be able to withstand a certain degree of additional loading without capsizing.

It was previously normal practice to

examine this by means of calculations before the ship was taken into service. These calculations were then carried out for a number of typical load conditions, and the results were used onboard the ship as a guideline for the loading of the ship.

This procedure is not always sufficient today. There are types of ships for which it is nearly impossible to predict the loading cases, or for which the number of possible loading cases would be too big for convenience. This applies for instance to ships which are carrying deck cargoes and to ships with large, partly filled tanks. In these cases it would be more reasonable to carry out the calculations onboard the ship for the actual loading case.

This requires a fast and reliable computer program and a well-defined calculating procedure. In the following some methods for the calculation are described, and the definitions of some of the terms of the stability theory are reviewed.

### 2. TRADITIONAL TRANSVERSE STABILITY CALCULATION

Normally, a longitudinal and a transverse axis may be placed in a ship in a very obvious and well-defined way. The resistance against a rotation around the longitudinal axis is much less than around the transverse axis. Therefore, most attention has been paid to the first or these movements, which is called heel. The latter, which is called trim, will normally not be important in connection with the safety of the ship.

The forces which may cause trim or heel of a ship are normally gravitation forces, which are acting in a direction perpendicular to the water surface, or forces from

wind and waves which are acting in a direction parallel to the water surface.

It is therefore a very understandable simplification of the stability problem to use only one coordinate system for the calculations. This system may have the X-axis in the direction of the ship's longitudinal axis and parallel to the water surface, the Y-axis in the direction of the transverse axis of the ship and parallel to the surface, and the Z-axis perpendicular to the water surface. The relationship between hydrostatic forces calculated from the ship offsets to the outside forces is then correct, if the ship is upright on even keel, and nearly correct, if the heel and trim angles are small.

A further simplification can be made, if trim and heel are regarded separately.

This simplification has later been modified by demanding that longitudinal equilibrium shall be maintained during the calculation of the restoring moment for different heeling angles. This demand will only result in small trim angles during the heel of the ship, but the calculations become, nevertheless, so complicated and time-consuming that from then on calculations by hand were out of the question.

The determination of the equilibrium position of the damaged ship with a reasonable accuracy is still possible under the given conditions. The difficulties arise, however, if terms like metacentric height, righting lever, trimming and heeling moment are used on a damaged ship with both trim and heel, because these terms are no longer defined. The problems are caused by the fact that the ship coordinate system no longer coincides with the force coordinate system. The difficulties increase with increasing heel- and trim angles and make it impossible to talk of a moment to heel or a moment to trim separately, because any force in the force coordinate system will cause a rotation around both the longitudinal and the transverse axis of the ship.

In the following paragraphs a new calculation procedure to overcome these difficulties will be proposed together with an extension of the definition of the stability terms.

### 3. PROPOSED PROCEDURE FOR DAMAGE STABILITY CALCULATIONS

It is proposed that the coordinate system, in which the outside forces are acting, is separated completely from the ship offset coordinate system from the beginning of the calculations.

The ship coordinate system is then identical with the one which has been used previously. It will have a longitudinal axis close to the keel line, pointing in the forward direction of the ship. Further, there will be an axis perpendicular to the longitudinal axis, pointing upwards. The third axis will then be the transverse

axis of the ship. In the following the longitudinal axis, the transverse axis and the vertical axis will be denoted X-axis, Y-axis and Z-axis, respectively, and it is supposed that the origin is placed where the aft perpendicular intersects the keel line.

The angle of heel may then be defined as the angle between the Y-axis and the trace of the waterline plane in the YZ-plane, while the trim angle in a similar way is the angle between the X-axis and the trace of the waterline plane in the XZ-plane.

The shape of the ship will normally be given by points on curves (frames) in a number of planes parallel to the YZ-plane at different positions on the X-axis.

The second coordinate system, in which the forces are acting, has one axis perpendicular to the water surface and the two other axes in the waterline plane, perpendicular to each other. The direction of one of these axes has, however, to be further decided. There are two obvious possibilities: The trace of the XZ-plane and the trace of the YZ-plane in the waterline plane. A choice between these has to be made, and the authors have chosen the first possibility, because they feel that the stem-stern line in the waterline plane will always be regarded as the fore-and-aft plane, even for a damaged ship. The coordinate system is therefore placed in such a way that the longitudinal axis, which is denoted the P-axis, is parallel to the trace of the XZ-plane (lateral plane of the ship) in the waterline plane and pointing in the forward direction of the ship. The two other axes are denoted the Q-axis and the S-axis (pointing upwards). The origin is for convenience supposed to be in the centre of gravity of the waterline area of the ship (see Figure 1).

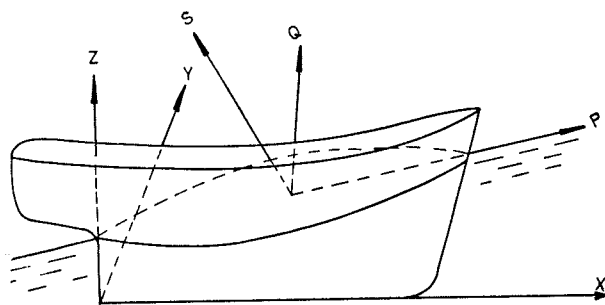


Figure 1: Ship Coordinate System (X,Y,Z) and Force Coordinate System (P,Q,S)

It is seen that it is now possible to talk of "trimming" and "heeling" forces and of a change of "trim" and "heel" angles with reference to the force coordinate system. These latter values must, however, not be mistaken for the trim and heel angles as defined previously. To avoid confusion it is proposed that the terms "longitudinal inclination" and "transverse inclination" from now on are used instead of "trim" and "heel" in the force coordinate system. The terms trim and heel will then from now on only be used in accordance with the definition.

#### 4. METHODS OF CALCULATION

##### 4.1 Outline

It is supposed that the ship is in equilibrium before and after the damage, and that the weight and the coordinates of the centre of gravity of the undamaged ship are known.

The problem is to find the mean draft and the angles of trim and heel of the damaged ship. This is done by means of hydrostatic calculations and iteration using 1st order restoring forces corresponding to the change of the mean draft and the angles of longitudinal and transverse inclination.

According to the proposed procedure, the hydrostatic calculations, including the calculations of the values related to the restoring forces, are carried out in the ship coordinate system based on the ship structure. The results are then transferred to the force coordinate system and corrected accordingly. The comparison of the forces and the determination of the next step in the iteration to equilibrium is performed in this coordinate system.

Each part of this procedure is treated separately in the following paragraphs.

##### 4.2 Definition of the Ship Structure

The structure is defined in the following iterative way:

- i) Each calculation unit, called a room (i.e. the entire hull, a damaged room or a tank), consists of a number of subrooms.
- ii) Each subroom is the volume of a section bounded by a number of longitudinal bulkheads and two transverse bulkheads.
- iii) Each section consists of a number of frames. (The division of the hull into sections is based on the assumption that the frames vary in a continuous manner within each section, cf. below).

Stated in reversed order, the geometry is defined by means of the following elements:

- i) All frames, longitudinal and transverse bulkheads are defined and each identified by a number. The complete definition of these elements is described in Appendix 1. Here, it suffices to state that a frame is defined by some points and the spline curve combining these points. A longitudinal bulkhead is defined by its equation in XYZ-space (defining the positive side of the bulkhead), and a transverse bulkhead is defined by its x-value.
- ii) Each section is defined by an interval of frame numbers.
- iii) Each subroom is defined by the number of the section containing the subroom, the numbers of the longitudinal bulkheads that bound the subroom and, for each bulkhead, a sign, indicating the side of the bulkhead that is part of the subroom, and finally the numbers of the two transverse bulkheads.
- iv) Each room is defined by the numbers of the subrooms constituting the entity.

This data structure is chosen because of its flexibility, its ability to represent even complicated structures by a rather limited amount of data, the size of the program that treats these data in the damage stability calculations, and the speed of these calculations.

Of course, one can not optimize all these measures simultaneously, but experience has shown that the selected method meets the requirements in a satisfactory way.

Regarding the conversion of the initial information concerning the ship into this data representation, two aspects should be mentioned in which an approximation is inherent:

- (1) The representation of the frames by parameter spline curves. This causes no serious problems, since even rather few points describe the curves with sufficient accuracy. Thus, for ordinary frames, some 10 points will do.
- (2) The number of frames chosen in a section. Here, one must keep in mind that the various hydrostatic data (volume, waterline area etc.) are obtained by longitudinal integration of functions that are known only at the frame stations (cf. below). Also, the relative importance of the subrooms in question must be taken into account.

##### 4.3 Calculation of Hydrostatic Data

A basic calculation unit is a subroom, given by a section and the longitudinal and transverse bulkheads that define the subroom.

For each subroom and a given waterline plane, the following ten quantities are calculated:

volume  
 volume moments (3 numbers);  
 waterline area;  
 longitudinal moment of  
 waterline breadth;  
 longitudinal moment of inertia  
 of waterline breadth;  
 area of waterline breadth squared  
 (for calculation of transverse  
 area moment);  
 longitudinal moment of waterline  
 breadth squared (for calculation of  
 product of inertia);  
 area of waterline breadth cubed  
 (for calculation of transverse  
 moment of inertia).

These quantities are then summed (after multiplication with the appropriate permeabilities) yielding the required quantities for the structure in question.

For the basic unit, the calculation proceeds in the following steps:

- (1) Each frame, consisting of one or more closed domains, is successively modified according to the intersection between the frame and the trace in the frame plane of the longitudinal bulkheads and the waterline plane. Note that after each intersection, the frame again consists of one or more closed domains. Because the last intersection is that given by the waterline, the waterline breadth and related quantities are readily obtained, cf. Appendix 2.
- (2) For the resulting modified frame, the frame area and the moments around the Y- and Z-axis are calculated, cf. Appendix 2.
- (3) Steps (1) and (2) are carried out for all frames in the section.
- (4) The six functions obtained for each frame are integrated longitudinally, yielding the above-mentioned ten quantities. In short, each of the six quantities is represented by a spline function of the longitudinal coordinate, and this function (together with the product of the function and  $x$  to the first and/or second power for some of the functions) is integrated analytically between the limits defined by the transverse bulkheads. If, for each integral, the value differs more than a certain amount from the value obtained by applying the trapezoidal formula, this last value is used, the reason being that the function values are such that a spline description is inappropriate. (This may occur for instance if a part of the section is immersed and hence all the waterline breadth functions are zero for a number of frames but different from zero for the remaining part).

#### 4.4 Correction of the Waterline Results

As mentioned previously, the force

coordinate system has its origin in the centre of gravity of the waterline area of the ship, a longitudinal axis and a transverse axis. The 1st order coefficients to the restoring force and the restoring moments corresponding to pure immersion and pure rotation around the longitudinal axis and the transverse axis will, according to traditional stability theory, be the waterline area,  $AW$ , the waterline area moment of inertia with respect to the longitudinal axis,  $IT$ , and the waterline area moment of inertia with respect to the transverse axis,  $IL$ . Further, a rotation around the longitudinal axis may also cause a moment around the transverse axis and vice versa. The coefficient of this moment is the product of inertia,  $MTL$ .

The four values for determining the coefficients of the restoring force and moments,  $AW$ ,  $IT$ ,  $IL$  and  $MTL$ , may be found from the waterline results of the hydrostatic calculations mentioned in paragraph 4.3, as described in Appendix 3.

#### 4.5 Determination of the Condition for Equilibrium

It is straight forward to determine the resulting forces and moments on the ship, when the weight and the buoyancy are transferred to the force coordinate system. If equilibrium is not achieved, the change of the condition of the ship is easily found from the resulting force and moments and the coefficients of the restoring force and moments.

The calculations are fully described in Appendix 5.

Normally, the condition of equilibrium is found after 2 to 3 iterations, or it is found that equilibrium is impossible (the ship will either capsize or sink).

The procedure is based on the assumption of linearity, but it will also work if the restoring force and moments are monotone functions of the movements. This condition is, however, not always fulfilled for the restoring moment corresponding to transverse inclination. It is therefore recommended to perform calculations of the total moment for transverse inclination corresponding to a number of heeling angles in order to find a preliminary angle of equilibrium by interpolation. This angle may then be used as a starting point for a final calculation as described above.

The intermediate results of the total moment for transverse inclination may, together with the corresponding metacentric heights, which are also found, also be used to determine the curve of righting levers, the so-called GZ-curve. This curve is used for the examination of the stability properties of the damaged ship in the state of equilibrium.

#### 4.6 Effect of Partly Filled Tanks

The free surfaces of partly filled

tanks will cause two effects, when the ship is trimmed and heeled:

- (1) The centre of gravity of the tank content will move.
- (2) The restoring moments for the ship will be changed.

This is taken into account in the following way:

The initial volume and centre of gravity of the contents of the tanks are known. Hydrostatic calculations are then carried out for the contents of the tanks, as soon as a new waterline plane is determined in the iteration procedure. The changes of the longitudinal and transverse weight moments are then summed up for all the tanks in question, and the initial weight moments for the ship are corrected accordingly. The calculated values corresponding to the free surfaces in the tanks are summed up, too, and are later on subtracted from the corresponding ship values immediately before the calculation of the restoring forces.

## 5. CONCLUSION

Damage stability calculations for actual load cases may improve both the security and the efficiency of a ship, and a calculation procedure which is suitable for a small onboard computer is described.

During the development of the method it has been found necessary to give a more accurate definition to the terms heel and trim and to define new terms to describe the moments, forces and movements which are relative to the surface.

It has further been found necessary to develop a simple way of dividing and describing the hull form and of improving the hydrostatic calculations in order to save computer time and reduce computer size. A method for exact calculation of the 1st order of the restoring forces, which reduces the number of iterations to equilibrium significantly, has also been described.

The methods are mainly developed for damage stability calculations of ships. They will, however, be applicable to all floating objects and will especially be suitable for constructions, for which stability in the longitudinal direction and in the transverse direction is of equal importance.

## APPENDIX 1. Definition of Frames and Bulkheads

### I) FRAMES.

Both symmetrical and asymmetrical frames are considered, a frame being symmetrical with respect to the Z-axis, if the first and last points are different.

For an asymmetrical frame, all points must be given, while only the starboard part of symmetrical frames need be given.

To each point, five numbers are attached: the y- and z-coordinate, the accumulated chord length, and the y- and z-moment in the spline description of y and z as functions of the accumulated chord length.

Besides ordinary points, this description includes 3 kinds of points of different discontinuity:

- branch points: y and/or z are discontinuous;
- knuckle points: tangent direction discontinuous;
- shear points: curvature discontinuous.

Before a symmetrical frame is used in the calculations, its point set is expanded by elementary operations to give a description of the complete closed frame.

Finally, an x value is attached to each frame.

### II) LONGITUDINAL BULKHEADS.

Each bulkhead is described by an equation of a plane:

$$ax + by + cz + d = 0$$

where b and/or c is  $\neq 0$ .

Besides ordinary longitudinal bulkheads, all planes that are not parallel to the YZ-plane may be described as a longitudinal bulkhead, f.inst. decks.

The vector (a, b, c) indicates by definition the positive side of the bulkhead.

## APPENDIX 2. Calculation of Frame Functions

As explained elsewhere, a frame, possibly modified by its intersection with longitudinal bulkheads and waterline, consists of a number of closed curves, described by parameter splines with the accumulated chord length as parameter.

### 1) Calculation of Waterline Breadth and Related Quantities

Suppose that the waterline actually cuts the frame and let the equation of the line be

$$(y, z) = (0, z_0) + t(dy, dz)$$

where the vector (dy, dz) is normalized.

In the case of two intersection points with line parameter values  $t_1, t_2$ , the quantities are:

waterline breadth :  $t_2 - t_1$

waterline breadth squared :  $-(t_2^2 - t_1^2)$

waterline breadth cubed :  $t_2^3 - t_1^3$

Generally, if more than one line segment contribute to the final closed curve(s) these expressions are summed for all segments.

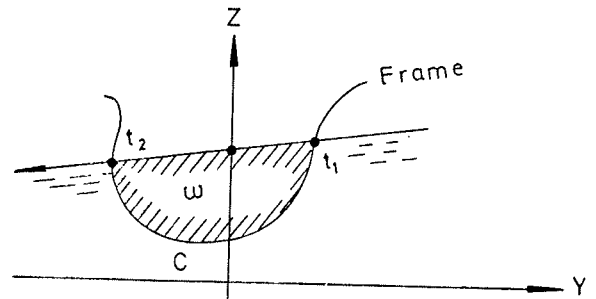


Figure 1: Frame Plane with Intersecting Waterline

### 2) Calculation of Frame Area and Moments

The identity

$$\int_{\omega} L'_z dy dz = \int_C L dy$$

where C is a closed curve, described counterclockwise, and  $\omega$  is the domain bounded by the curve, is used for

$$L = z, yz \text{ and } z^2/2 :$$

$$\text{area} = \int_{\omega} dy dz = - \int_C z dy$$

$$z \text{ moment} = \int_{\omega} y dy dz = - \int_C yz dy$$

$$y \text{ moment} = \int_{\omega} z dy dz = - \int_C z^2/2 dy$$

These three contour integrals are evaluated setting  $dy = (dy/ds) \cdot ds$ , where s is the spline parameter.

Since y and z are third degree polynomials of s in each interval of consecutive points, the integrals are polynomials of degree 5, 8 and 8 respectively. The integrals are calculated by means of Gauss-Legendre's 5 point formula, thus yielding exact results.



### APPENDIX 3. Correction of the Waterline Results

The hydrostatic results, which are related to the waterline, are

Waterline area,	AW
Transverse moment of waterline area,	MT
Longitudinal moment of waterline area,	ML
Product of inertia of waterline area,	MTL
Transverse moment of inertia of waterline area,	IT
Longitudinal moment of inertia of waterline area,	IL

These values may be calculated directly from the ship offsets, but have to be corrected before they may be used for the calculation of the 1st order restoring force and moments, which are used for the determination of the state of equilibrium.

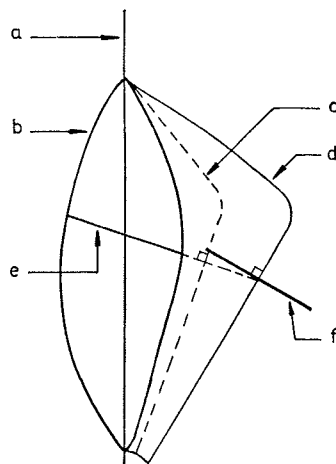


Figure 1: Waterline Plane (PQ-Plane)

The reason for the correction may be seen from Figure 1, which shows

- the trace of the XZ-plane in the waterline plane (a),
- the intersection between the ship and the waterline plane (b),
- the projection of the profile of the ship on the waterline plane (c),
- the profile of the ship, corresponding to a heel angle of 90 deg. (d),
- a waterline breadth, as it appears in the waterline plane (e) and as it is taken into account in the hydrostatic calculations (f).

The waterline breadth, which is situated on the trace of the frame plane in the waterline plane, is perpendicular to the projection of the keel line (X-axis) on the waterline plane. The reason for this is that the keel line is perpendicular to the frame plane and therefore perpendicular to

all lines in this plane. Hence the keel line is situated in a plane perpendicular to the waterline breadth, and the trace of this plane in the waterline plane will then be perpendicular to the waterline breadth.

All the hydrostatic values related to the waterline have been calculated under the assumption that the waterline breadths are perpendicular to the keel line. Two errors are therefore introduced in the calculations:

- (1) All longitudinal distances are too small. They have to be increased corresponding to a division by cosine to the trim angle VT.
- (2) All waterline breadths are too big. They have to be reduced corresponding to the angle VK, which is the angle between the trace of the XZ-plane in the waterline plane and the projection of the keel line on the waterline plane. For the calculation of this angle, see Appendix 4.

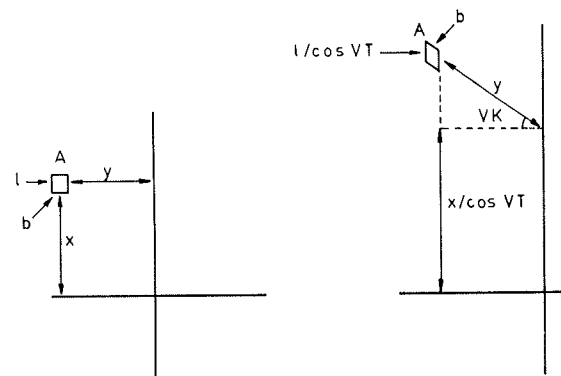


Figure 2: Corrections in the Waterline Plane

The corrections may be illustrated by Figure 2, which shows a small area before and after the correction. The designations on the figure are used in the explanations of the corrections, and the indices U and C are used to indicate the uncorrected and the corrected values respectively.

Area AW:

$$AW_C = AW_U \cdot \frac{\cos VK}{\cos VT} \quad (1)$$

Transverse moment MT:

$$\begin{aligned} MT_C &= AW_U \cdot \frac{\cos VK}{\cos VT} \cdot y \cdot \cos VK \\ &= (AW_U \cdot y) \cdot \frac{\cos^2 VK}{\cos VT} \\ &= MT_U \cdot \frac{\cos^2 VK}{\cos VT} \end{aligned} \quad (2)$$

Longitudinal moment  $ML$ :

$$\begin{aligned}
 ML_C &= A_U \cdot \frac{\cos VK}{\cos VT} \cdot \left( \frac{x}{\cos VT} + y \cdot \sin VK \right) \\
 &= (A_U \cdot x) \cdot \frac{\cos VK}{\cos^2 VT} \\
 &\quad + (A_U \cdot y) \cdot \frac{\cos VK}{\cos VT} \cdot \sin VK \\
 &= ML_U \cdot \frac{\cos VK}{\cos^2 VT} \\
 &\quad + MT_U \cdot \frac{\cos VK}{\cos VT} \cdot \sin VK \quad (3)
 \end{aligned}$$

Product of inertia  $MTL$ :

$$\begin{aligned}
 MTL_C &= A_U \cdot \frac{\cos VK}{\cos VT} \cdot y \cdot \cos VK \cdot \left( \frac{x}{\cos VT} + y \cdot \sin VK \right) \\
 &= A_U \cdot x \cdot y \cdot \frac{\cos^2 VK}{\cos^2 VT} \\
 &\quad + A_U \cdot y^2 \cdot \frac{\cos^2 VK \cdot \sin VK}{\cos VT} \\
 &= MTL_U \cdot \frac{\cos^2 VK}{\cos^2 VT} \\
 &\quad + IT_U \cdot \frac{\cos^2 VK \cdot \sin VK}{\cos VT} \quad (4)
 \end{aligned}$$

Transverse moment of inertia  $IT$ :

$$\begin{aligned}
 IT_C &= A_U \cdot \frac{\cos VK}{\cos VT} \cdot (y \cdot \cos VK)^2 \\
 &= A_U \cdot y^2 \cdot \frac{\cos^3 VK}{\cos VT} \\
 &= IT_U \cdot \frac{\cos^3 VK}{\cos VT} \quad (5)
 \end{aligned}$$

Longitudinal moment of inertia  $IL$ :

$$\begin{aligned}
 IL_C &= A_U \cdot \frac{\cos VK}{\cos VT} \cdot \left( \frac{x}{\cos VT} + y \cdot \sin VK \right)^2 \\
 &= A_U \cdot x^2 \cdot \frac{\cos VK}{\cos^3 VT} \\
 &\quad + A_U \cdot y^2 \cdot \frac{\cos VK \cdot \sin^2 VK}{\cos VT} \\
 &\quad + 2 \cdot A_U \cdot y \cdot \frac{\cos VK \cdot \sin VK}{\cos^2 VT} \\
 &= IL_U \cdot \frac{\cos VK}{\cos^3 VT} \\
 &\quad + IT_U \cdot \frac{\cos VK \cdot \sin^2 VK}{\cos VT} \\
 &\quad + 2 \cdot MTL_U \cdot \frac{\cos VK \cdot \sin VK}{\cos^2 VT} \quad (6)
 \end{aligned}$$

The values are now transferred to a coordinate system, which has the trace of the  $XZ$ -plane in the waterline plane as longitudinal axis and a transverse axis perpendicular to the longitudinal axis. The origin will still be at the position, where the  $Z$ -axis intersects the waterline plane. In this coordinate system the centre of gravity of the waterline area will have the coordinates  $(aL, aT)$ , where

$$aL = \frac{ML_C}{A_C} \quad (7)$$

and

$$aT = \frac{MT_C}{A_C} \quad (8)$$

The moments of inertia and the product of inertia may now in the usual manner be transferred to the centre of gravity, which will be indicated by an index  $F$ :

$$IT_F = IT_C - AW_C \cdot aL^2 \quad (9)$$

$$IL_F = IL_C - AW_C \cdot aT^2 \quad (10)$$

$$\begin{aligned}
 MTL_F &= MTL_C - MT_C \cdot aL \\
 &\quad - ML_C \cdot aT + A_C \cdot aL \cdot aT \quad (11)
 \end{aligned}$$

The coordinates of the centre of gravity of the waterline plane  $F$  in the  $XYZ$ -coordinate system are

$$x_F = aL \cdot \cos VT \quad (12)$$

and

$$y_F = aT / \cos VK \cdot \cos VH \quad (13)$$

where  $VH$  is the angle of heel (see Figure 3).

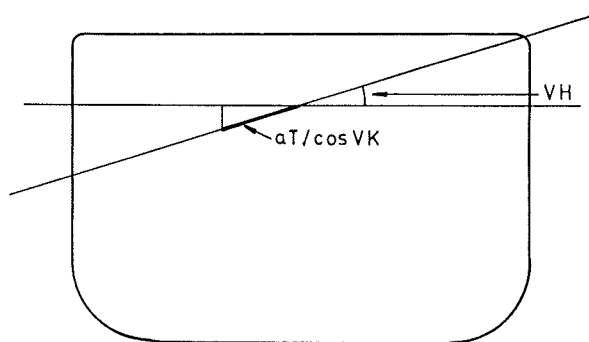


Figure 3:  $y$ -coordinate of the Waterline Centre of Gravity

#### APPENDIX 4. Calculation of the Angle VK

The angle VK between the trace of the XZ-plane in the waterline plane and the projection of the X-axis on the waterline plane, which is given by the normalized equation

$$Ax + By + Cz + D = 0 \quad (14)$$

is found in the following way:

A line perpendicular to the waterline plane through the point (0,0,0) has the equation:

$$(x,y,z) = (0,0,0) + t_0 \cdot (A,B,C) \quad (15)$$

The projection of the point (0,0,0) on the waterline plane,  $(x_0,y_0,z_0)$ , is found by inserting (15) in (14):

$$t_0 = -D \quad (16)$$

$$x_0 = -D \cdot A \quad (17)$$

$$y_0 = -D \cdot B \quad (18)$$

$$z_0 = -D \cdot C \quad (19)$$

The projection of the point (1,0,0) on the waterline plane  $(x_1,y_1,z_1)$ , is found in a similar way:

$$t_1 = -A - D \quad (20)$$

$$x_1 = 1 - A^2 - AD \quad (21)$$

$$y_1 = -AB - DB \quad (22)$$

$$z_1 = -AC - DC \quad (23)$$

The vector (1,0,0) on the X-axis has the projection

$$\begin{aligned} (x_1 - x_0, y_1 - y_0, z_1 - z_0) = \\ (1 - A^2, -AB, -AC) \end{aligned} \quad (24)$$

on the waterline plane.

After normalizing the vector will have the coordinates:

$$\left( \frac{1 - A^2}{\sqrt{1 - A^2}}, \frac{-AB}{\sqrt{1 - A^2}}, \frac{-AC}{\sqrt{1 - A^2}} \right) \quad (25)$$

A normalized vector in the trace of the XZ-plane in the waterline plane will have the coordinates

$$(\cos VT, 0, \sin VT) \quad (26)$$

where VT is the trim angle given by

$$\tan VT = -A/C$$

Hence (26) will be

$$\begin{aligned} & \left( \frac{1}{\sqrt{1 + \frac{A^2}{C^2}}}, 0, -\frac{\frac{A}{C}}{\sqrt{1 + \frac{A^2}{C^2}}} \right) \\ = & \left( \frac{-C}{\sqrt{C^2 + A^2}}, 0, \frac{A}{\sqrt{C^2 + A^2}} \right) \end{aligned} \quad (27)$$

The angle VK between the two vectors given by (25) and (27) may now be determined from the scalar product:

$$\cos VK = \sqrt{\frac{C^2}{(1 - A^2)(1 - B^2)}} \quad (28)$$

#### APPENDIX 5. Determination of the Condition of Equilibrium after Damage

The actual waterline plane is given by the equation

$$Ax + By + Cz + D = 0 \quad (29)$$

It is assumed that the equation is normalized, so that (A,B,C) are cosines for the normal N to the waterline plane. The sign of B is determined in such a way that N is directed downwards into the water.

The angle of heel, VH, may now be calculated from

$$\tan VH = -\frac{B}{C} \quad (30)$$

and the angle of trim, VT, from

$$\tan VT = -\frac{A}{C} \quad (31)$$

The following values are known from the hydrostatic calculations:

The weight of the undamaged ship, WW

The buoyancy of the damaged ship, WB

The centre of gravity, W, with coordinates  $(X_W, Y_W, Z_W)$

The centre of buoyancy, B, with coordinates  $(X_B, Y_B, Z_B)$

The waterline area for the damaged ship, AW

The centre of gravity, F, of the waterline area of the damaged ship with coordinates  $(X_F, Y_F, Z_F)$

$Z_F$  is found from (29) by insertion of  $X_F$  and  $Y_F$ , which are found in Appendix 3,

equations (12) and (13).

The lines perpendicular to the waterline plane through W and B are given by the equations:

$$\begin{aligned} (x_w, y_w, z_w) &= (x_W, y_W, z_W) \\ &+ t_W(A, B, C) \end{aligned} \quad (32)$$

and

$$\begin{aligned} (x_b, y_b, z_b) &= (x_B, y_B, z_B) \\ &+ t_B(A, B, C) \end{aligned} \quad (33)$$

The projections of W and B on the waterline plane are called W1 and B1. The coordinates to these points are found by insertion of (32) and (33) in (29).  $t_W$  and  $t_B$  may then be determined. These values are then inserted in (32) and (33) respectively, and the coordinates  $(x_{W1}, y_{W1}, z_{W1})$  and  $(x_{B1}, y_{B1}, z_{B1})$  to W1 and B1 are found:

$$t_W = -(A \cdot x_G + B \cdot y_G + C \cdot z_G + D) \quad (34)$$

and

$$t_B = -(A \cdot x_B + B \cdot y_B + C \cdot z_B + D) \quad (35)$$

$$\begin{aligned} (x_{W1}, y_{W1}, z_{W1}) &= \\ (x_W + t_W \cdot A, y_W + t_W \cdot B, z_W + t_W \cdot C) \end{aligned} \quad (36)$$

$$\begin{aligned} (x_{B1}, y_{B1}, z_{B1}) &= \\ (x_B + t_B \cdot A, y_B + t_B \cdot B, z_B + t_B \cdot C) \end{aligned} \quad (37)$$

The line P through the centre of gravity of the waterline plane F and parallel to the trace of the XZ-plane in the waterline plane is now found together with the line Q through F and perpendicular to P (see Figure 1).

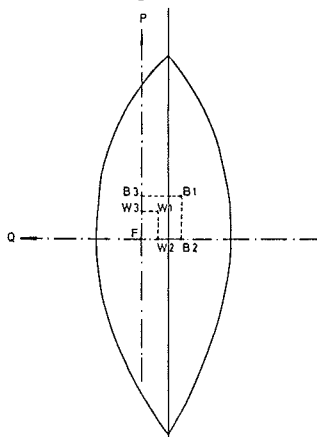


Figure 1: Waterline Plane

$$\begin{aligned} (x_p, y_p, z_p) &= (x_F, y_F, z_F) \\ &+ T_P(\cos VT, 0, \sin VT) \end{aligned} \quad (38)$$

$$\begin{aligned} (x_q, y_q, z_q) &= (x_F, y_F, z_F) \\ &+ T_Q((\cos VT, 0, \sin VT) \times (A, B, C)) \end{aligned} \quad (39)$$

Equation (39) may be changed to:

$$\begin{aligned} (x_q, y_q, z_q) &= (x_F, y_F, z_F) \\ &+ T_Q(-B \cdot \sin VT, A \cdot \sin VT - C \cdot \cos VT, \\ &B \cdot \cos VT) \end{aligned} \quad (40)$$

The projection of W1 on P is called W3. The distance FW3 is related to the coordinates  $(x_{W3}, y_{W3}, z_{W3})$  of W3 in the following way according to (38):

$$\begin{aligned} (x_{W3}, y_{W3}, z_{W3}) &= (x_F, y_F, z_F) \\ &+ FW3(\cos VT, 0, \sin VT) \end{aligned} \quad (41)$$

FW3 may now be found from the scalar product of  $(\cos VT, 0, \sin VT)$  and  $\overrightarrow{W3W1}$ :

$$\begin{aligned} ((x_F, y_F, z_F) + FW3(\cos VT, 0, \sin VT) \\ - (x_{W1}, y_{W1}, z_{W1})) \cdot (\cos VT, 0, \sin VT) \\ = 0 \end{aligned} \quad (42)$$

This gives

$$\begin{aligned} FW3 &= \cos VT \cdot (x_{W1} - x_F) \\ &+ \sin VT \cdot (z_{W1} - z_F) \end{aligned} \quad (43)$$

In a similar way it is found that

$$\begin{aligned} FB3 &= \cos VT \cdot (x_{B1} - x_F) \\ &+ \sin VT \cdot (z_{B1} - z_F) \end{aligned} \quad (44)$$

$$\begin{aligned} FW2 &= -(x_{W1} - x_F) \cdot B \cdot \sin VT \\ &+ (y_{W1} - y_F) \cdot (A \cdot \sin VT - C \cdot \cos VT) \\ &+ (z_{W1} - z_F) \cdot B \cdot \cos VT \end{aligned} \quad (45)$$

$$\begin{aligned} FB2 &= -(x_{B1} - x_F) \cdot B \cdot \sin VT \\ &+ (y_{B1} - y_F) \cdot (A \cdot \sin VT - C \cdot \cos VT) \\ &+ (z_{B1} - z_F) \cdot B \cdot \cos VT \end{aligned} \quad (46)$$

The resulting force, WN, the resulting longitudinal inclination moment, WML, and the resulting transverse inclination moment,

WMT, may now be determined:

$$WN = WW - WB \quad (47)$$

$$WML = WW \cdot FW3 - WB \cdot FB3 \quad (48)$$

$$WMT = WW \cdot FW2 - WB \cdot FB2 \quad (49)$$

Equilibrium is obtained if these values are negligible. If not, the position of the ship in the water has to be changed in such a way that the restoring force and moments equal WN, WML and WMT respectively.

The 1st order restoring force for immersion, RN, is given by the following equation:

$$RN = AW \cdot DT \quad (50)$$

where DT is the change of mean draft, perpendicular to the waterline plane. DT may be found from:

$$DT = WN/AN \quad (51)$$

The longitudinal inclination restoring moment is partly dependent on the positions of the weight centre of gravity, W, and the centre of buoyancy, B, partly on the change of the waterline area.

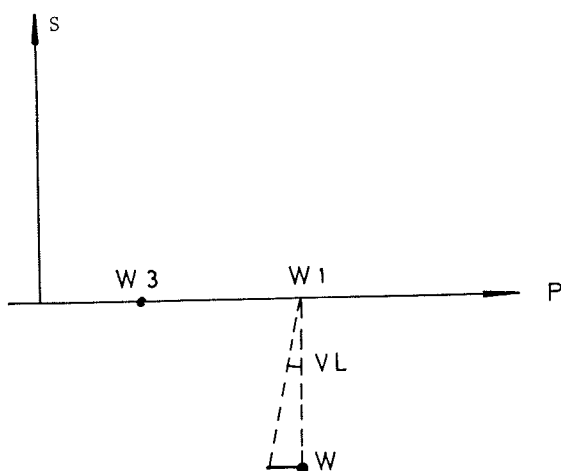


Figure 2.

From Figure 2 it is seen that a small longitudinal inclination angle, VL, will change the weight lever W3W1 by an amount of WW1 · VL.

Incidentally, WW1 is identical to  $t_W$  given in (34), though with opposite sign.

In a similar way it is seen that the buoyancy lever will be changed by BB1 · VL, where BB1 is identical to  $t_B$  given by (35) with opposite sign.

The change of waterline by a small longitudinal inclination angle, VL, causes according to well-known calculations a 1st order restoring moment of the magnitude  $IL \cdot VL$ .

The total restoring moment corresponding to a small longitudinal inclination

angle, VL, is then

$$VL \cdot (IL + WW \cdot WW1 - WB \cdot BB1) = VL \cdot RML \quad (52)$$

where

$$RML = IL + WW \cdot WW1 - WB \cdot BB1 \quad (53)$$

In a quite similar way a restoring moment coefficient, RMT, corresponding to a small angle of transverse inclination, VTV, is found:

$$RMT = IT + WW \cdot WW1 - WB \cdot BB1 \quad (54)$$

Further, a rotation around the longitudinal axis may give rise to a moment around the transverse axis and vice versa. This phenomenon requires one more 1st order restoring moment coefficient, RMTL, for which is known that

$$RMTL = MTL \quad (55)$$

Now the additional angles of longitudinal and transverse inclination, VL and VTL, may be found from the following equations, assuming linearity:

$$WML = RML \cdot VL + RMTL \cdot VTV \quad (56)$$

$$WMT = RMTL \cdot VL + RMT \cdot VTV \quad (57)$$

It should be mentioned that sine of the angles are normally used instead of the angles themselves. In this case it is, however, found more convenient to use the angles alone. It is justified by the fact that the equations are in any case only valid for small angles.

It should be stressed that VL and VTV are not identical to the additional angles to the trim angle and the heeling angle, VT and VH. New values of VT and VH have to be calculated from the coefficients of the new waterline plane, as described in (30) and (31).

The new waterline plane will contain the new centre of gravity of the waterline area, which has the coordinates

$$(x_F, y_F, z_F) = DT(A, B, C) \quad (58)$$

Further, the normal to the plane may be given by the vectorial product of two vectors in the new plane. These two vectors may be the direction vector of the line P after a rotation around the line Q corresponding to the angle VL, and the direction vector of the line Q after a rotation around the line P corresponding to the angle VTV.

The direction vectors of the lines P and Q will after rotation have the coordinates:

$$x_{PR} = \cos VT - A \cdot \sin VL \quad (59)$$

$$y_{PR} = -B \cdot \sin VL \quad (60)$$

$$z_{PR} = \sin VT - C \cdot \sin VL \quad (61)$$

and

$$x_{QR} = -B \cdot \sin VT - A \cdot \sin VTV \quad (62)$$

$$y_{QR} = A \cdot \sin VT - C \cdot \cos VT - B \cdot \sin VTV \quad (63)$$

$$z_{QR} = \sin VT - C \cdot \sin VTV \quad (64)$$

The normal vector to the proposed waterline plane will then have the coordinates (A1,A2,A3) equal to

$$(x_{QR}, y_{QR}, z_{QR}) \times (x_{PR}, y_{PR}, z_{PR}) \quad (65)$$

This gives

$$A1 = y_{QR} \cdot z_{PR} - z_{QR} \cdot y_{PR} \quad (66)$$

$$B1 = z_{QR} \cdot x_{PR} - z_{PR} \cdot x_{QR} \quad (67)$$

$$C1 = x_{QR} \cdot y_{PR} - x_{PR} \cdot y_{QR} \quad (68)$$

The proposed waterline plane will now have the equation

$$A1 \cdot x + B1 \cdot y + C1 \cdot z + D1 = 0 \quad (69)$$

where D1 is found by insertion of (58) in (69).

After normalizing, the proposed waterline plane is given by

$$A2 \cdot x + B2 \cdot y + C2 \cdot z + D2 = 0 \quad (70)$$

where the coefficients A2,B2,C2 and D2 are found from A1,B1,C1 and D1 from division by

$$\sqrt{A1^2 + B1^2 + C1^2}$$

The procedure is changed a little, if equilibrium for a fixed heeling angle is required. In that case, we know from Appendix 3 that a rotation around the Q-axis corresponding to an angle of 1 has to be followed by a rotation around the P-axis corresponding to an angle of  $-\tan VK$  in order to keep the angle of heel, VH, unchanged. The equa-

tions (56) and (57) are then reduced to

$$WML = (RML - RMT \cdot \tan VK) \cdot VL \quad (71)$$

VL may be found directly from this equation, and, as previously mentioned,

$$VTV = -\tan VK \cdot VL \quad (72)$$

The metacentric heights for longitudinal and transverse inclination may sometimes be useful. For the longitudinal direction it is found by putting  $WML = 1$  and  $WMT = 0$ . When VL is found, the metacentric height,  $GM_L$ , is found from

$$GM_L = 1/VL \quad (73)$$

In a similar way the metacentric height for transverse inclination,  $GM_T$ , is found by putting  $WMT = 1$  and  $WML = 0$  and calculating VTV:

$$GM_T = 1/VTV \quad (74)$$

Again it is important to realize that these metacentric heights correspond to change of angles in the force coordinate system which follows the water surface.

The procedure to find the metacentric heights corresponding to the trim and heel angles are a little more complicated. For the metacentric height corresponding to heel it may be done in the following way:

A WMT-value corresponding to a small angle of VTV (f.inst. 2. deg.) and  $VL = 0$  is found from (57). This WMT-value is then used again together with  $WML = 0$  in (56) and (57) in order to find the corresponding values of VL and VTV. A new waterline plane is then found using the equations (59) to (70), and the additional value of the heel angle, DVH, may then be calculated from

$$DVH = \arctan(-B2/C2) - \arctan(-B/C) \quad (75)$$

The metacentric height corresponding to heel is then

$$GM_H = WMT/\sin(DVH) \quad (76)$$

The metacentric height corresponding to trim may be found in a similar way.

## Discussion

C. Kuo (University of Strathclyde, UK)

Very briefly I would like the authors to indicate what experience they have gained on this program. Further more, how do they link it with the total design procedure of ship. Do you think this is the way ahead

in the future application of damaged stability computation?

### Author's Reply

A damage stability calculation program

based on the described method is in use on a number of ships of the product-carrier type as a part of the KOCKUMATION loadmaster system. It is known that this has reduced the number of pre-calculated loading cases in the loading manual considerably.

A program like this may also be used as a part of the design procedure to ensure that the requirements of the authorities are fulfilled. This was however not the intention of the authors. The method has been developed in order to give the captain a possibility, not only of checking if the requirements are fulfilled for a certain loading condition, but also to ensure that the ship may survive after other non standard types of damages, if he feels that this may add to the safety of the ship. For such

cases there will be no fixed limits on the heel and trim angles in the final condition, as long as the ship does not sink or capsize, and it is therefore required that the method works also for large angles. Further, if a ship with an onboard computer for the calculation of damage stability has been damaged, it may be useful to know within a couple of minutes, whether the ship is bound to sink or capsize or if something may be done, which will keep it afloat, even in some extreme position. The possibility of onboard calculation of damage stability using a method, which can cope even with large angles of trim and keel, may therefore increase the safety of the ship. This is in the authors opinion the most important feature at the described method.

# PREDICTION OF MOTION OF SHIPS IN DAMAGED CONDITION IN WAVES

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## ABSTRACT

A theoretical method of predicting the motions of asymmetrically flooded ships in waves is presented. The method is developed by using strip theory for computing hydrodynamic coefficients. The theoretical prediction can be applied to any type of flooding conditions which would induce neutral heel, sinkage and trim. The motion is computed by solving fully coupled linear equations of motion for five degrees of freedom motion neglecting the surge motion. Computed results are presented for a ship-like body at various neutral heel angles and wave headings at the ship speeds of zero and five knots. One case of combined heel and trim condition is also computed. Although the computed results are limited to one ship-like geometry, it is found that a ship in a neutral heel condition is likely to undergo larger roll motion when subject to the waves incident from the opposite side of the heel than from the heeling side. It is hoped that model experiments to verify the present theory could be conducted in the future.

## NOMENCLATURE

A: incident wave amplitude  
g: gravitational acceleration  
j: imaginary unit,  $\sqrt{-1}$   
 $K_0$ : wave number ( $=\omega_0^2/g=2\pi/\lambda$ )  
L: ship length  
Oxyz: right-handed Cartesian coordinate system; see the definition in Section 2.1  
U: ship speed  
 $\alpha$ : trim angle  
 $\theta$ : neutral heel angle  
 $\lambda$ : incident wave length

$\mu$ : wave heading angle; head wave ( $\mu=\pi$ ), and beam wave from port to starboard ( $\mu=\pi/2$ )  
 $\xi_i(t)$ : displacement of ship in the  $i$ th mode of motion from equilibrium position; surge ( $i=1$ ), sway ( $i=2$ ), heave ( $i=3$ ), roll ( $i=4$ ), pitch ( $i=5$ ), and yaw ( $i=6$ ).  
 $\bar{\xi}_i$ : complex amplitude of the  $i$ th mode of motion  
 $\bar{\xi}_R$ : complex amplitude of relative motion  
 $\phi_i$ : complex velocity potential for incident wave ( $i=0$ ), radiated waves for the  $i$ th mode of motion for  $i=1,2,\dots,6$ , and diffracted waves ( $i=7$ ).  
 $\omega$ : wave encounter frequency  
 $\omega_0$ : incident wave frequency

## 1. INTRODUCTION

Ships are normally designed to possess sufficient restoring moments to right themselves when their equilibrium conditions are disturbed. International classification societies and governmental regulatory bodies of maritime nations have developed stringent requirements for minimum metacentric heights to ensure the safety of ships at sea. Nevertheless, despite the precautionary measures present in the ship design process, loss of life and ships due to capsizing in bad weather is still occurring. Several investigations in the past into incidents involving the capsizing of small ships have revealed that the IMCO criteria [1]\* may be inadequate [2,3].

\*Numbers in the bracket indicate References listed on page 12.



Thus, it appears that significant progress must be made before we attain an adequate understanding of the basic mechanisms affecting the stability of a ship in waves. A recent investigation by Morrall [4] into the causes of the loss of a large stern trawler, GAUL, demonstrates how difficult it is to pinpoint the exact causes of capsizing of a ship which was presumably designed to have a sufficient restoring moment.

Except for extreme cases, such as encountering large breaking waves [2], or the presence of peculiar stern wave conditions [5], it is more likely that most mishaps could have been inflicted by progressive flooding due to green water shippings on the decks. When a ship is partially flooded, or subjected to steady wind forces, its upright mean equilibrium position is shifted to a heeling position, and the static as well as the dynamic stability characteristics of the ship are altered. For most ships, unless the heeling angle is beyond a certain limit, the roll restoring moment increases with heeling angle. Therefore, from the standpoint of static stability one could speculate that a slight heeling could increase the roll stability. However, from the dynamic standpoint, an increase in the roll restoring moment causes a decrease in the roll natural period, and in turn, increases the probability of resonant roll excitation by shorter waves. Furthermore, a ship in an equilibrium heeling condition has an asymmetric athwartship underwater geometry which could significantly alter its hydrodynamic properties, and hence, its motion characteristics.

The purpose of the present study is to develop an analytical method for predicting the wave-excited motions of ships which have an equilibrium heel angle. Such a prediction method could be utilized to investigate the safety margins of ships with respect to sea conditions when the ship is forced to assume a neutral heeling position due to transverse asymmetric flooding.

Development of the analytical prediction method is based on the linear wave excitation-to-ship response theory formulated in the frequency domain. While it is quite probable that the roll response of a ship in heeling condition is nonlinear, it is assumed that a linear analysis provides most of the essential features of the motion characteristics. If deemed necessary, a future investigation incorporating nonlinear effects may be undertaken to extend the present theory.

The hydrodynamic coefficients in the equations of motion are obtained by strip theory. Due to the asymmetry of the underwater cross section geometry, the convenient decoupling of the vertical and horizontal plane motion, as applied in the usual case of an undamaged vessel, can no longer be assumed. Consequently, the number of hydrodynamic coefficients in the

equations of motion is significantly increased. Furthermore, the computation of sectional hydrodynamic quantities, such as added mass and damping, is more complicated than for the usual case involving symmetric geometric properties.

In the first part of the paper, the formulation of the equations of motion and the associated hydrodynamic coefficients are described. The second part is devoted to the presentation of computed motion results for a selected ship at various equilibrium heeling angles (including the upright condition) for various wave headings and ship speeds. Results are presented, along with pertinent discussions, followed by concluding remarks.

## 2. FORMULATION

### 2.1 Equations of Motion

Consider the motion of a slender body having arbitrary cross sections which are not necessarily symmetric about a center-plane moving in a free surface. It is assumed that the motion response of the body to an incident sinusoidal wave is linear. Thus, the motion of the body in irregular waves can be expressed as a linear superposition of the responses to each harmonic component of the waves. The fluid surrounding the body is assumed inviscid, incompressible and its motion irrotational. The depth of the water is infinite. No wind or current exists, and the body is either at zero or constant mean speed.

The frame of reference for body motion is a right-handed Cartesian coordinate system originating at the intersection of the calm waterplane and the vertical line passing through the center of gravity of the body in its mean equilibrium condition and translating with the body in a straight line at constant speed. The  $Ox$ -axis is aligned in the direction of the mean path, and the  $Oz$ -axis is directed vertically upward (see Fig. 1).

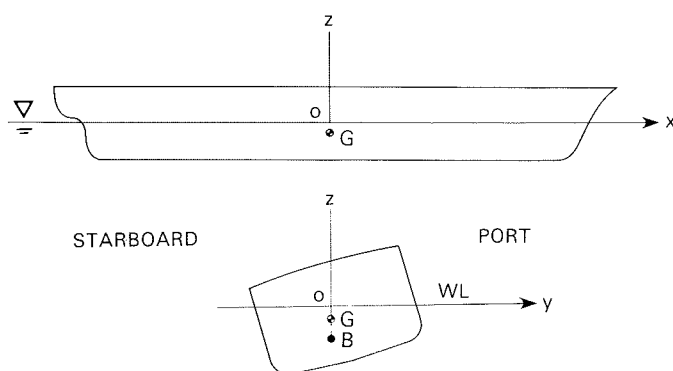


Fig. 1 Description of Coordinate System

Ships to be considered in this study are assumed to be symmetric about their centerplanes when they are intact. The asymmetry of the immersed cross sections is caused by heeling and trimming due to flooding of certain compartments. In the following formulation the weight of the flooded water is assumed to be part of the ship's weight. Also, as a result of the slender body assumption in the present study, the surge motion is assumed to be negligible compared to the other five modes of motion.

Under these assumptions, the equations of motion for a ship with asymmetrically immersed cross sections responding linearly in five-degrees-of-freedom to incident sinusoidal waves can be expressed as

$$\sum_{k=2}^6 [(M_{ik} + A_{ik}) \ddot{\xi}_k + B_{ik} \dot{\xi}_k + C_{ik} \xi_k] = F_i^{(e)} e^{-j\omega t} \quad \text{for } i=2,3,\dots,6 \quad (1)$$

where, the subscript  $i$  represents the five modes of motion, i.e., sway ( $i=2$ ), heave ( $i=3$ ), roll ( $i=4$ ), pitch ( $i=5$ ), and yaw ( $i=6$ ),  $M_{ik}$  the generalized body-mass matrix,  $B_{ik}$  the damping matrix,  $C_{ik}$  the restoring matrix,  $F_i^{(e)}$  the complex amplitude of the wave-exciting force or moment in the  $i$ th mode of motion,  $j$  the imaginary unit associated only with a harmonic-time function,  $\omega$  the wave-encounter frequency, and  $\xi_k$  the displacement of the ship from its mean position in the  $k$ th mode. The dots indicate the time derivatives of the variables.

If the center of gravity is located at  $(0,0,z_0)$ , the generalized body-mass matrix is given by

$$M_{ik} = \begin{bmatrix} M & 0 & -Mz_0 & 0 & 0 \\ 0 & M & 0 & 0 & 0 \\ -Mz_0 & 0 & I_x & -I_{xy} & -I_{xz} \\ 0 & 0 & -I_{yx} & I_y & 0 \\ 0 & 0 & -I_{zx} & 0 & I_z \end{bmatrix} \quad (2)$$

where  $M$  is the mass of the ship plus the mass of the flood water;

$$I_{x_i} = \iiint_V \rho_m (x,y,z) (r^2 - x_i^2) dv, \quad \text{for } i = 1,2,3$$

in which  $x_1=x$ ,  $x_2=y$ ,  $x_3=z$ ,  $\rho_m$  is the point mass-density of the body,  $r$  is the distance from the origin of the coordinate system,

$$I_{xy} = I_{yx} = \iiint_V \rho_m xy \, dv$$

$$I_{xz} = I_{zx} = \iiint_V \rho_m xz \, dv,$$

and the integration is performed over the mass distribution. For normal ships,  $I_{xy}$  and  $I_{xz}$  would be an order of magnitude smaller than the diagonal terms of the mass inertia; hence they will be neglected in the subsequent analysis.

While certain restoring coefficients do not exist, the non-vanishing  $C_{ik}$ 's are as follows:

$$\begin{aligned} C_{33} &= \rho g \iint_{S_w} ds & C_{34} &= \rho g \iint_{S_w} y \, ds \\ C_{43} &= C_{34} & C_{35} &= -\rho g \iint_{S_w} x \, ds \\ C_{44} &= \rho g \overline{GM} & C_{45} &= -\rho g \iint_{S_w} xy \, ds \\ C_{53} &= C_{35} & C_{54} &= C_{45} \\ C_{55} &= \rho g \overline{GM}_\ell \end{aligned} \quad (3)$$

where  $S_w$  is the waterplane area,  $\rho$  the mass density of water,  $g$  the gravitational acceleration,  $V$  the displaced volume of ship, and  $\overline{GM}$  and  $\overline{GM}_\ell$ , the transverse and the longitudinal metacentric height, respectively.

Since we assumed a linear relationship between wave-exciting forces and ship responses, we can formally express the ship motion as

$$\xi_k = \bar{\xi}_k e^{-j\omega t} \quad (4)$$

where  $\bar{\xi}_k$  represents the complex amplitude of the motion in the  $k$ th mode. Thus, expanding Equation (1) into the individual modes of motion, we have

$$\begin{aligned} (M+A_{22}) \ddot{\xi}_2 + B_{22} \dot{\xi}_2 + A_{23} \ddot{\xi}_3 + B_{23} \dot{\xi}_3 \\ + (A_{24} - Mz_0) \ddot{\xi}_4 + B_{24} \dot{\xi}_4 + A_{25} \ddot{\xi}_5 \\ + B_{25} \dot{\xi}_5 + A_{26} \ddot{\xi}_6 + B_{26} \dot{\xi}_6 \\ = F_2^{(e)} e^{-j\omega t} \end{aligned} \quad (5a)$$

$$\begin{aligned} A_{32} \ddot{\xi}_2 + B_{32} \dot{\xi}_2 + (M+A_{33}) \ddot{\xi}_3 + B_{33} \dot{\xi}_3 \\ + C_{33} \xi_3 + A_{34} \ddot{\xi}_4 + B_{34} \dot{\xi}_4 + C_{34} \xi_4 \\ + A_{35} \ddot{\xi}_5 + B_{35} \dot{\xi}_5 + C_{35} \xi_5 + A_{36} \ddot{\xi}_6 \\ + B_{36} \dot{\xi}_6 = F_3^{(e)} e^{-j\omega t} \end{aligned} \quad (5b)$$

$$\begin{aligned}
& (A_{42}-Mz_0)\ddot{\xi}_2 + B_{42}\dot{\xi}_2 + A_{43}\ddot{\xi}_3 + B_{43}\dot{\xi}_3 \\
& + C_{43}\xi_3 + (I_x+A_{44})\ddot{\xi}_4 + B_{44}\dot{\xi}_4 \\
& + C_{44}\xi_4 + A_{45}\ddot{\xi}_5 + B_{45}\dot{\xi}_5 + C_{45}\xi_5 \\
& + A_{46}\ddot{\xi}_6 + B_{46}\dot{\xi}_6 = F_4^{(e)}e^{-j\omega t} \quad (5c)
\end{aligned}$$

$$\begin{aligned}
& A_{52}\ddot{\xi}_2 + B_{52}\dot{\xi}_2 + A_{53}\ddot{\xi}_3 + B_{53}\dot{\xi}_3 \\
& + C_{53}\xi_3 + A_{54}\ddot{\xi}_4 + B_{54}\dot{\xi}_4 + C_{54}\xi_4 \\
& + (I_y+A_{55})\ddot{\xi}_5 + B_{55}\dot{\xi}_5 + C_{55}\xi_5 \\
& + A_{56}\ddot{\xi}_6 + B_{56}\dot{\xi}_6 = F_5^{(e)}e^{-j\omega t} \quad (5d)
\end{aligned}$$

$$\begin{aligned}
& A_{62}\ddot{\xi}_2 + B_{62}\dot{\xi}_2 + A_{63}\ddot{\xi}_3 + B_{63}\dot{\xi}_3 \\
& + A_{64}\ddot{\xi}_4 + B_{64}\dot{\xi}_4 + A_{65}\ddot{\xi}_5 + B_{65}\dot{\xi}_5 \\
& + (I_z+A_{66})\ddot{\xi}_6 + B_{66}\dot{\xi}_6 = F_6^{(e)}e^{-j\omega t} \quad (5e)
\end{aligned}$$

## 2.2 Hydrodynamic Coefficients

The hydrodynamic coefficients appearing in Equations (5) are the added masses,  $A_{ik}$ , damping coefficients,  $B_{ik}$ , and wave-exciting forces,  $F_i^{(e)}$ . These hydrodynamic coefficients can be obtained in terms of the velocity potential

$$\phi(x, y, z, t) = \text{Re}\{\phi(x, y, z)e^{-j\omega t}\} \quad (6)$$

where  $\text{Re}$  indicates the real part of a complex function. The complex potential,  $\phi$ , can be further decomposed into

$$\phi = \sum_{i=0}^7 \phi_i \bar{\xi}_i \quad (7)$$

Here,  $\phi_0$ , the complex potential representing the incident regular waves, can be expressed as

$$\phi_0 = -\frac{jgA}{\omega_0} e^{jK_0(x\cos\mu - y\sin\mu) + K_0 z} \quad (8)$$

where  $A$  is the amplitude,  $\omega_0$  the frequency, and  $K_0 = \omega_0^2/g$  the wave number,  $\mu$  the wave heading angle with  $\mu=\pi$  indicating the head waves;  $\phi_7$  is the complex potential representing diffracted waves, and is related to  $\phi_0$  by

$$\phi_{7n} = -\phi_{0n}$$

on the hull surface  $S_0$  under the calm waterplane at the mean equilibrium position of the ship; the subscript  $n$  indicates a normal derivative;  $\phi_i$  ( $i=1,2,\dots,6$ ) is

the complex potential representing the fluid disturbances created by the  $i$ th mode of motion;  $\bar{\xi}_0=\bar{\xi}_7=1$ ; and  $\bar{\xi}_1=0$ , i.e., zero surge motion.

As shown by Lee [6], we can define the added mass and damping coefficients in terms of these potentials by

$$A_{ik} = \text{Re} \left\{ -\frac{\rho}{\omega^2} \iint_{S_0} H_i(x, y, z) \phi_k ds \right\} \quad (9)$$

$$B_{ik} = \text{Im} \left\{ -\frac{\rho}{\omega} \iint_{S_0} H_i \phi_k ds \right\} \quad (10)$$

where  $\text{Im}$  indicates the imaginary part, and

$$H_i = \phi_{in} + \frac{2U}{j\omega} \phi_{3n\delta_{i5}} - \frac{2U}{j\omega} \phi_{2n\delta_{i6}} \quad (11)$$

where  $U$  is the constant forward speed of the ship and  $\delta_{ik}$  is Kronecker's delta. The wave-exciting forces are given by

$$F_i^{(e)} = \rho \iint_{S_0} \left\{ j\omega_0 n_i + H_i \frac{\partial}{\partial n} \right\} \phi_0 ds \quad (12)$$

where  $n_i$  (for  $i=1,2,3$ ) represents the  $i$ th component of a unit normal vector directed into  $S_0$ ; and  $n_i$  (for  $i=4,5,6$ ) is the  $i$ th component of the vector  $(\vec{r} \times \vec{n})_{i-3}$  for the position vector  $\vec{r}$ . In deriving the foregoing expressions, the so-called Haskind-Newman relation [7] was used.

For the purpose of evaluating Equations (9) through (12), the velocity potentials,  $\phi_i$ , for  $i=1,2,\dots,7$  are considered to be two-dimensional; i.e.,  $\phi_i$  is a function of  $y$  and  $z$  for any given section located at  $x$ . This approach is well known as strip theory. Thus, an integral such as  $\iint_{S_0} f(x, y, z) ds$ , for any arbitrary function,  $f$ , can be approximated by

$$\iint_{S_0} f(x, y, z) ds \approx \int_L dx \int_{C(x)} f(y, z; x) d\ell$$

where  $\int_L dx$  indicates the integral over the ship length;  $\int_{C(x)} d\ell$  indicates the integral along the immersed contour of the cross section located at  $x$  at the mean position of the body; and  $f(y, z; x)$  represents an approximation of  $f(x, y, z)$  by treating  $x$  as a parameter rather than a variable. Accordingly, the added mass coefficient,  $A_{ik}$ , can be approximated by

$$A_{ik} = \int_L a_{ik}(x) dx$$

where  $a_{ik}(x)$  is the sectional added mass obtained under the assumption of two-dimensional flow at station  $x$ .

In this study the velocity potentials  $\phi_i(x, y, z)$  for  $i=2,\dots,7$  are approximated by

the two-dimensional potentials  $\tilde{\phi}_i(y, z; x)$  which are obtained by the method of source distribution over the immersed contour  $C(x)$ . That is,

$$\tilde{\phi}(y, z; x) = \int_{C(x)} Q(\ell) G(\ell; y, z) d\ell \quad (13)$$

where  $Q$  is the unknown strength of the source,  $G$ , which is given by (see, e.g., pp. 479-481 of Wehausen & Laitone [8])

$$G(\eta, \zeta; y, z) = \frac{1}{2\pi} \left[ \log \frac{(y-\eta)^2 + (z-\zeta)^2}{(y-\eta)^2 + (z+\zeta)^2} - 2 \int \frac{e^{k(z+\zeta)} \cos k(y-\eta)}{k - K} dk \right] - j e^{K(z+\zeta)} \sin K(y-\eta) \quad (14)$$

where  $K$  is the wave number based on the encounter frequency,  $\omega$ , and  $f$  indicates a Cauchy principal-value integral.

A numerical procedure for obtaining the unknown source strength  $Q$  is given, for example, by Frank [9]. The expressions for  $A_{ik}$  and  $B_{ik}$  in terms of the sectional added mass,  $a_{ik}$ , and damping,  $b_{ik}$ , are given in Appendix I, together with the expressions for the wave-exciting force,  $F_i^{(e)}$ .

The foregoing description is given under the assumption of ideal fluid. However, it is well known that the roll motion cannot be properly predicted near its resonant frequency without including viscous damping effects in the equations of motion. In the present study the viscous damping effects are included in  $B_{ik}$  and  $F_i^{(e)}$  following the method given by Lee and Curphey [10]. The method in [10] is based on the empirical analysis of Thwaites [11] in which the viscous side force per unit length at the station  $x$  of a slender body with a yaw angle of  $\beta$  is given by

$$F_s = \frac{1}{2} \rho H(x) (U^2 a_0 \beta + C_D v|v|)$$

where  $a_0$  is often called the viscous-lift coefficient and  $C_D$  is the cross-flow drag coefficient;  $H(x)$  is the local draft of the body; and  $v$  is the relative fluid velocity with respect to the body at the station  $x$  and given by

$$v = U(\beta + \xi_6)$$

$$\beta = -\xi_6 + (\dot{\xi}_2 + x\dot{\xi}_6 + d_2\dot{\xi}_4 - \dot{\xi}_H)/U,$$

$d_2$  = one-half the draft at the station  $x$

$\dot{\xi}_H$  = transverse fluid velocity at  $z = -d_2$  due to the incident wave

The nonlinear term  $v|v|$  is approximated by

$$v|v| \approx \frac{8}{3\pi} v_0 v$$

where  $v_0$  is the amplitude of  $v$ . This approximation is often called the equiline- arization method, and is frequently used for a dynamic system with weakly nonlinear behavior.

The viscous lift force per unit length at the station  $x$  due to an angle of attack  $\alpha$  can be obtained similarly by

$$F_v = \frac{1}{2} \rho B(x) (U^2 a_0 \alpha + C_D w|w|)$$

where  $B(x)$  is the local beam of the body,

$$w = U(\alpha - \xi_5),$$

$$\alpha = \xi_5 + (\dot{\xi}_3 - x\dot{\xi}_5 - \dot{\xi}_V)/U$$

$\dot{\xi}_V$  = vertical fluid velocity at  $z = -d_2$  due to the incident wave.

The moments arising from  $F_s$  and  $F_v$  are also obtained by introducing proper moment arms.

The coefficients associated with the velocities  $\xi_i$  are lumped with the corresponding  $B_{ik}$ , and the coefficients associated with  $\dot{\xi}_H$  and  $\dot{\xi}_V$  are lumped with the corresponding  $F_i^{(e)}$ .

### 3. RESULTS AND DISCUSSIONS

The body plan of a ship selected for sample calculations is shown in Figure 2. It is a modified Mariner Class ship with the parallel middle body extending to sixty percent of the ship length. The principal characteristics of this ship are presented in Table 1.

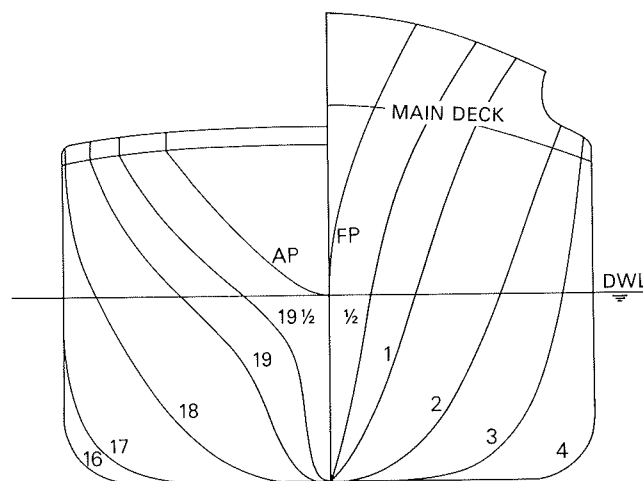


Fig. 2 Body Plan

Table 1 - Principle Characteristics of Ship  
at Intact Condition

Length, LBP	161.0 m (528.0 ft)
Breadth at midship, B	23.2 m (76.0 ft)
Draft at midship	8.2 m (27.0 ft)
Displacement	24166.0 long tons
Block coefficient, $C_B$	0.80
Pitch radius of gyration	0.247 LBP
Yaw radius of gyration	0.247 LBP
Transverse radius of gyration	0.354 B
Metacentric height, GM	2.09 m (6.86 ft)
Longitudinal center of buoyancy	82.3 m (270.0 ft) aft of F.P.
Vertical center of buoyancy	4.3 m (14.0 ft) above the keel

Motion calculations were carried out for the intact condition at a design waterline for several typical wave headings, i.e.,  $\mu = 90^\circ$  (beam waves from port to starboard),  $135^\circ$ ,  $180^\circ$  (head waves),  $225^\circ$  and  $270^\circ$  (beam waves from starboard to port) for both 0 and 5 knots forward speed. Similar calculations were then performed for the neutral heel angles of 5, 10 and 20 degrees. It is assumed that heeling of the ship is caused by a complete flooding of a starboard side wing compartment at mid-ship, without the creation of an internal free surface. In this study, the flooded water is treated as ballast.

If we assume that the center of the flooded water is located toward starboard at transverse distance  $h$  from the longitudinal centerplane, then the added water weight,  $w$ , due to the starboard flooding can be obtained by

$$w = \frac{\Delta \overline{GM} \sin \theta}{h \cos \theta - \overline{GM} \sin \theta} \quad (15)$$

where  $\Delta$  is the intact displacement of the ship, and  $\theta$  is the neutral heel angle. The transverse center of gravity,  $G_t$ , measured from the longitudinal centerplane at the equilibrium heel angle is given by

$$G_t = \frac{wh \cos \theta}{\Delta + w} \quad (16)$$

In order to simplify the present calculations,  $h$  is taken to be  $B/4$ , where  $B$  is the beam of the ship. In order to examine the effect of both initial heel and trim, calculations are made for 10 degrees of heel in combination with 2 degrees of trim by bow.

In Figure 3 immersed midship cross sections at  $\theta = 5, 10$  and  $20$  degrees are shown together with the location of the centers of buoyancy and gravity. In Table 2, the principal characteristics of the ship at the heel angles of 5, 10 and 20 degrees are presented.

It should be understood that the purpose of presenting the computed results for a chosen phantom ship in this paper is to demonstrate the typical computations of practical interest which can be performed by the analytical method presented in this paper.

The static stability characteristics of the ship are presented in Figure 4 by the curve of righting moment arm  $\overline{GZ}$  ( $= \overline{GM} \sin \theta$ ) versus heel angle  $\theta$ . The calculation is based on the intact condition of the ship.

For a symmetric ship in an upright condition and subject to regular head waves, one normally would not expect an occurrence of roll motion. However, if the ship is asymmetrically flooded, there is no reason to rule out an occurrence of roll in head waves because the ship no longer has the symmetric hydrodynamic properties. In Figure 5 the nondimensional roll amplitudes,  $\xi_4 = |\xi_4| / (K_0 A)$ , in heeled conditions is shown against the nondimensional wave length,  $\lambda/L$ , where  $L$  is the ship length. The results are given for the neutral heel angles  $\theta = 5, 10$  and  $20$  degrees at the ship speeds of 0 and 5 knots. The results show that the roll amplitudes increase as the neutral heel angle increases and as the neutral heel angle decreases, the resonant wave length increases. The values of the viscous coefficients,  $a_0$  and  $C_D$ , used in the present computations are 0.03 and 0.5, respectively. The maximum roll amplitude at  $\theta = 20$  degrees and  $U=0$  is about 9.7 degrees per meter of the incident wave amplitude. Thus, if the ship encounters, for example, head-on swell-like waves of about 230 m in length ( $\approx 1.5L$ ) and 4 m in amplitude when she is at a neutral heel angle of 20 degrees in stationary floating condition, the maximum heel to the starboard can be as large as 60 degrees. From Figure 4 we can observe that the ship may still possess a sufficient restoring moment to right herself.

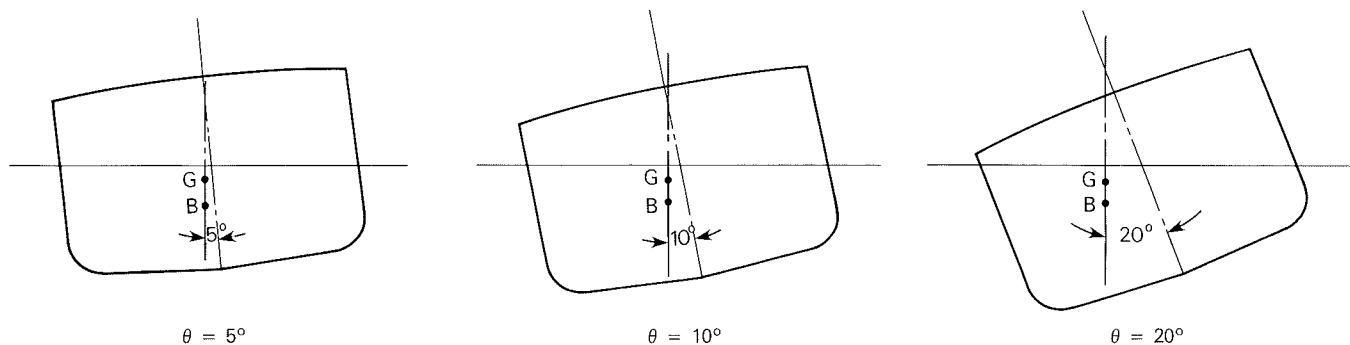


Fig. 3 Midship Sections at Various Neutral Heel Angles

Table 2 - Principle Characteristics of Ship at Neutral Heel (Starboard Side) and Trim (Bow Down) Conditions

		$\theta=5^\circ$ $\alpha=0^\circ$	$\theta=10^\circ$ $\alpha=0^\circ$	$\theta=20^\circ$ $\alpha=0^\circ$	$\theta=10^\circ$ $\alpha=2^\circ$
LBP,	m (ft)	161.0 (528.0)	161.0 (528.0)	161.0 (528.0)	161.0 (528.0)
Max. Beam,	m (ft)	23.3 (76.3)	23.5 (77.2)	24.7 (80.9)	23.5 (77.2)
Max. Draft,	m (ft)	9.2 (30.2)	10.1 (33.2)	11.7 (38.3)	11.9 (38.9)
$C_B$		0.734	0.679	0.594	0.680
Displacement, long tons		24848	25587	27054	25667
GM,	m (ft)	1.91 (6.27)	1.88 (6.15)	2.14 (7.03)	1.63 (5.36)
TCB*,		-0.47 (-1.54)	-0.91 (-2.98)	-1.78 (-5.83)	-0.97 (-3.17)
VCB**,	m (ft)	4.42 (14.50)	4.60 (15.09)	5.08 (16.65)	4.34 (14.23)
LCB†,	m (ft)	82.3 (270.1)	82.3 (270.1)	82.6 (271.0)	75.0 (246.1)

\* Transverse distance measured from the centerplane at intact condition; the minus sign means toward the starboard.  
 \*\* Distance above the keel.  
 † Distance aft of F.P.

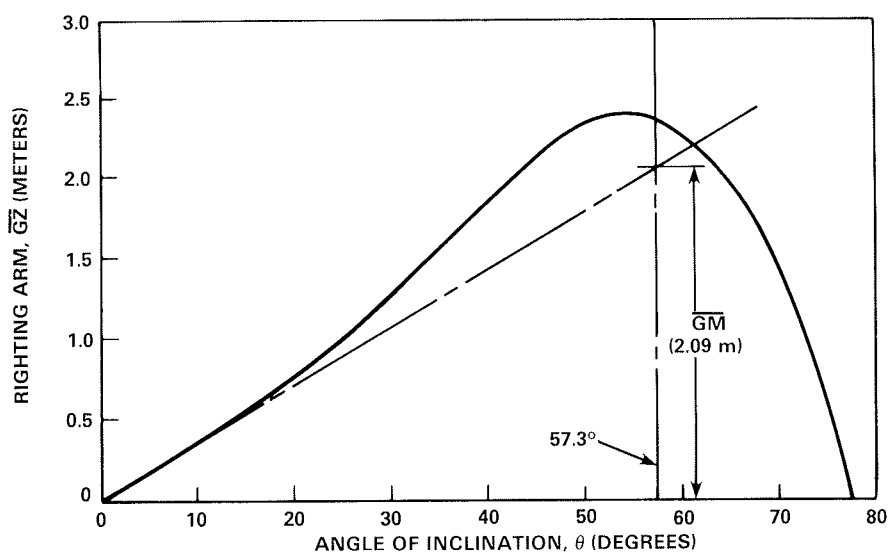


Fig. 4 Righting Moment Arm Versus Heel Angle

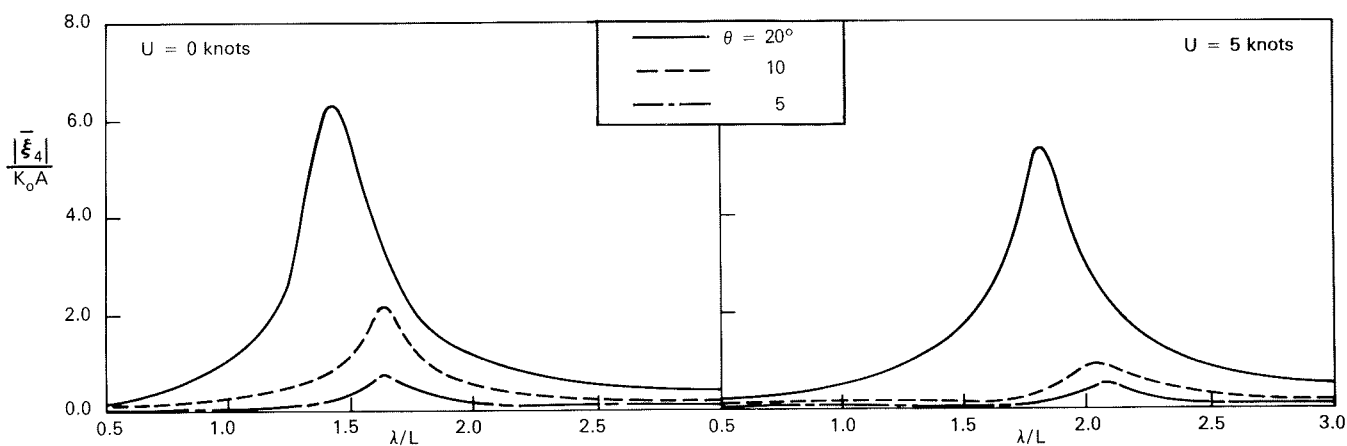


Fig. 5 Roll Amplitudes in Head Waves ( $\mu = 180^\circ$ ) at Various Neutral Heel Angles at Ship Speeds of 0 and 5 Knots

In Figure 6, the results of  $\xi_4$  for  $\theta = 5$  degrees are presented. The beam and bow-quartering wave headings from the port and starboard sides are shown for  $U=0$  and 5 knots. For comparison sake, the corresponding results for the upright condition ( $\theta=0$ ) are also shown. The wave headings  $\mu = 90$  and  $135$  degrees represent the waves incident from the port side in the beam and bow-quartering directions, respectively, and  $\mu = 270$  and  $225$  degrees represent the corresponding wave headings from the starboard side. It is interesting to note in Figure 6 that the maximum roll

amplitude for  $\mu = 90$  degrees is significantly greater than that for  $\mu = 270$  degrees. That is, when the ship has a neutral heel angle toward the starboard, the beam waves incident from the port side excite larger roll oscillation than the beam waves from the starboard do, and vice versa. Similar trends hold for bow-quartering waves but with a lesser degree. The results also indicate that the effect of ship speed on the reduction in roll amplitude is more pronounced for the heeled condition than the upright condition.

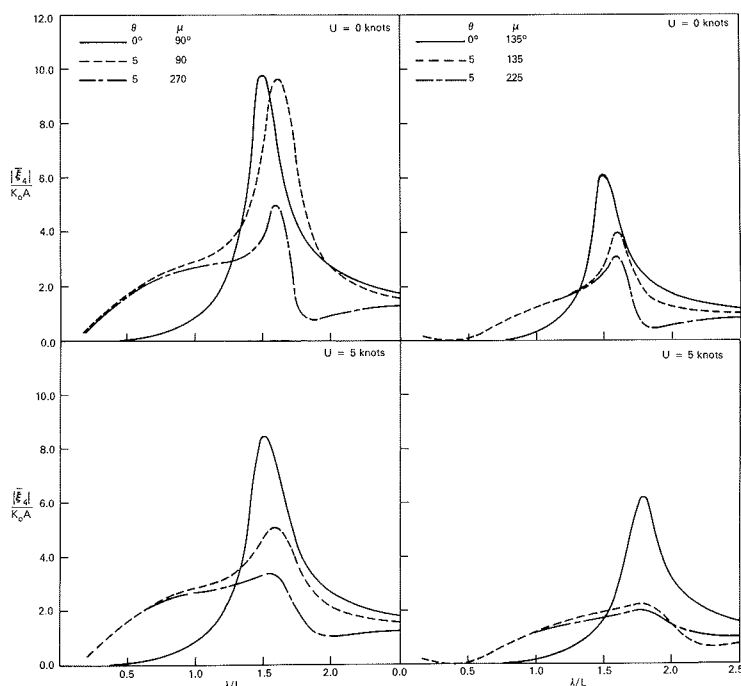


Fig. 6 Roll Amplitudes in Beam and Bow-Quartering Waves at Neutral Heel Angle of 5 Degrees at Ship Speeds of 0 and 5 Knots

Similar results for  $\theta = 10$  and  $20$  degrees are presented in Figures 7 and 8, respectively. From Figure 7, one can observe that in the port-to-starboard beam waves, the heeled condition has larger maximum roll amplitudes than the upright condition does at both 0 and 5 knots, whereas the same is not true for the bow-quartering waves. For  $\theta = 20$  degrees the results shown in Figure 8 indicate that the greater maximum roll occurs for the heeled condition when the waves are incident from the lee side of the heeling; however, with the magnitudes of the roll amplitudes at their peak values shown in the figure, the ship may no longer be able to keep itself from capsizing for even moderate sea conditions.

When a ship is flooded, it is more likely that the ship would assume a new equilibrium position with a neutral heel angle accompanied by sinkage and trim. In Figure 9, the roll amplitudes for the neutral heel angle of  $10$  degrees toward the starboard and the bow-down trim angle  $\alpha = 2$  degrees are shown for  $\mu = 90, 135, 225$ , and  $270$  degrees and  $U=0$ . The resulting new displacement is increased by  $819$  long tons from the intact condition. For comparison sake, the results for the condition of  $\theta = 10$  degrees and no trim ( $\alpha=0$ ) are also shown in Figure 9. One can observe that the roll-resonant wave length increases gradually as the ship takes a neutral heel and, then, a trim. Computations similar to the one shown in Figure 9 can be made for any combination of neutral heel, trim and sinkage for roll motion as well as the other modes of motion.

It is interesting to examine if a neutral heel and trim would have a significant effect on the heave and pitch motion characteristics. In Figure 10, the amplitudes of heave and pitch motions which are nondimensionalized by  $A$  and  $K_0 A$ , respectively, are shown for the upright condition and the combined starboard heel ( $\theta = 10$  degrees) and bow-down trim ( $\alpha = 2$  degrees) condition in head waves. The trends of the heave and pitch amplitudes are about the same for both conditions. The heel and trim condition shows a slightly increased pitch motion for the wave lengths greater than the ship length. It appears that the effect of a trim is greater on roll motion than on heave and pitch motions.

One of the critical measures of sea-keeping qualities of a ship is its vertical motion relative to the free-surface waves. This motion is often referred to as relative motion. The relative motion is a criterion for assessing the deck wetness, bottom slamming, or propeller and rudder emergence. For instance, if a ship encountering a swell has the relative motion amplitude greater than the free board or the keel draft in the bow region, it can be anticipated that the ship would experience deck wetness or keel emergence. The relative motion  $\bar{\xi}_R$  given in the form of complex amplitudes is defined by

$$\bar{\xi}_R(x,y) = \bar{\xi}_3 + y\bar{\xi}_4 - x\bar{\xi}_5 - \zeta(x,y) \quad (17)$$

where  $\zeta$  is the free-surface elevation which is assumed to be the same as the undisturbed incident wave given by

$$\zeta = A e^{jK_0(x\cos\mu - y\sin\mu)} \quad (18)$$

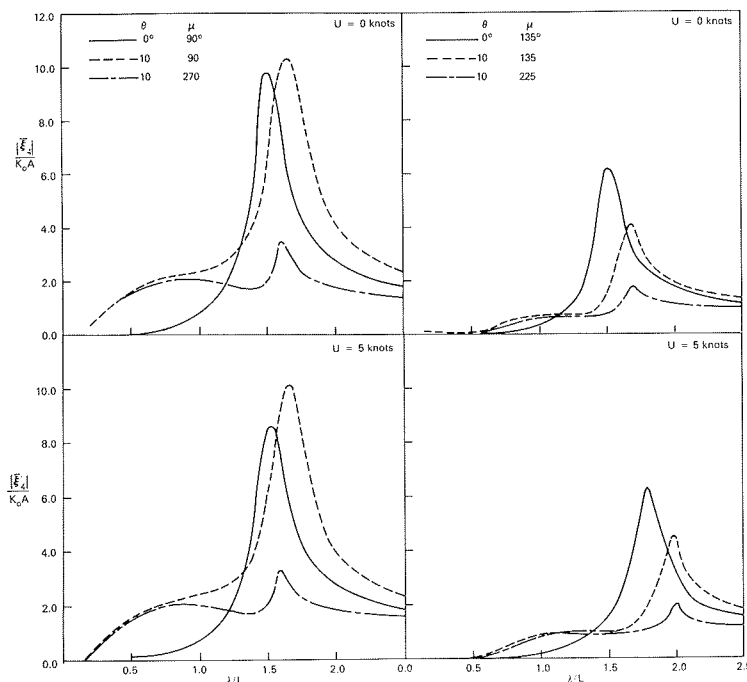


Fig. 7 Roll Amplitudes in Beam and Bow-Quartering Waves at Neutral Heel Angle of 10 Degrees at Ship Speeds of 0 and 5 Knots



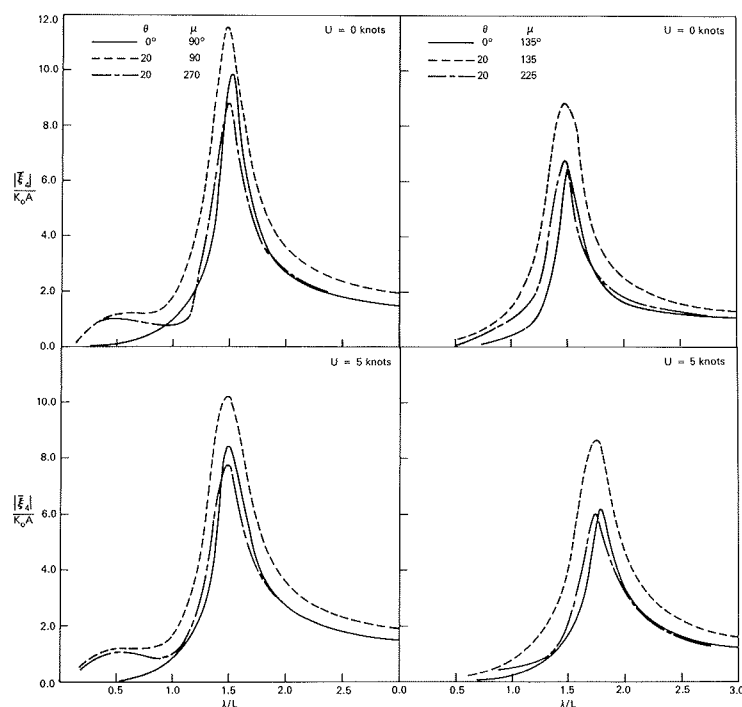


Fig. 8 Roll Amplitudes in Beam and Bow-Quartering Waves at Neutral Heel Angle of 20 Degrees at Ship Speeds of 0 and 5 Knots

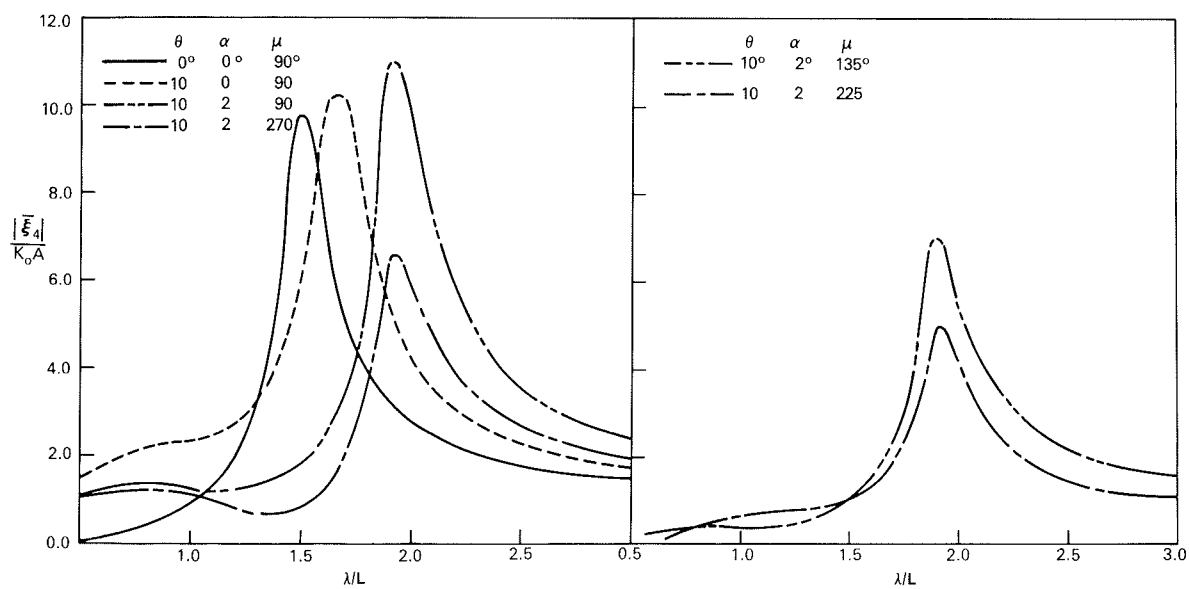


Fig. 9 Roll Amplitudes in Beam and Bow-Quartering Waves at Neutral Heel Angle and Trim at Zero Speed

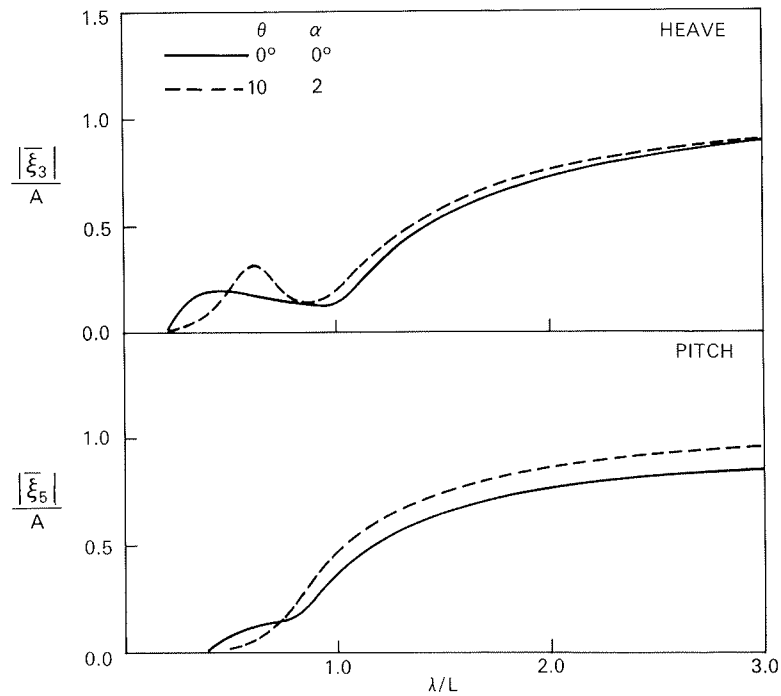


Fig. 10 Heave and Pitch Amplitudes in Head Waves at Neutral Heel and Trim at Zero Speed

In Figure 11 the amplitudes of the relative motion at the one-tenth of ship length after the fore peak (Station 2) is shown for the intact condition and the combined heel and trim condition ( $\theta = 10$  degrees to starboard and  $\alpha = 2$  degrees bow-down). The results are shown for  $\mu = 135$  and  $225$  degrees for the points on the port and starboard deck edges when the ship speed is zero. The double-peak behavior of the curves are the results of the

contribution from the heave and pitch motion for the first peak in the shorter wave lengths and that from the roll for the second peak in the longer wave lengths. The changes in the wave lengths for the second peaks between the intact condition and the heel and trim condition are directly related to the shift in the wave length at which the maximum roll occurs.

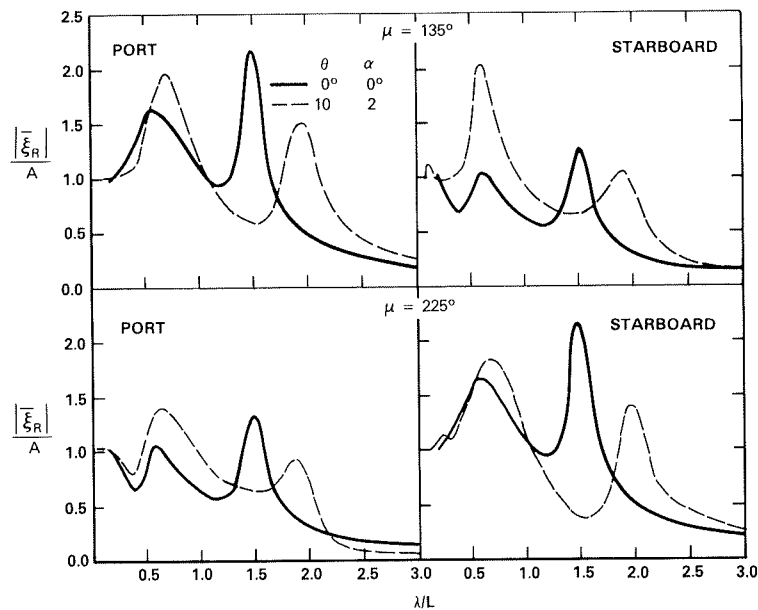


Fig. 11 Relative Motion Amplitudes at Station 2 in Bow-Quartering Waves at Neutral Heel and Trim at Zero Speed

#### 4. CONCLUDING REMARKS

From the present theoretical work, the following findings were made:

1. A ship in a neutral heel condition due to an asymmetric flooding can be excited to a large roll motion by head waves.

2. For small neutral heel angles, the roll amplitudes could be reduced; however, the roll amplitudes could be increased for large neutral heel angles.

3. For a ship in a neutral heel condition, the waves incident from the opposite direction to the heel could excite larger roll motion than the waves incident from the heeling side.

4. If a flooding results in a large trim in addition to a heel, the roll motion characteristics can be further altered by the trim; however, the effect of a trim on the heave and pitch motions would be less than that on the roll motion.

#### ACKNOWLEDGEMENTS

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#### APPENDIX I - EXPRESSIONS FOR $A_{ik}$ AND $B_{ik}$

From Equations (9) and (10), we have

$$A_{ik} = \operatorname{Re}_j \left[ -\frac{\rho}{\omega^2} \iint_{S_0} \left( \phi_{in} + \frac{2U}{j\omega} \phi_{3n\delta i5} - \frac{2U}{j\omega} \phi_{2n\delta i6} \right) \phi_k ds \right]$$

$$B_{ik} = \operatorname{Im}_j \left[ -\frac{\rho}{\omega} \iint_{S_0} \left( \phi_{in} + \frac{2U}{j\omega} \phi_{3n\delta i5} - \frac{2U}{j\omega} \phi_{2n\delta i6} \right) \phi_k ds \right]$$

$$\text{where } \iint_{S_0} ds = \int_L dx \int_{C(x)} d\ell$$

Using the strip approximation, we assume that at given  $x$  the velocity potential,  $\phi_i$ 's, are functions of only  $y$  and  $z$ . Then at the station  $x$ , we can define sectional added mass  $a_{ik}$  and damping  $b_{ik}$  by

$$a_{ik} = \operatorname{Re}_j \left[ -\frac{\rho}{\omega} \int_{C(x)} \phi_{in} \phi_k d\ell \right] \quad (I.1)$$

$$b_{ik} = \operatorname{Im}_j \left[ -\frac{\rho}{\omega} \int_{C(x)} \phi_{in} \phi_k d\ell \right] \quad (I.2)$$

In terms of  $a_{ik}$  and  $b_{ik}$  we can express the added mass and damping coefficients as follows

$$A_{22} = \int_L a_{22}(x) dx$$

$$B_{22} = \int b_{22} dx$$

$$A_{23} = \int a_{23} dx$$

$$B_{23} = \int b_{23} dx$$

$$A_{24} = \int a_{24} dx$$

$$B_{24} = \int b_{24} dx$$

$$A_{25} = -\int xa_{23} dx - \frac{U}{\omega^2} B_{23}$$

$$B_{25} = -\int xb_{23} dx + UA_{23}$$

$$A_{26} = \int xa_{22} dx + \frac{U}{\omega^2} B_{22}$$

$$B_{26} = \int xb_{22} dx - UA_{22}$$

$$A_{32} = A_{23}$$

$$B_{32} = B_{23}$$

$$A_{33} = \int a_{33} dx$$

$$B_{33} = \int b_{33} dx$$

$$A_{34} = \int a_{34} dx$$

$$B_{34} = \int b_{34} dx$$

$$A_{35} = -\int xa_{33} dx - \frac{U}{\omega^2} B_{33}$$

$$B_{35} = -\int xb_{33} dx + UA_{33}$$

$$A_{36} = -A_{25}$$

$$B_{36} = -B_{25}$$

$$A_{42} = A_{24}$$

$$B_{42} = B_{24}$$

$$A_{43} = A_{34}$$

$$B_{43} = B_{34}$$

$$A_{45} = -\int xa_{34} dx - \frac{U}{\omega^2} B_{34}$$

$$B_{45} = -\int xb_{34} dx + UA_{34}$$

$$A_{46} = \int xa_{24} dx + \frac{U}{\omega^2} B_{24}$$

$$B_{46} = \int xb_{24} dx - UA_{24}$$

$$A_{52} = -\int xa_{23} dx + \frac{U}{\omega^2} B_{23}$$

$$B_{52} = -\int xb_{23} dx - UA_{23}$$

$$A_{53} = -\int xa_{33} dx + \frac{U}{\omega^2} B_{33}$$

$$B_{53} = -\int xb_{33} dx - UA_{33}$$

$$A_{54} = -\int xa_{34} dx - \frac{U}{\omega^2} B_{34}$$

$$B_{54} = -\int xb_{34} dx - UA_{34}$$

$$A_{55} = -\int x^2 a_{33} dx + \frac{U^2}{\omega^2} A_{33}$$

$$B_{55} = -\int x^2 b_{33} dx + \frac{U^2}{\omega^2} B_{33}$$

$$A_{56} = -\int x^2 a_{23} dx - \frac{U^2}{\omega^2} A_{23}$$

$$B_{56} = -\int x^2 b_{23} dx - \frac{U^2}{\omega^2} B_{23}$$

$$A_{62} = \int xa_{22} dx - \frac{U}{\omega^2} B_{22}$$

$$B_{62} = \int xb_{22} dx - UA_{22}$$

$$A_{63} = -A_{52}$$

$$B_{63} = -B_{52}$$

$$A_{64} = \int xa_{24} dx - \frac{U}{\omega^2} B_{24}$$

$$B_{64} = \int xb_{24} dx + UA_{24}$$

$$A_{65} = A_{56}$$

$$B_{65} = B_{56}$$

$$A_{66} = \int x^2 a_{22} dx + \frac{U^2}{\omega^2} A_{22}$$

$$B_{66} = \int x^2 b_{22} dx + \frac{U^2}{\omega^2} B_{22}$$

$$F_2^{(e)} = - \frac{j\rho g A}{\omega_0} \int_L dx e^{jK_0 \cos \mu}$$

$$\int_{C(x)} \left\{ j\omega_0 N_2 + K_0 (-jN_2 \sin \mu + N_3) \phi_2 \right\} e^{K_0 (z - j y \sin \mu)} d\ell$$

$$F_3^{(e)} = - \frac{j\rho g A}{\omega_0} \int_L dx e^{jK_0 \cos \mu}$$

$$\int_{C(x)} \left\{ j\omega_0 N_3 + K_0 (-jN_2 \sin \mu + N_3) \phi_3 \right\} e^{K_0 (z - j y \sin \mu)} d\ell$$

$$F_4^{(e)} = - \frac{j\rho g A}{\omega_0} \int_L dx e^{jK_0 x \cos \mu}$$

$$\int_{C(x)} \left\{ j\omega_0 (y N_3 - z N_2) \right\}$$

$$+ K_0 (-jN_2 \sin \mu + N_3) \phi_4 \left\{ \right.$$

$$e^{K_0 (z - j y \sin \mu)} d\ell$$

$$F_5^{(e)} = - \frac{j\rho g A}{\omega_0} \int_L dx e^{jK_0 x \cos \mu}$$

$$\int_{C(x)} \left\{ -j\omega_0 x N_3 + K_0 (-jN_2 \sin \mu + N_3) \left( \frac{U}{j\omega} - x \right) \phi_3 \right\} e^{K_0 (z - j y \sin \mu)} d\ell$$

$$F_6^{(e)} = - \frac{j\rho g A}{\omega_0} \int_L dx e^{jK_0 x \cos \mu}$$

$$\int_{C(x)} \left\{ j\omega_0 x N_2 - K_0 (-jN_2 \sin \mu + N_3) \left( \frac{U}{j\omega} - x \right) \phi_2 \right\} e^{K_0 (z - j y \sin \mu)} d\ell$$

where  $N_2$  and  $N_3$  are respectively the  $y$ - and  $z$ -component of the unit normal vector in the  $y$ - $z$  plane.

## Discussion

Y. TAKAISHI (Ship Research Institute, Japan)

It is very interesting to see the calculated results on responses of inclined ship in waves. In SRI, a model experiment of inclined states ( $\pm 10$  degree) which simulate the cargo shift condition has recently carried out. [T. Haraguchi, 1982]\* In this case, the rolling period became shorter for the inclined condition because of increased GM value and the amplitude became smaller.

As to the relative wave elevations measured at midship part showed similar tendency as shown by the author in Fig. 11, qualitatively, i.e. the relative motion at lee side showed almost the same response characteristics in both wave directions, and this was also true for the weatherside. But slight difference appeared in short wave length range where the effects of the diffraction of waves by the ship are presumed to become rather large.

The relative wave elevations at short wave length range is very important from the viewpoint of shipping water as shown both by this calculation and by the experiment of SRI.

Therefore, the diffraction effect should be also taken into account for estimating the relative wave elevations induced on the inclined ship.

\* Report of Ship Research Institute  
(to be published)

K. Kudo (University of Tokyo, Japan)

Thank you Dr. Lee, for your most inspiring paper. I was especially interested in your result that for a ship in a neutral heel condition, the large roll motion can be excited by head waves, and the roll amplitudes depend strongly on the direction of the incident waves.

Now I have a question to ask. Your calculation generally shows that for a ship heeled starboard larger roll amplitudes are excited by the port-to-starboard waves, and this result is reasonable judging from the experience on the wave power devices such as Salter duck. However, in the combined heel and trim condition, as Fig. 9 shows, such relation is reversed, and I would like to ask further comments on this matter.

Setting aside this question, I should like to comment here on the steady turning moment. As your paper shows, the turning direction of a ship heeled in head waves has a great influence upon the rolling motion and then the capsizing. The steady turning moment may be generated by various causes, but the gyroscopic moment induced by the coupling of rolling and pitching can become the significant one in this case, especially because in a damaged condition it is probable to say that the handling of ship is also limited. I think it is important to study further into such problem.

#### Author's Reply

I am glad to hear that Mr. Haraguchi of SRI has carried out an experiment on the motion of an inclined ship model. I look forward to reading his report when it is published. As to Dr. Takaishi's comments on the importance of the diffracted wave effect on the relative motion, I fully agree with his view. In fact, I have presented a paper at the 14th Symposium on Naval Hydrodynamics held in Ann Arbor last August

concerning the effect of the diffracted waves and motion-generated waves on the relative motion. I have not included those effects in the present study; however, when a more reliable prediction method is developed, the actual free-surface motion at the sides of a ship should be included in the relative-motion prediction.

Dr. Kudo rightly pointed out an irrational trend shown in Figure 9. I apologize for the confusion caused by an error in the definition of the curves in the figure. The symbols of the bottom two curves in the left-hand side of the figure should have been interchanged. I hope this will be corrected in the final transaction. I am not certain if I have understood Dr. Kudo's comment on the effect of the turning direction on the roll motion correctly. I wonder if Dr. Kudo's turning direction means the wave heading. I agree with Dr. Kudo's opinion that a gyroscopic moment induced by the coupling between the roll and pitch would result in yawing moment even when a ship is subject to head waves. This may be a good subject to be investigated in the future. I appreciate both discussers' interesting comments on our paper.

*Session VIb*

## Stability of Special Ocean Craft (Part I)

*Chairmen*

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S VI - 1 b

# A MODAL APPROACH TO THE LATERAL PLANE STABILITY AND MOTIONS OF SUBMERSIBLES

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## ABSTRACT

The paper presents the results of a modal investigation into the lateral plane stability and response of submersibles. The approach adopted differs from previous work in that roll is not assumed to be decoupled from sway and yaw in the stability analysis and the well established modal lateral treatment of aircraft dynamics is used. The inclusion of roll coupling has provided a better understanding of the stability and motions in high speed manoeuvres and shown that the traditional approximations do not adequately describe the behaviour of a vehicle in that situation.

Specifically, the approach has provided an improved understanding of the 'snap roll' phenomenon and highlighted the means to provide enhanced lateral plane performance in future vessels.

Recommendations are made on how the lateral plane motion can be adequately modelled using a lower order system obtained from a low frequency modal approximation. Possible applications to other fields are suggested.

## NOMENCLATURE

Notation based on (Refs. 1 and 2).

### Body Characteristics

$m$	Mass of submersible.
$I_x, I_y, I_z$	Moments of inertia of body about $x, y, z$ axes.
$l$	Length of submersible.
$\overline{BG}$	Distance between centres of buoyancy and gravity.

$\delta_R$  Rudder angle.

$\rho$  Density of sea-water.

### Mathematical Model

$\phi, \theta, \psi$  Modified Euler angles of roll, pitch and yaw.

$U, V, W$  Velocity components.

$P, Q, R$  Angular velocity components.

$X, Y, Z$  Force components.

$K, M, N$  Moment components.

### Disturbed Motion

$\phi, \theta, \psi$  Roll, pitch and yaw increments.

$u, v, w$  Velocity component increments.

$p, q, r$  Angular velocity component increments.

$\delta_r$  Rudder deflection increment.

$Y_v, K_p, N_r$  etc Where velocity or acceleration components appear as subscripts to the force and moment components, these terms are the so-called hydrodynamic or stability derivatives.

### Dressings

The 'dot' denotes differentiation with respect to time eg  $\dot{v}$ .

The 'bar' denotes the value of a variable in steady state motion eg  $\bar{U}$ .

The 'prime' denotes a non-dimensional quantity based on the units of



length  $l$ , mass  $\frac{1}{2}\rho l^3$  and time  $l/U$   
 eg  $N_v' = N_v / \frac{1}{2}\rho U l^3$ .

$$c_1 = \frac{-Y_v'}{(I_x' - Y_{\dot{v}}')}; \quad c_2 = \frac{-Y_p'}{(I_x' - Y_{\dot{v}}')};$$

$$c_3 = \frac{-(Y_r' - m')}{(I_x' - Y_{\dot{v}}')}; \quad c_4 = \frac{Y_{\delta r}'}{(I_x' - Y_{\dot{v}}')};$$

$$d_1 = \frac{-K_v'}{(I_x' - K_{\dot{p}}')}; \quad d_2 = \frac{-K_p'}{(I_x' - K_{\dot{p}}')};$$

$$d_3 = \frac{K_r'}{(I_x' - K_{\dot{p}}')}; \quad d_4 = \frac{K_{\delta r}'}{(I_x' - K_{\dot{p}}')};$$

$$e_1 = \frac{-N_v'}{(I_z' - N_{\dot{r}}')}; \quad e_2 = \frac{-N_p'}{(I_z' - N_{\dot{r}}')};$$

$$e_3 = \frac{-N_r'}{(I_z' - N_{\dot{r}}')}; \quad e_4 = \frac{-N_{\delta r}'}{(I_z' - N_{\dot{r}}')};$$

$$\mu = \frac{m'g \overline{BG}/U^2}{(I_x' - K_{\dot{p}}')}$$

## 1. INTRODUCTION

The need to review the currently accepted means of assessing the stability of submersibles in the vertical and lateral planes has been appreciated for some time by those involved in the specification of the criteria on which the designs of such vehicles are based. The topics have been the focus of recent attention at AMTE(H).

The objective of this paper is to treat the submersible lateral plane dynamics in a similar way to aircraft practice, to demonstrate that the motion can be split into similar modes. This has provided a deeper understanding of the mechanics of submersible lateral plane motion, and indicated the magnitudes and relative importance of each modal contribution.

To this end the paper has been divided as follows. The background to the new approach and the rationale behind it are explained in Section 2, prior to the introduction of the linearised lateral plane equations of motion in Section 3. Three modal contributions are identified in Section 3 and their significance is discussed. Lower order approximations to the Modes are introduced in Section 4 and their validity is demonstrated in Section 5 via comparisons in both the frequency and time domains and an explanation of 'snap roll' is presented.

The new concepts are reviewed in Section 6 and their application is discussed. The relevance of the approach to the allied fields of surface ship and airship dynamics is postulated and ways to

improve the dynamic performance of submersibles using knowledge of the Modes are suggested.

## 2. BACKGROUND

The techniques adopted for the analysis of stability and control for submersibles owe much to those that have been in use for many years in aircraft practice.

The difference between submersible and aircraft dynamics is the surrounding medium and the need to consider buoyancy effects. Therefore terms which are significant for the submersible can be neglected for aircraft and vice versa.

Traditionally lateral plane stability work for submersibles has been based on the linearised equations of motion and the method of analysis that has been used depends upon the basic assumption that rolling motion does not affect the motions in yaw and sway and that the converse is also true. Consequently the roll equation has been treated separately from those governing yaw and sway (Refs. 3 and 4).

In reality, it is recognised that both sway and yaw do cause rolling and the current stability and control parameters, derived from the uncoupled equations, offer no clues as to the likely means of controlling the problems that can arise. However the treatment of the lateral plane stability of aircraft is well understood whereby motion essentially consists of contributions from three modes (Ref. 5).

- (a) Lateral Oscillation or 'Dutch Roll'.
- (b) Rolling Convergence.
- (c) Spiral Mode.

In this paper the effect of including those terms necessary to retain roll coupling is investigated so that a modal approach similar to that in the aircraft world can be made.

It is recognised that the adopted linear approach would not give an accurate description of the coupled motions in the lateral plane, but it is commonly accepted that it provides the means to assess stability.

## 3. THE EQUATIONS AND CHARACTERISTIC MODES

To determine the characteristic modes of motion we shall now consider the solution of the linearised equations of motion in the lateral plane. The control surface (rudder) is kept at mid-ships thus the solution is representative of the 'stick fixed' stability of the submersible. Decoupling is not assumed, leading to the incorporation of the terms  $Y_p$  and  $N_p$  which have commonly been neglected.

### 3.1 The Fundamental Equations

The linearised equations in non-dimensional canonical form, neglecting terms which are small, are:-

$$\begin{bmatrix} \dot{v}' \\ \dot{p}' \\ \dot{r}' \\ \dot{\phi} \end{bmatrix} = \begin{bmatrix} -c_1 & -c_2 & -c_3 & 0 \\ -d_1 & -d_2 & d_3 & -\mu \\ -e_1 & -e_2 & -e_3 & 0 \\ 0 & 1 & 0 & 0 \end{bmatrix} \begin{bmatrix} v' \\ p' \\ r' \\ \phi \end{bmatrix} \quad (1)$$

where the various terms are defined in the nomenclature.

The differentiation is with respect to non-dimensional time  $\tau$  ie  $\dot{r} = dr/d\tau$  and  $\tau = Ut/l$ .

### 3.2 The Characteristic Equation and Conditions for Stability

The eigenvalues of the state matrix are the roots of the following 4th order characteristic equation.

$$\begin{aligned} \lambda^4 + [c_1 + d_2 + e_3] \lambda^3 &+ [(c_1 d_2 - c_2 d_1) + (c_1 e_3 - c_3 e_1) \\ &+ (d_3 e_2 + d_2 e_3) + \mu] \lambda^2 \\ &+ [c_1 (d_3 e_2 + d_2 e_3) - c_2 (d_1 e_3 + d_3 e_1) \\ &+ c_3 (d_1 e_2 - d_2 e_1) + \mu(c_1 + e_3)] \lambda \\ &+ \mu(c_1 e_3 - c_3 e_1) = 0 \end{aligned} \quad (2)$$

The necessary and sufficient conditions for stability are that all the roots (or real part of the roots in the case of complex conjugate pairs) must be negative. Thus by writing the characteristic equation in the form:

$$\lambda^4 + F_1 \lambda^3 + F_2 \lambda^2 + F_3 \lambda + F_4 = 0 \quad (3)$$

Routh's criteria show that the submersible is stable if the following inequalities are satisfied:

$$\left. \begin{aligned} F_1, F_2, F_3, F_4 &> 0 \\ F_3(F_1 F_2 - F_3) - F_1^2 F_4 &> 0 \end{aligned} \right\} \quad (4)$$

Before continuing with the full stability analysis it is worthwhile examining the characteristic equation (2) to determine the range of validity of previous assumptions concerning the lateral motion in yaw and sway.

### 3.3 The Traditional Approximation for Yaw and Sway

Within the characteristic equation, there exists a multiple of a second order polynomial viz:

$$\mu \lambda^2 + \mu (c_1 + e_3) \lambda + \mu (c_1 e_3 - c_3 e_1) \quad (5)$$

At low speeds,  $\mu \gg 1$ ; thus it appears that the terms in  $\mu$  dominate all other terms in the characteristic equation and a reasonable approximation to the motion can be found by equating (5) to zero. This has the effect of decoupling the roll equation. The characteristic equation then reduces to:

$$\lambda^2 + (c_1 + e_3) \lambda + (c_1 e_3 - c_3 e_1) = 0 \quad (6)$$

This approximation appears valid below 4 m/s for typical high speed submersible forms. (See Section 4.2, Tables 1 and 4.) For stable roots:

$$\begin{aligned} c_1 + e_3 &> 0 \\ c_1 e_3 - c_3 e_1 &> 0 \end{aligned} \quad (7)$$

which become, after suitable re-arrangement:

$$\left. \begin{aligned} (I_z' - N_{\dot{r}}') Y_v' + (m' - Y_{\dot{v}}') N_r' &< 0 \\ (m' - Y_r') N_v' + Y_v' N_r' &> 0 \end{aligned} \right\} \quad (8)$$

The inequalities (8) are exactly the same conditions for stability as deduced in (Ref. 3) and elsewhere. Hence, the approximation to horizontal stability that has been used to date is evidently only valid at low speeds. For an improved description of the stability at higher speeds the full fourth order equation (2) should be used.

### 3.4. The Characteristic Modes

The roots of (2) take the form of a complex conjugate pair and two real roots throughout the operating speed range of most submersibles. Their variation with speed is shown in Table 1. For identification purposes, the complex pair will be called Mode 1, the larger real root Mode 2, and the smaller real root Mode 3. Mode 1 is a damped oscillation whilst Modes 2 and 3 are convergent aperiodic motions.

It is necessary to know which are the dominant motions within each Mode. A means of examining the relative contributions to the modes is to inspect the eigenmatrix of eigenvectors shown in Table 2, which has been calculated for a speed of 12 m/s.

TABLE 1. The Variation of Eigenvalues with Speed for a Typical Submersible

Speed m/s	Eigenvalues		
	Mode 1	Mode 2	Mode 3
2	- 1.70 ± i 21.4	- 2.63	- 0.241
4	- 1.71 ± i 10.6	- 2.62	- 0.241
6	- 1.71 ± i 6.97	- 2.60	- 0.242
8	- 1.72 ± i 5.11	- 2.58	- 0.244
10	- 1.74 ± i 3.95	- 2.54	- 0.246
12	- 1.76 ± i 3.16	- 2.50	- 0.249
14	- 1.79 ± i 2.55	- 2.43	- 0.252

TABLE 2. Eigen Matrix for a Typical Submersible at 12 m/s

	Mode 1	Mode 2	Mode 3
$v'$ (- $\beta$ )	- 0.972 x 10 <sup>-3</sup> ± i 0.108 x 10 <sup>-1</sup>	- 0.079	0.196
$p'$	0.955 ± i 0.186 x 10 <sup>-13</sup>	0.505	0.191
$r'$	- 0.291 x 10 <sup>-1</sup> ± i 0.746 x 10 <sup>-1</sup>	- 0.835	- 0.589
$\phi$	- 0.152 ± i 0.242	- 0.204	- 0.760
$\psi$	0.235 x 10 <sup>-1</sup> ± i 0.451 x 10 <sup>-2</sup>	0.338	2.347
$y'$	- 0.200 x 10 <sup>-2</sup> ± i 0.671 x 10 <sup>-2</sup>	- 0.105	- 10.13

In addition to the basic eigenmatrix, Table 2 contains a further two rows corresponding to the values of the two

state variables  $\psi$  and  $y$ , which can be found from:

$$\left. \begin{aligned} \psi &= \frac{r'}{\lambda_j} \\ y' &= \frac{(v' + \psi)}{\lambda_j} \end{aligned} \right\} \quad (9)$$

It will now be used for a brief discussion of the Modes.

#### 3.4.1 Mode 1

The first column of the eigenmatrix shows the relative contributions of the states of this mode.

The Argand diagram for this mode is given in Figure 1, and shows the negligible contribution from  $v'$ ,  $\psi$  and  $y'$ . The motion is therefore a damped oscillation consisting mainly of roll ( $\phi$ ), rate of roll ( $p'$ ) and some yaw rate ( $r'$ ), with  $p'$  lagging  $r'$  (approximately 110 degrees phase), but leading  $\phi$  (approximately 120 degrees phase) for this case.

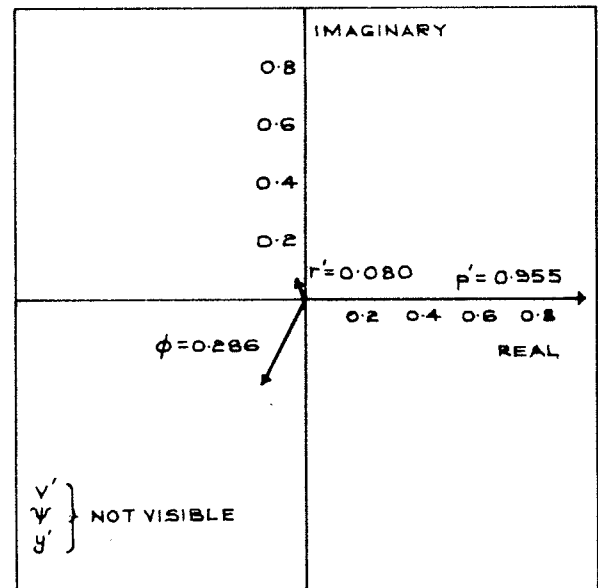


FIG.1 ARGAND DIAGRAM FOR MODE 1

The relatively large and negative real part of the root (Table 1 at 12 m/s), indicates that the motion decays fairly rapidly and is stable. The imaginary part of the root indicates an oscillation of relatively large frequency and is the damped rolling frequency for this submersible.

#### 3.4.2 Mode 2

It is interesting to compare the ratio of the 'angle' variables, taken from the second column of Table 2,

$$\beta : \phi : \psi = 0.23 : -0.60 : 1.00 \quad (10)$$

(where  $\beta \approx -v'$ , the angle of sideslip).

This mode evidently has coupling with respect to all three motions. However the main contribution to the mode comes from yaw and roll, and the relatively large and negative value for the real root (Table 1, 12 m/s), indicates that the motion decays fairly rapidly and is stable. The hydrodynamic forces and moments depend upon the state velocities, which are in the following ratio:

$$v' : p' : r' = 0.09 : -0.60 : 1.00 \quad (11)$$

The above confirms that the forces and moments have a strong dependence on yaw and roll with a negligible sideslip contribution. This mode is aperiodic.

### 3.4.3 Mode 3

The magnitude of the root for this mode, from Table 1 at 12 m/s, shows that it is the 'slowest' of the three modes and therefore the motion takes longer to decay, but it is stable.

The 'angle' variables are in the following ratio:

$$\beta : \phi : \psi = -0.08 : -0.32 : 1.00 \quad (12)$$

The coupling is much weaker, the main motion being that of yaw, with some roll. The state velocities are in the ratio,

$$v' : p' : r' = -0.33 : -0.32 : 1.00 \quad (13)$$

and show that the forces and moments have equal dependence upon the sideslip and roll derivatives, but, as in Mode 2, the yaw derivatives are dominant.

## 4. MODAL APPROXIMATIONS

Section 3.4 gave an insight to the main contributory motions to the modes in a qualitative manner. For a more concise and quantitative representation it is advantageous to make approximations in order to discover which derivatives have the most significant effect on each mode. It may then be possible to make changes to these derivatives and thus the modes in order to improve specific aspects of performance if required.

### 4.1 Mode 1 Approximation

It was shown in Section 3.4.1 that the sideslip contribution  $v'$  is negligible in comparison with  $r'$ ,  $p'$  and  $\phi$ . A good approximation to this mode was found by neglecting the side force equation and the  $v'$  derivatives. The resulting cubic characteristic equation may then be expressed as:

$$\lambda^3 + (d_2 + e_3) \lambda^2 + (d_2 e_3 + d_3 e_2 + \mu) \lambda + \mu e_3 = 0 \quad (14)$$

For stability, the roots of equation (14) must be negative. Alternatively the satisfaction of Routh's criteria (Ref. 5) gives the necessary and sufficient conditions for stability, thus:

$$d_2 + e_3 > 0 \quad (15)$$

$$\mu e_3 > 0 \quad (16)$$

$$(d_2 e_3 + d_3 e_2 + \mu)(d_2 + e_3) - \mu e_3 > 0 \quad (17)$$

In derivative form (15) and (16) effectively reduce to the following conditions:

$$K_p' < 0, N_r' < 0, \overline{BG} > 0 \quad (18)$$

which are the conditions for positive roll and yaw damping respectively, together with the well known condition for hydrostatic stability. At high speeds ( $\mu \rightarrow 0$ ) equation (17) provides the following inequality:

$$K_p' N_r' - K_r' N_p' > 0 \quad (19)$$

An identical expression to this is used in aircraft work as a criterion for spiral stability. All submersibles travelling at high speed should satisfy (19). Over the operating speed range of most submersibles, equation (14) takes the form of a complex pair and a real root. Comparison of Tables 1 and 3 shows that the complex pair give a good approximation to Mode 1.

TABLE 3. The Variation of Eigenvalues With Speed for the Mode 1 Approximation for a Typical Submersible

Speed m/s	Eigenvalues	
	Mode 1	Mode 2
2	- 1.70 ± i 21.5	- 2.10
4	- 1.71 ± i 10.7	- 2.08
6	- 1.72 ± i 7.04	- 2.06
8	- 1.74 ± i 5.20	- 2.02
10	- 1.77 ± i 4.08	- 1.97
12	- 1.80 ± i 3.31	- 1.90
14	- 1.85 ± i 2.75	- 1.81

At the highest speed in Table 1, the error in approximating the real part of Mode 1 is 3 per cent and the error in approximating the imaginary part of Mode 1 is 8 per cent, whereas the approximation is almost exact at low speed. The real root in Table 3 is a poor approximation to Mode 2 (a better approximation to this mode is given in the next sub-section).

#### 4.2 Mode 2/3 Approximation

Unlike the approximation to Mode 1, that to Modes 2 and 3 is not immediately apparent.

The basis of it originates from postulating that the effects of the hydrostatic restoring moment in roll with the addition of the rolling moments due to sideslip and yaw, combine to produce the same roll angle that would exist with  $p' = 0$ , ie the rate of roll is induced from a combination of yaw and sideslip. In addition to this basic assumption, several iterations were performed to find the best approximation to the modes. The closest modal approximation was found when  $Y_p'$  and surprisingly  $K_p'$  were ignored. This gave the following characteristic equation.

$$\begin{aligned} &(\mu + d_3 e_2) \lambda^2 \\ &+ [\mu(c_1 + e_3) + e_2(c_1 d_3 + c_3 d_1)] \lambda \\ &+ \mu(c_1 e_3 - c_3 e_1) = 0 \end{aligned} \quad (20)$$

Before proceeding further, it is relevant to note how this approximation fits in with the traditional approximation discussed earlier. At low speed ( $\mu \gg 1$ ) the  $\mu$  term in equation (20) acts as a magnification factor, reducing the characteristic equation to exactly that of equation (6), where the roll is uncoupled from the motion in yaw and sideslip. However at the other end of the operational speed scale the terms in  $\mu$  are of the same order as the other terms in the coefficients of equation (20) and roll becomes important. Herein lies the basic reason why the assumption concerning the uncoupling of roll (Refs. 3 and 4) is unsatisfactory. Application of Routh's criteria to equation (20) reveals the following:

$$\left. \begin{aligned} \mu + d_3 e_2 &> 0 \\ \mu(c_1 + e_3) + e_2(c_1 d_3 + c_3 d_1) &> 0 \\ \mu(c_1 e_3 - c_3 e_1) &> 0 \end{aligned} \right\} \quad (21)$$

At low speeds the above inequalities reduce to inequalities (7) and (8), but at high speed they reduce to the following additional requirements:

$$\left. \begin{aligned} K_r' &> 0, N_p' < 0 \\ K_v'(Y_r' - m') - Y_v' \cdot K_r' &> 0 \end{aligned} \right\} \quad (22)$$

Over a typical speed range, the solution of equation (20) takes the form of two real roots, giving the approximation to Mode 2/3, shown in Table 4.

TABLE 4. The Variation of Eigenvalues with Speed for the Mode 2/3 Approximation for a Typical Submersible

Speed m/s	Eigenvalues	
	Mode 2	Mode 3
2	- 2.63	- 0.240
4	- 2.62	- 0.239
6	- 2.61	- 0.237
8	- 2.60	- 0.235
10	- 2.58	- 0.232
12	- 2.56	- 0.228
14	- 2.54	- 0.224

Comparison of Tables 1 and 4, shows that the error in approximating the modes increases with speed, reaching 5 per cent for Mode 2 and 11 per cent for Mode 3 at the highest speed. At low speed the approximation is almost exact.

#### 5. VALIDATION

In this section a comparison of the responses in the frequency domain to actuation of the controls will be made between the total 4th order system and systems obtained from the modal approximations. This is followed by an explanation of the snap roll phenomenon from consideration of the modes in the time domain.

##### 5.1 Frequency Response Comparison

The following reduced system equations are obtained by using the assumptions made in Section 4.

(a) Mode 1

$$\begin{bmatrix} S+d_2 & -d_3 & \mu \\ e_2 & S+e_3 & 0 \\ -1 & 0 & S \end{bmatrix} \begin{bmatrix} p' \\ r' \\ \phi \end{bmatrix} = \begin{bmatrix} d_4 \\ -e_4 \\ 0 \end{bmatrix} (\delta_r) \quad (23)$$

where  $S \equiv \frac{d}{dt}$ , non-dimensional Laplace operator.

(b) Mode 2/3

$$\begin{bmatrix} S+c_1 & 0 & c_3 & 0 \\ d_1 & 0 & -d_3 & \mu \\ e_1 & e_1 & S+e_3 & 0 \\ 0 & -1 & 0 & S \end{bmatrix} \begin{bmatrix} v' \\ p' \\ r' \\ \phi \end{bmatrix} = \begin{bmatrix} c_4 \\ d_4 \\ -e_4 \\ 0 \end{bmatrix} \quad (\delta_r) \quad (24)$$

The range of frequencies where these system approximations were found to be valid are demonstrated in the Bode diagrams of Figures 2 to 5 using data for the typical submersible.

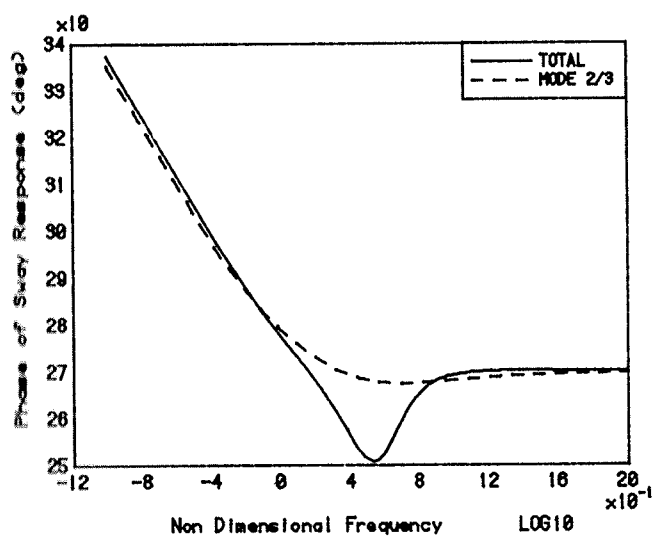
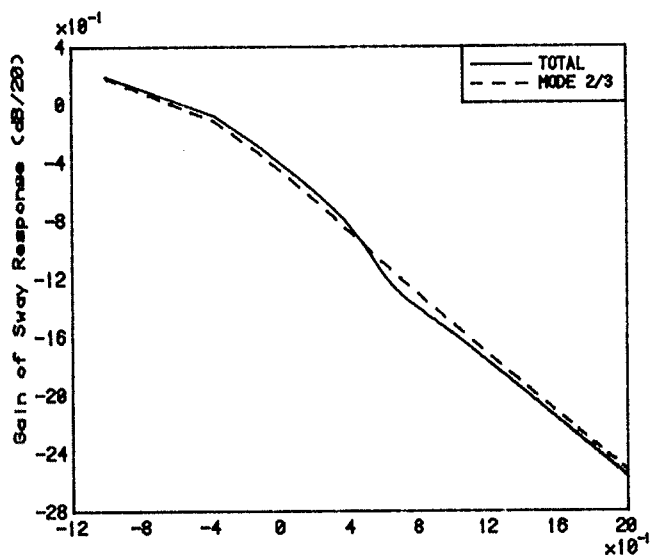


FIG 2 COMPARISON OF SWAY GAINS AND PHASES AT 12 M/S

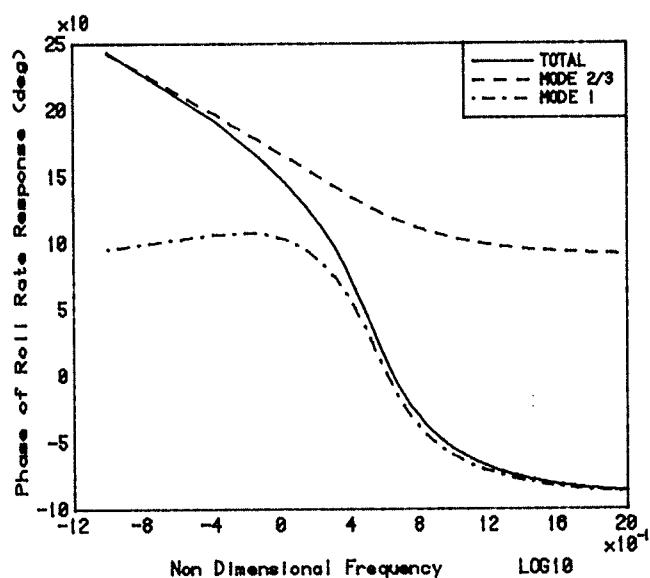
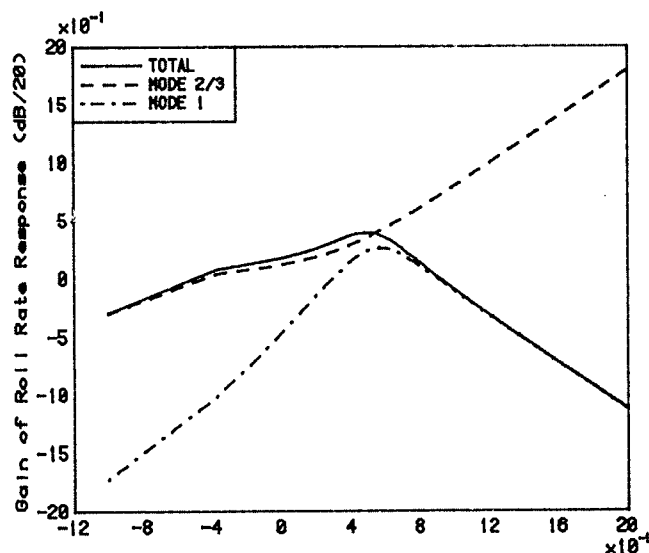


FIG 3 COMPARISON OF ROLL RATE GAINS AND PHASES AT 12 M/S

Comparison of the Mode 1 and Mode 2/3 approximations with the total system frequency responses reveals two areas of good agreement whilst there are ranges of frequency where neither is satisfactory. Two general observations can be made:

- The Mode 1 approximation is correct for the higher frequencies.
- The Mode 2/3 approximation is correct in the low frequency range.

This is an analogous result to that obtained when aircraft vertical plane motions are split into the phugoid and short period modes and their lateral plane

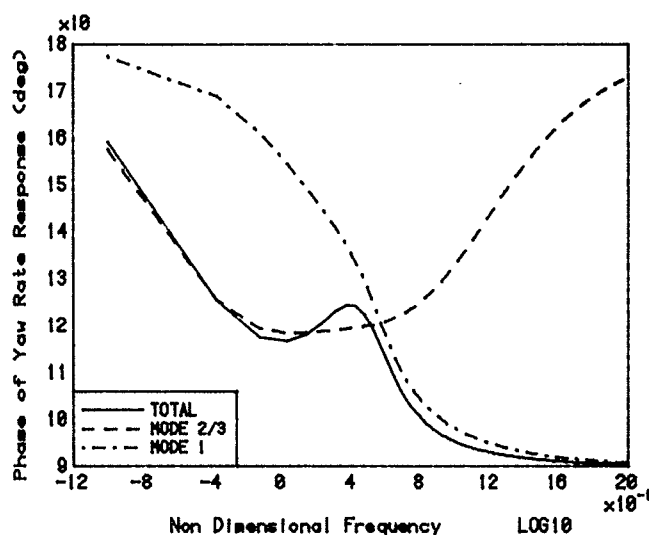
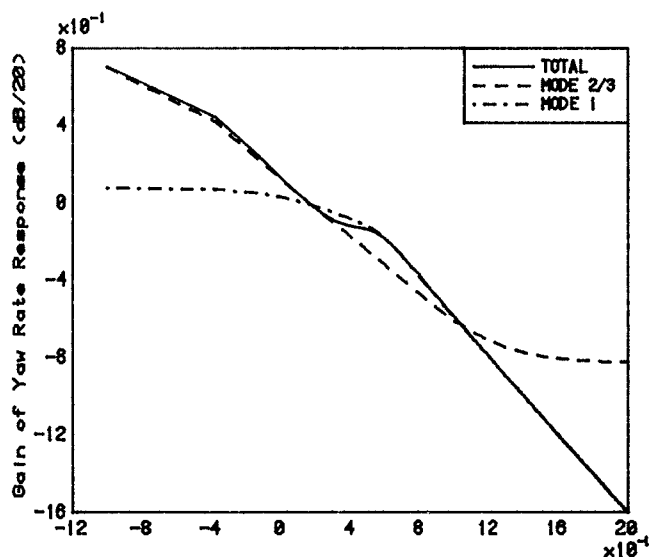


FIG 4 COMPARISON OF YAW RATE GAINS AND PHASES AT 12 M/S

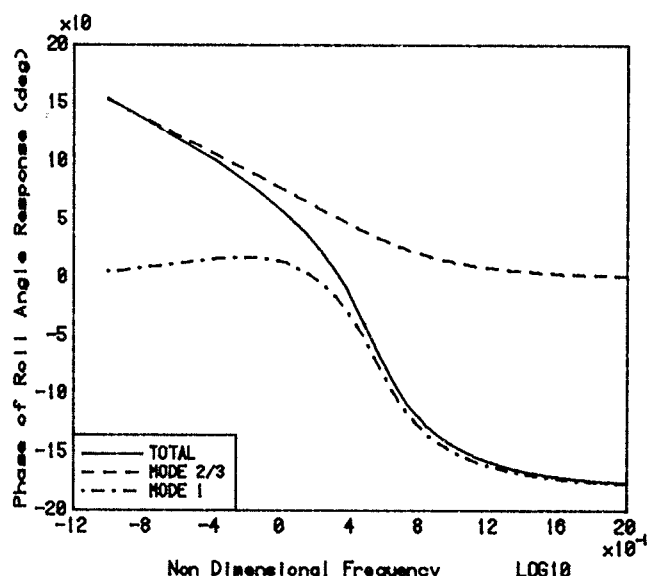
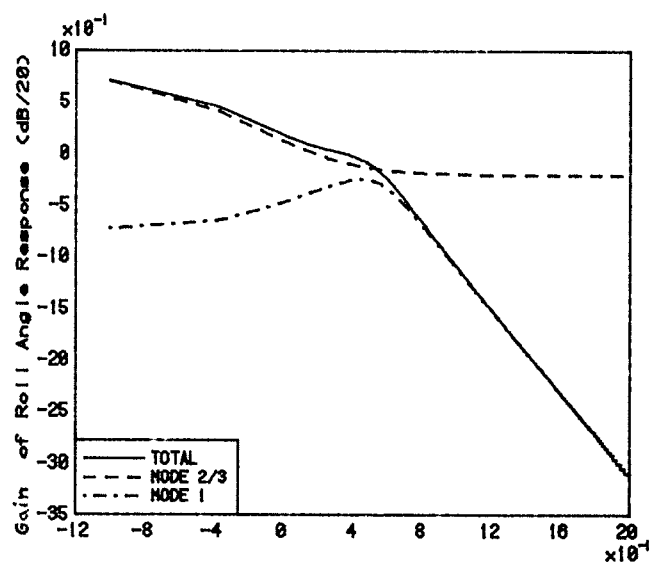


FIG 5 COMPARISON OF ROLL ANGLE GAINS AND PHASES AT 12 M/S

motions are split into the Dutch Roll and spiral/roll approximations (Ref. 5). However the correspondence between the aircraft and submersible in the lateral plane, is not exact. This is because the equations which are used to approximate the aircraft Dutch Roll (oscillatory motion) are in fact the ones used to find the submersible Mode 2/3 approximation (non-oscillatory), whereas those which are used for the submersible Mode 1 approximation (oscillatory) are used to determine the aircraft spiral/roll approximations (non-oscillatory). This arises from the differences in derivative values which in turn, are functions of body geometry.

## 5.2 Comparisons in the Time Domain

In addition to the frequency domain comparison it is also relevant to compare the total system indicial admittances with those from the modal approximations since it is in the time domain that the submariner experiences the response to actuation of the controls.

Figures 6 and 7 illustrate the  $v'$ ,  $p'$ ,  $r'$  and  $\phi$  time responses to a 25 degree step input to the rudder applied to starboard at a speed of 12 m/s and were obtained using the modal approximations and the total system equations. The application of a 25 degree rudder angle is outside the linear regime, but in this example serves to magnify the differences between the various responses. The salient features in the figures are:

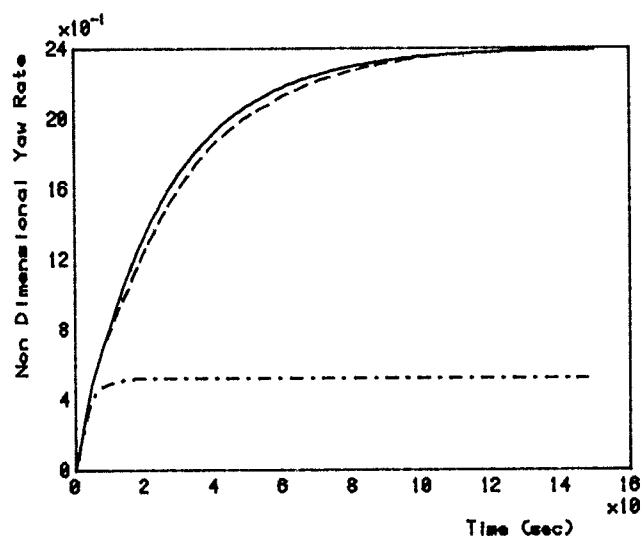
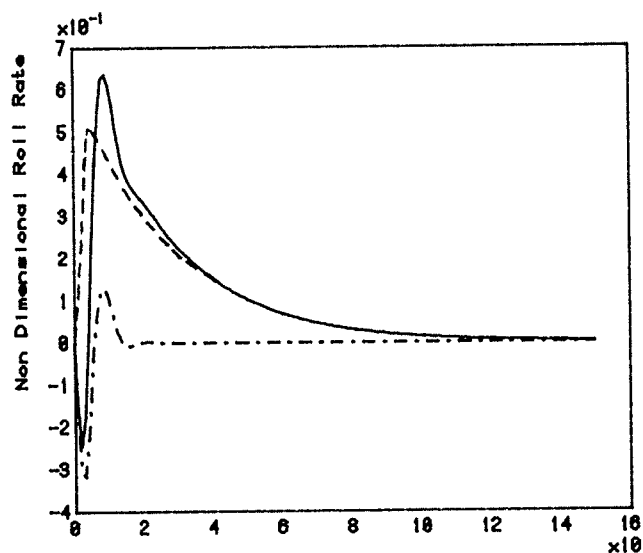
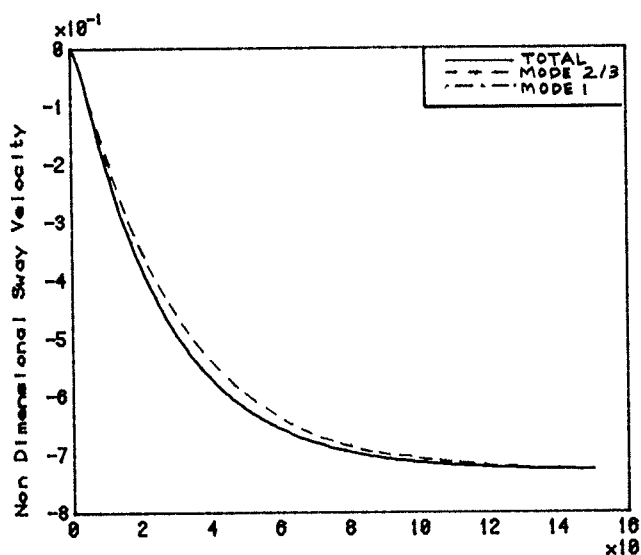


FIG 6 COMPARISON OF SWAY, ROLL RATE & YAW RATE  
INDICIAL ADMITTANCES AT 12 M/S

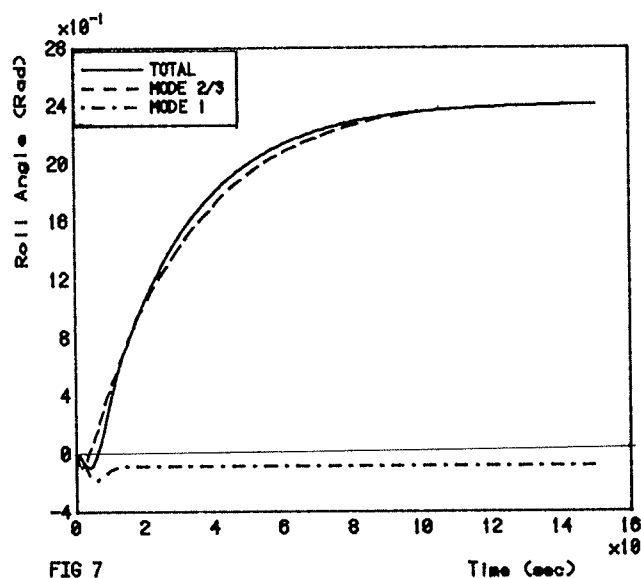


FIG 7  
COMPARISON OF ROLL ANGLE INDICIAL ADMITTANCES AT 12 M/S

- (a) The Mode 2/3 approximation gives a more accurate account of the motion over a significantly longer period than does the Mode 1 approximation, but it is inferior in the initial few seconds for roll responses. The high frequency motion (Mode 1) deteriorates rapidly and explains why the Mode 1 approximation is only accurate in the initial seconds of motion - except for the sway response where it is assumed to be zero.
- (b) The steady state roll angle response is large because the response was calculated assuming a constant speed of 12 m/s. This gives a totally unrealistic steady state value of 2.4 radians but, in reality, the severity of this manoeuvre would always result in a large reduction in forward speed, and therefore the steady state roll angle would be very much smaller due to the influence of  $\mu$ . (See equation (2))
- (c) 'Snap roll' is not apparent in the roll response. This will be discussed later.
- (d) Figures 6 and 7 show that the response in Mode 2/3 is established more quickly than in Mode 1, which is mainly due to the Mode 2 component having a larger negative real root than the real part of Mode 1 as shown in Table 1 at 12 m/s. The damped roll oscillation which exists in the response of Mode 1 is therefore not immediately apparent in the total system response.



### 5.3 Snap Roll Phenomenon

Snap roll is the term used to describe an undesirable facet of submersible response at the beginning of a high speed turn manoeuvre, characterised by a large transient roll angle into the turn. An explanation of the mechanisms involved in this phenomenon is provided in this section.

The roll angle responses of Figure 7 show that the total system response is composed of two component rolling motions from the Mode 1 and Mode 2/3 approximations, Mode 1 is a small amplitude damped oscillation whilst Mode 2/3 is a larger amplitude aperiodic motion of longer time constant. Neither appears to be responsible for the snap roll phenomenon. It is therefore essential to consider whether the marked reduction in forward speed known to occur during a high speed turn would alter not only the magnitude but also the form of the roll responses. To this end a suitable non-linear X force equation was implemented and the results are presented in Figure 8. It is emphasised that the incorporation of this equation serves only to illustrate a trend and not the exact response. It can be seen in Figure 8 that the introduction of the forward speed loss has a marked effect on the total and Mode 2/3 responses, which have had their steady states reduced from 2.4 to .12 radians. The total and Mode 2/3 responses now both exhibit a snap roll amounting to approximately 21 degrees, whereas the Mode 1 roll response is virtually unaffected by the reduction in speed. From examination of the sign and magnitude of the roll responses it is concluded that Mode 2/3 can adequately describe the snap roll phenomenon.

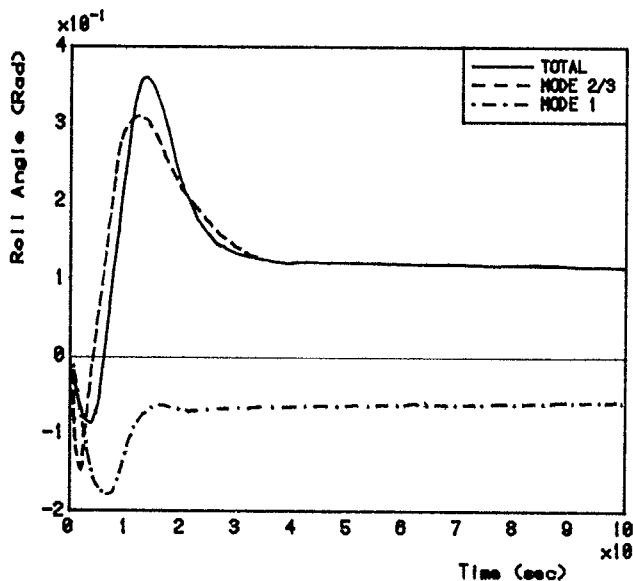


FIG 8 COMPARISON OF ROLL ANGLE INDICIAL ADMITTANCES  
(INCORPORATING FORWARD SPEED LOSS)

From the foregoing a rational explanation for the snap roll phenomenon can be postulated. At high speed the sway, yaw and rudder hydrodynamic rolling moments dominate the hydrostatic restoring term and hence, from the evidence of Figure 7, a large roll angle starts to develop. However the effect of yaw, sideslip and rudder then combine to reduce the forward velocity of the submersible and consequently the relative influence of the static term increases until it balances the hydrodynamic effects and limits the roll excursion. The subsequent further reduction in forward speed allows the hydrostatic term to dominate and hence the roll angle reduces to a small steady state value in the fully developed turn as demonstrated in Figure 8.

### 6. DISCUSSION

The foregoing sections have revealed that the stability analysis that has been used in the aircraft world is directly applicable to the submersible. The retention of roll coupling in the total linear system description of the responses in the lateral plane has highlighted that previous traditional assumptions concerning roll de-coupling are valid only at low speeds. Consideration of the coupled equations has enabled the identification of three modal contributions to the total motion. Examination of Eigenvalues and Eigenvectors has demonstrated that Mode 1 is a damped oscillatory mode and both Modes 2 and 3 are damped aperiodic modes.

The adoption of similar ideas to those in aircraft practice has provided the means to obtain approximations to the modes. These have been shown to provide a good account of the total response in specific areas. In particular, the examination of frequency response plots has demonstrated that Mode 1 is correct for the higher frequencies and Modes 2/3 are correct in the lower frequencies, whilst comparisons of the total responses with the approximations in the time domain confirm this. A notable feature from Mode 2/3 is that rolling motion appears to be totally induced from a combination of yaw and sway motions.

Further examination of the Mode 2/3 approximation has emphasised the importance of roll coupling at higher speeds and has also enabled a postulation to be made concerning the mechanisms behind the snap roll phenomenon. From the stability analysis and the discussion on the nature of the Modes, it is concluded that the reduced order model corresponding to the Mode 2/3 approximation for the coupled motions in yaw, sway and roll can be used with reasonable confidence if a low order system is required. This would be of particular benefit in the design of autopilots, when a simpler representation of the submersible is preferred to the complete fourth order system.

The remainder of this section will seek to assess the possible benefits that may accrue from the foregoing stability analysis. There are two conflicting requirements in the lateral plane namely:

- (a) The reduction of snap roll.
- (b) The improvement of turning response.

This conflict is due to the presence of yaw/roll coupling in the damping of the modes and leads to the usual trade off between improved roll response at the expense of turning or vice versa.

A feasible attempt at improving submersible roll response is suggested from examination of Figure 8. There it is shown that the rolling motion from Mode 2/3 resulting from a rudder deflection, opposes that of Mode 1. Since it has been demonstrated that Mode 2/3 characterises the snap roll phenomenon it is conceivable that if the response of the Mode 2/3 contribution were to be decreased at the same time that the Mode 1 roll response were to be increased, a possible overall reduction in roll response and thus snap roll, may be achieved.

Turning performance of a submersible is also essentially governed by the low frequency Mode 2/3 (see Figure 6) and therefore may be changed significantly by alteration of the Mode 2/3 stability and response.

With this new modal understanding of the lateral plane behaviour of submersibles a judicious arrangement of the strength of each modal contribution is possible. This could improve submersible lateral plane performance, but a compromise is clearly necessary if snap roll is not to be unduly affected by improvements in turning ability.

Changes to the modal contributions in effect mean changes to the hydrodynamic derivatives. Thus certain derivative

combinations could exist which would improve a particular response without other undesirable consequences, and knowledge of these, if they exist, would be beneficial. It is suggested that the application of Parameter Identification techniques (Ref. 6) to low order modelling would enable the identification of the derivative values to be changed and their quantification to achieve a desired response.

Should physical limitations preclude the achievement of particular derivative changes by geometrical variations, the adoption of techniques such as Stability Augmentation (Ref. 7) to synthesise them is envisaged.

The analysis that has been presented in this paper is not necessarily unique to the submersible problem and of course is directly applicable to a number of other vehicles such as SWATH, surface ships, underwater weapons and airships.

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## Discussion

R. Barr (Hydronautics, Incorporated, USA)

The author refers to the influence of forward speed on motions and stability and to the strong coupling between motions (yaw, roll and yaw) and speed loss with large motions, speed losses can become large enough to change the basic behavior of the submersible in my experience the coupling of other motions to surge is highly non-linear, and I wonder if the author could comment on the utility of linear methods, as employed, for dealing with cases involving large speed losses.

The methods described by the authors should be of significant value for under-

standing the effect of design parameters on stability and is a welcome addition to the literature.

#### Author's Reply

We are very grateful for Dr. Barr's comments and we thank him for his kind remarks.

We agree with Dr. Barr in that large speed losses can change the basic behaviour of the submersible. However it was not our intention to simulate the motion under these circumstances, but to understand which mode was responsible for triggering the mechanism behind the Snap Roll phenomenon. Indeed, we

point out in §5.3 of the paper that a non-linear X force equation had to be implemented just to resemble the time history of Snap Roll. As far as understanding the phenomenon is concerned, the linear approach was found to be successful.

However, the authors would certainly doubt the validity of utilizing linear methods in cases involving large speed losses, since the X forces are at least second order due to symmetry.

A.Y. Odabasi (The British Ship Research Association, UK)

I would like the author to clarify two points if possible:

(1) His definition of low and high frequency differs from the one employed in seakeeping. If my understanding is correct, his high frequency limit corresponds to roll natural frequency which is considered low frequency in seakeeping terms.

(2) If one considers the complete form of the canonical equations, the corresponding equilibrium state is obtained as:

$$(y = ? , p = 0)$$

$$(\phi = 0 , q = 0)$$

$$(\psi = ? , r = 0)$$

i.e. there is actually one equilibrium position corresponding to roll whereas there are equilibrium axes corresponding to yaw and sway. Can the author clarify how his results relate to the stability of these three different equilibria?

#### Author's Reply

In reply to the two points of clarification requested by Dr. Odabasi:

1) The terms low and high frequency are related to the frequency range of the submersible and are therefore used in a relative sense. The roll/yaw mode (Mode 1) is a high frequency motion in the case of a submersible, whilst the yaw/roll/sway mode (Mode 2/3) is of lower frequency. As for seakeeping, Dr. Odabasi is correct in saying that the rolling motion is considered as low frequency. However, we would like to suggest that the analysis would be of value in seakeeping, although the resulting modal approximations may be different due to differences in the characteristics of submersibles and surface vessels.

2) The equilibrium position used in the paper corresponds to a constant forward velocity, with all other states initialised at zero. The body axis system is employed and therefore the stability analysis and the results obtained correspond to this axis system. Finally, we would like to thank Dr. Odabasi for his contribution.

R.E.D. Bishop & W.G. Price (Brunel Univ., UK)

Of course there is much to discuss by way of detail in this paper. But we merely wish to congratulate the authors. It is our view that papers of this sort in which the elegant techniques of aircraft Stability and Control theory are employed in submarine dynamics are very welcome.

#### Author's Reply

The authors would like to thank Prof. Bishop and Prof. Price for their kind remarks.

# CAPSIZE SAFETY OF JACK-UP CARRIERS

JOHN KOCH NIELSEN

Danish Ship Research Laboratory

Denmark

## ABSTRACT

Heavy lift ships differ in many ways from conventional ships.

The hull shape is unusual due to the design for a very high centre of gravity. The high centre of gravity and the often relatively large moment of inertia about the roll axis increases the risk of a sub-harmonic coupling of the roll motion with the heave motion. The large difference in mass moments of inertia between the yaw and the pitch mode will cause large gyroscopic heeling moments at simultaneous pitch and yaw motion in oblique waves.

These topics are described in the paper by maths and graphs and are analysed in terms of general mass dynamics, hydrodynamics, and model tests with the ship type.

## NOMENCLATURE

$x, y, z$	= coordinate axes fixed in ship.
$\phi, \theta, \psi$	= rotations about $x, y$ , and $z$ -axes.
$I_{xx}, I_{yy}, I_{zz}$	= mass moments of inertia of ship about $x, y$ , and $z$ -axes.
$F(t)$	= excitation moment.
$B_1, B_2$	= damping coefficients.
$\Delta$	= displacement of ship.
$d$	= draught.
$GM$	= transverse metacentric height.
$GZ$	= righting arm.
$K_{z\phi}$	= coupling coefficient from heave to roll motion.

$\phi_0$	= roll amplitude.
$z_0$	= heave amplitude.
$\omega_\phi$	= roll angular frequency.
$\omega_z$	= heave angular frequency.
$\delta, \epsilon$	= parameters in the Mathieu equation.
$\tau$	= non-dimensional time.
$(\cdot)$	denotes differentiation with respect to the time.

## 1.0 INTRODUCTION

Heavy weight carriers are a new ship type, which entered the transport market only a few years ago. The ships, of which more than a dozen are operating today, are especially designed to carry heavy loads on the deck, such as large offshore structures, factory modules, jack-ups, and semi-submersible platforms. Besides providing transit at normal ship speed, which results in considerable savings in off-hire time and insurance costs, the heavy weight carriers can also serve as floating dry docks in areas where repair facilities are scarce.

Unusual hull shapes and loading conditions for the heavy weight carriers makes it relevant to consider the stability aspects, gyroscopic coupling, and Mathieu instability, which are normally of no importance for conventional ships.

At the Danish Ship Research Laboratory model tests have been carried out with three types of heavy lift ships. Totally, seven different loading conditions were tested, and the present paper is based on the results and experience obtained from these tests.

## 2.0 HEAVY LIFT SHIP CHARACTERISTICS

The loaded heavy lift ship differs from conventional ships in many ways. The hull form is unusual due to the design for a very large deck area and a very high centre of gravity. The often large moments of inertia for roll, and the usually small metacentric height, gives the heavy lift ships very slow and soft roll motions, resulting in a low athwartships acceleration level. However, the long period roll motion increases the risk of subharmonic coupling from heave to roll.

Another characteristic for this type of ships is the shape of the GZ-curve, which is typically as shown on Fig. 1.

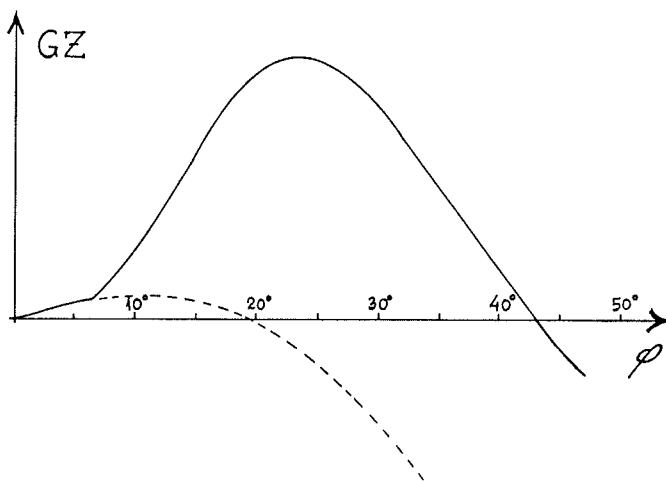


Fig. 1: Righting Arm Curves.

There is a characteristic crack on the curve at the point at which the jack-up or semisubmersible platform dips into the water. The dotted curve corresponds to the theoretical situation in which the volume of the platform is not taken into account.

## 3.0 EQUATION OF MOTION

A fairly complete differential equation which describes the roll motion is given by:

$$I_{xx}\ddot{\phi} + B_1\dot{\phi} + B_2|\dot{\phi}|\dot{\phi} \quad (1a)$$

$$+ (I_{zz} - I_{yy}) \dot{\psi} \dot{\theta} \quad (1b)$$

$$+ \Delta GM\phi + \frac{\partial(\Delta GM)}{\partial z} \cdot z \cdot \phi \quad (1c)$$

$$= F(t)$$

where (1a) represents the basic acceleration and damping terms, (1b) is the gyroscopic coupling moment, and (1c) is the linear and non-linear restoring moments.

Contrary to usual ships, the gyroscopic coupling and the non-linear restoring moments can be of importance when considering heavy weight carriers depending on the actual loading condition.

## 3.1 Gyroscopic Coupling

The gyroscopic coupling moment given by:

$$(I_{zz} - I_{yy}) \dot{\psi} \dot{\theta} \quad (2)$$

consists of three factors, which shall all be non-zero simultaneous in order to cause a coupling moment.

The condition  $(I_{zz} - I_{yy}) \neq 0$  is fulfilled for many types of loading conditions, and two examples of loads, for which the moment of inertia about the sway axis is greater than the moment of inertia about the yaw axis, are sketched in Fig. 2.

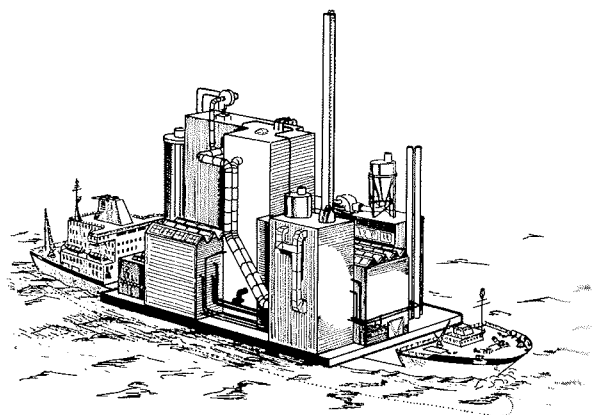
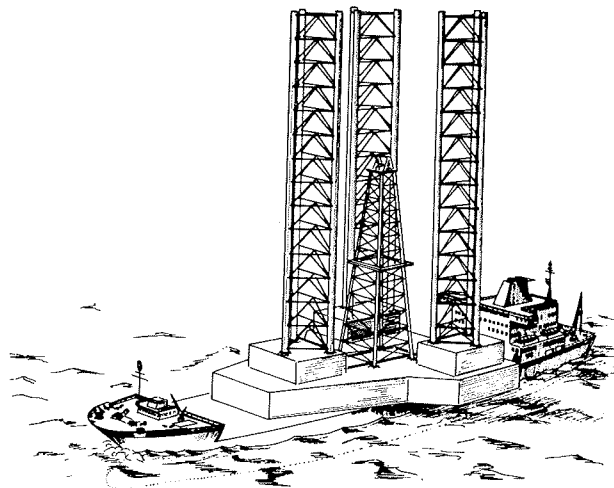


Fig. 2: Loading Conditions where  $I_{yy} > I_{zz}$ .

However, for conventional jack-ups, where three legs are placed at the corners of the drilling rig, giving the rig a rela-

tively large moment of inertia in yaw, the difference between the moments of inertia may be so small that the coupling moment is insignificant.

The other requirement, that the yaw rate and the pitch rate are sufficiently large, is usually fulfilled in oblique waves, especially in bow-quartering seas.

The influence from this coupling on the roll motion is easiest determined by performing model tests where the pitch and yaw rates are measured.

Model tests have shown that for some of the tested heavy lift ships there appeared gyroscopic coupling moments of magnitudes corresponding to the righting moment at a heeling angle of one degree. Furthermore, the tests showed that the phases between roll angle, pitch rate, and yaw rate are randomly distributed. This means that the gyroscopic coupling moment either subtracts or adds to the righting moment, and it is therefore important to evaluate the magnitude of this contribution when approving new loading conditions.

### 3.2 Mathieu Instability

The other phenomenon, which is of special interest, is described by:

$$(\Delta GM + \frac{\partial \Delta GM}{\partial z} z) \phi \quad (3)$$

where the second term expresses the variation of the righting moment with the relative heave motion. The relative heave motion, which in practice is wave induced, is defined as the vertical motion relative to the wave surface counted positive downwards. If only the non-linear heave-roll coupling is considered, the equation of motion takes the form:

$$I_{xx} \ddot{\phi} + \Delta GM \phi + K_{z\phi} \cdot z \cdot \phi = 0 \quad (4)$$

where

$$K_{z\phi} = \frac{\partial \Delta GM}{\partial z} \quad (5)$$

The heave motion is assumed to be harmonic and described by:

$$z = Z_0 \cos(\omega_z t) \quad (6)$$

This results in the Mathieu Equation:

$$\ddot{\phi} + \frac{\Delta GM}{I_{xx}} \phi + \frac{K_{z\phi}}{I_{xx}} Z_0 \cos(\omega_z t) \cdot \phi = 0 \quad (7)$$

which can be reduced to the standard form by introducing the transformation  $\omega_z t = \tau$ . One finds:

$$\frac{\partial^2 \phi}{\partial \tau^2} + (\delta + \epsilon \cos \tau) \phi = 0 \quad (8)$$

where

$$\delta = \frac{(\frac{\Delta GM}{I_{xx}})}{\omega_z^2} = \frac{\omega_\phi^2}{\omega_z^2}, \quad \epsilon = \frac{Z_0 K_{z\phi}}{\omega_z^2 I_{xx}} \quad (9)$$

Equation (8) is known to have unstable solutions for certain values of  $\omega_z$ ,  $\omega_\phi$ ,  $K_{z\phi}$ , and  $Z_0$ . The stability of its solutions may be deduced from a stability chart in the  $\epsilon$   $\delta$  plane. The stability chart of the Mathieu equation is shown in Fig. 3, where the shaded regions are the stable domains.

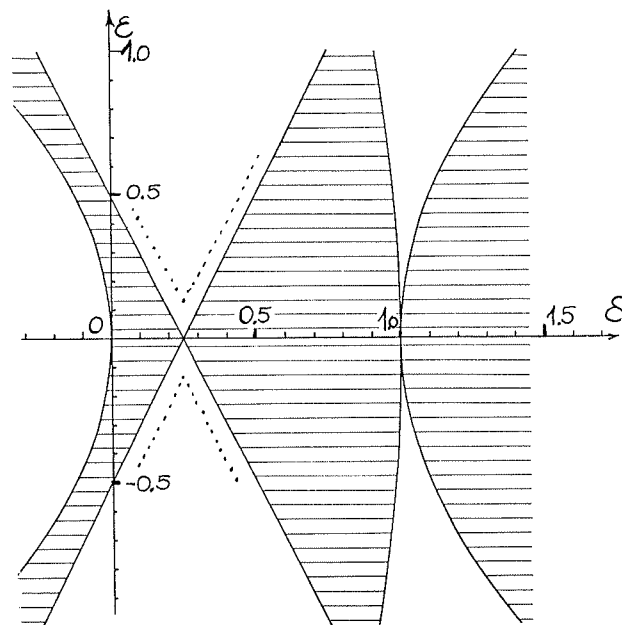


Fig. 3: Stoker's Approximation (Ref. 3) of Stable and Unstable Regions for the Mathieu Equation.

Fig. 3 shows that unstable roll may be excited by vanishingly small heave motion, if the natural frequency of roll is half of that of heave:

$$\omega_\phi = \frac{1}{2} \omega_z \quad (10)$$

When the natural frequencies are nearly, but not exactly of the sort as in (10), the motion can no longer be vanishingly small if it is to produce instability. Let

$$\omega_\phi = (1/2 + \mu) \omega_z, \quad |\mu| \ll 1/2 \quad (11)$$

Then it can be shown (see Ref. 2) that unstable roll occurs whenever the amplitude of the heave motion is:

$$Z_0 \geq \mu \cdot \frac{\omega_z^2 I_{xx}}{K_{z\phi}} \quad (12)$$

### 3.3 Added Unstable Work

So far we have omitted the damping effects in the equation of motion. The inclusion of damping effects would increase the necessary heave amplitude of which instability would occur for fixed values of  $\omega_z$  and  $\omega_\phi$  as indicated by the dotted curves in Fig. 3. In the following an expression for the damping work, which is required in order to limit the unstable roll motion, is derived as a function of the heave motion.

Threshold value for the damping work just allowing unstable roll motions is equal to the added work due to heave coupling. This added work can be evaluated from Fig. 4, which shows the variation of  $\Delta GZ$  with the heave motion.

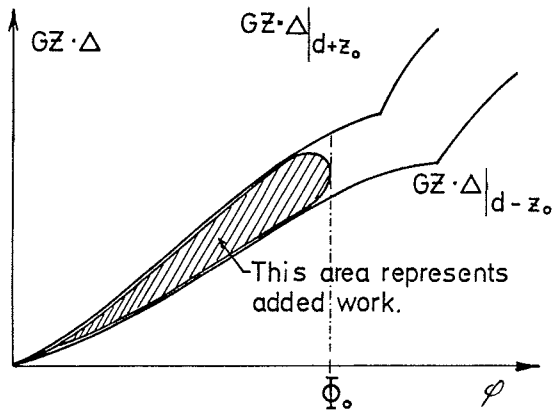


Fig. 4: Added Work Due to Heave Coupling.

The shaded area in Fig. 4 is strongly dependent on the mutual "phase" relationship between heave and roll. The phase between two quantities of different frequencies is not well defined. However, an easy way to determine the "phase" relationship between heave and roll is to plot the heave motion versus the roll motion, resulting in a so-called Lissajou figure as shown in Fig. 5.

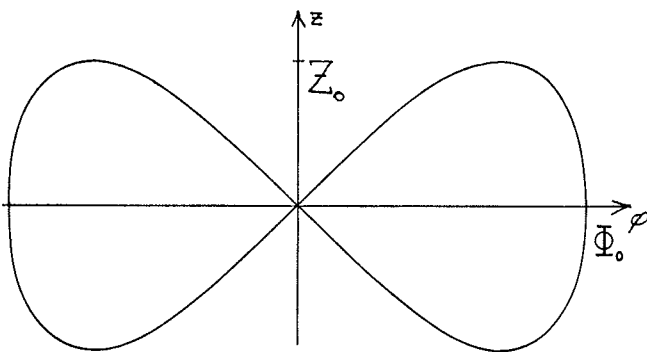


Fig. 5: Lissajou Figure.

The heave dependent part of the curve in Fig. 4 is expressed by

$$\Delta GZ|_{d+z} - \Delta GZ|_d \quad (13)$$

and this curve is related to the curve in Fig. 5 by the ordinate scaling factor

$$\frac{\partial (\Delta GZ)}{\partial z} \approx \frac{\partial (\Delta GM)}{\partial z} \phi \quad (14)$$

This leads to the following relation between the two curves.

$$\Delta GZ|_{d+z} - \Delta GZ|_d = \frac{\partial (\Delta GM)}{\partial z} \cdot \phi \cdot z \quad (15)$$

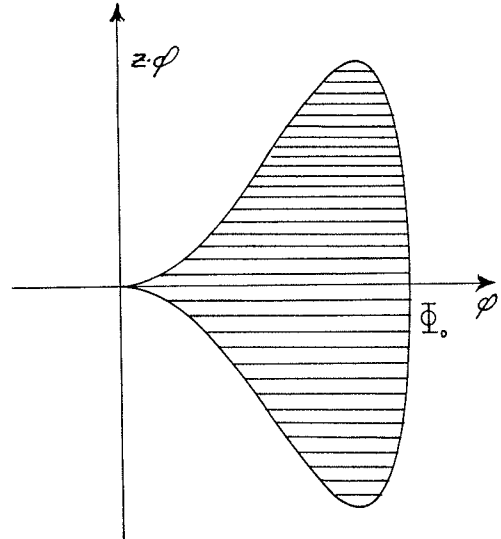


Fig. 6: Heave \* Roll Versus Roll.

The function  $\phi \cdot z$  is shown in Fig. 6, and the area encircled by each half roll period is related to the shaded area in Fig. 4 by the relation:

$$\text{Area (Fig. 4)} = \frac{\partial (\Delta GM)}{\partial z} \cdot \text{Area (Fig. 6)} \quad (16)$$

Assuming harmonic oscillations for both heave and roll, we get

$$z = z_0 \sin(2\omega_\phi t + \alpha) \quad (17)$$

$$\phi = \phi_0 \sin(\omega_\phi t) \quad (18)$$

The phase angle  $\alpha$ , which is related to the heave motion time variable  $\omega_\phi t = 2\omega_\phi t$ , is determined from the Lissajou Figure in Fig. 5 as

$$\alpha = \text{Arcsin}\left(\frac{a}{z_0}\right) \quad (19)$$

where  $a$  is the distance from origo to the curve crossing point on the heave axis. In this case, where the unstable coupling work is maximum, the curve crossing point and the origo are identical, and  $\alpha$  is therefore equal to zero.

The shaded area of Fig. 6 is given by:

$$\begin{aligned}
 & 2 \int_0^{\phi_0} z \cdot \phi \cdot d\phi \\
 & = 2 \int_0^{\phi_0} z_0 \sin(2\omega_\phi t) \cdot \phi_0 \sin(\omega_\phi t) d(\phi_0 \sin(\omega_\phi t)) \\
 & = 4 \int_0^{\frac{\pi}{2}} z_0 \sin^2(\omega_\phi t) \cdot \phi_0^2 \cos^2(\omega_\phi t) d(\omega_\phi t) \\
 & = z_0 \phi_0^2 \frac{\pi}{4} \quad (20)
 \end{aligned}$$

When this result is inserted in (16) we get the following expression for the added unstable work due to heave coupling:

$$\text{Added work} = \frac{\partial \Delta GM}{\partial z} \cdot z_0 \phi_0^2 \frac{\pi}{4} \quad (21)$$

When the two motions do not oscillate harmonically, or the phase angle is different from zero, the area of the curve corresponding to that in Fig. 6 is easiest determined by means of numerical integration. The added work can then be calculated by multiplying this area by the coupling coefficient  $K_{z\phi}$ , which can be determined from the hydrostatic data for the ship. The added work found in this way is equal to the damping work, which should be introduced in the system in order to prevent the undesired coupling from heave to roll.

#### 4.0 EXPERIMENTAL RESULTS

From model tests with a heavy lift ship curves corresponding to those given in Fig. 4, 5, and 6 have been generated. Two examples are plotted in Appendix A, and the Mathieu Instability phenomenon is clearly demonstrated.

The heave motion was excited by exposing the ship to regular beam waves, which makes it easy to achieve the correct heave frequency. In the first example the roll motion was excited manually, but here the waves did not have the correct frequency (see Fig. A1). This results in a "phase" angle of about 85 degrees (see Fig. A2) between heave and roll. The added work is therefore very small, and the roll motion died out quickly.

In the second example (Fig. A5 - A8), the heave frequency is almost exactly twice the natural roll frequency, and the phase angle is about 15 degrees. This results in an added work, which exceeds the damping work, and the roll motion is therefore increasing to an angle where the added work equalizes the damping work.

The unstable roll motions can be very serious. However, for a heavy lift ship with GZ-curve as indicated in Fig. 1 the unstable roll motion will be destroyed whenever the platforms or pontoons dips into the water.

Due to the extreme damping forces, which act on the out hanging parts of the platform, the roll motion will be stopped

almost immediately when the platform hits the water, and due to the large restoring forces which act on the platform the ship will be forced to roll harmonic to the waves in heavy weather.

An investigation of the sizes of these restoring and slamming forces has been carried out. The excitation forces from the jack-up, which act on the heavy lift ship, were measured by means of three strain-gauge balances, (see Fig. 7).

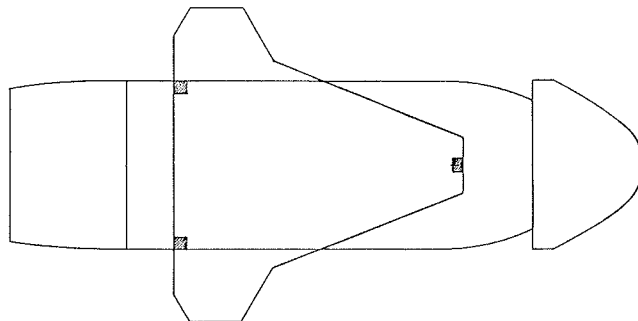


Fig. 7: Positions of Strain-gauge Balances.

These measurements showed that up to 60 per cent of the static force on the up-sea balance was removed in beam seas when the significant wave height was about 9 metres.

For smaller roll angles there is only the usual damping present. However, when the centre of gravity is placed very high above the water surface the roll motion results in a sort of sway motion for the underwater part of the hull, which results in quite a good damping due to generation of waves.

#### CONCLUSIONS

This paper has discussed some of the stability aspects, which are of special interest when considering heavy weight carriers. The gyroscopic coupling moment, which can be of considerable magnitude depending on the actual load, is always important to bear in mind when approving new loading conditions. The Mathieu Instability, which can arise only when the wave encounter frequency is nearly twice the natural frequency of roll, has been examined, and a method to calculate the necessary damping work, which prevents the unstable roll motion, has been presented.

#### ACKNOWLEDGEMENTS

The author wishes to express his thanks to Kai Kure, Bent Kofoed Jacobsen, and Jens U. R  meling, senior Naval Architects at the Department of Ship and Ocean Engineering, Danish Ship Research Laboratory, for their interest and advice, and to Sonja



Davidson for her keen interest in the presentation and the diligent efforts in preparing the cameraready manuscript.

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# APPENDIX

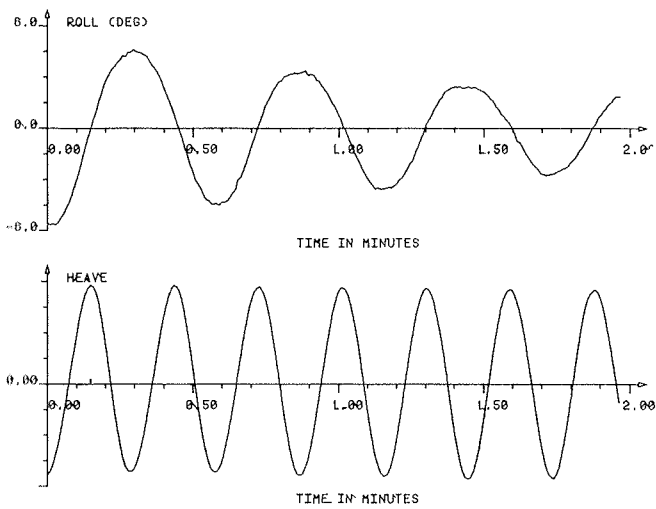


Fig. A1

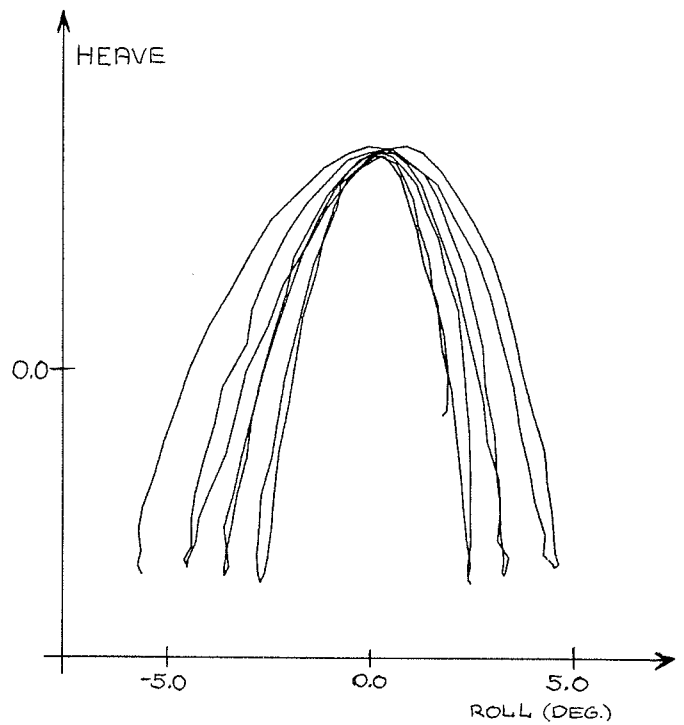


Fig. A2

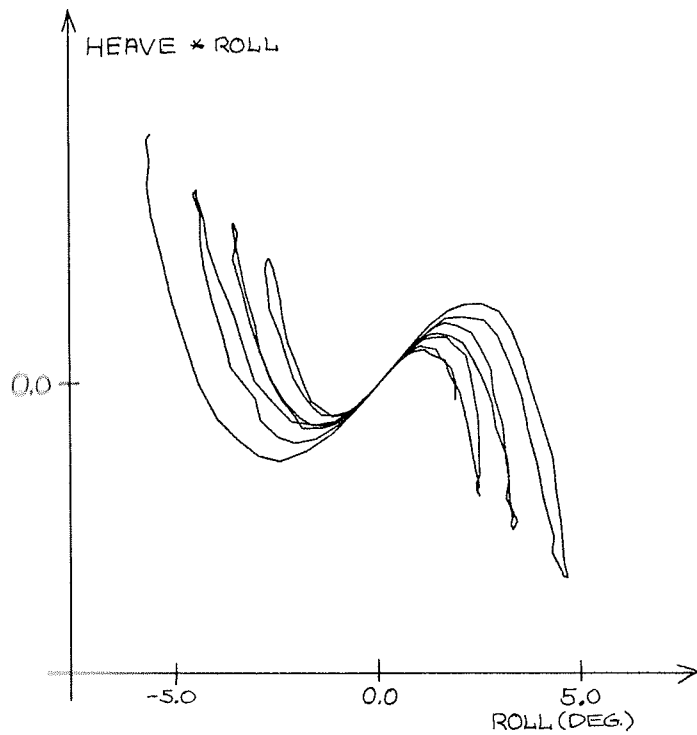


Fig. A3

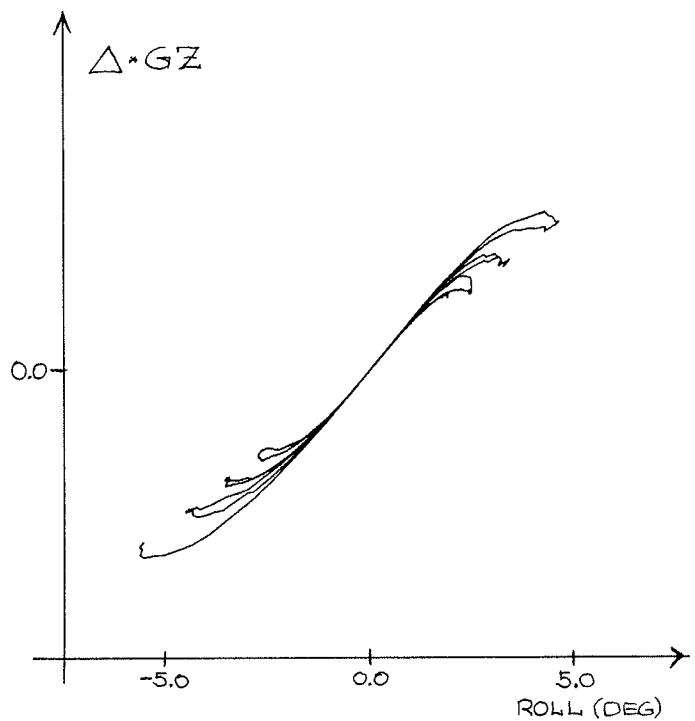


Fig. A4

Fig. A1 - A4 describes a situation where the roll motion is excited manually, but here the waves do not have the correct frequency, and the added work due to unstable roll is very small. Therefore the roll motion is quickly dying away.

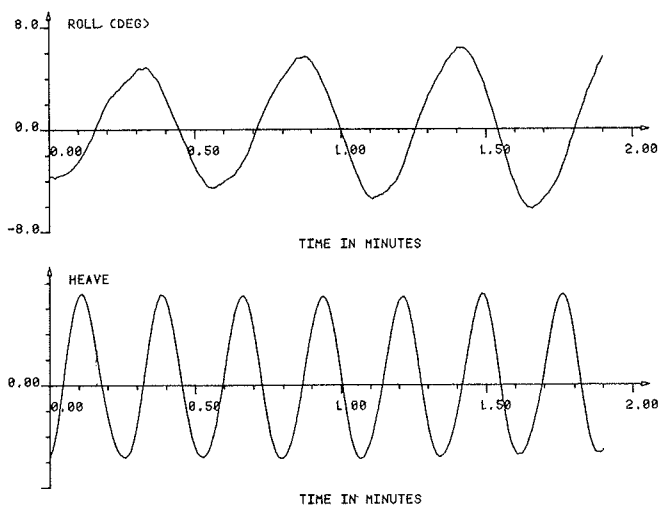


Fig. A5

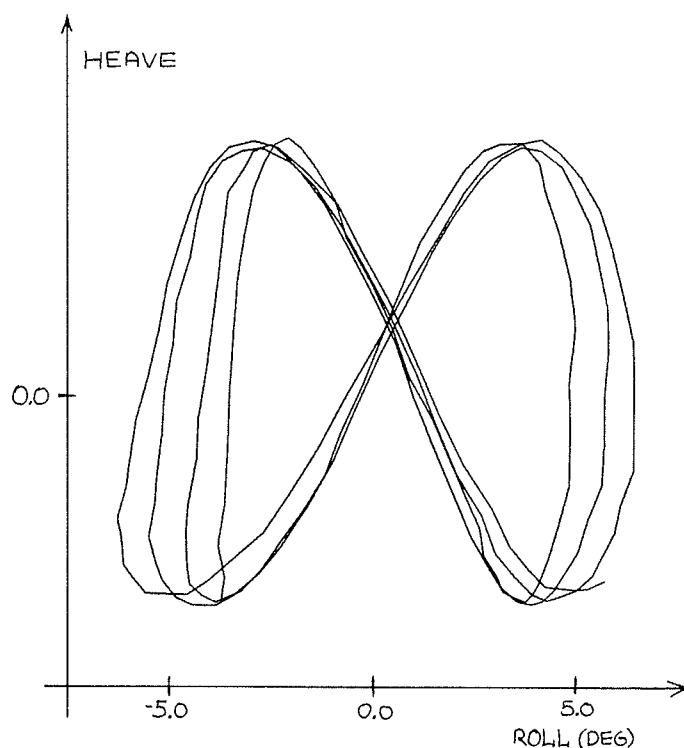


Fig. A6

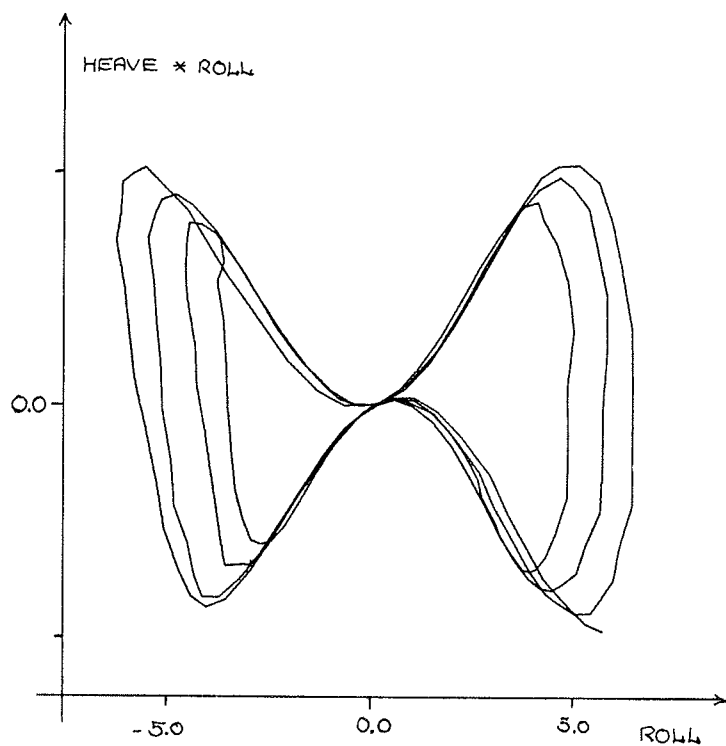


Fig. A7

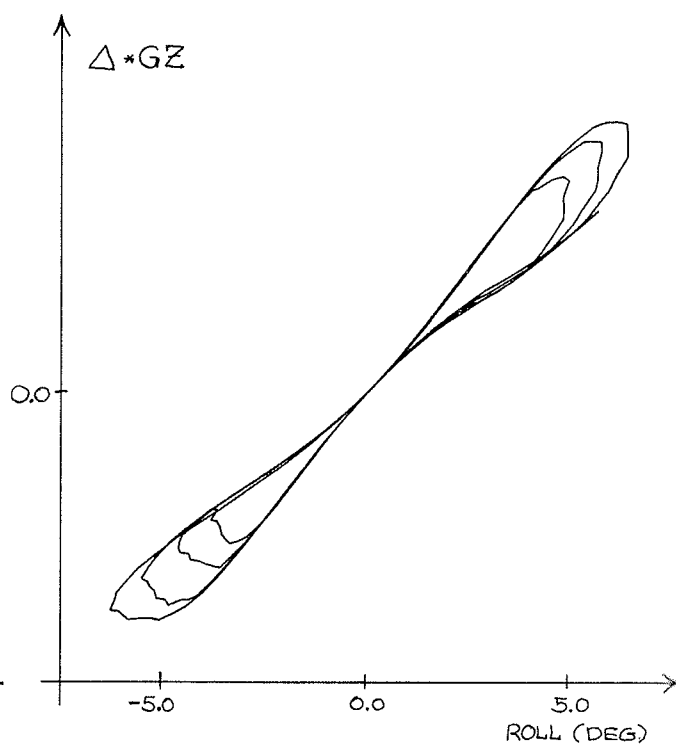


Fig. A8

Fig. A5 - A8 describes a situation where the added work due to the subharmonic coupling from heave exceeds the damping work, and the roll angle is increasing to an angle where the added work equalizes the damping work.

## Discussion

N.G. Skomedal (The Norwegian Institute of Technology, Norway)

I would like to thank the author for a clear presentation and an excellent paper on an extremely interesting subject.

1) In course of this conference we have heard about a number of cases where parametric excitation have been seen upon as a very important contribution to the total roll response especially in following and quartering seas. The probability for such excitation could be significantly reduced causing improved safety for sensitive ship types, if the damping is increased. How important do the author think Mathieu instability is in the assessment of safety against capsize?

2) In the paper jack-up carriers are considered. How will such ships act if their cargo have no overhang which kills every large roll response?

### Author's Reply

First I would like to thank Mr. Skomedal for his discussion.

For ships having a large roll period and with the center of gravity placed high above the water surface, it is our experience that it is highly relevant to consider Mathieu Instability in the assessment of safety against capsize.

If there is no overhanging parts of the load, heavy lift ships will act like conventional ships, with respect to the actual moments of inertia, restoring and damping forces.

*Session VIIa*

## Stability Devices

*Chairmen*

Prof. Odo Krappinger  
Hamburg Ship Model Basin  
F.R.G.

Dr. Yoshifumi Takaishi  
Ship Research Institute  
Japan

S VII - 1 a

## ON A MICRO-COMPUTER BASED CAPSIZE ALARM SYSTEM

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\*University of Tokyo, \*\*Japan Craft Inspection,

\*\*\*Toshiba Corporation

Japan

### ABSTRACT

The development of a micro-computer based capsize alarm system for small crafts and fishing boats was made under the sponsorship of Japan Craft Inspection (JCI).

The system issues the capsize alarm on a stochastic basis using the data processing ability of a micro-processor. The following items are taken into account in issuing the alarm:

- (1) Original stability characteristics of a ship.
- (2) Variation of the location of the center of gravity.
- (3) Constant heeling angle.
- (4) Weather conditions.

Pendulum systems were employed to measure the ship motion with lower cost. The difficulty in measuring the ship motion by pendulum system was studied extensively.

Full scale measurements were made for two fishing boats. Each trial was continued for a month respectively. It was deduced from these experiments that the present system will be very promising as long as the lower ship speed is concerned. Some more studies are required to get the reasonable alarm at higher speed.

### 1. INTRODUCTION

Knowing the stability characteristics of small crafts and fishing boats is the most important concern in their operation. Those ships are not designed as all weather type so that they have to come

back to harbors when the weather is getting rough. The way of ship operation affects the safety of ships much more than that of larger ships from the stability view point. A large catch of fishes may raise up the center of gravity of a boat, or a careless helming may cause the large heeling.

Therefore a reliable capsize alarm which tell the skipper "better to come back to the harbor" or "center of gravity is too high" will be very helpful for the safe operation of small ships.

Japan Craft Inspection(JCI) sponsored the development of a prototype of a micro-computer based capsize alarm system from this view point. It was aimed from the beginning of this project to make the system as inexpensive as possible in order to be accepted by smaller boats.

The development of capsize alarm devices had been tried several times already. But none of them was accepted for practical use yet. The main reason of failure is that the capsize alarm of those devices are dependent on the deterministic criteria. The typical criteria of those alarm are maximum rolling angle experienced, average heeling angle of a ship and so on. Therefore, the alarm rings all the time if the threshold level is set low or it never rings before the capsizing if it is set high.

A metacentric height meter was tried to develop elsewhere. The idea was based on the fact that the mean period of rolling in irregular waves is very close to the synchronous rolling period of a ship. Then, if we know the mean rolling period, the location of the center of gravity can be obtained using the estimated displacement and the radius of gyration of a ship.

The reliable capsize alarm must be based on the stochastic criteria. The expected extreme rolling angle is much more reliable information than the maximum rolling angle which was happened to be measured. For this purpose, some ability of data processing is required in a capsize alarm device. An appropriate length of time series of data is necessary. On the otherhand, if there was a time series data available, mean period of rolling and average heel angle can be calculated much more accurately.

It used to be impractical for small boats to have a data processing ability on board. But nowadays the cost of microprocessor is negligible. The material cost of the present system, for example, is less than \$500. Consequently, each ship may have a data processing ability for capsize alarm only, if it is useful.

## 2. OUTLINE OF THE SYSTEM

### 2.1 Principle of the System

The stability standard of the Japanese Government for passenger ships is based on the following principle.

Assume a ship to be rolling synchronously around  $\phi_m$  with the amplitude of  $\phi_s$ . Where  $\phi_m$  is the constant heeling angle due to the specified wind pressure by the regulation and  $\phi_s$  is the synchronous rolling amplitude which can be obtained by maximum expected wave slope and the damping factor of a ship.

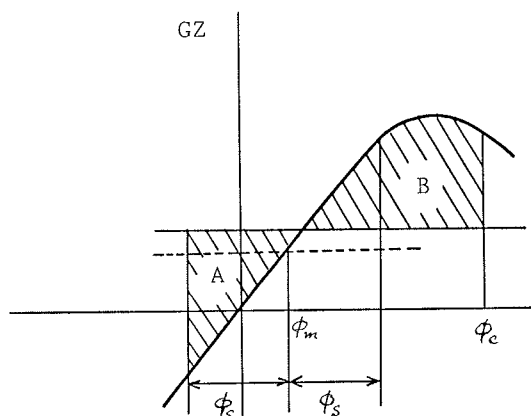


Fig. 1 The Principle of C- coefficient

Next, assume the wind pressure to be increased 50% when a ship rolls to weatherside. Then the subsequent rolling angle to leeward should not exceed  $\phi_c$  where a non-watertight opening of a ship exists. All of this scenario is illustrated in Figure 1.

To fulfill the above requirement, The area A in Figure 1 should be less than the area B. The ratio of areas B and A is called C-coefficient and the Japanese Government requests the C-coefficient of a passenger ship should be larger than 1.0.

The same type of regulation is applied for fishing boats over 20 gross tons.

We tried to borrow this principle to our capsize alarm system as the generalized index for capsizing with some modifications.

As we are aiming at the system to judge the stability of a ship online, the alarm should be based on the conditions of a ship at each time on the contrary to the regulation which is based on the existable extreme condition.

In Figure 1, GZ curve varies according to the variation of the center of gravity above keel and loading condition. The average heeling angle depends on the wind pressure and loading condition at that time. If we use the expected extreme rolling angle in stead of the synchronous rolling angle, it varies with the weather conditions as well as the change in stability characteristics of a ship.

If these time varying parameters are used for the stability criteria, the C-coefficient will be time dependent value as well. We call this time varying value  $C(t)$ .

We thought that the  $C(t)$ -coefficient such defined will be one of the most promising index for capsize alarm.

It is necessary to measure the following parameters to calculate the  $C(t)$ -coefficient:

- (1) Average rolling period in irregular waves: from which we are going to estimate the synchronous rolling period and then estimate the location of the center of gravity of a ship.
- (2) The draft condition of a ship.
- (3) The root mean square of rolling angle at that time: from which we estimate the expected extreme rolling angle.
- (4) The average heeling angle of a ship at that time.

The cross curve of a ship is assumed to be calculated beforehand and be stored in a computer to obtain the GZ curve at that time.

### 2.2 System Configuration

According to the principles above mentioned, a capsize alarm system was designed as shown in Figure 2. Figure 3 illustrates the picture of this system on board at the full scale trial. The components of the system are as follows:

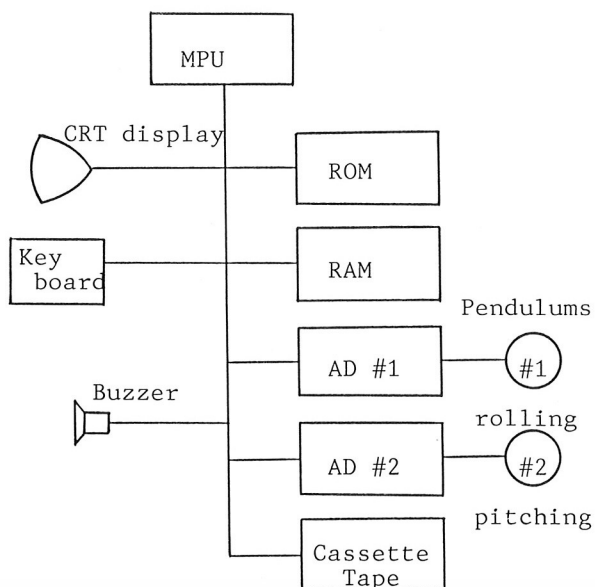


Fig. 2 Configuration of the prototype system

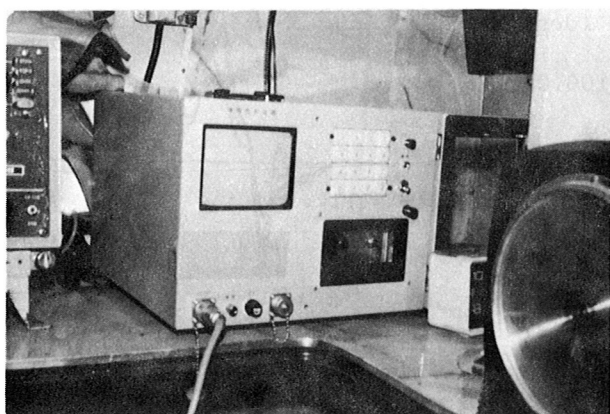


Fig. 3 Capsize alarm system on board

**MPU:** An Intel's 8085 compatible micro processor unit was employed to control the whole system and to analyze data. Programs were written in Intel's PL/M language.

**MEMORIES:** 8KB read only memories(ROM) were used to store the control programs and the necessary data for analysis as the cross-curves of a ship. 12KB random access memories(RAM) were installed for

temporary data.

**CRT DISPLAY:** A 5-inches CRT display was used to show the analysed results and to help the interactive data input of an operator. This CRT displays 32\*16 alpha numeric characters.

**KEY BOARD:** A neumeric keyboard was arranged on the front pannel of the system to input the necessary data like draft conditions.

**PENDULUMS:** Two pendulum units were used to mesure the rolling and pitching angles of a ship. This pendulum unit consists of a pendulum, an oil damper and non-contact type potentiometer. The measurement range of this unit is 45 degrees in both side. The rolling and pitching angle of a ship are read by computer with 8-bits AD converters.

**BUZZER:** The system ring the capsie alarm with this buzzer when it is necessary.

**CASSETTE TAPE:** As this system is a prototype, a digital cassette tape unit was equipped to store the raw data during the full scale trials. those data can be reanalyzed afterwards with a larger computer. About 500KB data can be stored in a single tape.

**ELECTRIC POWER UNIT:** 24V DC power supply from a ship is converted to stabilized 5V, 12V and -15V DC power supply for the system.

All of these components are built in a water proof case as shown in Figure 3.

### 3. MEASUREMENT OF SHIP MOTION BY PENDULUM

#### 3.1 Difficulty in Measurement

It is well known that the oscilation angle of a pendulum is much larger than that of a ship if it is located far from the center of oscilation of a ship. The ratio of oscilation angle varies according to the location of a pendulum, frequency of oscilation and the characteristics of pendulum itself. The natural frequency of the pendulum which is used here is 4Hz(0.25 second).

Figure 4 shows the oscilation angle ratio of pendulum and ship. The effects of the location of a pendulum are shown as a parameter. In case of the distance of 1.0m, experimental results are shown simultaneously comparing with the calculated results.

As the measurement of rolling angle is the most important value of the system,



it must be compensated properly.

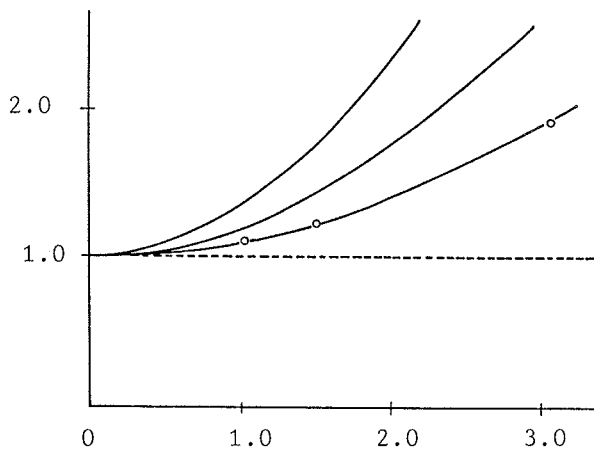


Fig. 4 Amplitude ratio of pendulum and rolling angle

Although the best way to measure the rolling angle is using a vertical gyro instead of a pendulum, it will be too expensive for small crafts to use it in common. There can be two ways to compensate the effects of pendulum:

- (1) To apply an online digital filtering in time domain: The weighting function which has the inverse characteristics of Figure 4 can be obtained easily. Then the actual rolling angle is calculated by convolution integral.
- (2) To apply a frequency domain compensation: If the spectra of the oscillation of pendulum are obtained, the actual rolling angle spectra can be calculated dividing them by the square of amplitude ratio.

Another difficulty in measuring the rolling angle with a pendulum is that a pendulum reacts the swaying acceleration of a ship as well as the rolling. There are no way to compensate this phenomena because no information is available. Using an accelerometer for swaying will increase the cost of capsizing alarm system too much.

Model tests were carried out in irregular waves to see how large the effect of swaying is. The principal particulars of a model are as follows:

L :	2.500m
B :	0.365m
D :	0.200m
d :	0.120m
Δ :	56.156Kg

A typical example of obtained spectrum is shown in Figure 5 comparing

with the rolling spectrum which was measured by vertical gyro.

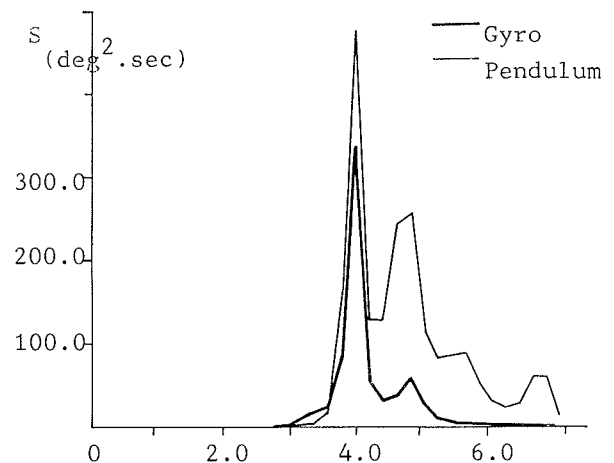


Fig. 5 Difference in spectrum between rolling and pendulum

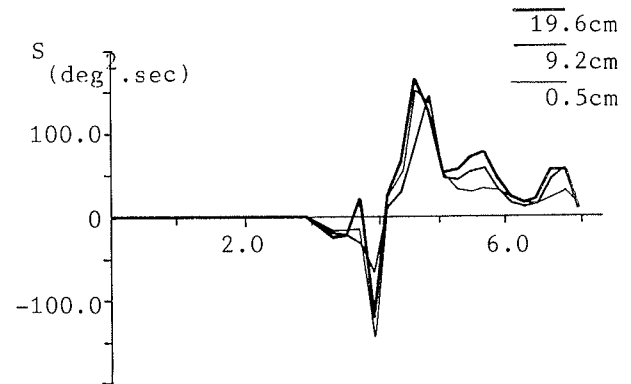


Fig. 6 Effect of swaying to the angle of pendulum

Some compensation technique is definitely needed because the difference in high frequency range is very large.

Figure 6 illustrates the effects of swaying. These are obtained from the following calculations:

$$S_{ss} = S_{pp} - S_{\phi\phi} |H_{p\phi}|^2 \quad (1)$$

where

$S_{ss}$  : effect of swaying  
 $S_{pp}$  : spectrum of pendulum  
 $S_{\phi\phi}$  : spectrum of rolling  
 $H_{p\phi}$  : amplitude ratio of pendulum and rolling

It can be seen that the effects of

swaying seems to be independent from the location of pendulum above the center of rolling and that the effects of swaying are large in high frequency domain.

From the above consideration, we realize that if we compensate the effect of rolling by the amplitude ratio of Figure 4, it reduce the effect of swaying at the same time as the side effect. The higher the location of a pendulum above the center of rolling is, the larger the reduction of swaying effect is because the swaying effect is independent from the location of a pendulum. The higher location of a pendulum is preferable on the contrary to our common understanding as long as the spectrum of rolling is concerned. Although this compensation can be attained either by online filtering in time domain or by frequency domain, the frequency domain compensation will be preferable for accurate calculation.

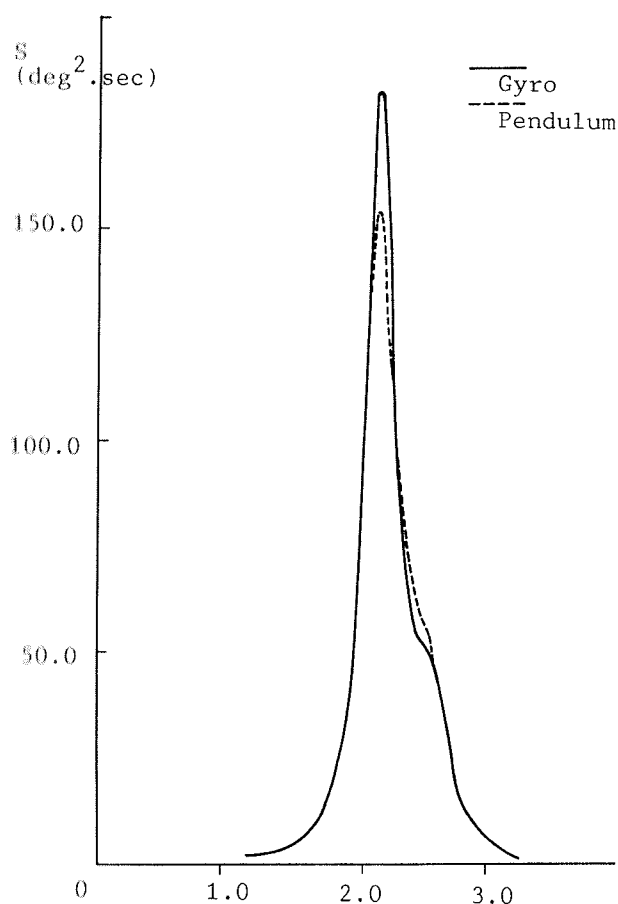


Fig. 7 Spectrum of rolling  
(0 knot, Head sea)

This sort of expectation is not rational theoretically. But it will be a very good compromising practically. We need only a simple pendulum to obtain the approximate rolling spectra. FFT(Fast Fourier Transform) routine was made to

calculate the spectra.

### 3.2 Preliminary Full Scale Test

Preliminary full scale tests were carried out in and offshore of Hamanako Lake area. The objectives of the tests were:

- (1) To see the characteristics of pendulum unit at the full scale environment in waves. For this purpose, rolling and pitching of ship were measured by a vertical gyro simultaneously.
- (2) To know the possibility in estimating the synchronous rolling period by mean rolling period in irregular waves.

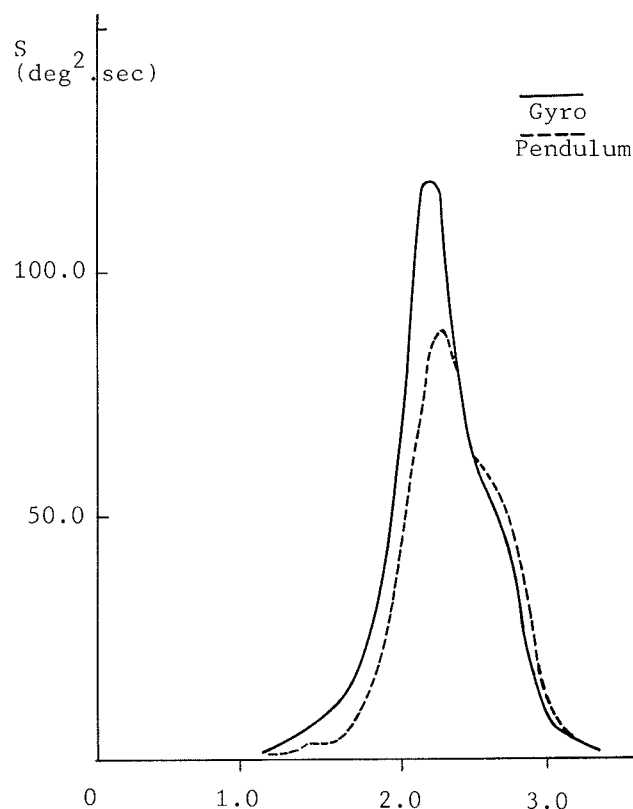


Fig. 8 Spectrum of Rolling  
(0 knot, beam sea)

The principal characteristics and condition of ship were as follows:

L :	7.550m
B :	2.000m
D :	0.680m
d :	0.408m
trim :	0.415m(aft)
$\Delta$ :	25.110t
KM :	1.380m
KG :	0.895m

GM : 0.485m  
T(sync): 2.66sec

The test were carried out under the following cases and weather conditions:

ship speed: 0, 4 and 12 knots  
wave direction: 0, 45, 90, 135 and 180 degrees  
wind speed: 4 to 6 m/sec  
wave height: 0.5 to 1.0 m  
wave length: 5 to 10 m

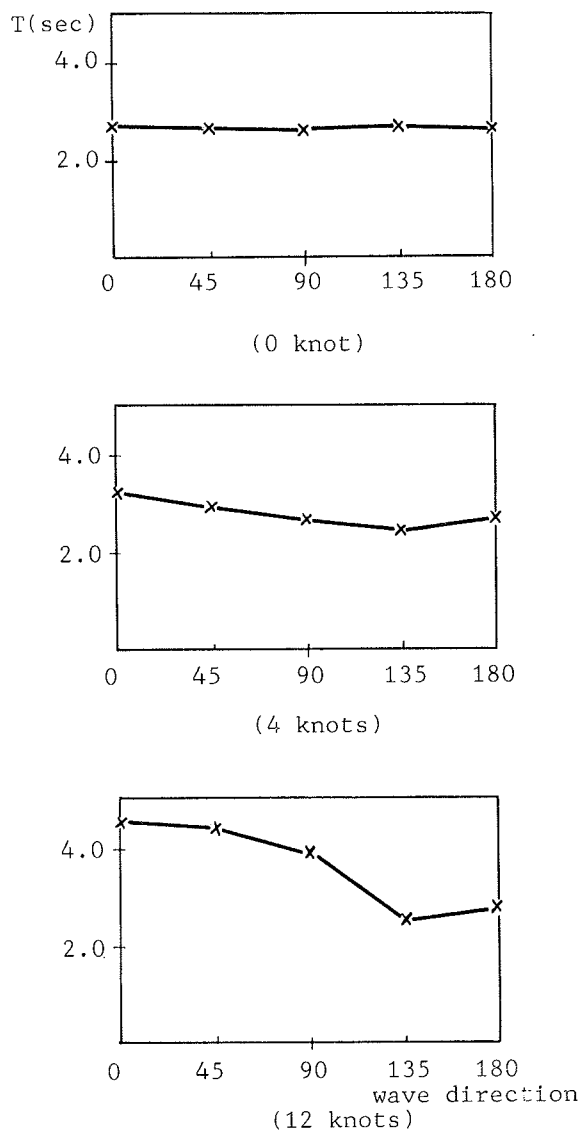


Fig. 9 Mean period of rolling

Figures 7 and 8 show the comparison of the spectra obtained from the measurement of vertical gyro and the compensated measurements of pendulum unit.

The results of other cases show more or less similar tendency as these figures. It can be said that the spectra which were obtained by pendulum unit can be used as the spectra of rolling if they are compensated properly.

On the other hand, there are too many problems in estimating the synchronous rolling period. Figure 9 shows how the mean period of rolling varies according to the ship speed and incident wave direction. The mean period of rolling was calculated here using the 0-th and 2-nd moment of the spectrum.

$$T = 2\pi \sqrt{m_{\phi} / m_z} \quad (2)$$

In the case of  $v=0$  knot, results are fine. The mean period of rolling is almost constant for any wave direction and its value is quite similar to the synchronous rolling period which was obtained from the free rolling test in still water.

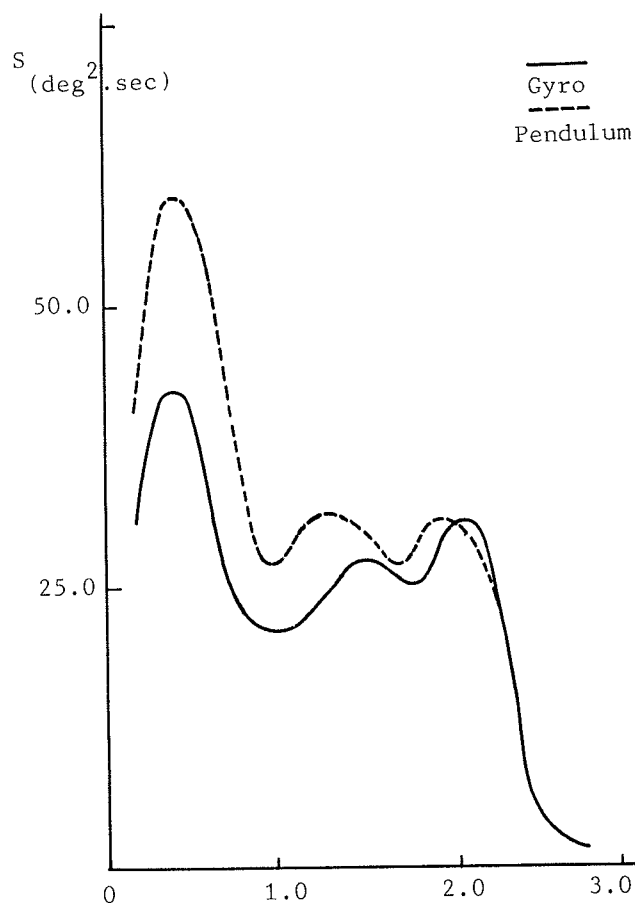


Fig. 10 Spectrum of rolling (12 knots, Quartering sea)

Although the results are still admissible in case of 4 knots, they are almost hopeless in case of 12 knots. The

mean rolling period is 4.5 second in following sea which is almost 70% more than the synchronous rolling period. It can be seen in the spectrum of rolling that the effect of the frequency of encounter due to ship speed is much larger than that of the narrow band characteristics of rolling (Figure 10).

Although several algorithms were tried to obtain the zero-up cross period of rolling, the mean period such obtained had the same tendency which was obtained by equation (2).

Consequently, it must be deduced that the mean period of rolling is affected by the incident wave spectrum. Therefore it will be close to the synchronous rolling period only if the frequency of the spectrum peak of waves is not far from the natural frequency of rolling.

Concluding the above, the rolling spectrum can be measured by pendulum unit but the synchronous rolling period can not be estimated by the mean period of rolling when the ship speed is high.

Although it is not implemented yet, the synchronous rolling period estimation routine of the capsize alarm system must be switched off when a ship is at high speed. This can be done by connecting the system with the position of a engine telegraph.

#### 4. FULL SCALE TESTS FOR EVALUATION OF THE SYSTEM

##### 4.1 The Method of the Tests

Although several difficulties remain unsolved, full scale tests were carried out for two fishing boats to evaluate how the system works at this stage. Each trial continued for a month.

Trials were executed automatically. The skippers were asked to set a cassette tape and to switch on the system every morning. A short comment of the weather was also requested to take note.

The capsize alarm system had run all the day by itself showing the 1/100 expected value of rolling angle as a reference. Raw data of each day were stored in a cassette tape for the offline analysis which would be done afterwards. A single side of cassette tape can store the raw data for 7 hours which was just enough for one day's experiment.

The system worked out without any troubles through the two full scale trials.

##### 4.2 The First Evaluation Test

The first evaluation test was executed at Kushiro, Hokkaido from October 5 to November 11, 1981 on a 9.97GT type small trawler.

The principal particulars and the conditions of the ship when we set up the system on board were as follows:

L :	12.600m
B :	3.200m
D :	1.080m
GT :	9.970t
△ :	2.730t
d :	0.84m
trim :	0.64m(aft)
KM :	2.163m
KG :	1.383m
GM :	0.780m
K/B :	0.53
T(sysc) :	3.8sec

Every day the ship leaves the port at about 5:30 in the morning for fishing and come back home after the noon. The average catch of fishes were 0.3 to 1.5 tons each day.

The ship had encountered very rough weather once during the test. On October 6, the ship left port at 6:00 in morning. The fishermen's union ordered to stop working because of rough weather at 6:40, so that the ship had come back to the harbor at 7:30.

Figure 11 illustrates the ship condition on that day. The curves represent the variation of mean rolling period, root mean square of rolling angle and C(t) coefficient such defined in Section 2.1 during this 90 minutes.

The variation of mean period of rolling seems to be reasonable in this case. The root mean square of rolling angle reached to the maximum value at 35 minutes after the departure. At this time the skipper might decided to come back. C(t) coefficient shows the reversed variation to the root mean square. In this case 1/100 expected value of rolling was used for the maximum rolling angle. It is not approved to use the value of 1/100 for the maximum expected rolling angle so that the absolute value of C(t) coefficient has no specified meaning. It must be considered as relative value.

According to the skipper's comment, he felt the weather to be rough when the root mean square of rolling angle was more than 4 degrees and to be calm when it is less than 2 degrees. In Figure 11, the root mean square of rolling exceeded 4 degrees at 30 minutes after the departure and was still increasing. It seems to be quite reasonable that the fishermen's union ordered to stop working on that day.

##### 4.3 The second Evaluation Test

The second evaluation test was carried out at Musashi-Cho, Ohita Prefecture from November 16 to December 16 with a 2.39GT type pole and line fishing boat.

The principal particulars and the condition of the ship when we set up the system on board were as follows:

L :	7.95m
B :	2.16m

D : 0.726m  
 GT : 2.39t  
 $\Delta$  : 3.01t  
 d : 0.307m  
 trim : 0.231m(aft)  
 KM : 1.457m  
 KG : 0.715m  
 GM : 0.742m  
 K/B : 0.599  
 T(sync) : 2.80sec

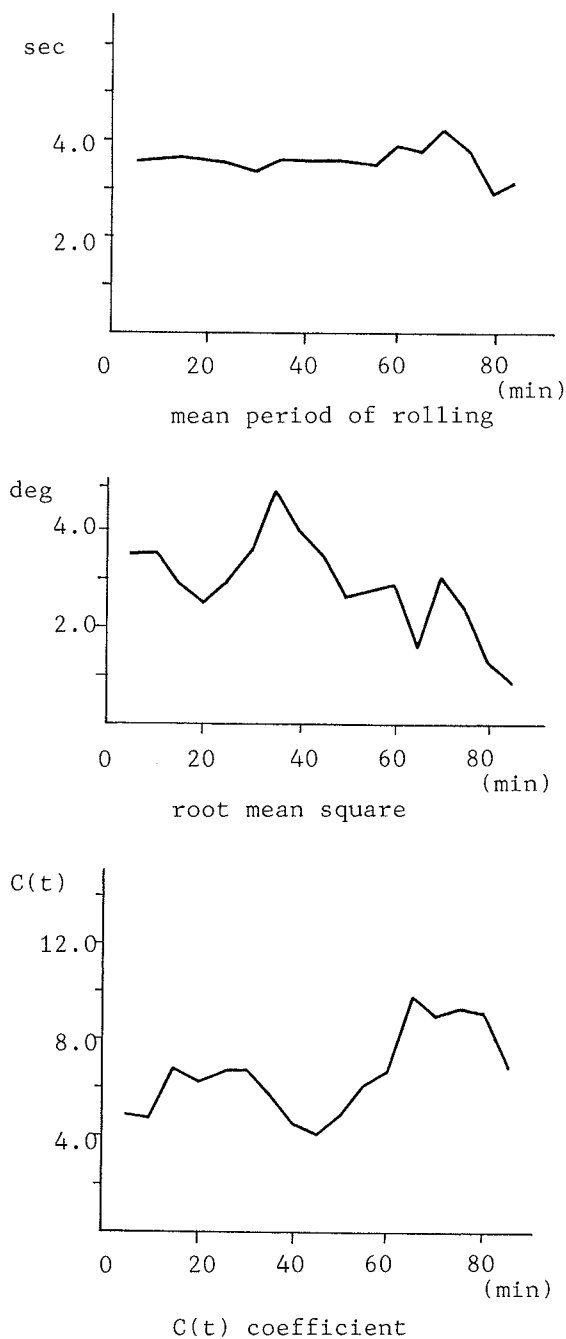


Fig. 11 First evaluation test

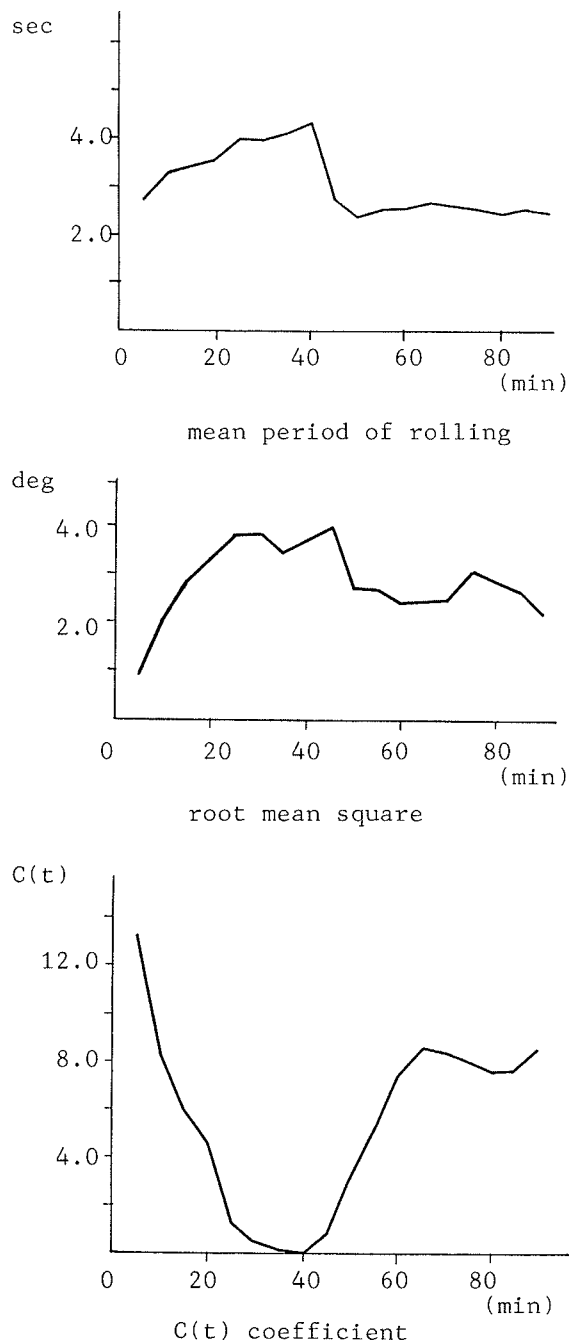


Fig. 12 Second evaluation test

The ship leaves the port at about 6:30 every day and come back in the afternoon. Fishing in this area is made in a rather protected area. The catch of fishes was about 30 to 120Kg each day. The ship had encountered rough weather once in this trial too. On December 13, the ship had to come back the port after 90 minutes of the departure due to the rough weather.

Figure 12 shows the time history of

this process. After the ship left the port, the root mean square of rolling angle got increased and reached to 4 degrees. The ship would be running in following sea because the mean period of rolling increased from 2.8 to 4.3 seconds. These variations decreased the  $C(t)$ -coefficient very abruptly and it went down as low as 0. The longer mean period of rolling apparently indicated the smaller metacentric height in this case.

The skipper changed the course at 35 minutes so that the mean period of rolling decreased to the level of 2.5 seconds. At 45 minutes the root mean square of rolling had begun to decrease and the  $C(t)$  coefficient recovered accordingly.

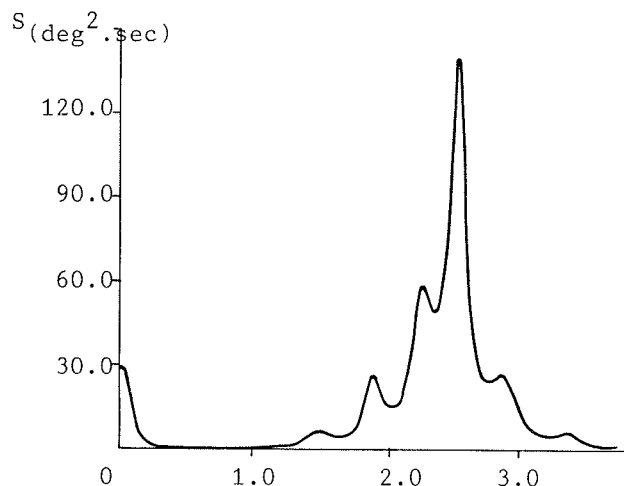


Fig. 13 Spectrum of rolling

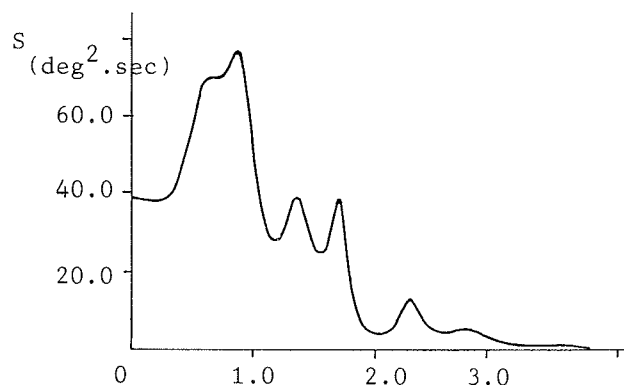


Fig. 14 Spectrum of rolling  
in following sea

The difference between head sea and following sea in spectra is clearly shown in Figures 13 and 14. In head sea, the peak of spectrum is located close to the synchronous frequency of rolling but the

spectrum has very concentrated power in low frequency range in following sea. This makes the mean period of rolling much longer as already mentioned in 3.2.

#### 4.4 Considerations

Summarizing the above results of full scale trials, it seems very difficult to specify the generalized index for capsize alarm as  $C(t)$ -coefficient. The  $C(t)$ -coefficient is fully dependent on the synchronous period of rolling which we expected to be able to estimate with the mean period of rolling in irregular waves. It becomes very clear that the mean period of rolling varies significantly according to the frequency of encounter of waves.

However, there are several useful values for a capsize alarm which can be measured by this system.

- (1) Root mean square of rolling angle: According to the skipper's comment the root mean square of rolling angle was very close to thier feeling of the weather condition. It seems to be possible to find the threshold level which indicates the critical roughness of weather. The time trend of the root mean square is also promising index for capsize alarm. If the root mean square reaches the certain level and if it is still increasing, the ship will be better to come back. Although this criterion is not general because it does not take into account the ship's own condition like  $C(t)$  coefficient, it still indicate the very important feature in capsize alarm.
- (2) Mean period of rolling: It was found very clearly by full scale trials that the synchronous period of rolling can not be estimated by the mean period of rolling at high speed. Especially in following sea, the mean period of rolling will get much longer than the synchronous period. On the other hand, the estimation can be done satisfactory at low ship speed. Mean period of rolling can be used to alarm the high location of the center of gravity if it is measured at low speed. It may be used to warn the broaching-to as well. The broaching-to is one of the most important concern for the safety of small ships because the Froude number of those ships are usually very large. Although the criteria which indicates the danger of broaching-to properly is not clear yet, the simultaneous occurrence of long period of rolling and some level of root mean square of rolling must be a very good index for the broaching-to.
- (3) Average heeling angle: This is certainly the important index for the safety of small ships. They must be

taken into account much more than larger ships.

Using the above values which are available by the present system, following types of alarms can be used practically:

- (1) LOWER THE CENTER OF GRAVITY: This will be issued by too long mean period of rolling only at low speed.
- (2) ADJUST THE HEELING ANGLE: This will be issued by too large and continuous average heeling angle.
- (3) SLOW DOWN THE SPEED: This will be issued by simultaneous occurrence of the certain level of root mean square angle and long period of rolling.
- (4) COME BACK TO THE PORT: This will be issued by large root mean square of rolling angle modified with its time trend.

The tentative threshold for these types of alarm could be specified now, but extensive experiments are requested to make them more reasonable.

## 5. CONCLUSIONS

A prototype of micro-computer based capsize alarm system was developed and its characteristics were studied. It can be said that the data processing ability of

micro-computer makes the capsize alarm system much more flexible and reasonable than it used to be.

Following types of alarm are able to be issued with this system:

- (1) Lower the center of gravity.
- (2) Adjust the heeling angle.
- (3) Slow down the ship speed.
- (4) Come back to the port.

The reasonable threshold levels for these types of alarm must be specified by continuing the extensive full scale trial in future.

A pendulum unit was used to measure the rolling angle in this system to reduce the cost. Although the amplitude ratio of pendulum and ship rolling is large, it can be compensated satisfactorily as far as the spectra of rolling angle are concerned.

## ACKNOWLEDGMENT

This project was organized by the committee of stability meter development at JCI. Authors acknowledge the members of committee for their very kind cooperations. Authors thank skippers also who were very cooperative at full scale trials and made very useful comments for the principle of the capsize alarm system.

## Discussion

### C. Kuo (University of Strathclyde, UK)

This is a most interesting paper on the use of advances in microcomputer technology to aid ship performance and to improve ship stability. Based on the experience gained, would the authors give their views on the following points:

- a) What is a typical cost of such a system?
- b) How reliable is the hardware if it is used over longer periods at sea and would it be possible to replace pendulum by other systems?
- c) Have you considered alternative criteria in place of the area ratio function  $C(t)$ ?

### Author's Reply

Thank you very much for your comments and questions Professor Kuo.

(a) I have no reliable estimation of a typical cost of this system in market. But the material cost of the present system was \$500 without a cassette tape handler. So it will be possible to produce this system less than \$500 in mass production.

(b) Only one part in this system which is mechanical is a pendulum. This can be replaced by U-tube, water bubble on liquid on a rate transducer. However we have no doubt on the reliability of a pendulum at all.

(c) We settled the  $C(t)$  coefficient just to configure the system at the beginning of this project. Now we think it is better several criteria for every specific aim of warnings and alarms to make them instructive for skippers. We should find the proper threshold levels in future.

### W.A. Cleary, Jr. (U.S. Coast Guard, USA)

Prof. Koyama and his associates are to be congratulated for the development of a capsize alarm system which at \$500 is of low cost such that it might be actually used. In 1968 the cost of a similar system was \$15,000. It is important that the cost be kept low or small vessels operators will not buy it.

Also if the system becomes an annoyance to the operators it will be removed (or simply turned off).

The system must be able to accept not only fish loads, and changes in fuel oil during the voyage but also ice loads and an allowance for fishing gear heeling moments or it will give the fishermen (operator) false confidence.

Finally, we must TRAIN the operator (perhaps by videotape of model experiments) so that he will understand the limit of use of the system. Only by training will he

gain full confidence with the system.

Author's Reply

Thank you very much for your very important comments to make the system be accepted by fishermen.

We are thinking the way to estimate the location of the center of gravity of ships. It can be done satisfactorily when the ship

speed is very low. So it will be necessary to link the system with the information like the position of engine telegraph.

I agree that the operator must be trained to understand the limit of use of the system. At the same time we should make the system as comprehensive as possible. Many Japanese fishermen want to have some reasonable standard which tell them the timing to go back to port nowadays.



## THE THEORETICAL AND EXPERIMENTAL RESEARCH IN A SEMI-ACTIVE ANTI-ROLL TANK

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### ABSTRACT

A new type of Semi-Active Anti-Roll Tank is presented in this paper. This type of tank seems to have the advantage of both passive and active tanks, thus making full use of their passive effect so as to save power. The method of control in the system is phase-control.

Theory and test have proved that in obtaining the same anti-roll effect as active-tanks, the power needed by Semi-Active Tanks can be greatly reduced and the size of pumps can also be diminished.

Results of hydrodynamic model test and theory of anti-rolling of ships with this new tank are presented in the paper. It is proved that the anti-roll efficiency of this new tank will reach about sixty per cent. Being affected by winds and waves, the ships in navigation or working in mooring conditions would be subjected to pitch, roll and heave motion of six degrees. The rolling is most harmful, because it makes crews and passengers uncomfortable, affects the safety of transport and the measuring accuracy of scientific research work, and decreases the percentage of hits in fighting. Consequently, to decrease the roll of ships is one of the most important tasks in shipbuilding. There are two ways to stabilize roll, one of which is to use fin stabilizers. Since they were used in 1923, they have become more and more perfect. In China quite a few of fin stabilizers have been installed in ships. However, this stabilizer is suitable only for ships navigating at speed not lower than 16 knots. The other kind of stabilizers is the anti-roll tank which may be used in ships at lower speed or mooring condition. Generally, according to the working principle, anti-roll tanks can be divided into two kinds—the active and

passive. The active anti-roll tank requires additional energy (pump) and the passive tank employs the motion of water during the roll of ships without the need of any energy. The stabilizing efficiency of the passive tank is relatively poor and difficult to be controlled. But the active tank is also seldom used, being large in size complicated in structure and expensive in cost. For instance, according to calculation, in condition of sea state 6, to stabilize a ship with displacement of ten thousands tons to a roll angle of 5 degrees requires a power of about 1500 kW. It is difficult to get such a large pump and strong power.

In view of this situation, we have investigated a new type of anti-roll tank by the principle both of the active and passive tanks namely, the Semi-Active Anti-Roll Tank, which might greatly reduce the size of pump and provided power.

### 1. The Principle and Construction of Active and Semi-Active Anti-Roll Tanks

1.1 The construction and principle of general active Anti-Roll Tank is shown in fig. 1

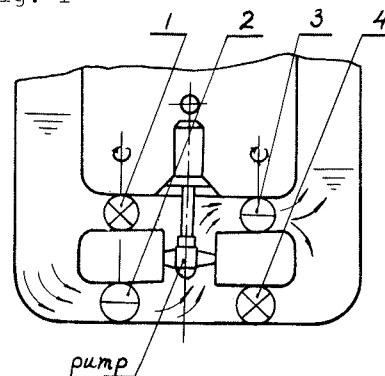


Fig. 1

A pump, driven by a motor or other engine, located in the ship, conveys the water from port to starboard so that it creates a difference of water level in the tanks. In the meantime the valves 1, 4 are shut and 2, 3 are open. In consequence, a stable moment acting on the ship in clockwise order is created to balance the disturbing moment from the sea. On the contrary, when the disturbing moment is in the opposite direction, valves 1, 4 should be shut and 1, 3 open. Valves 1, 2 should set up on one shaft and 3, 4 on the other. Each valve driven by a hydraulic cylinder is constructed from several vanes of air-foil section. Since the stable moment produced by the pump is independent of the speed of the ship, the active anti-roll tank is particularly suitable to stabilize ships working in mooring position or sailing at low speed.

## 1.2 The Construction and Principle of Semi-Active Anti-Roll Tank.

Fig. 2 illustrated a new construction of semi-active tank and its working principle.

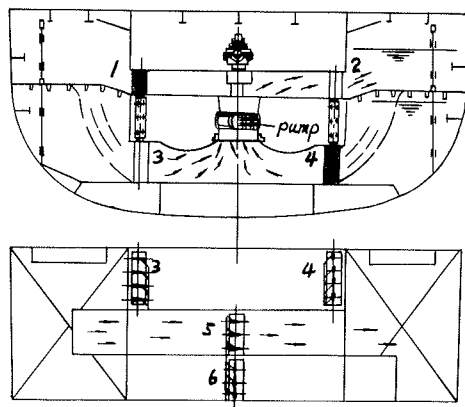


Fig. 2

The difference between the semi-active tank and the active is that the water tanks of each board of the ship are separated into upper and lower tanks by horizontal spacers. Two bypass pipes are installed between both tanks. The upper part of the water tank in the port board is linked with the lower part of the water tank in the starboard by bypass pipe possessing valve 5. Furthermore, the upper part of the tank in the starboard is connected with the lower part of the tank in the port board also by another bypass pipe with valve 6. These two valves 5 and 6 located in bypass pipes are controlled by one hydraulic cylinder. When the right tank is required to be filled with water, we shut the valves 1, 4 and open valves 2, 3 as well as 5, which makes the water in upper part of the tank in the port board move to the lower part of the tank in starboard.

The valve 6 is only controlled in this way that when the valve 5 is in the open

position, it should be in the closed position so that the water moving from the left to right is composed of two parts, one of which is transported by the pump, and the other flows through bypass pipe by self-weight. In consequence, to produce the same stabilizing moment in a certain time requires the quantity of water provided by the pump to be equal to the half of that given by the pump in the active tank. It reduces the capacity of the pump nearly to fifty per cent, which is the main advantage of the semi-active tank. When the state of wind and wave varies, adjusting the quantity of water entering the lower tank may be achieved by regulating the stroke of the actuator of the bypass valves. To make these valves work in various degree of opening results in the convenient controlling of the moving water in semi-period of rolling. The pump may be made with fixed blades to reduce the cost of it.

## 2. Hydrodynamic Research of Semi-Active Anti-Roll Tank

A number of tests have been carried out for obtaining the hydrodynamic characteristics of the semi-active anti-roll tank. These characteristics involve:

- (1) The relation between the water quantity moving in the tank with the semi-period of rolling and the position of the pump blade  $V = f(T, \phi)$  as well as the maximum capacity of the pump  $N_{max} = f(\phi)$ .
- (2) The discharge coefficients of bypass valve under its various opening  $\mu = f(\alpha)$ .
- (3) The Dynamic moment characteristics of the main valve and the bypass valve during opening and shutting.

A model pump of specific No. 700 with diameter of the runner 300 MM and 680 R.P.M. was used in testing. Each volume of the four water tanks in both boards is 1.2 x 1 x 0.6 M (length x width x height). The size of bypass valve is 0.315 x 0.315 M (width x height). The inlet and outlet valves on both sides of each tank are operated by actuators respectively. The vanes of the two bypass valves on pivots, which should be operated simultaneously, are coupled individually to one regulating link operated by a hydraulic cylinder as shown in Fig. 2.

The action period of the above mentioned valves are controlled by an electromagnetic valve operated by a timing relay with regulable time of action. The time of intermittent signal is measured by an oscillograph. The action time of valves depends on the pressure of oil in the cylinder. In our testing the action time was held about 0.4 second. The water depths in boxes are measured by level sensing devices. The depth multiplied by the area of the box equals the quantity of water.

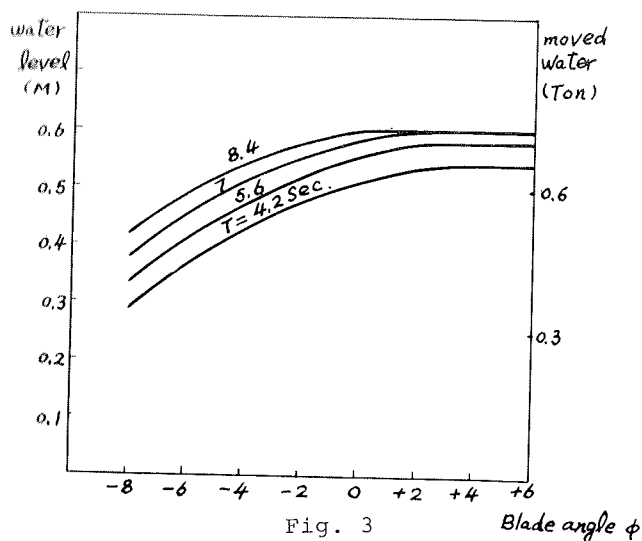


Fig. 3

Fig. 3 shows the curves of semi-rolling period  $T$  obtained by adjusting the time opening main valves and the head of water (the difference of water level in both sides). As shown in Fig. 3, along with the increase of semi-period and the angle of pump blade, the head is rising. For every semi-period, particularly the longer period, the water level does not continuously rise until the angle increases up to a certain position (in Fig. 3  $\phi = 0$ ). The reason is that when the head is 0.5 M, the water surface at the inlet of the pump is lower than the highest point of the upper flange of the passage. Air is drawn into the pump, which causes the flow to be broken. This phenomenon will affect the flow of the next period. As the valve is opened, the water is unable to fill the inlet of the pump immediately. It reduces the quantity of water removed by the pump. Such phenomenon can be cured by lifting the spacer to increase the volume of the lower water tank. Fig. 4 shows the curves of the power output of the pump. From these curves we can see that the power output depends only on the position of its blades and is independent of the level of water. The upper flange of the outlet located in the highest point of the water level of the tank caused the above mentioned situation. Though the water level in the inlet varies, as a result of disturbance of the wave, the fluctuation of the pump output is not so obvious. The data in curves represent the maximum value of the pump capacity.

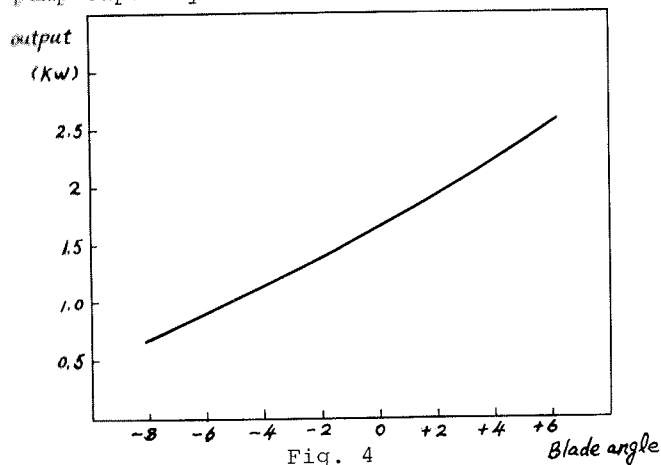


Fig. 4

The testing for measuring the flow quantity coefficient has been carried out to get the passage characteristics of the bypass valves. Keeping a certain difference of water level in both tanks, we open the valve at once to make the water flow horizontally. According to the time recorded during the whole process, the flow quantity coefficient  $\mu$  can be counted by the following equation.

$$T = 2F_1F_2/F_1 + F_2 \cdot \sqrt{H}/\mu f\sqrt{2g} \quad (1)$$

where  $F_1$ ,  $F_2$  ----- the areas of horizontal section of the tank,  $f$  --- the area of the bypass pipe; Curve  $\mu = f(\alpha)$  is illustrated in Fig. 5

Using this curve, the quantity of water moved by pump can be counted, if the adjustment of bypass valves is to be used (the rpm and blade of the pump are unchangeable). At this moment a part of the quantity of water  $V$  pumped by the pump in semi-period

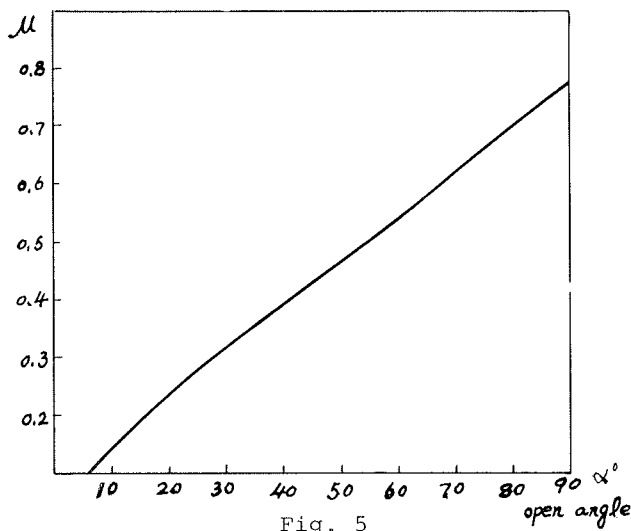


Fig. 5

will pass away through the bypass valve with degree of opening  $\alpha$ . By the end of period  $T$  the water remained in the upper tank will equal the half of the quantity needed to move. The other half will be obtained by gravity flow from the other upper tank through the other bypass pipe. The calculating formula is:

$$V = 2\mu f\sqrt{2gH_0} T \quad (2)$$

$H_0$  ----- The average head of water tank equal to half of the maximum head by the end of semi-period.

Owing to the moment acting on the vanes of the valve arrange on the pivots with eccentric the flow will push vanes to reduce the moment provided by the actuator.

$$M = C_m \frac{a^2 b}{8} H \gamma \quad (3)$$

where  $M$ -hydraulic moment acting on the valve.

$a$  - The width of the vane.

$b$  - The length of the vane.

$H$  - Operating head.

$\gamma$  - Specific weight of water.

$C_m$ - Moment coefficient of vane.

Fig. 6 shows the curve of moment coefficient of the valve under various degrees of opening. As shown in the figure, the value of  $C_m$  does not equal to zero in the full opening ( $\alpha = 90^\circ$ ). The reason is that when the water runs out from the tank, the irregular disturbance of fluid field causes the unsymmetrical flow passing the vanes.

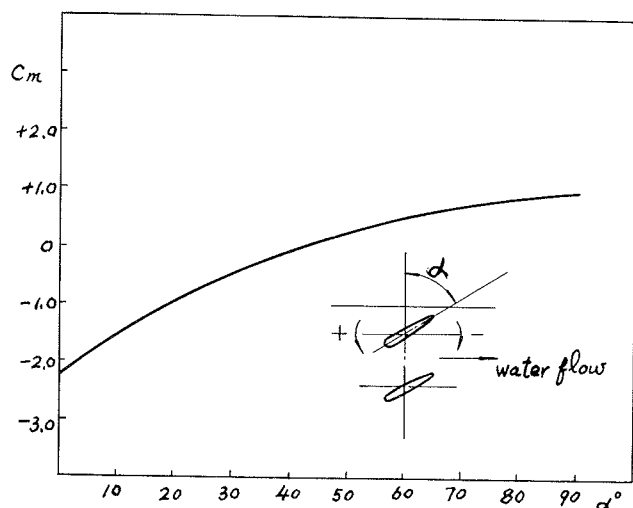


Fig. 6

#### Scalar Conversion

According to the principle of hydraulic similarity, the results of model testing may be used for prototype through scalar conversion which are shown as following. The scale of geometric length

$$\lambda_L = l_1/l_2 \quad (4)$$

The scale of rolling period of the ship

$$\lambda_T = \lambda_L^{1/2} \quad (5)$$

The scale of water head in the tank

$$\lambda_H = \lambda_L \quad (6)$$

The scale of the diameter of the pump

$$\lambda_D = \lambda_L \quad (7)$$

The scale of rpm of the pump

$$\lambda_n = \lambda_H^{1/2} \cdot \lambda_D^{-1} \quad (8)$$

The scale of opening-shutting period of valves

$$\lambda_{t_1} = \lambda_n^{-1} \quad (9)$$

The scale of the operating time of valves

$$\lambda_{t_2} = \lambda_{t_1} \quad (10)$$

The scale of power of the pump

$$\lambda_N = \lambda_n^3 \cdot \lambda_D^5 \quad (11)$$

### 3. The Installation of Bench Test of Semi-Active Anti-Roll Tank

The addition to above hydrodynamics testing, the dynamic simulated testing of the whole system should be carried out. The aims of research are:

(1) The existence of possibility of opera-

tion from gyrosignal to the electro-hydro-control system as well as the operating mechanism.

(2) The anti-rolling ability of the new type tank under various frequencies of disturbing force.

(3) The influence upon the effect of anti-rolling by using method of phase-lead and the best phase angle for opening valves under various frequencies of disturbing force. The model of the tank is supported on the supporting stand with bearings. Two actuators are linked with both boards of the tank on each side. The stretching out and drawing back of the actuator cylinders make the tank itself roll round the center of gravity. Outside disturbing moment given by a generator of low-frequency is fed to an analogue computer, which simulates the motion of a ship in the sea. Depending on the rolling formula of the ship, the computer gives an output, through modulation and amplification, to an electro-hydro-servo-control system which operates those two actuator cylinders and results in a rolling angle  $\theta$  corresponding with the outside disturbing moment. This angle  $\theta$  equals that angle of roll without stabilization. Meanwhile, operating the pump to move the water, the direction of which is controlled by valves, results in different levels of water in each side of the tank. Consequently a stabilizing moment is established. This moment can be calculated by measuring the volume of water with a level sensing device and be converted into electric quantity. After comparison of the above electric quantity with the disturbing moment the result should be given to the computer as a feedback. The eventual rolling angle produced by the controlling system represents the residual rolling angle with stabilization. Control of valves is executed by means of the phase-lead device and the solenoid control valve. The synthesized quantity of the rolling angle, angle velocity and angle acceleration measured by the gyro is used as the signal of the control. (Fig. 7)

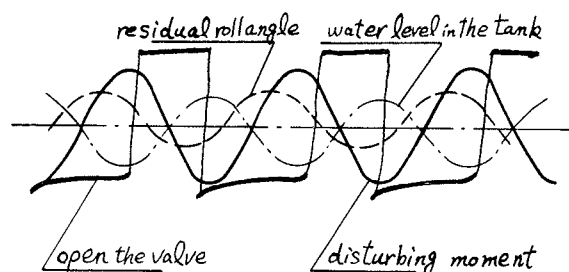


Fig. 7

In order to fit the complicated sea state and the disturbance of irregular waves, the method of control used in fin stabilizer, i.e. the angle  $\theta$ , angular velocity  $\dot{\theta}$  and angle acceleration  $\ddot{\theta}$ , a three-signal-control is adopted here. Using this control we can obtain, under any disturbing moment

with various frequency of wave an efficient effect, provided the chosen rule of signals is suitable. As the valves are controlled, the head  $Z$  in water tank is controllable. Using the method to control the phase of opening valves, we can arbitrarily adjust the head  $Z$  which is the main purpose of researches on the semi-active anti-roll tank.

The curve shown in Fig. 8 indicates the behavior between the roll angle  $\theta$  (double amplitude) and the variety of the capacity of phase lead, while the resonance frequency is 1.16. Obviously, the stabilizing efficiency depends greatly on the controlling phase.

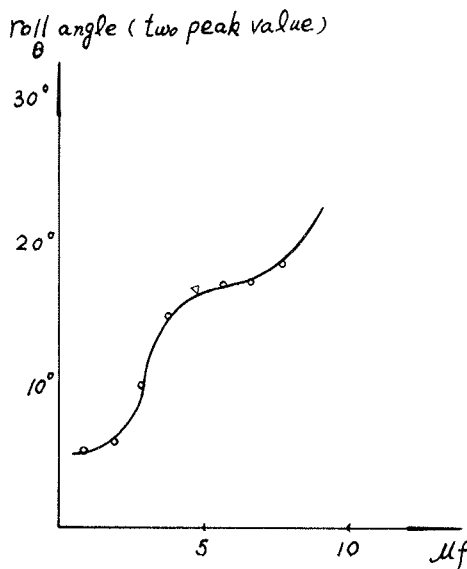


Fig. 8

The efficiency in low frequency is fairly good, when the capacity of phase-lead and the angle signal are a bit larger (Fig. 9)

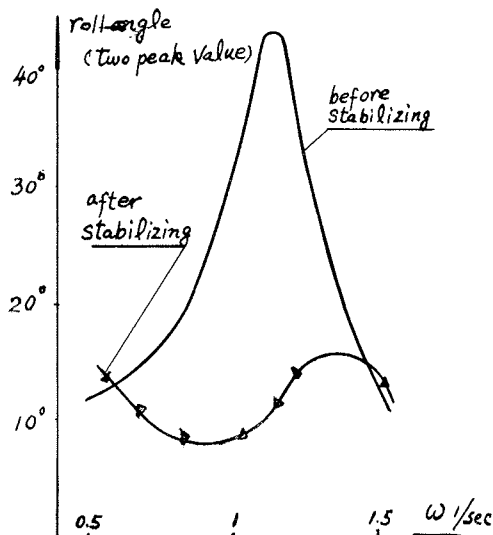


Fig. 9

Under the smaller capacity of phase lead and the larger angle velocity as well as the angle acceleration the efficiency in

high frequency will be better. (Fig. 10)

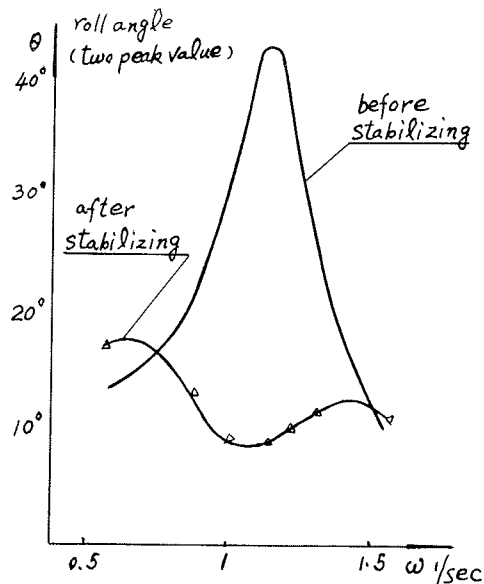


Fig. 10

#### 4. Conclusion

Main items which may be deduced from the results and discussions are summarized as follows.

1. The efficiency nearby the resonance region is rather ideal. The roll angle (double amplitude) is reduced from about 40° to 9°. The efficiency may approach 80%. But in the range of the lowest frequency the roll angle rises on the contrary. In the average the stabilizing efficiency of this new tank may amount to 60%.

2. The power needed by the semi-active anti-roll tank is smaller than that of the active tank by approximately fifty percent.

3. This research work is done only under conditions of regular waves. The work concerning unregular waves will be done in the future.

## Discussion

G.G. Cox (David Taylor Naval Ship R & D Center, USA)

The authors present an interesting idea for application to the future design and evaluation of antiroll tanks. I wish to make two comments.

a) The transfer functions shown in Figures 9 and 10 are fairly typical for antiroll tanks and should not be taken as indicative of effectiveness for an actual installation aboard a ship at sea. This requires evaluation for a ship in realistic short crested seas and is only possible using the type of simulation approach discussed in the Seakeeping Committee Report of the 15th International Towing Tank Conference. Model tests using a small model of the tank in a model ship running in longcrested irregular waves raises questions of scale effects, impossibility of sufficient test runs and unrealistic wave conditions for antiroll device evaluation.

b) It should also be realized that any antiroll tank causes a reduction in GM (this concept has four free surfaces). Hence it is not a suitable application for any ship where risk of capsizing is of concern, e.g. fishing boats.

In conclusion I wish to thank the authors for a new approach to the antiroll tank concept and hope they will continue with further development and evaluation.

### Author's Reply

I would like to thank Mr. Cox for his valuable comments.

I agree with your first view that the curves in figs. 9 and 10 are typical. However these data in pictures, which were directly got from my test may state the efficiency of this tank.

Concerning the second comment I fully agree with you.

# MAKING EFFECTIVE USE OF SHIP-STABILISATION DEVICES

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## ABSTRACT

This paper deals with the mechanism of ship stabilisation systems and its aim is to enhance the understanding of the way in which roll reduction is achieved. A brief review is first made of the available systems before the bilge keel is taken as an example to demonstrate theoretically the importance of the combined effects of damping and added moment of inertia on the roll response of the system. The theoretical techniques are then discussed and the method based on a distribution of sources and vortices is highlighted. Good agreement was obtained from model experiments using a two-dimensional section to check the theory. Finally the practical applications of the research findings are considered and illustrated.

## NOMENCLATURE

B	Beam of a ship
$b_t$	Tank breadth
D	Depth
$GM_T$	Transverse metacentric height
g	Acceleration due to gravity
h	Tank breadth
KB	Keel to buoyancy centroid distance
KG	Keel to centroid distance
L	Length (ship/model)
$l_t$	Tank length
R	Roll reduction factor
$S_R(\omega)$	Roll response spectrum
$S_\omega(\omega)$	Wave spectrum
T	Period
$T_d$	Draught
$\Delta$	Displacement
$\gamma_t$	Tank damping coefficient
$\gamma$	Ship damping coefficient
$\lambda$	Draught to beam ratio
$\sigma$	Sectional area coefficient
$(\zeta_{1/3})_s$	Significant Roll Amplitude (stabilised)
$(\zeta_{1/3})_u$	Significant Roll Amplitude (unstabilised)
$\omega$	Excitation frequency (radians)
$\omega_0$	Natural frequency of ship

## 1. INTRODUCTION

The terms "stability and "stabilisation" are closely related and have a wide range of meanings, depending on the basic discipline and experience of the user. In the ship design context they imply that a vessel must be able to operate safely in hazardous seaways, either on the basis of its design features alone, or because stabilisation devices have been installed for this purpose.

The stability of a ship depends on many factors, ranging from its design characteristics, such as the shape of the hull, to the efficiency of the crew handling its operations, and not all the parameters are within the designer's control.

Because of the geometrical shape of traditional hull forms, the most significant motion of a ship, especially from the capsizing safety point of view, is without doubt the rolling motion. This is one of the reasons why traditional "stability calculations" are usually concerned with determining the self-restoring capacity of a hull when subjected to heeling.

Regulations on the stability of ships are at present based solely on statistical considerations although extensive work is now in progress which should lead to the incorporation of the effects of ship motions into stability assessment criteria.

This will be a logical step forward since ships in seaways are, more frequently than not, subject to such motions, but the problem is complex and it is not expected that stability criteria which are meaningfully advanced and practical will be devised in the near future. Designers have found it possible, however, to reduce the magnitude of rolling motion in the meantime by the installation of both active and passive stabilisation devices on their vessels.

Roll reduction is usually achieved by either passive devices such as bilge keels or, passive tanks, or active devices, e.g. controlled tanks, and active fin-stabilisers, and current understanding of their effects can be summarised as follows:

- (1) Bilge keels - reduce roll motion by drastically increasing roll damping
- (2) Passive tanks - achieve the same result by draining roll energy from the ship, or by providing a roll opposing moment when it is needed, or both
- (3) Active devices - generate a couple to counteract the roll excitation moment.

In spite of the success that has been claimed for most of the available stabilisation systems or devices, there is enough evidence to suggest that some of them are not always fully effective.

For this reason a research project was initiated at the University of Strathclyde in 1978 to examine the problem of ship roll stabilisation, and to evaluate, in particular, the influence of

bilge keels on rolling motion.

The present paper outlines some of the research findings and considers the practical implications.

## 2. BASIS OF ROLL STABILISATION

The amplitude of roll is closely related to human comfort and work efficiency on board, possible shifting of cargoes, and the safety of the ship as a whole, so that it has always been the designer's first task to ensure that a vessel has "good rolling characteristics".

A ship attains large amplitudes in the rolling mode even though the external couples acting on it may not be too extensive, because the inherent damping properties of most hulls are very low. Since this is the case it appears to be relatively easy to conceive a system that can produce a moment to counteract that induced in the environment. Such systems are commonly known as "roll stabilisers" and their function is to minimise roll motion, rather than to enhance transverse stability as their name might imply.

On the basis of the foregoing it may not seem a very demanding task to ensure that the rolling amplitudes of a ship operating in the seaways do not exceed an acceptable level. However, when we take into consideration the need to fulfil the criteria for cost effectiveness in all the elements making up a "good design", the number of constraints is sufficient to limit freedom of action.

Thus, the designer may try to produce a hull form that will not resonate in the seaways of interest, or, in the belief that the ship will resonate in any case in a random environment, he may increase damping in roll so that the roll response is reduced. In fact a ship will only respond in the frequency of excitation when regular steep waves act upon it for a sufficiently long time. In seaways a ship tends to roll in her own natural frequency and many believe that there is a considerable increase in roll motion in the presence of seas with predominant wave frequencies in the resonant region of ship roll response. This opinion, however, is not universal and fearing that it would be very difficult to avoid resonance over a wide range of frequencies and that this could lead to severe operating penalties for the ship, designers have endeavoured to increase roll damping by introducing stabilisation devices.

With these points in mind, the principles of roll stabilisation will now be considered in relation to bilge keels and passive tank stabilisers. The choice of these two systems is based on their suitability for demonstrating an alternative use of their roll reducing capabilities.



## Bilge Keels

Bilge keels are structural members having a depth from around 1% to 5% of the ship's beam, placed in the bilge region of the hull to "obstruct" transverse flow around the section.

In their simplest form they are made of flat bars welded to the turn of the bilge, extending longitudinally from one-third to two-thirds of the ship's length.

Their principle of operation consists of extracting energy from the rolling ship and dissipating it, primarily in the form of vortices shed from the edge of the keel plate.

Bilge keels are commonly regarded as a means of radically increasing roll damping and thus most of the effort to assess their hydrodynamic action has been related to their effect on roll damping. They generally increase roll damping by a factor of 3 at zero speed and on an average by a factor of 2 at forward speeds.

Theoretical treatment of the effects of bilge keels on roll damping remains a very difficult task because of the viscous nature of the flow and numerous empirical or semi-empirical relations have been developed to estimate the bilge keel contribution to roll damping for practical applications, see Ikeda et al [1] and [2].

These investigations have yielded data on the magnitude of damping (equivalent linear) brought about by bilge keels of varying sizes, and the resulting roll response curve shows in most cases only a reduction in the height of the roll resonant amplitude response. The implication is that the sole effect of bilge keels is to increase the magnitude of roll damping and hence reduce the amplification factor at resonance.

The technical literature also suggests that the bilge keels' contribution to roll moment of inertia is of some significance, see for example Iddle and Baker [3], Wendel [4], Szajnberg et al [5]. In spite of these findings however, this latter effect seems not to have received the attention it deserves in design and the ship motion studies.

## Free Surface (Passive) Tank Stabilisers

There are two distinct schools of thought regarding the roll-reducing action of passive tanks and their relevant arguments are as follows:

### Watts-type Tanks [6]

Linearised roll motion resembles very closely a second order mechanical oscillator which foreshadows the phenomenon of resonance. The most important features, in relation to the present context, are the peakiness of the response for systems with small damping, rolling being one such system, and that there is a 90° phase lag at resonance between the motion of the

system and a harmonic input.

It is logical to assume that the mass of water contained in a free surface tank is also a second order mechanical oscillator and it will slosh back and forth when disturbed, at a frequency depending upon the dimensions of the tank and the depth of water it contains.

It is, therefore, possible to install such a tank on a ship and to adjust its natural frequency so that it coincides with the natural frequency of the ship.

When a synchronous harmonic input is then applied to the ship the resulting ship motion which lags the excitation by 90° will be the excitation to the tank system. Hence the water motion in the tank will lag by 90° the ship motion and thus by 180° the environmental excitation moments.

The moment exerted on the ship by the displaced mass of water will, therefore, be in opposition to the applied external excitation.

The passive tank stabiliser takes some energy from the ship to reduce the roll motion but its major effect is achieved by the opposing moment, and the internal damping of the tank is considered to be relatively unimportant.

With Watts-type tanks two requirements must be fulfilled for maximum stabilisation:

- (1) The frequency of the passive tank stabiliser should be the same as the ship roll natural frequency.
- (2) The tank turning moment should be in antiphase with the wave excitation moment.

This implies that a deviation from a 90° phase lag between wave-ship and ship-tank may even result in an increased turning moment and such a deviation is almost unavoidable in practice as phase-lag is appreciably affected by a number of factors.

Does this mean that Watts-type tanks are likely to destabilise the ship? To answer this properly the coupling effects must be taken into account. Although strictly incorrect the usual treatment is to consider the tank system as an equivalent damped vibration absorber coupled to the main system, the ship. If a ship is resonating with a harmonic excitation and a passive tank stabiliser is tuned to that frequency it will have the following effects:

- (1) A drastic reduction of the response of the stabilised vessel, at the resonant frequency of the same vessel when unstabilised.
- (2) Theoretically there are now two new resonances but experimentally only one peak at a lower frequency is indicated.
- (3) This new resonant response of the stabilised vessel is always lower than that of the same vessel unstabilised because the coupled system is better damped.

The two latter effects mean that even though the overall roll response will improve, an increase in the roll amplification factor may result, at frequencies

lower than the uncoupled resonant frequency. Therefore, since it is not possible to maintain the tank opposing moment in anti-phase with the wave excitation moment, tuning the tank to the resonant frequency at the unstabilised ship will not necessarily give the minimum amplification factor for the coupled system since other factors are also important.

#### Flume Tanks [7]

For the second school of thought the tank systems are relatively heavily damped so that at resonance the roll energy of the ship is transferred to the stabiliser which converts this energy into heat and hence the roll motion is greatly reduced.

For the flume tanks, tuning seems to be the major requirement. The traditional means of tuning a damped vibration absorber is to ensure that the two peaks in the frequency response curve are almost equal. If the damping in the system is the optimum, then the response curve in the region between the two peaks is nearly flat. For maximum efficiency, there must be ideal values for depth of water, tank size, set of restrictions in the tank and optimum damping for every ship condition, notwithstanding other design and operational constraints regulatory body requirements, and so on.

Achieving all these optima would probably be uneconomical and a design is usually chosen which gives the minimum possible peak response for one or a number of applied wave slopes. Thus, the frequency response curve of the stabilised ship will again have, most of the time, a single resonant peak at a frequency lower than the resonant frequency of the unstabilised vessel.

Both tank types, therefore, affect roll response in two distinct ways: the roll amplification factor is drastically reduced and the resonant peak is shifted to lower frequencies, usually by a large amount.

If we accept for the moment the damping effect of passive tank stabilisers, an alternative means of reducing the ship roll response is possible. By using the frequency shift as a means of avoiding the resonant bandwidth of the sea spectrum it should be possible at either the design stage or the operational stage to prevent the ship from being excited at resonance by large environmental moments. In other words it should be possible to "detune" the ship from its operating environment. On the other hand, if this effect is not properly considered, some of the shortcomings of passive tank stabilisers can easily be explained.

According to the above discussion, there are three possible methods of reducing rolling motion:

- (1) Drastically increase damping (damping stabilisation). This is effective especially at resonance, where damping

has its maximum effectiveness. (e.g. Use of bilge keels, flume tanks, etc.).

- (2) Directly oppose the excitation moment and hence reduce the disturbing effect of the environment (e.g. using Watts-type tank stabilisers). This, however, is difficult to achieve without proper control.
- (3) Introduce detuning effect in conjunction with the other two factors.

### 3. METHODS OF ASSESSMENT

In spite of recent advances, both design criteria and criteria for assessing stabiliser performance are anything but specific, and allow wide discretion in determining whether or not a specific roll stabiliser is satisfactory. Most of the available methods for assessing the effectiveness of roll stabilisers are based on criteria which are related directly or indirectly to roll frequency, or damping ratio, or both. The most popular ones are:

#### (a) Response Function

The usual practice is to find the percentage reduction in the roll response at resonance by comparing the response function of the stabilised and the unstabilised systems

#### (b) Statistical Averages

The usual statistical averages, calculated from roll response spectra, are those of 1/nth highest roll amplitudes and in particular "significant roll amplitudes" (i.e. the average of 1/3 higher roll amplitudes). A commonly used statistical measure of effectiveness is the roll reduction factor, defined by

$$R = 1 - \frac{(\zeta_{1/3})_s}{(\zeta_{1/3})_u}$$

where  $(\zeta_{1/3})_s$ ,  $(\zeta_{1/3})_u$  are stabilised and unstabilised significant roll amplitudes respectively.

#### (c) Percentage Exceedance Curves

The percentage of rolls exceeding certain roll amplitudes is calculated and curves are drawn for the stabilised and unstabilised systems.

#### (d) Critical Damping Ratios

Another way of determining the effectiveness of a stabiliser is by using the ratio of critical damping for the ship with and without stabiliser. This method is independent of the seaway and measurements made during calm water trials can be used.

Of the four methods employed in practice (a) and (d) overestimate the actual effectiveness of the stabiliser since the equation is not totally correct, i.e. the amplitude

of roll is reduced at a given frequency but that level of the reduction is not achieved at other frequencies of importance. Method (b) provides an overall assessment and method (c) involves extrapolation which is often not justified.

Furthermore, because of the variability in the characteristics of sea spectra only peak frequencies and the associated energy in that vicinity can be considered.

#### 4. PROPOSING AN ALTERNATIVE APPROACH

Since diminution of roll amplitudes has been the major concern of designers, most stabilisation devices have been treated simply as "roll dampers". Adequate attention has not been given to the other important effects of stabilisers on roll response. For example, it is known that bilge keels may substantially increase roll period and yet this effect is usually ignored. Similarly, stabiliser tanks produce a significant shift of the frequency response and this has been observed by many but no comments have been made on this phenomenon. It is our firm belief that not enough attention has been paid to these important effects and that they can provide a means of assessing the effectiveness of stabilisers and an explanation of this effectiveness. To emphasise this point we would suggest that:

"The installation in a ship of a roll reducing device will have a two-fold effect:

- (a) a reduction in the magnitude of the response
  - (b) a frequency shift of the response
- Both effects can be of equal importance.

To make effective use of such a device in practical design it is essential always to relate the roll response of the vessel to the sea state(s) in which it is planned to operate".

The practical implications of this hypothesis will now be examined.

#### Practical Implications

The roll response of a ship, both with and without stabilisers (bilge keels or passive tanks), can be illustrated graphically from the above hypothesis as shown in Figure (1).

The curve of the ship roll RAO without stabiliser will be similar to curve  $B_1$  with peak frequency  $f_1$ . The introduction of a stabiliser would first cause the peak response to drop from  $R$  to  $S$ , and second, shift the peak frequency from  $f_1$  to  $f_2$ , with the roll RAO illustrated by curve  $B_2$  and the new peak value now located at  $P$ .

The practical implications of the results shown in Figure (1) can be demonstrated with the aid of the spectral method, introduced by St. Denis and Pearson [8] in which the ship rolling is

is represented as follows:

$$S_R(\omega) = S_W(\omega) |H_R(\omega)|^2$$

where,  $S_R(\omega)$  roll response spectrum

$S_W(\omega)$  one-dimensional sea or wave spectrum for a given area of operation

$|H_R(\omega)|^2$  roll response amplitude operator

$\omega$  excitation frequency (radians)

If we can define the predominant sea spectrum, it is possible to compare the stabilised and unstabilised systems.

In the first example an unstabilised ship is operating in a given sea state in which the peak roll response frequency  $f_1$  coincides with the region of the sea spectrum where energy content is maximum, as represented by curve  $A_1$  in Figure (2). Here the roll response spectrum determined by equation (1) would be large, as illustrated by the curves  $A_1$ ,  $B_1$ , and  $C_1$ , for sea spectrum, roll RAO and roll response spectrum respectively.

When a stabiliser is installed, the original roll RAO curve will, according to our hypothesis, move to curve  $B_2$  with the resulting roll spectrum now showing a much lower value due to both effects, that is, increased damping and frequency shift.

In the second example the same ship, unstabilised, is operating in a sea state where the peak frequency of the predominant sea is at  $f_2$ , to the left of the original RAO curve  $B_1$ , as shown in Figure (3).

Curves  $A_2$ ,  $B_2$  and  $C_3$  represent respectively the sea spectrum, roll RAO and roll response spectrum. Curve  $C_4$  illustrates the effect of installing a stabiliser, and it can be seen that under the conditions postulated, the rolling response of the ship is increased.

The sea spectrum in Figures (2) and (3) is a JONSWAP spectrum which is used widely as an input to studies of vessels operating in the North Sea.

Although the conditions in the foregoing were selected to illustrate the point in a simple manner the importance of the practical implications can nevertheless, not be overemphasised.

The implications also suggest that the frequency shift of the roll response may be used as a means of enhancing the stabilisation effect, and that the degree of this "detuning effect" may be used as a measure for assessing the effectiveness of the stabilisation device for a combined ship-device-seaway system.

So far the argument is based on established theories and practical evidence. The next step is to evaluate the amount of added moment of inertia introduced to cause this shift in roll frequency by devices

such as bilge keels or passive stabiliser tanks.

## 5. EXAMPLES OF APPLICATION

The practical applications of bilge keels and passive tank stabilisers are now considered separately.

### Bilge Keels

#### Theoretical Analysis

For the theoretical analysis it was decided, in view of the difficulty of the problem, to consider a two-dimensional form and to use an idealised potential flow theory to evaluate the bilge keel contribution to roll moment of inertia.

The drawback of such an approach is that the physical problem becomes highly idealised and the whole exercise appears to be directed towards solving a mathematical rather than an engineering problem. However, this type of theoretical calculations would provide firm indications as to the nature and the order of the bilge keel's contribution. Thus the two parameter Lewis-form cross-sections were chosen for the theoretical analysis.

Figure (4) shows the various contributions of the bilge keels to roll hydrodynamic coefficients and the methods adopted for evaluating the various effects. The approach entitled the "Source-vortex method" is the novel method presently developed to calculate the hydrodynamic effect of bilge keels on rolling motion.

The Source-Vortex method employs the close-fit procedure, see Frank [9], but two singularities are used instead of one and they are distributed along the segments which define the body contours. These are wave sources distributed on the section contour and vortices distributed over the keels and their images. The generalised geometry of the boundary value problem is shown in Figure (5).

It should be noted that the term "image keel" in this instance refers to the symmetric keel about the local keel plate in the bilge region and not the inverse, since the inverse cannot be considered for a normal ship section. The boundary value problem is solved by the integral equation method using Green's functions in a similar way to Frank's source method.

The results of calculation are shown in Figure (6) for the roll added moment of inertia and damping moment coefficients over a range of non-dimensional frequencies

$\Omega (= \omega \sqrt{I_d/g})$ , where  $I_d$  is the sectional draught and  $g$  is acceleration due to gravity.

The added moment of inertia is non-dimensionalised by the factor  $[\frac{1}{4} \rho \pi^4 \Omega^2]$  and the damping moment by  $[\frac{1}{4} \rho \pi^4 \Omega]$ , where  $\rho$  is water density.

The term  $K_0$  refers to a Lewis form section with  $\lambda = 0.9$  and  $\sigma = 0.7$  without bilge keels and the terms  $K_1$ ,  $K_2$ , and  $K_3$  refer to the same section with bilge keel sizes  $B/40$ ,  $B/20$ , and  $B/10$ , respectively, where  $B$  is the sectional beam. See Figures (7) to (10).

These results suggest that the bilge keel contribution to roll added moment of inertia is substantial and approximately constant over the whole frequency range, and there is a significant change in the roll hydrodynamic damping moment as well.

The same trends have been observed in calculations carried out by Seto [10], Bishop et al [11], and Wendel [4].

#### Model Experiments

A model, with the same two-dimensional form as that used for the theoretical analysis, was also constructed to allow experiments to be performed. The model was made of wood, and three sizes of aluminium strip bilge keels were used. The dimensions of the model were:

L (Length)	=	1100 mm
B (Beam)	=	250 mm
D (Depth)	=	202.5 mm
$T_d$ (Draught)	=	112.5 mm

(Draught to half beam ratio) = 0.9  
(Cross-sectional area coefficient) = 0.7

$\Delta$ (Displacement)	=	21.63 Kg
KB	=	68 mm
KG	=	125 mm
$GM_T$	=	9 mm

Natural roll period ( $\omega_y$ ) = 5.33 rad/s  
(0.88 Hz)

The bilge keels extended the full length of the model and the three depths used were again  $B/40$ ,  $B/20$  and  $B/10$ , in order to facilitate investigation of the effect of different bilge keel depths on the roll hydrodynamic coefficients.

The model experiments were performed in the department's towing tank, the dimensions of which are 24 m (length) by 1.5 m (width) by 0.6 m (depth). At one end of the tank there is a "rolling seal wave-maker" unit consisting of five flaps and at the other end there is an absorbing beach made of wedge shaped frames filled with aluminium mesh.

The model was placed at the mid-point of the tank and was secured by elastic lines from the side walls of the tank, attached to the model at the centre of gravity level through universal joints. The effects of the lines were minimised experimentally. Regular waves were used for the experiments, their frequency being controlled through a frequency analyser which can generate signals from 0.001 - 999 Hz.

The elevation of the waves was measured by a turn wire wave probe situated near the model, and the roll motion by a motion monitor. Both signals were monitored on a digital storage oscilloscope and recorded on a pen recorder.

Four model conditions were tested: model without bilge keels, and model with each of the three sizes of bilge keel. The frequency range of regular waves was 0.5 Hz to 1.1 Hz and every experiment was repeated a number of times.

Figures (11) to (14) show the response for each condition. The reduction of the peak response and the frequency shifts are clearly indicated.

Roll decay experiments were also carried out for each of the test conditions and the best fit damping models are shown on Figure (15).

Since the emphasis on the present work is directed at determining the added moment contribution of bilge keels, it was decided to use the roll damping obtained from free oscillation tests in order to construct the theoretical responses. Figures (16) and (17) show the same trends experimentally and theoretically. These trends are confirmed by similar measurements carried out by Yokoyama et al [12] and by Szajnberg et al [5].

From the present research findings and also from information generally available it can be suggested that bilge keels may increase the roll added moment of inertia by a factor of 3 or more so that the roll natural frequency will alter by an amount great enough to affect accuracy in predicting the roll response.

The effect of bilge keels on the roll frequency response and the sea state at the area of operation may be such that when both are taken into consideration, the designer may reject the bilge keels on the grounds that they are ineffective or even harmful.

#### Installation of Passive Tank Stabilisers

With the installation of passive tank stabilisers large resonant frequency shifts become feasible and the ship can then be detuned from its operating environment at either the design or operation stages, or this can even be done by the ship's master, in the case of multiple intended-areas-of-operation or routes. To achieve this effectively, design charts have to be constructed which will indicate the maximum detuning effect in relation to the main tank and ship parameters.

To illustrate this point a typical stern trawler will be considered and suggestions will be made as to the choice of tank parameters so as to achieve maximum resonant frequency shift for fixed ship parameters along with one operational constraint and one requirement related to stability.

Goodrich's results [13] indicate that the tank parameters which would affect roll natural frequency are: tank frequency, tank size, tank position, and tank damping. In his calculation the effect of each parameter was considered while keeping the other parameters constant but this does not affect the quali-

ative nature of the results. Without doubt, however, the most important parameter is the tank natural frequency given by the relation

$$\omega_t = \frac{\pi}{b_t} \sqrt{gh} \quad (2)$$

where,  $h$  is water depth and  $b_t$  is tank breadth

The obvious adverse effect of passive tank stabilisers on a ship is on transverse stability in relation to tank size, tank position, and water depth. This effect will be one of the limiting parameters in the present calculations and a limitation that has been proposed is that the water in the tank should not decrease the meta-centric height GM by more than 40%. The percentage free surface loss is given in [13] by the relation

$$\mu * = \frac{l_t b_t^3}{12 GM} \quad (3)$$

where,  $l_t$  is tank length and  $\Delta$  is displacement of the ship

It will be assumed, for simplicity, that the tank is located at the centre of gravity of the ship.

Another arbitrary choice is related to the capacity of the tank whereby the tank should be able to roll at least  $10^\circ$  without the water touching the top or without the bottom becoming partially dry. This will be the second limiting parameter.

The ship parameters and the resulting tank parameters, based on the foregoing, are then as follows:

#### Ship parameters

$L_{BP}$	=	60.3 m
$B(mld)$	=	12.0 m
$D(mld)$	=	7.66 m
$T_{mean}(mld)$	=	5.07 m
$\Delta$	=	1878.0 tonnes
GM	=	1.32 m
$\omega_\varphi$	=	0.75 rad/sec
$\gamma_\varphi$	=	0.037

#### Tank parameters

$l_t$	=	6.718 m	from (3)
$b_t$	=	12.0 m	
$h$	=	1.0 m	
$\omega_t$	=	0.82 rad/sec	from (2)
$\gamma_t$	=	0.212	

It was assumed that the maximum possible depth of the tank was approximately 2.0 metres and the tank linearised damping ratio  $\gamma_t$  was calculated from the following empirical formula, derived in [14] and based on bench tests

$$\gamma_t = 0.1 + 0.025 \times \left(\frac{h}{b_t}\right)^{-1.57} \times \omega_t \frac{b_t}{g} \times a_a$$

where,  $a_a$  = tank amplitude at resonance (assumed 0.1 radius).

Using this data and the linear mathematical model presented by Goodrich, the solution in the form of an amplification factor is shown in Figure (24), where it will be noted that the resonant frequency shift is 38.7%.

The simple calculations used in this section to illustrate how the maximum resonant frequency shift may be calculated could become an elaborate design procedure given all the constraints and requirements, and also the freedom to change ship parameters.

#### Combinations of Stabilisation Devices

It has become very popular to use combinations of stabilisation systems in order to achieve maximum roll minimisation for the various operating conditions or routes. For example, since bilge keels are effective at any speed and also at any roll amplitudes, especially large ones, it is customary to install them on ships as a stabilisation system supplementary either to passive tank stabilisers whose efficiency tends to diminish at large roll angles, or to active fins, which are ineffective when the ship is stationary.

If any of these alternatives is contemplated it would be desirable to perform calculations to predict the response of the combined system. The effect of the combined system on the ship rolling motion will include increased damping, increased weight, increased moment of inertia and coupling effects from both tank and/or fin stabilisation systems.

The possibility also exists of installing two or more passive tank stabilisation systems on board ships.

In the light of the present research, it is clear that these combinations offer the designer a range of alternatives for achieving maximum roll reduction.

#### 6. DISCUSSION

The research findings appear to substantiate the suggestion which was put forward and also to emphasise the significance of the practical implications.

It must be made absolutely clear that the performance of ships, whether stabilised or unstabilised, cannot be judged in isolation from that environment as conclusions so drawn may be misleading; indeed for a more effective use of any stabilisation device it is essential to relate the vessel's response to the seaway(s) in which it is intended to operate. The validity of this recommendation may be open to question for vessels operating over a number of routes, some of which may not be very well defined. Nevertheless, even in this case it is necessary to understand the two-fold effect of stabilisation devices on the roll motion. Furthermore, if the detuning effect is explored during the design process it may

be possible to introduce some form of frequency shift more effectively and more quickly than is possible at present.

The popular belief that to increase the damping of the system is all that matters fails to take advantage of the full potential available for dealing with the stabilisation problem. Damping is, of course, very important but it may not always be the controlling factor.

A fuller use of the detuning principle is by no means uncommon in relation to other problems of marine-vehicle dynamics. This principle is used in the design of both semi-submersibles and tethered buoyant platforms in order to control heave motion, and these examples may encourage possible changes in the current design of ship-stabilisers, provided that the end result is achieved in a cost effective manner.

#### 7. CONCLUSIONS

On the basis of the research findings the following conclusions can be drawn:

- (a) The attachment of bilge keels to a ship will result in both increased damping and increased added moment of inertia, and the influence of the latter can, in certain cases, be greater than that of damping.
- (b) The research findings are based on the use of the "source-vortex" method for predicting the magnitude of bilge keel moment of inertia, and application of this technique will assist in the understanding of the roll contribution of bilge keels.
- (c) To make effective use of any stabilisation device or combination of devices, it is essential to relate the response of the vessel to the seaway(s) in which it is intended to operate.

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#### ACKNOWLEDGEMENTS

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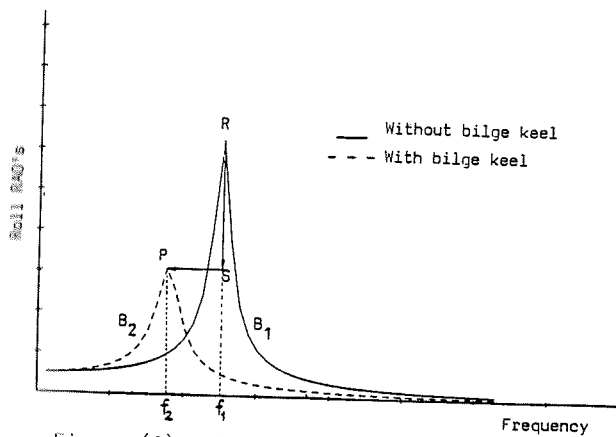


Figure (1) : Influence of bilge keel on roll response

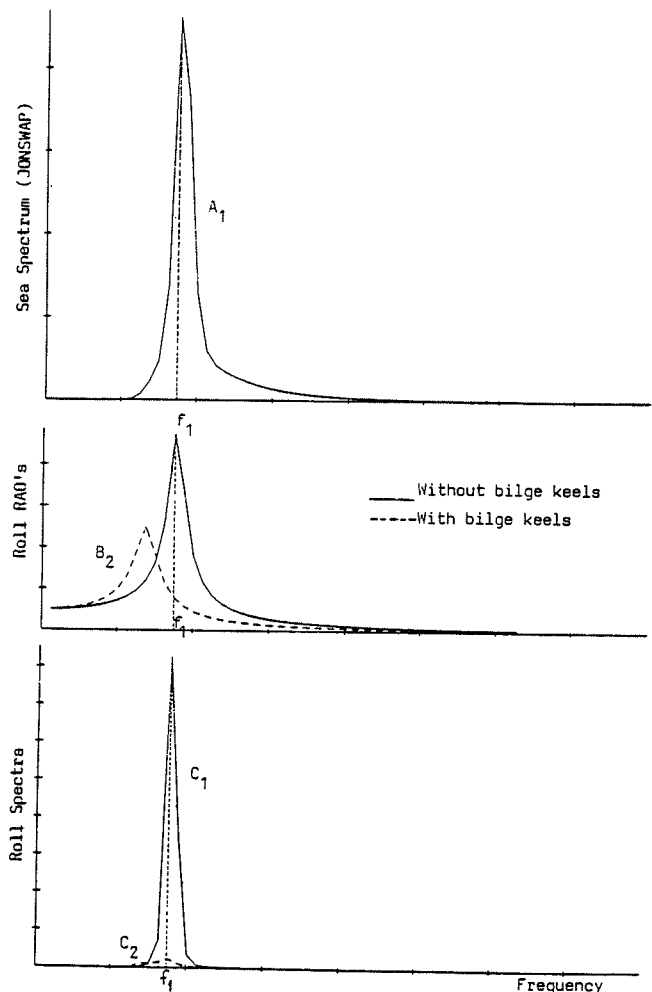


Figure (2) : Positive effect of bilge keel on roll response

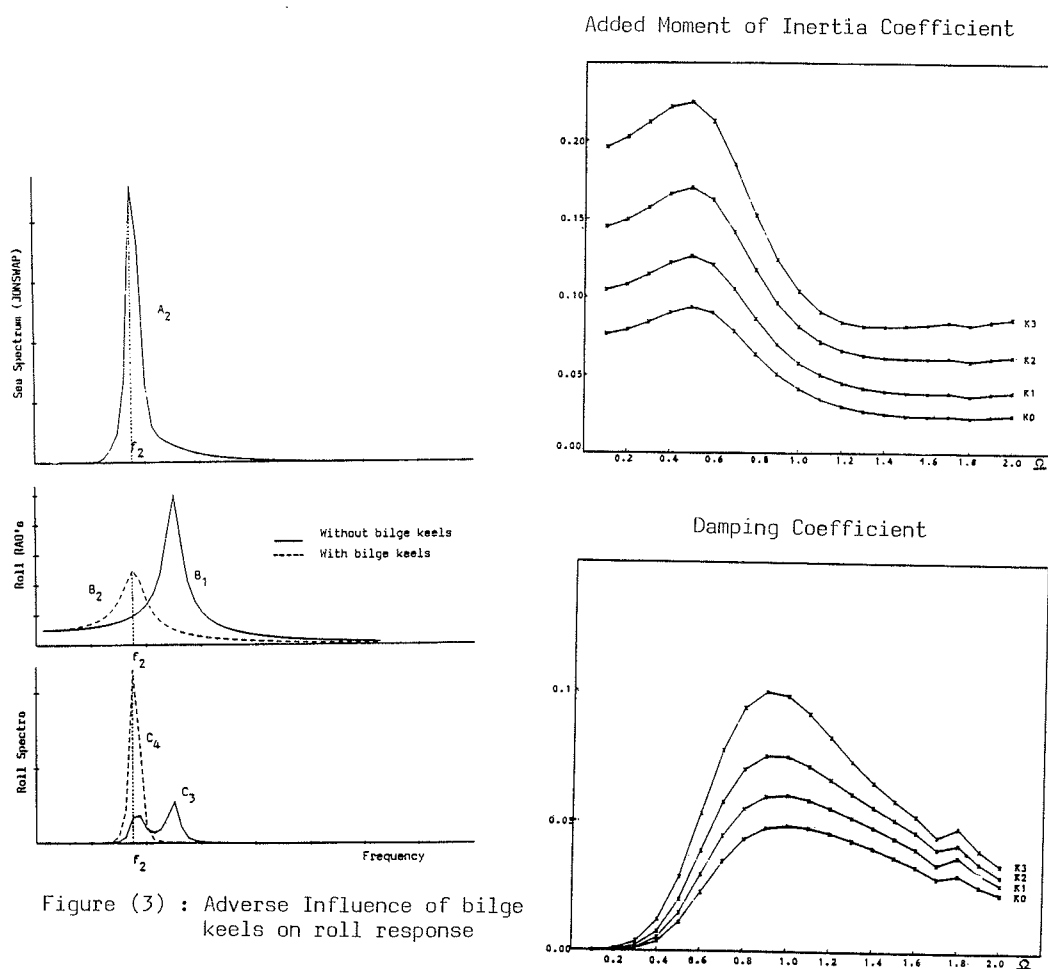


Figure (3) : Adverse Influence of bilge keels on roll response

Figure (6) : Variations of inertial and damping coefficients with bilge keel depth

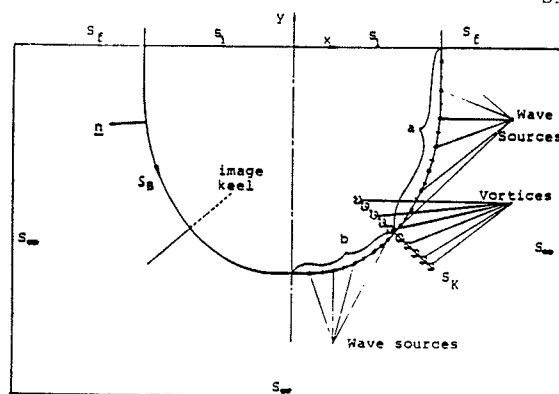


Figure (5) : Sources and Vortices distributions on the section



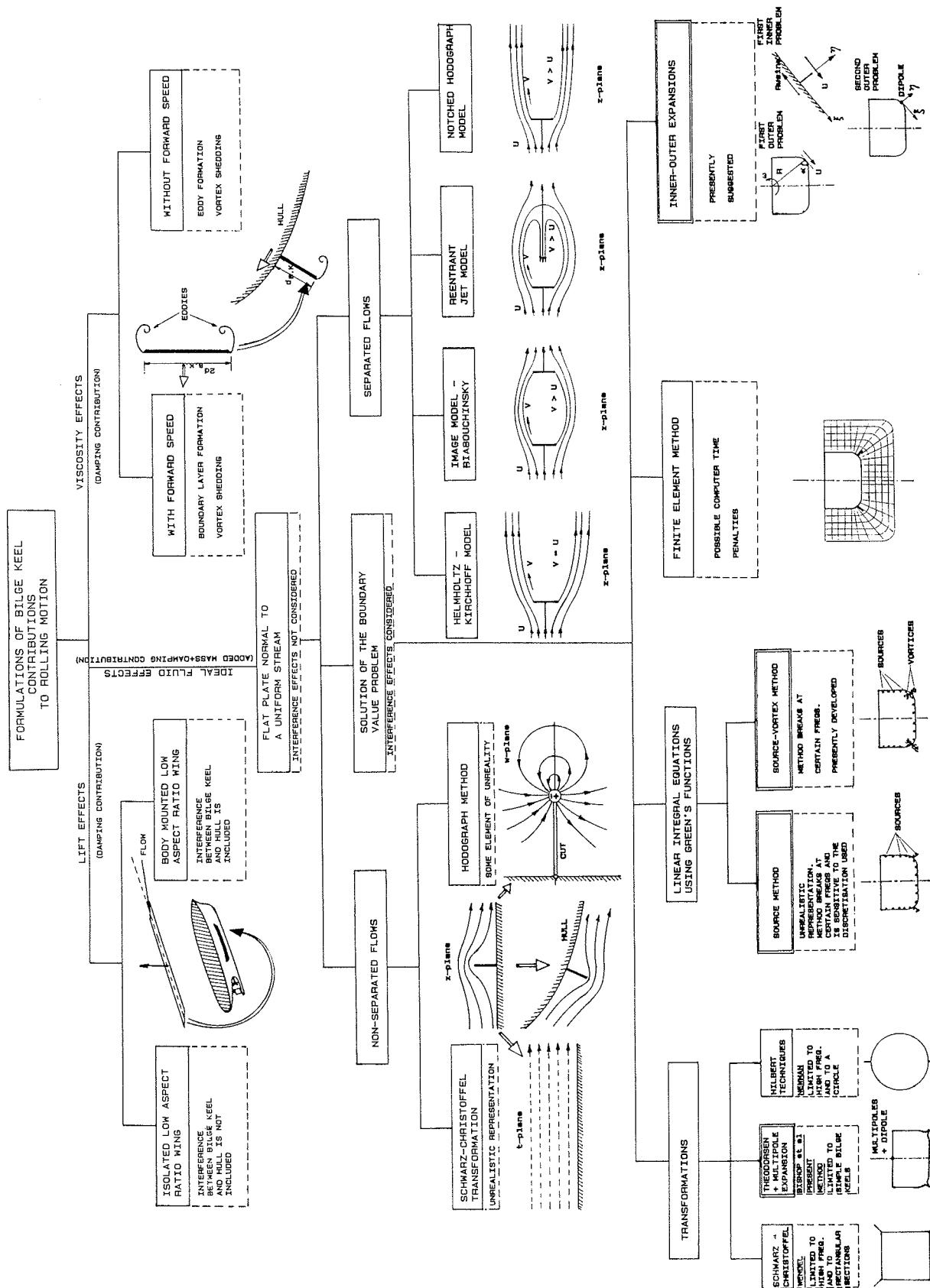


Figure (4) : Flow Diagram of the Theoretical Techniques

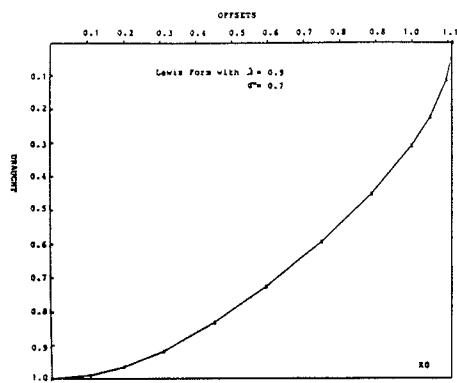


Figure (7) : Lewis form without bilge keel

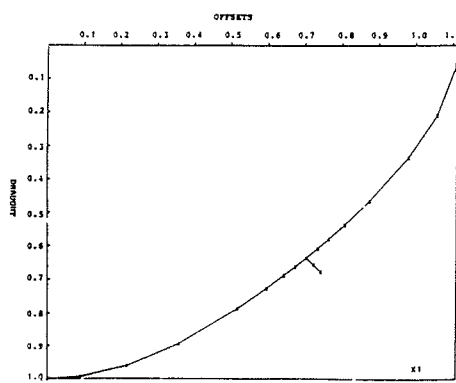


Figure (8) : Lewis form with bilge keel (depth is  $B/40$ )

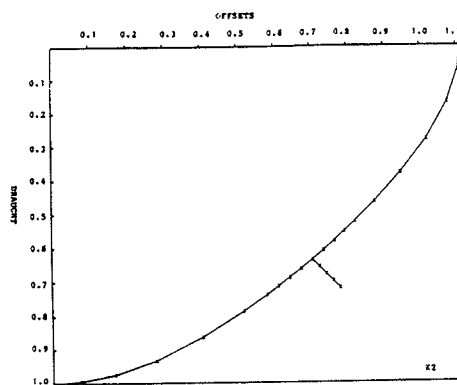


Figure (9) : Lewis form with bilge keel (depth is  $B/20$ )

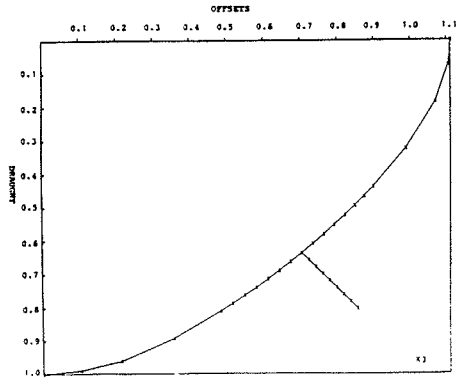


Figure (10) : Lewis form with bilge keel (depth is  $B/10$ )

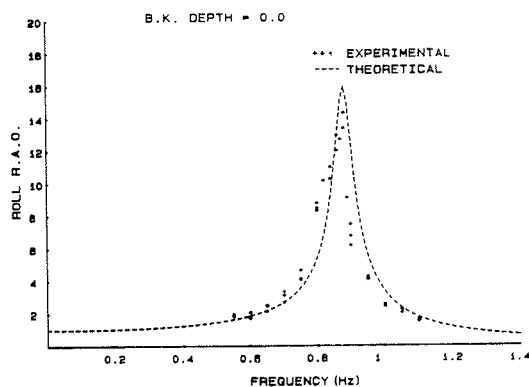


Figure (11) : Roll responses of Lewis form - without bilge keel

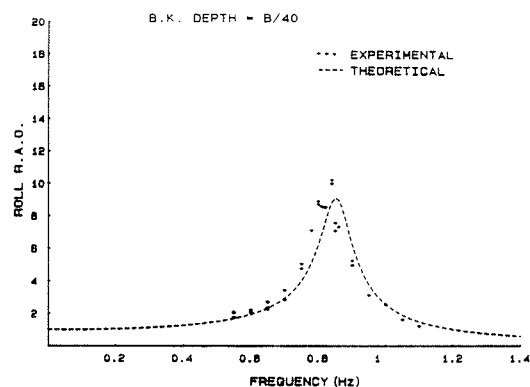


Figure (12) : Roll responses of Lewis form with bilge keel (B/40)

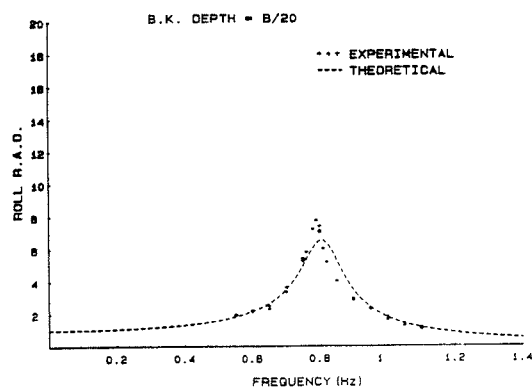


Figure (13) : Roll responses of Lewis form with bilge keel (B/20)

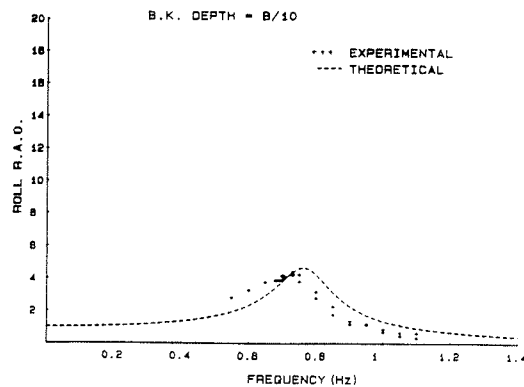


Figure (14) : Roll responses of Lewis form with bilge keel (B/10)

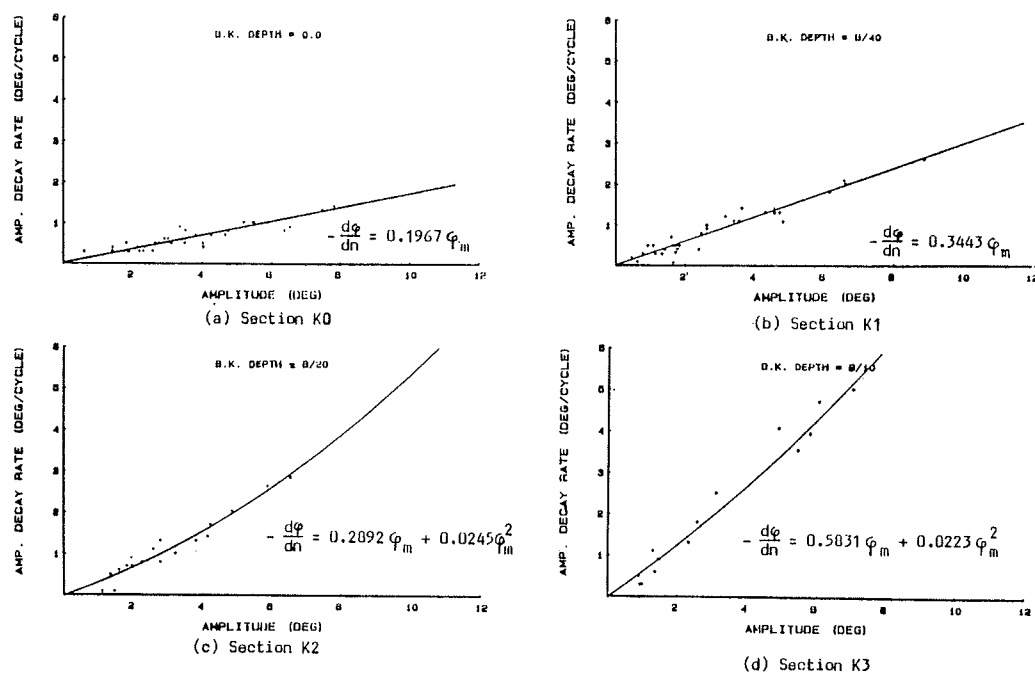


Figure (15) : Amplitude decay rate versus Mean amplitude of roll

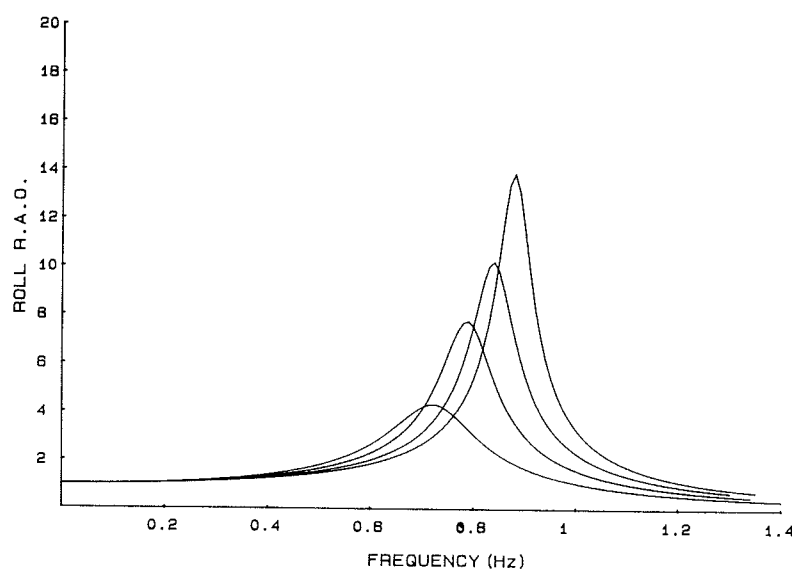


Figure (16) : Experimental results showing trends of roll responses

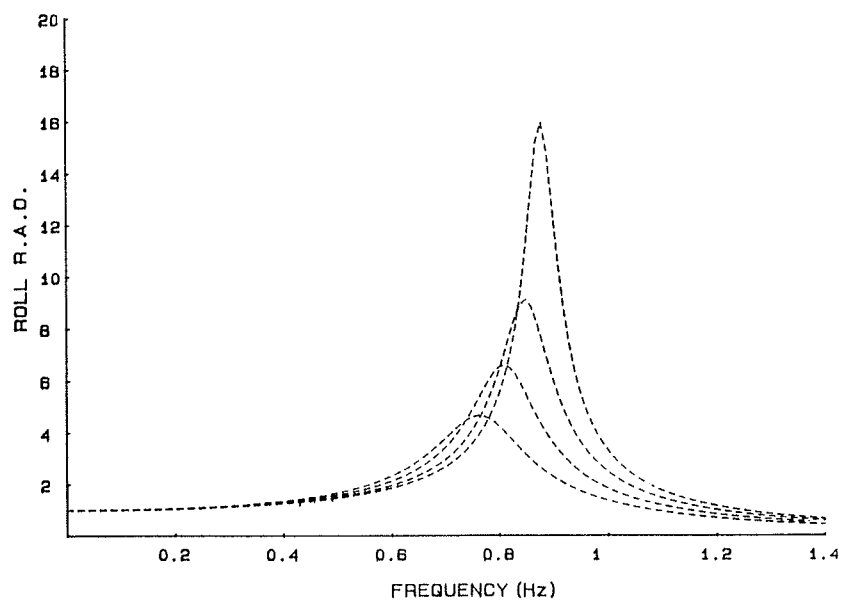


Figure (17) : Theoretical results showing trends of roll response

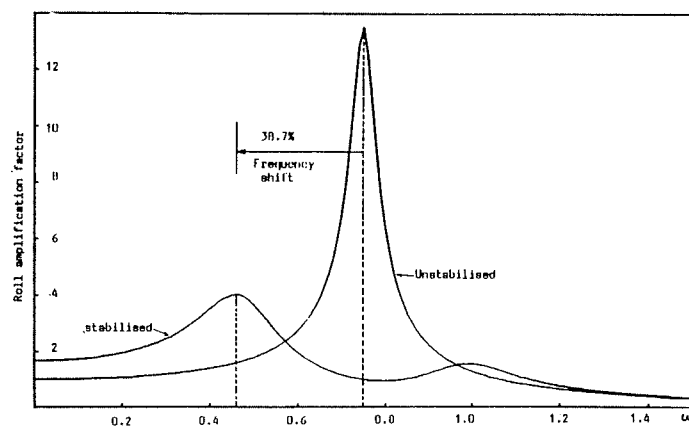


Figure (18) : Frequency shift of a stern trawler free surface tank stabiliser

## Discussion

Y. Ikeda (Univ. of Osaka Prefecture, Japan)

Thank you very much for your nice presentation.

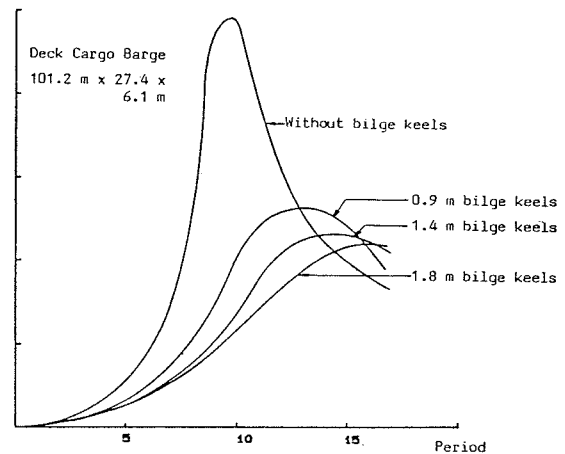
In this paper, the authors pointed out the importance of added moment of inertia due to bilge keels. According to our experience of many roll motion tests, however, the change of the natural roll frequency due to bilge keels is usually very slight in the case of conventional cargo ships with normal size bilge keels. Though theoretical calculations and experiments in this paper suggest significant effect of added moment of inertia on roll motion, I think this is because a round hull model ( $\sigma=0.7$ ) with huge size bilge keels so chosen as shown in Figs. 7 thru. 10. In general, a round hull has a small added moment of inertia. Moreover, since bilge keels are located near the free surface, the added moment of inertia and the roll damping due to bilge keels are large. Then, I think the added moment of inertia due to bilge keels takes an important role on roll motion only in very special situations.

I would greatly appreciate the comments of the authors on this point.

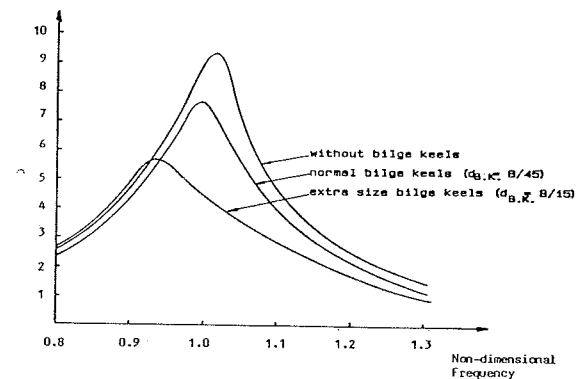
### Author's Reply

Thank you Dr. Ikeda for your contribution.

We have followed with interest your research as well as other Japanese studies in this area. Although we agree with your suggestions regarding round hulls and position of bilge keels, a round hull was chosen because it closely approximates the fishing ships we are studying and the bilge keels were positioned in accordance to these vessels. In general, the depths of bilge keels that you use are of the orders of  $B/40$  or smaller, and this would account for the small increase in added moment of inertia that you mentioned. With deeper bilge keels, however, the effect is more pronounced. For your information we include figures (19) & (20) taken from references 5 and 12, respectively.



Figure(19): Variation of roll R.A.O.'s for bilge keels of various depths



Figure(20): Variation of roll R.A.O.'s with varying bilge keel depth

M.G. Honkanen (Eng. Co. M.G. Honkanen Ltd., Finland)

I cannot agree with the authors that the shift of the natural roll period to the lower frequencies using a passive tank stabilizer is to be considered as good seamanship. In Fig. 18 you show a very typical roll response for a very lightly damped ship. In order to increase the comfort of the crew in resonant beam seas you have in practice doubled the roll in following seas, i.e. at low frequencies. In my opinion this can mean nothing but to increase the risk of capsizing. This could be avoided by the use of a controlled-passive tank, a type not mentioned in your paper.

I also consider the division of Watts and Frahm tanks as physically artificial, since both can be described with the same second order differential equation with frequency dependent coefficients and with cross-couplings to roll, sway and yaw. The

fluid damping should be considered quadratic. The coefficients may be obtained from semi-empirical formulae or by direct bench test measurements, see Reference [1].

- [1] M. Honkanen: On the Wave Induced Motions of Ships. The Swedish Academy of Engineering Sciences in Finland, Report No. 30, 1976.

#### Author's Reply

Thank you Dr. Honkanen for your contribution.

This paper aims to emphasize the role of frequency shifting effect so that in the future one could try to enhance it and make proper use of it rather than omit it or dismiss it. The suggestions made by us, therefore, were related to the concept we proposed and not to a particular design. The reasons for choosing a passive free-surface tank, as explained during our presentation, are: 1) It is a popular device; 2) The literature is replete with contradictions regarding its usage, and 3) It is a suitable device to illustrate the "detuning" effect and its importance.

Regarding the artificial division between Watts and Frahm tanks I would refer Dr. Honkanen to the discussions of Dr. Vughs and Mr. Field in reference 13 and to the battles over the McMullen patents in U.K. in order to prove the opposite.

We would agree that extra care should be taken in quartering seas although the arguments of Dr. Honkanen support our suggestions in that the detuning effect should not be overlooked. Couplings should also be considered.

T. Takahashi (Nagasaki Experimental Tank, MHI, Japan)

The authors should be congratulated for the presentation of this interesting paper. We have a lot of experiences in designing passive type anti-rolling tanks. I would like to explain our present status of ART design.

Our policy is to design a tank with natural period shorter than that of ship's rolling. Then, response amplitude operator becomes as shown by a dotted line in Fig. A. Although we have to expect small peak in the lower frequency region, the effect of this peak is not so large as expected from the non-dimensional expression like Fig. A, because wave slope is smaller in the lower frequency region. This policy is quite effective in the cases of ships with sufficiently large GM. In these cases, ship's rolling period is rather short, and sometimes severe rolling motion is experienced.

Then, we can equip the ship with ART as high on the deck as possible. By the effect of ART, peak of response amplitude operator is shifted to the lower frequency region. At the same time, GM is reduced by the weight of ART attached on the high deck, and roll natural period is shifted by this effect, too. In this way, "detuning" recommended by the authors has already been achieved quite effectively.

In the cases of ships with relatively small GM such as high speed container ships, however, several problems emerge. We have to examine intact stability seriously with free surface effect of ART, and try to avoid uncomfortable rolling of long period. Unstable rolling by the coupling effect of heaving should also be considered. Therefore, our policy cannot necessarily applied directly to this type of ships.

This is my opinion from practical experiences. Any comment by the authors would be highly appreciated.

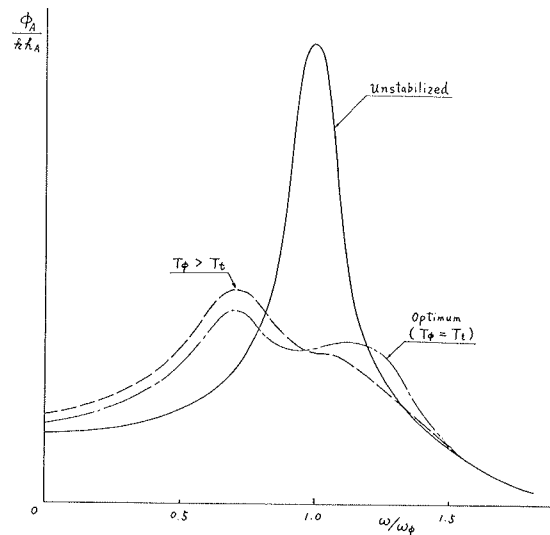


Fig. A Response Amplitude Operators of Rolling

#### Author's Reply

Thank you Mr. Takahashi for your kind words. It is very pleasing to see that the "detuning" effect is taken seriously by your company.

We believe that the reduction in GM can only be traded-off against reduced motions, and if the latter is not satisfactory then other measures should be considered.

*Session VIIb*

## Stability of Beam Trawlers and Experimental Techniques

*Chairmen*

Prof. Emil A. Dahle  
University of Fisheries  
Norway

Prof. Rihei Kawashima  
Hokkaido University  
Japan



# SAFETY OF BEAM TRAWLERS WITH A "FASTENING" NET

H. VERMEER\* AND J. A. KORTEWEG\*\*

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\*\*Delft University of Technology

The Netherlands

## ABSTRACT

The main theme of the first part of this paper is to outline the mathematical evaluation of the motions of a beam trawler and the tensile force in the warp, upon the occasion one of the beamtrawls becomes suddenly fastened in an abrupt manner on an obstruction on the seabed e.g. a wreck during the operation of trawling. The mathematical model has been developed for the vessel in still water and two alternatives have been investigated viz. with and without propeller thrust during the fastening of the beamtrawl.

In the second part of this paper a review will be given of the tests with a model of a beam trawler, which have been carried out at the Ship Hydromechanics Laboratory of the Delft University of Technology, Netherlands. The objective of these tests is to verify the aforementioned mathematical model with results of systematic model tests. Testequipment, testprogramme and procedures, which include the measurement of all relevant parameters, are described.

In the last part of this paper the calculations according to the mathematical model are compared with the model test-results which is followed by a discussion of the found discrepancies and an analysis of the trends observed. This analysis has revealed a number of preliminary conclusions relating to the stability of this type of vessel and the strength of fishing gear, which could well be of importance for both regulatory and design purposes. Finally this contribution is concluded with a few remarks with regard to the possible application of the findings of the investigation in order to improve the safety of beamtrawlers.

## NOMENCLATURE

$K_0, K_{OH}, K_{OV}$ (ton)	tensile force in the warp and its resp. components.
$K_1, K_{1H}, K_{1V}$ (ton)	additional tensile force in the warp due to fastening of a beamtrawl and its resp. components.
$S$ (ton)	thrust.
$W$ (ton)	resistance of ship and fishing gear.
$m_x$ (tonsec <sup>2</sup> m <sup>-1</sup> )	ship's mass including added mass
$I_\phi, I_\psi$ (tonsec <sup>2</sup> m)	ship's mass moment of inertia including added moment of inertia in the resp. direction.
$B_\phi$ (tonm)	restoring moment coefficient of the rolling motion.
$x, y, z$ (m)	longitudinal, resp. lateral and vertical transfer.
$\phi, \psi$ (rad)	rolling angle and yawing angle.
$\epsilon$ (-)	tg $\alpha$ , in which $\alpha$ is defined in the text.
$r$ (m)	distance from ship's centre line to point of application of warp pull.
$a, b$ (m)	vertical and horizontal distance from point of application of warp pull to centre of gravity.
$v$ (msec <sup>-1</sup> )	ship's advance speed.

$t$ (sec)	time.
$\alpha, \beta, \gamma$ (sec <sup>-1</sup> )	circle frequencies as defined in the text.
$\mu, \nu, \omega$ (tonm <sup>-1</sup> )	spring constant of the warp.
$\Delta l_1$ (m)	elongation of the warp due to $K_1$ .
$\overline{GM}$ (m)	initial metacentric height.
$\Delta$ (ton)	ship's displacement
$N$ (H.P.)	ship's power.
$c_0$	coefficient.

## 1. INTRODUCTION

Beamtrawling is a common fishing method in the Netherlands and to a less extent in the neighbouring countries. This fishing method is characterized by the occurrence of excessive dynamic external forces acting upon the fishing gear. The external moment for a highpowered beam trawler may under special circumstances endanger the ship's stability and lead to capsizing. Due to this phenomenon several fishing vessels have foundered especially during the sixties when the fishermen had no experience with the particular features of beamtrawling.

An investigation into these accidents revealed that two different dangerous situations can be distinguished:

- (1) A beam trawler attempting to release the beamtrawl after a fastener by hauling with the warp running over the fishing block, at the end of the boom without using a sliphook as required.

This case has been investigated in a quasi-static manner for a trawler in general and accordingly a proposal regarding stability criteria has been put forward to IMCO (Ref.1).

- (2) A beam trawler suddenly fastening in an abrupt manner on an obstruction on the seabed e.g. a wreck during the operation of trawling followed by a violent rolling and yawing motion due to the impact load exerted by the fastened warp.

This case will be investigated more thoroughly, taking into account the dynamics involved. This paper presents a description of this investigation by means of a mathematical model and extensive testing of a ship model in idealized tank-conditions.

## 2. MATHEMATICAL MODEL

### 2.1 Basic Assumptions And Development Of Mathematical Model

Prior to the set up of the fundamental equations of the mathematical model it is useful to outline the basic assumptions.

- (1) In order to avoid complication of the problem it is assumed that the process of fastening of the beam trawl occurs in still water, i.e. the effects of waves, wind and current are not considered. In this respect it should

be mentioned that generally a fishing vessel stops fishing approximately at Beaufort 8. So the possible adverse effect of wind and waves should be borne in mind when interpreting the results of this investigation.

- (2) In principle a fastener may occur when the beamtrawl is digging into a sand bank, which gives rise to a gradual increase of the pull in the warp (Ref.2). However, in this investigation abrupt fastening of the beamtrawl e.g. hooking a wreck is assumed being the most critical case. This has the additional advantage of being less complicated in the mathematical sense.
- (3) In the equations of motion damping and hydrodynamic coupling terms are ignored, which is justified, bearing in mind the time period of the forced oscillation.
- (4) Within this study no account is taken of the effect of a sudden "upcast" of the unfastened boom, so it is assumed that occurrence of this phenomenon is prevented by suitable means.

Prior to the occurrence of a fastener the following equation applies:

$$S-W-2K_{OH}=0 \quad (1)$$

When a fastener occurs the differential equations of motion for the  $x$ ,  $y$ ,  $z$ , and  $\phi$ -direction can be formulated using Fig.1, and are given in Ref.3.

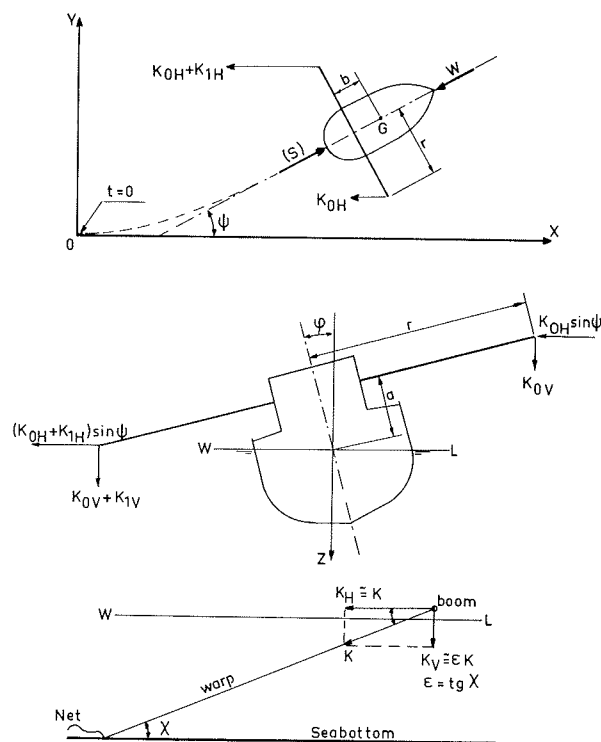


Fig.1. Coordinate system and forces acting upon a beam trawler during a fastener.

Using Fig.2 the pulling force in the warp due to the fastener can be expressed as:

$$K_1 = k \cdot \Delta l_1 = k \{x - r\psi - \varepsilon(z + r\varphi)\} \cos \chi \quad (2)$$

where it has been assumed that Hooke's law is applicable. In Ref.3 it is shown that the stress-elongation curve for a (used) warping line is a straight line for  $K_1$ -values occurring in practice.

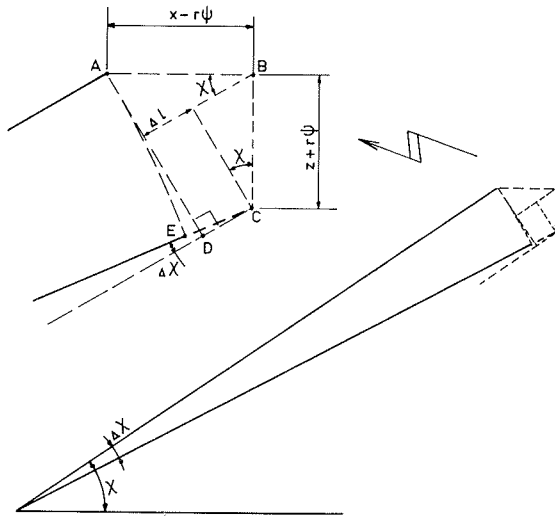


Fig.2. Elongation of the warp.

## 2.2 Simplification And Evaluation Of Mathematical Model

In order to solve the mathematical model, consisting of (1), (2) and the differential equations of motion, the following operations are carried out:

- (1) The differential equations of motion have been linearized.
- (2) Second order terms have been neglected.
- (3) The coupling term with  $z$  in the equation for  $K_1$  (2), has been ignored, because the resulting heave motion as such is negligible.

Using the boundary conditions:

$$t=0, \dot{x}=v, x=\psi=\dot{\psi}=\varphi=\dot{\varphi}=0 \quad (3)$$

the mathematical model can be solved for the various degrees of freedom. The procedure of the solution is presented in Ref.3 and the most important results are as follows:

$$\varphi = \frac{v}{\varepsilon r} \frac{(\gamma^2 - \omega_1^2)(\gamma^2 - \omega_2^2)}{\gamma^2(\omega_1^2 - \omega_2^2)} \left( \frac{\sin \omega_1 t}{\omega_1} - \frac{\sin \omega_2 t}{\omega_2} \right) \quad (4)$$

$$x = vt - r \frac{\alpha^2}{\beta^2} \psi \quad (5)$$

$$\psi = -\frac{v}{r} \frac{\beta^2}{\gamma^2} \left( \frac{\gamma^2 - \omega_2^2}{\omega_1^2 - \omega_2^2} \frac{\sin \omega_1 t}{\omega_1} - \frac{\gamma^2 - \omega_1^2}{\omega_1^2 - \omega_2^2} \frac{\sin \omega_2 t}{\omega_2} - t \right) \quad (6)$$

$$K_1 = k(x - r\psi - \varepsilon r\varphi) =$$

$$= \frac{kv}{\gamma^2} \left( \frac{\gamma^2 - \omega_2^2}{\omega_1^2 - \omega_2^2} \omega_1 \sin \omega_1 t - \frac{\gamma^2 - \omega_1^2}{\omega_1^2 - \omega_2^2} \omega_2 \sin \omega_2 t \right) \quad (7)$$

$$\text{with } \omega_{1,2} = \left[ \frac{1}{2}(\gamma^2 + \mu^2) \pm \frac{1}{2} \{ (\gamma^2 + \mu^2)^2 - 4\gamma^2 v^2 \}^{1/2} \right]^{1/2} \quad (8)$$

$$\text{and } \alpha^2 = \frac{k}{m_x}, \quad \beta^2 = \frac{kr^2}{J_\psi}, \quad \gamma^2 = \alpha^2 + \beta^2$$

$$\mu^2 = \frac{B_\varphi + kr^2 \varepsilon^2 - 2a\varepsilon K_0}{J_\varphi}, \quad v^2 = \frac{B_\varphi}{J_\varphi}$$

## 2.3 Result of Numerical Computations

The results stated in the previous sub-paragraph are valid for the particular case that the propulsion machinery is not stopped when the vessel gets a fastener. It is also possible to solve the mathematical model for the case:

$$t = 0, S = 0 \quad (9)$$

The solution for this particular case is given in the Annex of Ref.3, where substitution of  $S = 0$  yields the equations 4 - 7 inclusive.

In order to obtain an idea about the difference between these two methods a calculation example has been elaborated.

Table 1: Basic data for numerical calculation

Parameter	Value
$m_x$ (tonsec <sup>2</sup> /m)	31.2
$I_\varphi$ (tonsec <sup>2</sup> m)	333
$I_\psi$ (tonsec <sup>2</sup> m)	3470
$K_0$ (ton)	3.5
$k$ (ton/m)	13.33
$B_\varphi(\overline{GM})$ (tonm, m)	143.5 (0.47)
$v$ (m/sec)	2
$r$ (m)	9
$a$ (m)	3.5
$\varepsilon$ -	0.2
$S$ (ton)	8.5

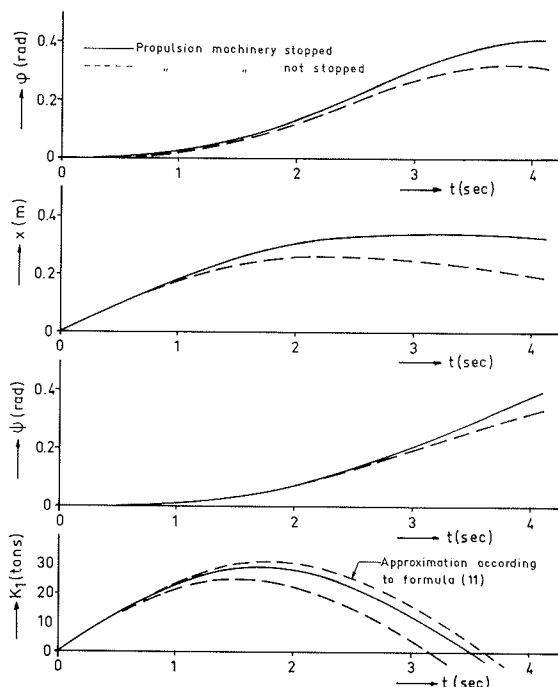


Fig.3. Results of numerical calculations.

The basic data are listed in Table 1 and the results are plotted in Fig.3. From Fig.3 it can be concluded that in case the propulsion machinery is stopped there is a reduction of 20% in terms of maximum heeling angle. Taking into account the significant time delay usual in practice to reduce the thrust it is proposed to consider only the most critical case i.e. "without human intervention", which has the additional advantage of being less complicated in application.

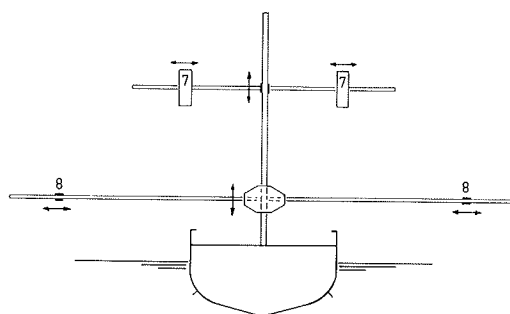
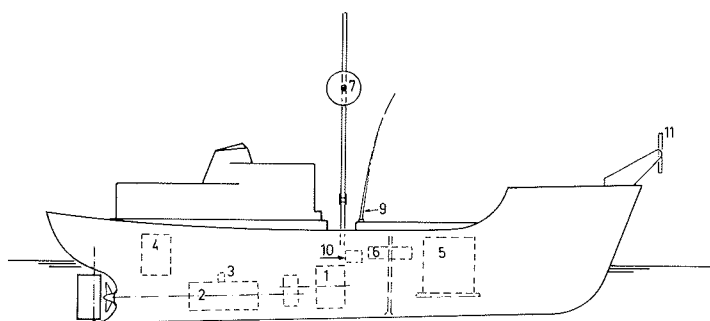
### 3. MODEL EXPERIMENTS

#### 3.1 Test Facilities, Model And Instrumentation

The model experiments have been carried out in the experimental tank of the Ship Hydromechanics Laboratory of the Delft University of Technology in the Netherlands. The dimensions of this tank are: length = 150 m, width = 4.2 m, depth of water = 2.55 m. The carriage can travel with speeds up to 6m/sec.

The model was a 1 : 15 scale model of the Dutch beam trawler IJM 44. Particulars of this vessel and the model are given in Table 2. The model was constructed of GRP and was fitted out with the following instruments and equipment (see Fig.4):

- propulsion motor, propeller shaft and propeller;
  - rudder with locking device;
  - yawing gyro (measurement of yawing angle);
  - rolling gyro (measurement of rolling angle);
  - a mast with a vertically adjustable beam.
- At the beam two horizontally adjustable attaching points for the fishing warps with dynamometers for measurement of the tensile forces in the warps;
- a second beam also vertically adjustable along the mast with two horizontally adjustable weights (adjustment of metacentric height and moment of inertia);
  - a vertically adjustable weight on a screw spindle (adjustment of metacentric height);



- |                                  |   |
|----------------------------------|---|
| 1. Propulsion motor.             | 7. Vertically and horizontally adjustable weights.                            |
| 2. Shaft dynamometer             | 8. Adjustable attaching points for fishing warps.                             |
| 3. Tachometer                    | 9. Cable connection to towing carriage.                                       |
| 4. Yawing gyro.                  | 10. Acceleration meters.  |
| 5. Rolling gyro.                 | 11. Rotating steel plate for electromagnetic connection with towing carriage. |
| 6. Vertically adjustable weight. |   |

Fig.4. Equipment and instrumentation of model.

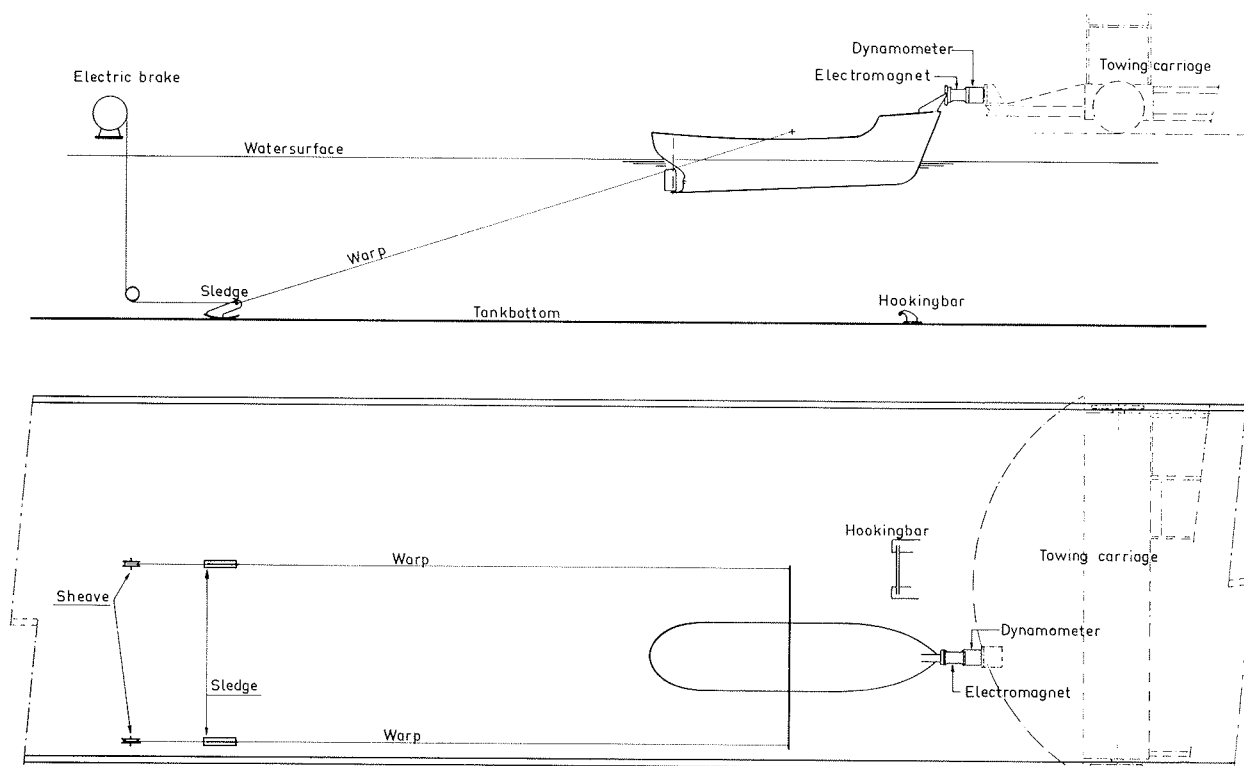


Fig.5. Arrangement of model in towing tank.

Table 2: Particulars of ship and model

		SHIP	MODEL 1 : 15
Length b.p.	(m)	23.25	1.557
Breadth mlded	(m)	6.40	0.427
Depth mlded	(m)	3.10	0.207
Draught fwd	(m)	1.95	0.130
Draught aft	(m)	2.63	0.175
Volume of displacement	(m <sup>3</sup> )	164.16	0.04864
Centre of buoyancy forward of L/2	(m)	0.615	0.041
Metacentre above base	(m)	3.335	0.222
1/2 angle of entrance of CWL		31.5°	31.5°
Block coefficient		0.48	0.48
Prismatic coefficient		0.645	0.645

- acceleration meters (measurement of linear acceleration in x, y and z direction of a coordinate system fixed to the ship);
- a small rotating steel plate on the stem-head for the electromagnetic connection with the towing carriage;
- bilge keels and bulwark with freeing ports.

Powersupply and control of propulsion motor were carried out from the towing carriage, which stayed close to the model during the test. The measurements were transmitted by flexible cables to the towing carriage and recorded on UV-recorders and instrumentation tape recorder.

### 3.2 Testarrangement And Program

The arrangement of the model in the towing tank is shown in Fig.5. The model was connected by warps with two sledges which were pulled over the tankbottom. The tensile force in the warps was supplied and adjusted by an electrical brake at each warp. As soon the model had travelled some distance to adjust speed, propeller revolutions and tensile force in the warps the port sledge was caught by a bar on the

bottom of the towing tank. During this fastener the angles of roll and yaw, the tensile force in the warps, and the linear accelerations in x, y and z directions were recorded.

At the start of every run the model was connected to the towing carriage by an electromagnet. The speed of the towing carriage was adjusted to the required speed of the vessel during fishing and the number of revolutions of the propeller motor was increased until the dynamometer between model and carriage showed zero force. Then the model was disconnected from the carriage and sailed free on her own. Some seconds later the port sledge hit the bar on the tank bottom and a fastener was initiated.

To investigate the conformity of the results of the model tests with the mathematical model it was expected that beside rolling angle and tensile force in the warp also x and y according to Fig.1 could be determined from the tests by mathematical treatment of the linear accelerations. Further a number of runs have been carried out simulating different situations. The main parameters and the corresponding quantitative variations are shown in Table 3.

Most tests have been carried out with the propulsion motor not stopped. Some however were carried out with a stopped motor at the moment of fastening. From the results it appeared that the differences between both tests are small in the order of less than 5%, which is within accuracy limits.

Table 3: Main parameters and their variations

Parameter:	Variation (for ship):
$\overline{GM}$ (m)	0.40 - 0.55 - 0.70(*) - 0.85 - 1.00
v (m)	1.5 - 2.0 - 2.5(*) - 3.0 - 3.5
r (m)	7.00 - 8.00 - 9.00(*) - 10.00 - 11.00
k (t/m)	4 - 8 - 12(*) - 16 - 20
$\epsilon$ (-)	0.25 - 0.275 - 0.30(*) - 0.325 - 0.35

Values signed \*) are average values

### 3.3 Preliminary Considerations And Measurements

Prior to the model tests in the towing tank the following subjects were considered.

(1) The tensile force in the fishing warps of a beam trawler of current practice and dimensions under normal conditions at different ship speeds during fishing was supplied by the Technical

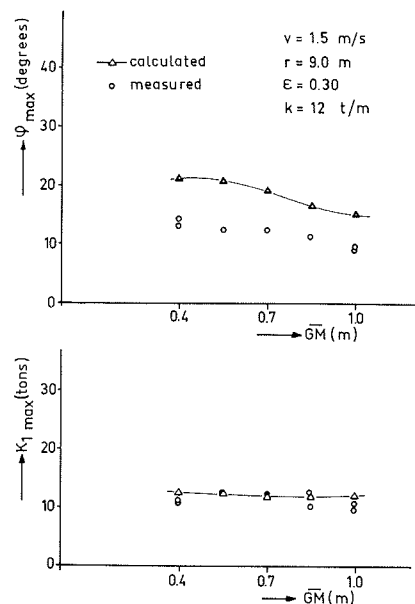


Fig.6.  $\phi_{max}$  and  $K_1_{max}$  as function of  $\overline{GM}$ .

Table 4: Tensile force fishing warps at different ship speeds

Ship speed (m/sec.)	Tensile force in warp $K_0(t)$
1.5	2.30
2.0	3.00
2.5	3.98
3.0	5.37
3.5	7.12

Research Department of the Netherlands Institute for Fishery Investigations (see Table 4).

(2) From calculations with the mathematical model it was shown that differences in mass and mass moment of inertia of  $\pm 15\%$  did hardly affect the resulting forces in the fishing warps at the fastening of a net, nor the rolling angle or the x- and y- coordinates. Therefore it was considered sufficient to measure  $f_1$  and  $f_3$  in the following expressions and to use for  $f_2$ ,  $f_4$  and  $f_5$  standard values of respectively 2.0, 1.95 and 0.25.

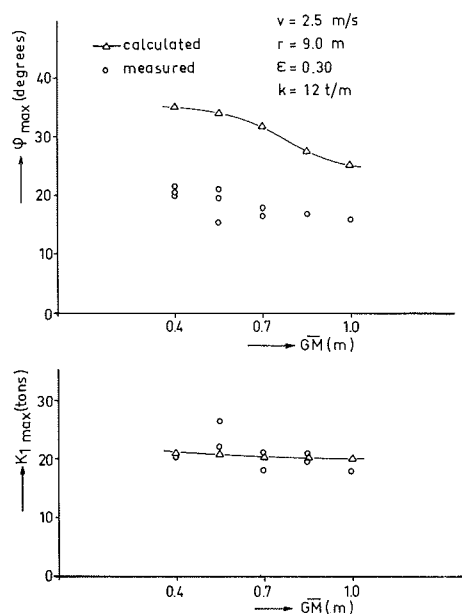


Fig.7.  $\phi_{\max}$  and  $K_1 \max$  as function of  $\overline{GM}$

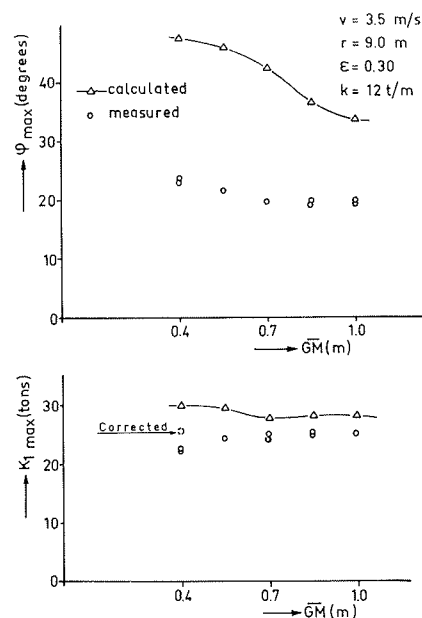


Fig.8.  $\phi_{\max}$  and  $K_1 \max$  as function of  $\overline{GM}$ .

$$m_x = f_1 m \quad (10)$$

$$m_y = f_2 m \quad (11)$$

$$I_\phi = f_1 m (f_3 B)^2 \quad (12)$$

$$I_\psi = f_4 m (f_5 L)^2 \quad (13)$$

The results of the measurements of  $f_1$  and  $f_3$  depending on the  $\overline{GM}$  are given in Table 3.

The coefficients were measured by rolling tests of the model in air and water.

(3) For the fishing warps was chosen "Nichrome Alloy V" wire of 0.4 mm diameter. To vary the elasticity of the fishing warps springs with different characteristics were connected between the warps and the dynamometers at the adjustable beam.

Table 5: Results of Measurement of  $f_1$  and  $f_3$

$\overline{GM}$ (m)	$f_1$	$f_3$
0.40	1.045	0.485
0.55	1.152	0.417
0.70	1.072	0.385
0.85	1.001	0.417
1.00	0.984	0.401

In this way the elasticity of the warps could be varied between spring constants of 4 and 20 ton/m (full scale).

(4) After fitting out the model with the required equipment and instruments as mentioned earlier it was shown that the required variation of  $\overline{GM}$  could be obtained by moving the adjustable weights.

### 3.4 Results Of Model Experiments

The model tests showed in general well reproducible results with a maximum deviation of 2.5° in  $\phi_{\max}$  and a maximum deviation of 2.5 ton in  $K_1 \max$ . All results are calculated from the model tests for ship size.

The calculation of x- and y- coordinates from the acceleration measurements turned out to be very laborious and did not give acceptable results. Therefore it was decided to focus on a comparison of  $\phi_{\max}$  and  $K_1 \max$  for mathematical model and model tests.

During the model tests according to Table 3 capsizing of the model did not occur. In order to find out at what combination of parameters the vessel would capsize the parameters were altered in an unfavourable fashion until capsizing did occur. The combination of parameters, which lead to capsizing are shown in Table 6. From this table it can be seen that only a very unfavourable combination of parameters values which lies beyond the conditions found in practice will lead to capsizing.

On the other hand it should be kept in mind that all tests were carried out without wind, waves or currents. In this way the danger of capsize under unfavourable operational conditions could differ quite considerably from that during the model tests.

Table 6: Combination of parameters at capsizal

Parameter (for ship)		Value
$\overline{GM}$	(m)	0.30
$v$	(m/sec)	3.5
$r$	(m)	10
$k$	(t/m)	20
$\epsilon$	-	0.35

During the model tests it was observed that in some cases permanent elongation of the warp took place (see Fig.15). In these cases  $K_{1max}$  lagged behind the expected values. Therefore  $K_{1max}$  has been corrected in these cases in the shown manner. These corrected  $K_{1max}$ -values and also the original values are shown in Fig.8 and 9.

(1) In Fig.6-8 the results are shown of the maximum rolling angle and the maximum tensile force in the warp at different values of  $\overline{GM}$ . The values of  $r$ ,  $\epsilon$  and  $k$  were kept constant, only the speed of the vessel has been varied: 1.5, 2.5, 3.5 m/sec respectively for the ship.

(2) In Fig.9-11 the results are shown of the maximum rolling angle and the maximum tensile force in the warp at different fishing speeds. The values of  $r$ ,  $\epsilon$  and  $k$  were kept constant, only  $\overline{GM}$  has been varied as 0.40, 0.70 and 1.00 m respectively.

(3) In Fig.12 the results are shown of the maximum rolling angle and the maximum tensile force in the warp at different values of the spring constant  $k$ . The values of  $v$ ,  $r$ ,  $\epsilon$  and  $\overline{GM}$  were kept constant.

(4) In Fig.13 the results are shown of the maximum rolling angle and the maximum tensile force in the warp at different values of  $r$ . The values of  $v$ ,  $\epsilon$ ,  $k$  and  $\overline{GM}$  were kept constant.

(5) In Fig.14 the results are shown of the maximum rolling angle and the maximum tensile force in the warp at different values of  $\epsilon$ . The values of  $v$ ,  $r$ ,  $k$  and  $\overline{GM}$  were kept constant.

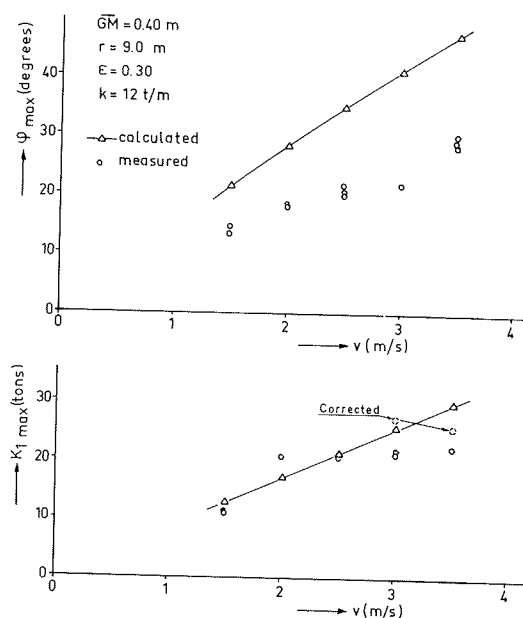


Fig.9.  $\phi_{max}$  and  $K_{1max}$  as function of  $v$ .

(6) In Fig.15 the results are shown of the rolling angle and the tensile force in the warp as a function of time. In this case the warp showed a permanent elongation and a flattened top of the  $K_1$ -curve. This curve has been corrected in the indicated way to obtain comparable results of  $K_{1max}$ .

#### 4. DISCUSSION AND ANALYSIS OF RESULTS

##### 4.1 Comparison Of Results Of Model Experiments With Systematic Calculations

(1) Reviewing the Figures 6-11 it can be stated that the conformity between the calculated tensile force in the warp from the mathematical model and those measured during the model tests is good. This refers to the conformity between the general appearance and character of the curves for calculated and measured forces as well as to the conformity of their values. One exception however is found in Fig.9 where the calculated and measured forces do not well agree. Some points in this figure had to be corrected because of permanent elongation of the warp so that in this way the measured values could be less reliable than usual.

(2) The conformity between the maximum rolling angles calculated from the mathematical model and those measured during the model tests is considerably less satisfactory. In all cases the measured angles are much smaller than the calculated angles. Though this is a reassuring result, efforts have been devoted to find out the origin of this discrepancy.



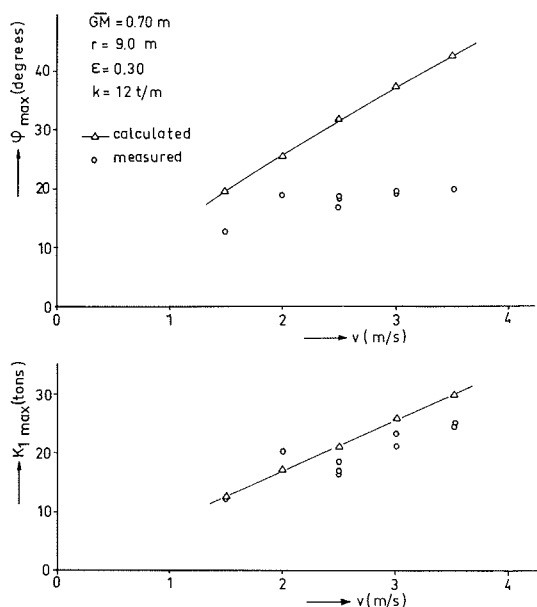


Fig.10.  $\phi_{\max}$  and  $K_{1\max}$  as function of  $v$ .

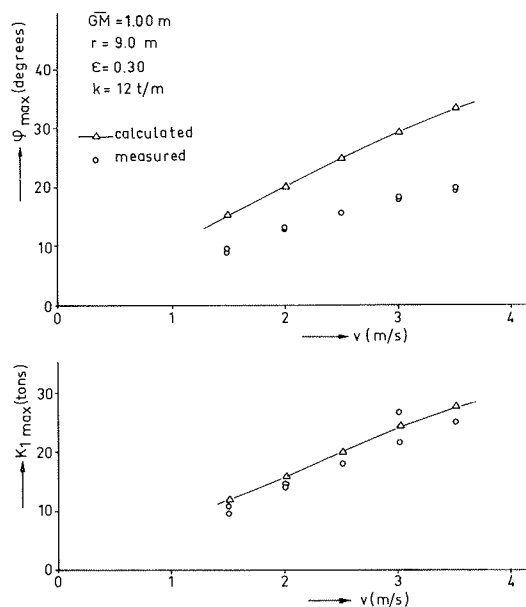


Fig.11.  $\phi_{\max}$  and  $K_{1\max}$  as function of  $v$ .

One of the causes of these differences could be expected to be found in the assumptions and approximations of the mathematical model. However as some of these will produce smaller rolling angles and others - for instance the linear restoring moment  $B_0 \phi$  - underestimate the maximum rolling angle the large discrepancies can hardly be explained on this basis.

Another cause of the forementioned discrepancies could have been the neglect in the mathematical model of the increase of the tensile force in the non-fastening warp. According to Ref.2 this force is also increasing to a considerable extent during a fastener. However the recordings of this force during the model tests did not show any substantial increase of the tensile force in the non-fastening warp. So the mathematical model was in this respect exactly in accordance with the model tests and gives no explanation for the found discrepancies.

A third possibility was the influence of the bilge and bar keels of the model at different speeds. These influences were not included in the mathematical model. Therefore some additional tests with the model of the beam trawler have been carried out without warps, but with a small weight at the end of one boom. The inclined model was towed by the towing carriage until the required speed was obtained by adjustment of the propeller revolutions. Then the small weight was dropped overboard and the successive rolling oscillations recorded. The differences between the rolling angles at different and zero speeds were in the order of 5 - 15% and are by no means

sufficient to give an explanation of the found discrepancies.

(3) Reviewing the Figures 12-14 it can be stated that the influence of the spring constant on the tensile force in the warp agrees well with the corresponding results of the calculations according to the mathematical model. The differences in  $\phi_{\max}$  are of the same order of magnitude as mentioned before.

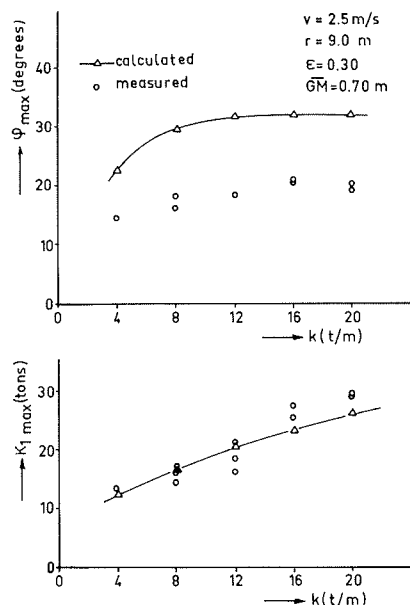


Fig.12.  $\phi_{\max}$  and  $K_{1\max}$  as function of  $k$ .

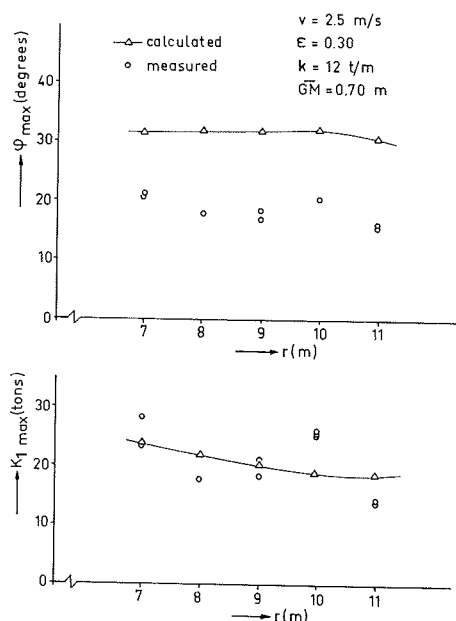


Fig.13.  $\phi_{max}$  and  $K_{1max}$  as function of  $r$ .

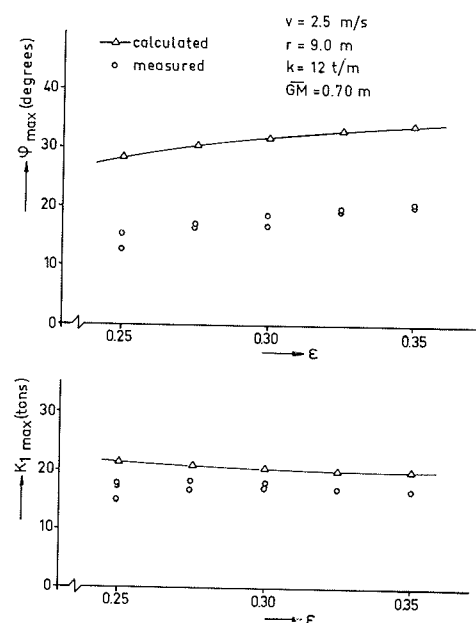


Fig.14.  $\phi_{max}$  and  $K_{1max}$  as function of  $\epsilon$ .

The influence of the length of the boom  $r$  is according to the model tests much more marked than according to the mathematical model. A rather pronounced minimum and maximum can be noted at resp.  $r = 8$  m and  $r = 10$  m (see Fig.13). This phenomenon could also clearly be observed during the model tests.

The influence of  $\epsilon$  - which refers to the steepness of the warps during fishing on the tensile force in the warp is very small. The influence of  $\epsilon$  on  $\phi_{max}$  is only slightly greater. Also in this case  $\phi_{max}$  from the model tests is about 50 - 60% of  $\phi_{max}$  according to the mathematical model.

(4) Reviewing Fig.15 where  $K_1$  and  $\phi$  have been presented as a function of time, it can be noted that  $K_1$  measured during the model tests takes some more time to reach its maximum than  $K_1$  calculated from the mathematical model. From analysis of equation (15) it is found that this difference must have been caused by differences in the elasticity of the warp and further structural parts between mathematical model and ship model. This could be caused by bending of the boom and mast including some elasticity in the connection of the mast foot to the deck of the model of the beam trawler. During the tests some bending of the booms has been observed and this situation has been improved by rigging wires from boom ends to stem head.

The great differences between  $\phi_{max}$  according to mathematical model and model tests are stated here once more. Also a considerable time lag for  $\phi_{max}$  can be observed.

#### 4.2 Observations On Gear Strength And Stability

Equation (7) does not offer the possibility to derive an analytical expression for the maximum value of  $K_1$ . However, application of the law of preservation of energy (Ref.3) leads to:

$$\frac{\sin \omega_1 t}{\omega_1} = \frac{\sin \omega_2 t}{\omega_2} = \frac{\sin \gamma t}{\gamma} \quad (14)$$

Substitution of (10) into (7) gives:

$$K_1 = \frac{kv}{\gamma} \sin \gamma t \quad (15)$$

Consequently the approximate value of the maximum value of  $K_1$ , reads:

$$K_{1max} = \frac{kv}{\gamma} \quad (16)$$

From subsequent calculations it can be shown that equation (16) produces a value for the maximum value of  $K_1$ , which exceeds the corresponding value of  $K_1$  according to equation (7) by about 10% (see e.g. Fig.3). Taking into consideration that  $K_0$  has a value in the order of 5 - 15% of  $K_1$  it is believed that this expression (16) provides an accurate basis for gear strength calculations. This result may also be taken into account when considering the effect of fishing gear load on submarine pipelines. An outline of this problem is presented in Ref.5.

Using equation (4) it can easily be demonstrated that the maximum heeling angle is represented by the expression:

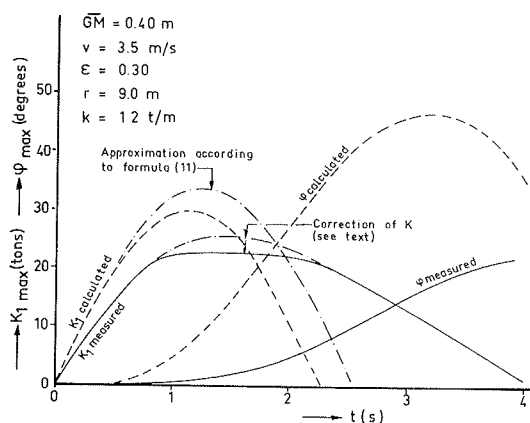


Fig.15.  $\Phi$  and  $K_1$  as function of time.

$$\Phi_{\max} = \frac{v}{\epsilon r} \frac{(\delta^2 \omega_1^2)(\delta^2 \omega_2^2)}{\delta^2(\omega_1^2 - \omega_2^2)} \left( \frac{\sin \frac{\omega_1}{\omega_1 + \omega_2} 2\pi}{\omega_1} - \frac{\sin \frac{\omega_2}{\omega_1 + \omega_2} 2\pi}{\omega_2} \right) \quad (17)$$

This relation has been used for systematic calculations, which revealed that for an arbitrary constant value of  $\Phi_{\max}$  approximately the following relation applies:

$$\frac{\overline{GM}}{GM_0} = \left( \frac{v}{v_0} \right)^2 \quad (18)$$

Where the subscript 0 refers to an arbitrary standard condition. Further ignoring the frictional resistance of the beamtrawls the following approximate expression is valid:

$$\frac{N}{N_0} = \left( \frac{v}{v_0} \right)^3 \quad (19)$$

From equations (18) and (19) it can be derived that the initial metacentric height meets the following relation with the ship's power:

$$\frac{\overline{GM}}{GM_0} = \left( \frac{N}{N_0} \right)^{2/3} \quad (20)$$

This theoretical result should be modified taking into account model test results and full scale measurements. From model test results it can be shown that the exponent of the ratio  $v/v_0$  of equation (18) is approximately equal to 1.5, which has been derived from Ref.4 (see e.g. Fig.6-8). Likewise it can be shown from full scale measurements (Ref.6) that the exponent of the ratio  $v/v_0$  of equation (19) is approximately equal to 1.5. For this purpose it has been assumed that an increase

of ship's power is consumed for 50% by an increase in ship's fishing speed and for 50% by an increase of the weight of the fishing gear, which is conform the existing practice. From these modifications it can be concluded that the initial metacentric height for the same  $\Phi_{\max}$  is approximately directly proportional to the ship's power in the range of actual fishing speeds i.e.:

$$\frac{\overline{GM}}{GM_0} = \frac{N}{N_0} \quad (21)$$

This basic concept of adaption of stability requirements to excessive propulsive power has been introduced in the national legislation for application to beam trawlers.

With regard to the above mentioned standard condition the following observations may be made:

- (1) According to the model experimental results and subsequent numerical calculations is  $\Phi_{\max}$  approximately independent of  $\epsilon$ ,  $k$  and  $r$  within the range of practical applications.
- (2) According to the model experimental results is  $\Phi_{\max}$  approximately proportional to  $v^{1/2}$ , whereas the theory predicts a linear relationship.
- (3) According to subsequent numerical calculations is  $\Phi_{\max}$  approximately proportional to  $\Delta^{-1/3}$  and both according to the model experimental results and subsequent numerical calculations is  $\Phi_{\max}$  approximately proportional to  $GM^{-1/3}$ .

It should be noted that these observations can be reflected in the following basic expression:

$$\Phi_{\max} = c_0' \frac{v^{1/2}}{(GM \cdot \Delta)^{1/3}} \quad (22)$$

Taking into account that  $N$  is approximately proportional to  $v^{3/2}$  (ref.6) equation (22) can be rewritten as:

$$N_0 = c_0 \cdot \overline{GM} \cdot \Delta \quad (23)$$

This relationship does actually give evidence that the standard condition, represented by the ship's power standard, can be expressed as a measure of the restoring moment, which is not an unexpected result.

## 5. CONCLUSIONS

1. Simulating a fastener of a beam trawler by model experiments is a feasible option, provided due attention is paid to the performance procedures.

2. The analytical solution of the simplified mathematical model produces sufficient accurate results for further analysis and drafting of regulatory requirements bearing in mind the observed discrepancies.

3. The maximum pulling force in the warp due to an abrupt fastener can be approximately estimated by means of equation (12), providing a basis for the assessment of damage to a submarine pipeline.

4. The maximum heeling angle due to fastening of one of the beamtrawls can be approximately estimated by means of equation (10), using a reduction factor of about 0.6.

5. It has been demonstrated that the initial metacentric height is approximately directly proportional to the ship's power assuming a constant maximum heeling angle, whereas for the reference value there is an indication that this standard condition can be described by the restoring moment coefficient.

6. The phenomenon of a fastener, which has been investigated for a beam trawler, can be extended to other types of fishing vessels although this represents a substantial less critical condition from the stability point of view because the r/B-ratio is reduced to less than one third of the value for a beam trawler.

## 6. ACKNOWLEDGEMENT

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## TRANSIENT AND IRREGULAR EXPERIMENTS FOR PREDICTING THE LARGE ROLLING IN BEAM IRREGULAR WAVES

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### ABSTRACT

As the one step before considering the ship capsizing, the prediction of large rolling motion in irregular waves must become possible satisfactorily. Of course the irregular waves are probabilistic phenomena but the rolling in given irregular wave is considered to be deterministic.

For this purpose, the damping and restoring force coefficients of a nonlinear equation of motion including the nonlinear wave exciting forces in irregular waves should be given.

But on the present stage, the purely theoretical estimation on all of those coefficients are not possible in regular waves, much less in irregular waves.

On the other hand, experimental studies in regular waves have been made but experimental studies in irregular waves, especially in severe waves, are very few.

So we show the experimental techniques for obtaining nonlinear damping coefficients by conducting large amplitude (35° in single amplitude) forced transient and irregular rolling experiments and for evaluating nonlinear wave exciting forces on a fixed model from experiments in high transient (19m in ship scale) and irregular waves. As a transient water wave (T.W.W), we used the so called concentrated T.W.W.

Obtained results are compared with calculated values by the so called strip method (here N.S.M) and also compared with some results of forced regular rolling or measured forces in regular waves.

We call those techniques as transient and irregular experiment like in the field of mechanical or electrical oscillation.

Finally, using above results, we show that the large amplitude rolling measured from the model experiments in high irregular

waves and in high concentrated T.W.W can be predicted in time domain by using the given time history of waves. And from this consistent procedure, obtained nonlinear characteristics are confirmed.

The reason why we adopted the concentrated T.W.W is that it will become an example of the technique of generating the wave group (or wave packet) part separately, including maximum wave height in an abnormal irregular waves. Usually the transient experiment is mainly used for linear phenomenon, but it seems to be useful for studying nonlinear phenomenon.

In this studies, we mainly consider deterministic, uncoupled rolling motion and deal with the model of a fine container ship. And adding this, some examples of a model of small fishing boat are also shown.

### NOMENCLATURE

$h(t)$	: Wave elevation
$H$	: Wave height
$\phi(t)$	: Roll angle
$\phi_0$	: Roll amplitude
$\phi$	: Roll double amplitude
$m(t)$	: Wave exciting roll moment
$y(t)$	: Displacement of sway
$f(t)$	: Wave exciting sway force
$S_h(\omega)$	: Power spectrum of $h(t)$
$S_\phi(\omega)$	: Power spectrum of $\phi(t)$
$M(\omega)$	: Fourier spectrum of $m(t)$
$G_{mh}(\omega)$	: $\equiv M(\omega)/H(\omega)$
$T_n$	: Natural period of roll
$\xi_1$	: $\equiv \sqrt{B/2g}$
$\xi_2$	: $\equiv \rho \nabla B^2$
$\hat{\omega}$	: Non dimensional angular frequency $\equiv \omega \cdot \xi_1$
$A_{44}$	: Non dimensional $A_{44}$ , $\equiv A_{44}/\xi_2$

- $B_{44}$  : Non dimensional damping coefficient  $\equiv B_{44} \cdot \xi_1 / \xi_2$   
 $B_{44}(2)$  :  $\equiv B_{44}(2) / \xi_2$   
 $C_{44}$  :  $\equiv C_{44} \cdot \xi_1^2 / \xi_2$   
 $\sigma_{\dot{\phi}}^2$  : Variance of  $\dot{\phi}(t)$

## 1. INTRODUCTION

About the rolling in the ship motions, the estimation of it by the practical method of so called Strip Method which is based on a linear theory is difficult because of over-estimation.

The weak damping makes the peak high and sharp in the frequency response function. And from this result, the effect of eddy making nonlinear damping and the nonlinear restoring force become strong relative to wave making damping force.

Further, that peak value is largely changed according to the magnitude of damping. So, the accurate estimation of a nonlinear damping is required. On the other hand, the nonlinearity of added mass moment of inertia is said to be small.

In order to overcome this difficulty, we usually introduce the damping coefficient in natural frequency obtained from a free rolling test into the Strip Method. And such an experimental formula including ship forward speed based on forced rolling experiment is also proposed. [1].

Recently, the 161 st Research Panel of The Ship Building Research Association of Japan (SR 161) made an energetic research on the roll damping.

And also a practical method of composing the damping coefficient from estimating its component is proposed [3] successfully in some extent.

On the other hand, as one example which measured damping in large rolling beyond the angle of deck edge immersion, there is an experiment [2] which used the gravity type forced rolling mechanism.

These are all regular experiments, and few experiments about the large amplitude irregular rolling are seen.

The investigations on the nonlinearity of damping coefficients based on irregular experiment, and on the relation between the results from irregular and regular experiment are very important for practical use.

We already investigated in such points in some extent [4, 5], but the consistent examination through making the prediction of maximum rolling time history in abnormal irregular waves was insufficient.

Further, in such a large motion case, the nonlinearity of wave exciting force which estimation is usually considered to be capable by linear theory, seems not to be ignored.

If investigations about these points are made and the reliable coefficients are obtained, the numerical examinations about the distribution of maxima of rolling in irregular waves [for example 6, 7] seems to become more practical.

In this paper we tried to show a

process consistently from the evaluation of roll damping coefficients to the time history prediction of maximum rolling.

For this purpose first we measure the rolling of a model ship in long crested irregular waves.

Used model is a container ship which seems to have a tendency to present large rolling. This ship is also used in the comparative studies of I.T.T.C Sea Keeping Committee. For comparison, some results obtained from the experiment of small fishing boat are shown. This experiments are made for the 17th regulation panel of the Shipbuilding Research Association of Japan (RR 17).

About experimental conditions, ship speed is zero, transverse wave condition and for simplification, and to maintain the consistency with forced pure roll the rolling with single degree of freedom case is treated.

The significant wave height of used irregular wave is about 11 meter in ship scale. This value corresponds to the abnormal wave of 3 years return period in the region of relatively severe seas in the Pacific Ocean.

Besides in irregular waves, we measure the rolling in concentrated transient water waves (Concentrated T.W.W). Concentrated T.W.W means a wave like the freak wave by the phase controlled wave generation. And for the high T.W.W used in this studies, wave breakers are seen in a wave front region or at the instance of concentration.

The reason why we use such a wave is as follows. It is convenient to be able to reappear isolately the part of wave packet including maximum wave height in an abnormal irregular wave.

And the technique demanded for realizing this wave is the method of generating arbitrary controlled T.W.W [8]. In this method, by controlling the phase of component wave constant at some position in a experimental tank, we can realize a maximum wave height within the specified Fourier spectrum.

This phenomena is called the concentration of T.W.W. Adding this, it is seemed that the nonlinearity of fluid forces become largest in this wave, so this artificial wave will give a kind of upper limit among maximum wave in a irregular waves with specified power spectrum.

In actual, strong nonlinearity is seen in the concentrated T.W.W. But in this considerations, the preceeding waves to the separated wave packet are explained to give initial values randomly to the rolling in this wave packet.

To the next, the transient experiment and the irregular experiment are made in order to grasp the nonlinearity of coefficients of the equation of motion and to give informations for response predictions.

This kind of experiments are very few.

First, large amplitude forced irregular roll and forced transient roll is made. And then the method of determining the coefficients which is more likely in actual motion from view point of equivalent linearized technique is shown.

For this purpose, we use a compact and simple forced roll mechanism which can be driven to the angle of  $40^\circ$  (single amplitude) according to the signal of arbitrary time history.

Next, we measure the wave exciting force in large amplitude irregular and concentrated T.W.W, and compare with the linear theory. These experiments are rarely seen in the past.

Finally, we make predictions about the roll time history in concentrated T.W.W using the results of transient experiment, the time history in high irregular waves using the irregular experiment, and the maximum roll time history in high irregular wave from transient experiment.

This time, for spectral analysis, we use the method of Fast Fourier Transformation (F.F.T.) and use the Q window.

And the values which are shown in ( ) following to some numerical figures mean that for actual ship.

## 2. MODEL AND EXPERIMENTAL CONDITIONS

The body plans of two models used in this study are shown in Fig. 1. To draw a distinction between two ships, each capitals [C] and [F] are written in the upper right hand side corner of following figures.

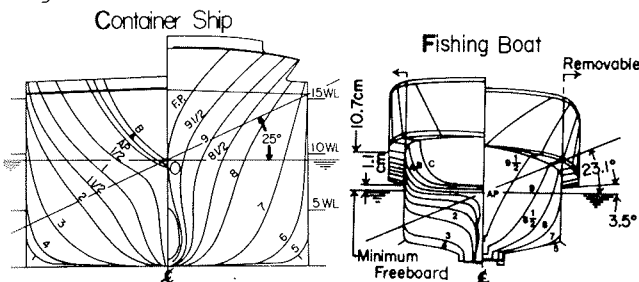


Fig. 1 Body Plans of Container Ship & Fishing Boat

Fishing boat is attached with box keel. And have the sponson (or over hanged deck). The lower face of this sponson touch the water level at  $3.5^\circ$  list. Further, this sponson is made to be removable.

Both are equipped with bilge keel (B.K). But without B.K. case is also experimented to some extent. And in this investigation, we do not treat the capsizing phenomena, so the superstructure is different from that of actual ships. About both ships, the upper face of the bulwark top are covered with acrylic plate. So there is few remaining water when there is shipping water.

Principal dimensions are shown in Table 1.

The rough sketch of ship motion measurement instrument is shown in Fig. 2. The sign in the figure define the directions of motion and external force. Model is setted right to the towing tank of 100 m in length and 8m in width.

Except surge and yaw, motions are made free. But most cases the sway is fixed. And for the case of sway free, the drift is supported by weak springs to make a similar condition of completely free.

Namely, the sway natural frequency by that spring is separated sufficiently from that of roll. The spring constants are shown in following Figure 6, 7. Of course in the theoretical calculations (New Strip Method = N.S.M) for comparison, the spring constants and the weight of sub carriage are included.

Table 1 Principal Dimensions & Experimental Conditions

Item	Container Ship [C] (SR-108-S7) scale 1/87.5		Fishing Boat [F] scale 1/8.47	
	Ship	Model	Ship	Model
Length-Lpp (m)	175	2.000	14.40	1.700
Length on Water Line-LWL	178.2	2.037	17.28	2.040
Breadth-B (m)	25.4	0.290	3.05	0.360
Depth-D (m)	15.4	0.176	1.38	0.163
Draft-d (m)	9.5	0.1086		
Displacement- $\nabla$ (m <sup>3</sup> )	24050 (Full)	0.0359	37.0	0.05939
dfore - d <sub>F</sub> (m)	9.5	0.1086	0.522	0.0616
daft - d <sub>A</sub> (m)	9.5	0.1086	1.597	0.1885
dmean - d <sub>M</sub> (m)	9.5	0.1086	1.059	0.1250
Trim (m) (%Lpp)	0.0	0.0	1.075 (7.47%)	0.1269
Block coefficient-C <sub>B</sub>	0.570	0.570	0.66	0.66
KG (m)	9.5 (=d)	0.1086	1.528	0.1804
KM (m)	10.52	0.1202	1.668	0.1969
GM (m)	1.02	0.0116	0.14	0.0165
GG (m) (+AFT)	2.48	0.0283	1.6	0.1889
Kxx/B (in Air)		0.2817		0.340
Kxx/B (in Water, with B.K.)		0.3027		0.432
Kxx/B (in Water, without B.K.)				0.411
Rolling period (in Water, with B.K.)	15.3	1.636	7.36	2.53
(sec) (in Water, without B.K.)				2.41
Length of Bilge Keel	0.25Lpp	0.25Lpp	0.5Lpp	0.5Lpp
Breadth of Bilge Keel (m)	0.45	0.005		0.0225
Rudder Area (cm <sup>2</sup> )		34.9		126
Note	without propeller		without propeller with sponson	

Wave height probe is putted on the sway sub carriage, so its position is always on the center line, and in order to measure the relative wave height, relative wave height probes are attached on both sides of each ship.

## 3. MEASUREMENT OF LARGE AMPLITUDE ROLLING

First, time history examples of rolling and heaving in transverse irregular waves of the container ship model are shown in Fig. 3.

The power spectrum of this wave is shown in Fig. 4 by a broken line. A P-M type spectrum is aimed.

The peak frequency of this spectrum is arranged to coincide with the roll natural frequency. But according to the breaking waves this spectrum is modified to some extent.

The solid line is a power spectrum of roll and becomes narrower than the wave.

The significant wave height is 10.2m in ship scale and the maximum height among

100 waves is 17.9m. This significant wave height corresponds to an abnormal irregular wave with 3 year return period from the data of a weather ship at the point "V" (34°N, 164°E) [9].

Looking at Fig. 3, it will be seen that the lump of waves (here we call it wave packet) which is formed from the envelope of the wave time histories make one to one correspondence with that formed from heave time history.

On the other hand, the packet formed from roll time history does not show one-to-one correspondence to that from wave. (For example © part in Fig. 3). This is because that the mean frequency of irregular wave are near to roll natural frequency and so a relatively large amplitude roll is always excited and the free roll is likely to remain.

But as to the part of wave, resulting fairly large roll (for example Ⓐ or Ⓓ in Fig. 3), we call these waves in abnormal irregular wave also abnormal wave, and there seems to exist correspondence between both packet.

The maximum roll angle in the abnormal irregular wave in Fig. 3 is 29.2 deg (Ⓐ part). In this connection, about the same type container ship as treated in this study, there is an estimated result on extreme maximum of 31° in 100 year return period and of 27° in 20 year return period according to the actual ship measurements [10]. Of course for this case, the sway is free.

The apparent periods (defined by the zero-up-cross method) of those abnormal waves are close to that of roll natural frequency. Converting this period into the wave length of a regular wave, the apparent wave height length ratio is about 1/13 for the case of Ⓐ in Fig. 3. This value is near the regular wave steepness limit.

Next, the measured example focusing on the abnormal wave packet like Ⓐ, Ⓓ in Fig. 3 is shown in Fig. 5.

The wave time history at the top of Fig. 5 is a concentrated T.W.W which is mentioned in Introduction and not the wave packet Ⓐ Ⓓ in Fig. 3 itself. But the degree of transient is strong.

About the roll in those waves like this T.W.W, there is a special feature of remaining free rolls after passing the wave packet.

In Fig. 5, Fourier Spectrum of wave and roll are also shown by  $H(\omega)$  and  $\phi(\omega)$ . Though the roll is large, the higher frequency components like  $2,3\omega_n$  are not seen in  $\phi(\omega)$ . And the narrow band feature of  $\phi(\omega)$  is outstanding.

The spectrum of T.W.W is aimed to become constant amplitude namely  $H(\omega) = \text{constant}$ . In this case, the amplitude of wave time history off the concentrated position (the duration or the sweep time become long) become constant approximately.

The central frequency of frequency range is chosen to coincide with  $\omega_n$ , so the apparent wave period at concentration be-

comes near to the natural period of roll.

Next, the frequency response functions of roll in above two kinds of waves are shown in Fig. 6. The result of irregular experiment shown by a broken line is obtained from the data of 120 second length (18 minutes in ship). In this nonlinear case, these response functions are considered to be equivalently linearized expression in some sense.

Calculated value by the New Strip Method (N.S.M) [11] are also shown. This is made in constant wave height condition and used nonlinear damping coefficients based on free rolls are shown in Table 2 with N coefficients.

Further in Fig. 6, the calculation by using nonlinear restoring moment is shown.

Comparing with irregular experiment and calculation, the peak value of calculation by the wave height equal to  $1.7 \times$  (significant wave height) — this value correspond to max height out of 200 waves — correspond to that of irregular experiment.

In the higher frequency range there are differences between calculation and experiment, and this seems to come from the difference in wave exciting roll moment.

This tendency in higher frequency range than  $\omega_n$  can be seen in Fig. 7(a) for fishing boat. In this figure the experimental result in a regular wave is also shown. But it is higher than calculation.

The frequency response function of roll in T.W.W like Fig. 5 is shown in Fig. 6. As we made  $H(\omega) \approx \text{constant}$ , it becomes similar to  $\phi(\omega)$ .

For this case the peak values are abnormally high. For the sway free case, this peak value is 2 times higher than that in irregular waves. And for the sway fix case, it is about 4.6 times. Of course the direct comparisons are not possible, but in usual sense, it will probable that the response in T.W.W will smaller because of its higher wave height.

This phenomena is seen for the fishing boat (Fig. 7(a)).

About the fishing boat, for the sway free case, the peak value in T.W.W is pushed down to the level of irregular experiment with sway fixed.

The reason why the peak become abnormally high will be as follows. Firstly, the wave exciting roll moment becomes large in T.W.W. Secondly, remaining free roll becomes more and more linear according to the time passed, and this part makes the peak value higher. So, it is questionable to use frequency response function obtained from such a T.W.W directly for predicting the large amplitude roll timehistory.

In the last, we show an example of relative wave height in Fig. 7(b). In T.W.W, weatherside, and for fishing boat case. For the sway free case, the amplitude in resonant frequency is small and for the sway fixed case there also seen the abnormal tendency including phase part.

The calculated values by N.S.M on



weather side are also shown by the broken line. This calculation is made by sway fixed condition and coincide with that of sway free experiment.

#### 4. MEASUREMENT OF COEFFICIENTS OF EQUATION OF MOTION

##### 4.1 EQUATION OF MOTION

In this paper we treat only the transverse wave condition. Then neglecting yawing motion, the roll  $(\phi)$  - sway  $(y)$  coupling equation of motion in general form are expressed as follows.

$$I_{22} \ddot{y} + D_{22} (\dot{y}) + [R_{22}(y)] + I_{24} (\ddot{\phi}) + D_{24} (\dot{\phi}) = f(t) \quad (1)$$

$$I_{44} \ddot{\phi} + D_{44} (\dot{\phi}) + R_{44} (\phi) + I_{42} (\ddot{y}) + D_{42} (\dot{y}) = m(t) \quad (2)$$

Where, suffix 2 means sway, 4 means roll. And I, D, R each means inertia force, damping, and restoring force. For example,  $D_{42}(\dot{y})$  means the roll moment depending on the sway speed.

Right hand side means wave exciting force and moment.  $R_{22}(y)$ , which appear in the case of sway supported by springs for suppressing drift, is putted into bracket.

For linear case, each term of roll is expressed as follows using frequency dependent coefficients and the similar expression is possible for sway.

$$I_{44} \ddot{\phi} = (J + A_{44}(\omega)) \ddot{\phi}, \quad D_{44}(\dot{\phi}) = B_{44}(\omega) \dot{\phi} \\ R_{44}(\phi) = C_{44} \phi, \quad (3)$$

$$I_{42} \ddot{y} = A_{42}(\omega) \ddot{y}, \quad D_{42}(\dot{y}) = B_{42}(\omega) \dot{y} \quad (4)$$

Where, J is moment of inertia of a ship herself, A is added mass moment of inertia, B is wave making damping coefficient, C is restoring moment coefficient and  $C_{44} = W \cdot GM$ .

In the case of large amplitude rolling, it is said that the nonlinearity of damping and restoring force is strong and the nonlinearity of added mass moment of inertia is weak. Further the coupling terms like  $D_{42}$  have nonlinearity, so we mainly treat roll with single degree of freedom for simplifying the consistent procedure of experiments and predictions.

As usually used nonlinear expressions of  $D_{44}$  and  $R_{44}$ , we adopt the following simple formula. Where,  $B_{44}E$ ,  $C_{44}E$  means the equivalently linearized coefficients.

$$D_{44}(\dot{\phi}) = B_{44}(1) + B_{44}(2) \dot{\phi} / \dot{\phi}_0 \\ \equiv B_{44}E(\omega, \dot{\phi}_0) \dot{\phi} \quad (5)$$

$$R_{44}(\phi) = C_{44}(1) \phi + C_{44}(3) \phi^3 \\ \equiv C_{44}E(\phi_0) \phi \quad (6)$$

Further as equivalently linearized expressions, we adopt the energetically equivalent formula for regular and transient motion [13], and also the formula evaluated from making the expectation of

square of difference between damping force by equivalently linearized method and by nonlinear expression minimum [14, 15] for irregular motion. These are as follows.

$$B_{44}E = B_{44}(1) + \frac{8}{3\pi} (\omega \dot{\phi}_0) B_{44}(2) \quad (\text{Regular or Transient}) \quad (7)$$

$$C_{44}E = C_{44}(1) + \frac{1}{2} C_{44}(3) \dot{\phi}_0^2 \quad ( " ) \quad (8)$$

$$B_{44}E = B_{44}(1) + \sqrt{\frac{8}{\pi}} \sigma_{\dot{\phi}} B_{44}(2) \quad (\text{Irregular}) \quad (9)$$

$$C_{44}E = C_{44}(1) + 3\sigma_{\dot{\phi}}^2 C_{44}(3) \quad ( " ) \quad (10)$$

$B_{44}(1)$  and  $B_{44}(2)$  in the formula (7), (9) are determined in a form of not depending on  $\omega$  by the analysis described in Appendix 1. Further, for the time history prediction, those coefficients are setted constant approximately as  $\dot{\phi}_0$  changes. Adding this  $A_{44}$  is considered to be constant in the base of  $\omega$ .

The fact that those coefficients are constant is convenient when we solve the differential equation numerically.

After all final expression mainly used in this paper is following equation of motion with constant coefficient.

$$(J + A_{44}) \ddot{\phi} + B_{44}(1) \dot{\phi} + B_{44}(2) \dot{\phi} / \dot{\phi}_0 + C_{44}(1) \phi + C_{44}(3) \phi^3 = m(t) \quad (11)$$

In equivalently linearized expression

$$(J + A_{44}) \ddot{\phi} + B_{44}E \dot{\phi} + C_{44}E \phi = m(t) \quad (12)$$

Finally we will consider the growing of an envelope of rolling time history response when a resonant sinusoidal wave moment is acted on a ship from the time zero. For this case the variation of envelope in time is expressed as shown in Appendix 2, and from this the time constant  $T_c$  of the envelope becomes as follows.

$$T_c = 2(J + A_{44}) / (B_{44}E) \quad (13)$$

##### 4.2 LARGE AMPLITUDE FORCED ROLL

In this section we show a method of obtaining the nonlinear coefficients in equation (11) considering equation (12) and by conducting the large amplitude forced transient and irregular pure roll. Used model is the container ship.

We write Fourier spectrum of roll  $\phi(t)$  as  $\phi = \phi_0(\omega) \exp(i\omega t)$  and Fourier spectrum of moment  $m(t)$  as  $M = M_0(\omega) \exp(i\omega t + \epsilon)$ . And from the Fourier transformation of equation (12), we can obtain the following relations. In this case,  $C_{44}E$  is able to be known from the stability curve.

$$J + A_{44} = [C_{44}E - (M_0 / \phi_0) \cos \epsilon] / \omega^2 \quad (14)$$

$$B_{44}E = [(M_0 / \phi_0) \sin \epsilon] \omega \quad (15)$$

The value  $M_0 / \phi_0$  and  $\epsilon$  which are function of  $\omega$ , can be obtained formally from the only one transient experiment.

So, using formulas (14) and (15), we can obtain the coefficients  $A_{44}$  and  $B_{44}E$  which are considered to be equivalently linearized in some sense. Of course this formulas can be used for regular experiment analysis. In the formula (14), if we use a familiar linear coefficient  $C_{44}(1) (\equiv W.GM)$  instead of  $C_{44}E$ , obtained  $A_{44}$  becomes smaller than it is. Because, in general,  $C_{44}(1) < C_{44}E$ .

First, we describe on the forced transient experiment. Before conducting experiment, we made numerical experiment about forced transient roll according to the equation (11). And confirmed the analyzing method and its accuracy. Namely, giving the coefficients of practical value, and changing  $\phi(t)$  like actual experiment, we calculated transient  $m(t)$  numerically. After that we applied formula (14) and (15), and checked the reappearance of those coefficients. From this numerical simulation, it is shown that even if the coefficients are constant, re-obtained  $B_{44}E$  makes hump and hollow on the base of  $\omega$ . And this phenomena is also seen in next experimental results. Of course, if we put  $B_{44}(2) = 0$  then this phenomena vanishes.

As the methods of obtaining  $B_{44}(1)$ ,  $B_{44}(2)$  from  $B_{44}E$  by the formula (15), following two are considered. As already mentioned in section 4.1, above two coefficients are expressed as functions of only  $\phi_0$ . So taking  $\phi_0$  constant in the formula (7),  $B_{44}E$  becomes linear function of  $\omega$ . Or approximating  $B_{44}E$  obtained by a transient experiment of constant  $\phi_0$ , by a linear function, the coefficients of this linear function gives  $B_{44}(1)$  and  $B_{44}(2)$ . For this case, two coefficients are obtained only one transient experiment. As described in the numerical simulation of forced transient roll, although there are hump and hollow, the mean line of  $B_{44}E$  can give  $B_{44}(1)$  and  $B_{44}(2)$ . To make apparent  $\phi_0$  constant, Fourier spectrum amplitude  $\phi_0(\omega)$  is to be made constant. And we call this a constant amplitude transient roll.

The second method is as follows. Taking  $\omega\phi_0$  constant in the formula (7), then  $B_{44}E$  becomes constant on the base of  $\omega$ . So by a forced transient roll with  $\omega\phi_0 = \text{constant}$ , we can obtain one  $B_{44}E$  as a mean value. Then conducting another  $\omega\phi_0 = \text{constant}$  experiment, we can obtain a relation between  $B_{44}E$  and  $\omega\phi_0$ . Approximating this relation by a linear function, we can obtain  $B_{44}(1)$  and  $B_{44}(2)$  according to the formula (7). This method is also checked by a numerical simulation. To make apparent  $\omega\phi_0$  constant, Fourier spectrum must be controlled like  $\omega\phi_0(\omega) = \text{constant}$ .

About this case,  $\phi_0(\omega)$  is diminished in higher frequency range, the phenomenon seems more close to the actual roll.

In Fig 8, a rough sketch of used experimental instrument for forced roll is shown.

This mechanism is composed of a low inertia servo motor connecting directly the roll axis through a reduction gear and strain gauge type dynamometer. So there is no extra source of friction.

Natural frequency of the dynamometer with model ship is about 10 Hz. And so, this frequency and the forced roll motion frequency can be separated without filter. Axis of rotation of this pure roll is coincide with the center of gravity.

Experiments are conducted with forward speed of  $Fn=0, 0.1, 0.2, 0.275$ , and the reflected wave is absorbed by the side beach.

About the container ship, the deck edge is immersed at a heel of 25 degree. So a superstructure of 60 cm in length and 6 cm in height is equipped. And then the experiments with maximum roll single amplitude of 35 degree are made. A photograph of an experiment is shown in Photo 1.

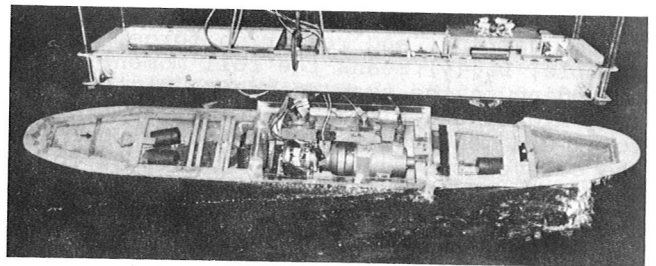


Photo 1. Forced Roll Experiment

Next in Fig 9, we show an example of forced transient roll time history. Speed is zero. This example contains two case with the same roll maximum amplitude about 15 degree but different duration (or sweep time). Fourier spectrum of the two case is similar with constant amplitude, so for the long sweep time case the apparent  $\phi_0$  become constant.

Roll time history of short sweep time correspond to the case of concentration in Transient Water Wave field.

Calculating  $M/\phi$  and  $\epsilon$  from the Fourier spectrum of these time history and applying the formula (15), obtained nondimensional  $B_{44}E$  is shown in Fig 10. And obtained nondimensional  $B_{44}(1)$  and  $B_{44}(2)$  from mean line approximation are also shown. The case of without bilge keel is shown in lower side.

It will be noticed that, looking at mean line, the effect of bilge keel is outstanding comparing to the effect of sweep time. Fig 11 show how the difference between with and without bilge keel effects on  $M/\phi$  and  $\epsilon$ , and it is said that phase difference is very important. The top figure is  $\phi_0(\omega)$ , and it will be seen as constant.

The case of  $Fn=0.275$  is shown in Fig 12.  $\phi_0(\omega)$  is also constant. Three experiments with different  $\phi_0$  are shown. If  $B_{44}(2)$  does not depend on  $\phi_0$ , the same value should be obtained from these experiment. But the analyzed results give a little different coefficient as shown in

next Fig 13. Further, as shown in Fig 12, the wave making damping by Strip Method is considerably small. In the bottom of Fig 12, the obtained value of mass moment of inertia by the formula (14) are shown. For this case, as  $C_{44}E$ ,  $W.GM(=C_{44}(1))$  is used. But if we use the expression (8) as  $C_{44}E$ , the corrected value approach to the theoretical one. This is shown by an arrow. And, in the result, the nonlinearity of  $A_{44}$  is considered to be small as said so far.

We also made experiments of the case  $\omega\phi_0 = \text{constant}$ . But here, we show the corrected results in Fig 13. The results of forced irregular experiment mentioned after are also included.

Fig 13 is drawn in the base of Froude Number, but when we see  $B_{44}E$  at one  $Fn$ , we can understand the effect of  $\phi_0$ . The coefficient  $B_{44}(1)$  can be arranged by a line with positive inclination. About the regular experiment, the data by Tasai [16] is quoted as reference, and this value occupy a little lower position. Where the  $\max \phi_0$  was about 15 degrees. Next  $B_{44}(2)$  can not be arranged by a line, and this show the effect of  $\phi_0$  on  $B_{44}(2)$ . But, as a whole, there is a tendency of decreasing as  $Fn$  increase. Namely as  $Fn$  increase, the nonlinearity is diminished. This phenomena is already pointed out [17] in the past.

The contour curves of  $B_{44}E(\omega, \phi_0)$  at specified  $B_{44}E$ , are shown in Fig 14.  $Fn=0$  case. Two figures in lower side is the section of AA' and BB'. From this figure it will be seen that the increase of  $B_{44}E$  is suppressed over 25 degree. This means that there is an inclination of decreasing  $B_{44}(2)$  as  $\phi_0$  increase.

For the motion prediction in T.W.W, the representative value of large amplitude forced transient roll is used. To put it concretely the value at 25 degree. Then for small  $\phi_0$ ,  $B_{44}(2)$  is evaluated smaller than it is, so the overestimation will appear.

Next, the results from forced irregular roll are shown. Fig 15 shows roll angle and reacted moment and power spectra. The power spectrum of  $\phi(t)$  is chosen as narrow banded so as to become more actual. The spectrum of  $\sigma\dot{\phi}$  is also shown because in order to apply the expression (9), the value of  $\sigma\dot{\phi}$  is needed.

An example of obtained  $B_{44}E$  from the formula (15) is shown in Fig 16.  $M/\phi$  and  $c$  are obtained from using the cross spectrum of  $\phi(t)$  and  $m(t)$ , and the auto spectrum of  $\phi(t)$ . In this figure, the coefficient  $A_{44}$  is neglected, but the results are the same as transient experiment.

Narrow banded roll spectrum results in a constant  $B_{44}E$  in the base of  $\omega$ . So from one experiment with one  $\sigma\dot{\phi}$ , a mean value of  $B_{44}E$  can be obtained. If we approximate the relation of  $B_{44}E$  and  $\sigma\dot{\phi}$  from many experiments as a straight line, we can obtain  $B_{44}(1)$  and  $B_{44}(2)$

applying the formula (9).

The relation between  $B_{44}E$  and  $\sigma\dot{\phi}$  is shown in Fig 17 for  $Fn=0, 0.275$  case. The variation of coefficients in  $Fn$  base is shown in former Fig 13.

In the former Fig 14, the irregular experimental results are shown by broken lines. This is obtained by substituting  $B_{44}(1)$ ,  $B_{44}(2)$  from irregular experiment into the formula (7). From this figure, over 25 degree, it will be seen that the  $B_{44}(2)$  from irregular experiment becomes greater than that from transient experiment. And this fact will results under estimation in time history prediction as one reason.

#### 4.3 MEASUREMENT OF WAVE EXCITING ROLL MOMENT

Used T.W.W is similar to that used for motion measurement, and the apparent  $\omega$  in concentration becomes  $\omega_n$ . Measuring instrument is the same as shown in Fig 2, and further a strain gauge type dynamometer is equipped at the bottom of the heaving rod. Heave is made free and bilge keels are attached. Sway is fixed. But when we support the sway motion by springs, the change in reaction force give the term  $I_{42}$  and  $D_{42}$ . However, this time, we used theoretical value as coupling coefficient.

Fig 18 shows experimental results. Vertical axis is the response function of wave exciting roll moment  $G_{mh} (=M(\omega)/H(\omega))$ . In the case 1, the signal of wave generator is kept the same, and the measured position is selected 3 points in both sides of concentration. It is the same about case 2.

And so, the wave height becomes maximum at concentration. In Fig 18,  $G_{mh}$  show a non-linear character of increasing as the apparent wave height increase. This result differs to that from regular response in small wave height and from theoretical calculation. From the theoretical calculation it is said that this experimental condition correspond to the minimum wave making damping. But in this calculation the effect of bilge keels are not considered.

It will be said that about the calculation of lateral forces by N.S.M, the Lewis Form approximation is used, but the Close Fit Method [18] will be better. But also by this case the nonlinearity shown in Fig 18 can not be explained.

Here we arrange experimental results as follows and used for time history prediction. It is clear that  $G_{mh}$  takes a form depending on  $\omega^2$  in low frequency range, so we approximated  $G_{mh}$  with next polynomials

$$\begin{aligned} |G_{mh}(\omega, H)| &= a(H)\omega^2 + b(H)\omega^3 \\ \arg(G_{mh}) &= C(H) \end{aligned} \quad (16)$$

From each experiment we can obtain  $a$ ,  $b$  and  $c$ . So plotting these coefficients on the

base of maximum wave height ( $H_{max}$ ), then the obtained result show that those coefficient change is proportional to  $H_{max}^2$ . From this result, rearranged result of  $a(H)$ ,  $b(H)$  on the base of  $H_{max}^2$  are shown in Fig 19. But, in detail, the case 1 result and case 2 result seems to have another tendency. But it is noticed that the concentration have not an extra tendency.

And then, from the mean lines of these points, we can obtain the coefficients  $a_0$ ,  $a_1$ ,  $b_0$ ,  $b_1$ . Judging from experiment, the phase difference is considered as constant on the base of  $\omega$ , and arranged by  $H_{max}^2$  on the base of  $H$ .

From this line approximation it will be said that the difference between experiments and mean lines become smaller as wave height increase.

The results about the fishing boat are shown in Fig 20. Curves in upper half of Fig 20 are approximated results by square functions. Measured position is 60% point from the wave maker when the concentrated position is considered as 100% point.

There may be the reason that the measured position is far from concentration calculated value by Strip Method is relatively close to the experiments than that of the container ship.

Coefficients for polynomial approximation of  $G_{mh}$  is arranged on the base of  $H_{max}$ , and these are shown in the lower part of Fig 20. Each coefficients seem to be function of  $H_{max}^2$  like the case of container ship. And this tendency is more strong in the case of sponsons are fitted. But it is noticed that the tendency of increase or decrease as  $H_{max}^2$  increase is not the same in both ships. On the other hand, the experimental results of  $G_{mh}$  in concentrated position shown as row result show an extraordinary tendency in lower frequency range. And this seems to be similar to the container ship result.

Further, also in irregular waves, we obtained  $G_{mh}$ . This is shown in Fig 18 by a broken line. Used irregular wave is corresponded to the wave shown in Fig 3. That broken line is approximated results as follows. First, obtained  $G_{mh}$  from an experiment is approximated by a function  $A\omega^2$ . Then another experiment of  $\sigma_h^2$  gives another A. So, if we approximate the relation between A and  $\sigma_h^2$  by a function  $A=a+b\sigma_h^2$ , then we can obtain an expression

$$G_{mh} = (0.0211 + 18.08\sigma_h^2) \quad (17)$$

The phase part is approximated by a similar method.

Looking at Fig 18, although the maximum wave height of that irregular wave is comparative with the T.W.W.,  $G_{mh}$  is not so high as in T.W.W. concentration. And this broken line is closer to that of T.W.W. with about half of that maximum height, namely case 2.

In any way it is shown that there are some

what large wave height effect about  $G_{mh}$ . Further, although we do not show the results of sway force response function, the Strip Method give a good estimation on sway force, both for container ship and fishing boat, obtained from experiment.

## 5. PREDICTION OF ROLL TIME HISTORY

### 5.1 CALCULATION METHOD

The method of calculating time history of roll-sway coupling motion against the incident wave like Fig 3 or Fig 5 is shown in Fig 21.

First, by  $\sigma_h^2$  from given irregular wave or by maximum wave height  $H_{max}$  from given Transient Water Wave, obtain  $G_{mh}(\omega)$  using the formula (16) or (17).

Next, by a inverse Fourier transformation of  $G_{mh}$ , calculate the impulse response function of roll moment  $P_m(t)$  against the incident wave.

Further from the convolution of incident wave height  $h(t)$  and  $P_m(t)$ , calculate the roll moment time history  $m(t)$ .

About sway force, we calculate  $f(t)$  from linear  $G_{fh}$  by the Strip Method (N.S.M). Using the obtained  $m(t)$  and  $f(t)$  as the right hand side of nonlinear equation (11) or linear equation for sway, solve those equations by the Runge-Kutta-Gill method. The coefficients of equation of motion for sway are also obtained from N.S.M.

If we require the final response only, the method of direct prediction of  $\phi(t)$  and  $y(t)$  will be considered, using the impulse response function of the transfer function between incident wave  $h(t)$  and roll  $\phi(t)$   $G_{\phi h} (= G_{\phi m} \cdot G_{mh})$  which include  $H_{max}$  (for example) as a parameter. But in this paper we examined the coefficients itself or the form of equation of motion, and so we do not adopt such a method. Further, for nonlinear case, that direct prediction method is not so simple for the coupling motion. We also do not adopt the equation of motion with equivalently linearized coefficients because it include  $\omega$  and  $\phi_0$  explicitly and this make the solution method complexed as mentioned [17].

In Fig 22, examples of impulse response functions by inverse transformation of  $G_{mh}$  and  $G_{fh}$  (response function of wave exciting sway force) are shown. Calculation is made of the interval of  $\pm 10$  sec. This case,  $H_{max}$  is chosen as 0.2m (17.5m in ship) and the T.W.W. case. Outstanding point is that comparing to the sway force the damping of moment is very large. By the way, calculation interval  $\Delta t$  is chosen as 0.05 sec following to the numerical simulation results. The coefficients which are obtained from transient experiment and used for time history predictions are shown in Table 3.

### 5.2 EXAMPLES OF PREDICTION

Here, we will show first the examples predicting roll time history in neighbour-

hood of concentration of T.W.W. using the coefficients obtained from transient experiment. Fig 23.  $H_{max}$  reaches 21m in actual ship scale. Fourier spectrum and frequency range is similar to that of Fig 5, and  $\phi_0(\omega) = \text{constant}$ . In (b), maximum is good prediction, and in (a) and (c) there are some differences in maxima, but this case become also satisfactory when we consider a small bias term.

So it will be said that the coefficients obtained from independent experiments and used equation form are suitable in such a strongly transient phenomena. On the other hand, there seems to exist some discrepancies in the small wave height region but it will be probable because the calculation method is aiming at the maximum amplitude prediction. After the wave passed, there remains free roll like Fig 5. Free rolls are not shown in Fig 23 but the prediction will not good considering the former discussion.

Next in Fig 24, an example of predicted irregular roll using the results of irregular experiment is shown. Though shown time history is only one part, the prediction is in good agreement with experiment as a whole, excepting the region of extreme maximum. This extreme part prediction shows 30% underestimation.

In order to improve this defect, next in Fig 25 we show the results of prediction of this extreme part by using coefficients and  $G_{mh}$  from transient experiment. As  $H_{max}$ , we adopted the wave height of extreme maximum.

The Fig 25(a) corresponds to Fig 24 and also to the part (A) in Fig 3. Fig 25(b) is quoted from another part of experiment of Fig 3. And, although there are no relation, except for the similar apparent frequency, between irregular wave and T.W.Wave, the prediction of extreme maximum of this time show good agreement with experiment.

The other part of prediction is not good showing overestimation. But this fact itself assert the nonlinear character in high waves. As already described, the damping coefficient in irregular roll is not so different from that in transient roll, so the main reason of former discrepancies will exist in the  $G_{mh}$  in Fig 18.

The difference between transient experiment and irregular experiment in Fig 18 seems to show that the  $G_{mh}$  obtained by evaluating whole data of irregular experiment cannot explain the characteristics of abnormal part.

Further in Fig 25, the Prediction 1 and 2 expression shows the difference of the point of starting time of prediction.

### 5.3 SOME EXAMINATION BY A NUMERICAL SIMULATION

It is already pointed out in former section that when we consider the rolling

in a separated wave packet including extreme maximum from an abnormal irregular wave, the final value of motion ( $\phi$  and  $\dot{\phi}$ ) in the preceeding wave packet becomes initial values. And this initial values are usually random.

Here the effects of these initial values on the transient response are examined by numerical simulation.

A concrete example is shown in Fig. 26. This wave is a sequence of three waves with  $H_w = 0.25m$  (21.9m in ship) and with  $T_n$  which is the same as period from free roll. As coefficients of equation of motion, we used the results from transient experiment.

This wave height is a little higher than the concentrated T.W.W used for transient experiment, but lower than the extreme maximum of about 30m observed in the past.

This wave is very steep, so each wave is expressed by 3rd order Stokes Wave.

If we do not use the method of simulation it will be difficult to examine initial value effect in such a model wave.

In Fig. 26, there are 9 combinations about initial value, and these are chosen as extraordinary values than usually expected.

Though, given such an extreme initial value, it will be seen that the effect of differences of initial values vanish within the range from the end of first wave and to the peak of the second wave. Namely it will be considered that the roll response in a large and near resonance period wave packet (we also call such a wave as abnormal wave) will reach the state of steady response or maximum amplitude rapidly, so the initial value in these wave packet or the final value in the former wave packet do not give much influence upon the extreme maximum roll angle.

Main reason of this phenomena is that the damping become so large and the apparent time constant  $T_c$  based on the formula (13) become so small. In this case the ratio  $T_c/T_n \div 0.6$ .

This means that the amplitude of the envelope of roll angle  $\phi(t)$  grows rapidly to 63% of steady state within 0.6 times roll natural period  $T_n$ .

Further using this method, the response function  $\Phi/kH$  at  $\omega = \omega_n$  is plotted in the Fig. 6. In this case, although the coefficients used are obtained from transient experiment, the abnormal high peak is not seen. And so, from this reason also, the very high peak obtained from concentrated T.W.W will be influenced by the remaining free roll with strong linearity.

Finally an example of the numerical simulation of roll-sway coupling motion in large wave is shown in Fig. 27. Here,  $T = 1.956$  ( $>T_n$ ) and  $H = 0.25m$ .

In this figure, roll exciting moment  $m(t)$  and sway exciting force  $f(t)$  are also shown. About this calculation, roll-sway coupling coefficients are obtained from linear Strip Method (N.S.M) and coupling effect is very small. But in experiment, there seems to be a fairly strong coupling effect, so about this motion there are much

problem to be improved.

## 6. CONCLUSION

In this investigation, we treated the large amplitude irregular rolling of a fine container ship, and investigated from the evaluation of damping coefficients by experiment to the prediction of maximum rolling in an abnormal irregular wave in time history domain consistently.

In this studies we made deterministic considerations because it is assumed that the incident irregular wave are given in time domain.

Here we mainly investigated pure roll with heaving free and only a few example are shown for roll-sway coupling motion.

The main conclusion is listed as follows.

1) By a compact forced roll device which can be driven by an arbitrary signal we made large amplitude forced irregular roll experiment and forced transient roll up to the angle of 35 degree in single amplitude. And the Froude number is changed from 0 to 0.275. This angle is as large as rarely seen in the past.

2) From this experiment we showed the method of obtaining nonlinear damping coefficients applying the expression of equivalently linearized formula (7), (9). For this expression the coefficients are made to be the functions of  $\phi_0$  only.

3) Obtained damping coefficients from two kinds of experiment showed about the same value. These coefficients are fundamentally identical, so this result means the appropriateness of the different expression of equivalently linearized coefficients.

4) The coefficient  $B_{44}(1)$  which is related to the damping proportional to angular velocity, increase linearly with the increase of Froude Number. On the other hand the  $B_{44}(2)$  of the term of the square of angular verocity decrease linearly.

5) The response function of the wave exciting roll moment in large amplitude irregular wave and concentrated T.W.W showed the rather strong nonlinearity especially for T.W.W increasing with the wave height. And this is different from the linear tendency as usually said.

6) When we evaluated the moment response function by coefficients including a maximum wave height as a parameter, they showed a tendency of changing depending on the square of wave height or the variance of wave elevation. This tendency is also seen in the results of a fishing boat.

7) About the wave exciting sway force, the nonlinearity of it is weak and can be estimated by a linear strip method (N.S.M).

8) From above results, the time history prediction in concentrated T.W.W with strong nonlinearity can explain the experiment well, using the coefficients from transient experiment. This fact means that the expressions of equation of motion

considered here and the obtained coefficients are reasonable.

9) Though, it must be noticed that the roll angle response function obtained from the measurement in concentrated T.W.W show a very large peak value in spite of large damping. But this phenomena will because that in remaining free roll after passed the T.W.W, the linearity become stronger and stronger and then the response function reaches to that of linear case.

10) The roll time history prediction in irregular wave using the coefficients from irregular experiments coincide with the experiment excepting the extreme maximum part (29 degree in half amplitude).

11) Further, the part of this extreme maximum, can be explained using the coefficients from transient experiments within the error of  $\pm 5\%$  in double amplitude.

12) The reason why we used the T.W.W which were used for linear case to the nonlinear phenomena is that we aimed the generation of only the separated wave packet with maximum wave height in abnormal irregular wave. And the result of this investigation will show the effectiveness of such a use.

13) Finally following fact will be said from some numerical simulation. In the rolling response in abnormal waves which result extreme maxima, the damping become very strong.

This further implies that when a separated wave packet with abnormal wave enters, the effects of damping on the maximum roll in this wave is larger than that of initial values. So, from this consideration, it will be said that for prediction of roll extreme maximum it is sufficient to predict the response in a T.W.W which are composed of a few waves including abnormal wave. The result mentioned in 11) is seemed to support this consideration.

## ACKNOWLEDGEMENT

First, the authors express their gratitude to Mr. K. Miyakawa and Mr. T. Takayama for their carrying out the experiments and for preparing the figures of this paper. They thank all the members who took part in this study and graduated from our laboratory, especially Mr. J. Haraguchi who improved the calculation program for numerical simulation. The authors also thank Prof. S. Motora and Prof. M. Fujino who lent the fishing boat model pleasantly for us. Finally they are grateful to Mr. Y. Yamakoshi who offered the data about the fishing boat.

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## APPENDIX 1

Here, we describe the relation between the expression of damping coefficients. The nonlinear damping expression which is usually used is as follows with frequency dependent coefficients  $B_1(\omega)$  and  $B_2(\omega)$ .

$$D_{44}(\dot{\phi}) = B_1(\omega)\dot{\phi} + B_2(\omega)\dot{\phi}^3 \quad (A-1)$$

The equivalently linearized expression of (A-1) is expressed as next formula.

$$B'_{44E}(\omega, \phi_0) = B_1(\omega) + \frac{8}{3\pi} \omega \phi_0 B_2(\omega) \quad (A-2)$$

Where  $\phi_0$  is roll amplitude.

Next we consider the Taylor expansion of  $B_1, B_2$  in  $\omega$  then

$$B'_{44E}(\omega, \phi_0) \approx B_1(0) + \frac{8}{3\pi} \omega \phi_0 (B_2(0) + \frac{3\pi}{8\phi_0} \cdot \frac{dB_1(0)}{d\omega}) + \dots \quad (A-3)$$

Where  $B_1(0)$  is considered to be zero at forward speed is zero.

On the other hand, the equivalently linearized expression at a specific frequency  $\omega$  of the formula (5) becomes as follows.

$$B_{44E}(\omega, \phi_0) = B_{44}(1) + \frac{8}{3\pi} \phi_0 B_{44}(2) \cdot \omega \quad (A-4)$$

where  $B_{44}(1)$  and  $B_{44}(2)$  is only the function of  $\phi_0$ .

So when we approximate  $B_{44E}$  which are obtained from the forced transient roll with constant amplitude  $\phi_0$ , as a linear function in the base of  $\omega$  according to the formula (A-4), the relation between obtained coefficients  $B_{44}(1)$ ,  $B_{44}(2)$  and  $B_1, B_2$  is as follows from (A-3) and (A-4),

$$B_{44}(1) = B_1(0) \quad (A-5)$$

$$B_{44}(2) = B_2(0) + \frac{3\pi}{8\phi_0} \frac{dB_1(0)}{d\omega} \quad (A-6)$$

$dB_1(0)/d\omega$  is usually positive, so  $B_{44}(2)$  which is the function  $\phi_0$  decrease as  $\phi_0$  increasing. Further at large  $\phi_0$   $B_{44}(2) \approx B_2(0)$ .

## APPENDIX 2

Here we describes apparent time constant briefly. Considering equation (12) and we put

$$\zeta = \frac{B_{44} E}{2\sqrt{(J+A_{44}) \cdot C_{44} E}}, \quad \omega_n = \sqrt{\frac{C_{44} E}{J+A_{44}}} \quad (A-7)$$

and when the sine wave exciting moment with resonance frequency is acted from the time  $t=0$ , then the solution becomes

$$\phi(t) \propto \frac{1}{2\zeta} [-\cos \omega_n t + e^{-\zeta \omega_n t} (\cos \omega_d t + \frac{\zeta}{\sqrt{1-\zeta^2}} \sin \omega_d t)] \quad (A-8)$$

where  $\omega_d = \sqrt{1-\zeta^2} \cdot \omega_n$ .

From equation (A-8) considering the envelope variation the apparent time constant  $T_c$  is expressed as follows.

$$T_c = 1/\zeta \omega_n \equiv 2(J+A_{44})/B_{44} E. \quad (A-9)$$

Table 2 Free Roll Results

Free Roll Results			
	Container Ship	C	Fishing Boat
$B_{44}(1)$	0.00197		0.0138
$B_{44}(2)$	0.0550		0.119
$N_{10}$	0.0201		0.0452
$N_{20}$	0.0136		0.0294

Table 3

Coefficients of Equation of Motion for Simulation

(Model Scale)			C
$J+A_{44}$	0.026	Kg·m·sec <sup>2</sup> /rad	
$B_{44}(1)$	0.0033	Kg·m/(rad/sec)	
$B_{44}(2)$	0.0251	Kg·m/(rad/sec) <sup>2</sup>	
$C_1$	0.4164	Kg·m/rad	
$C_3$	0.8738	Kg·m/(rad) <sup>3</sup>	

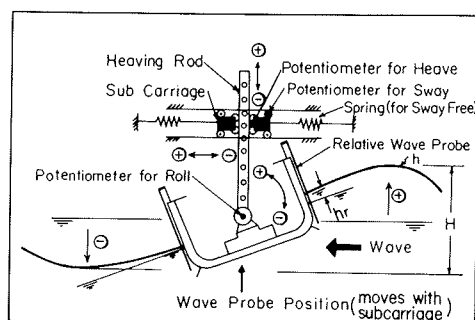


Fig.2 Rough sketch of motion measurement

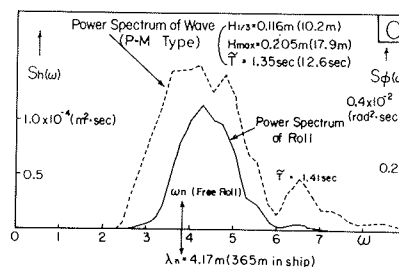


Fig.4 Power spectrum of the wave and roll of Fig.3

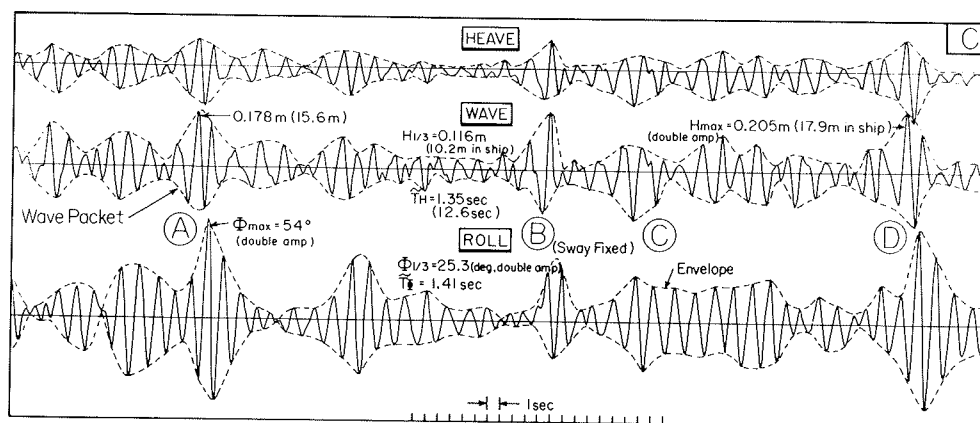


Fig.3 An example of Roll and Heave in a transverse severe irregular wave. (Experiment)



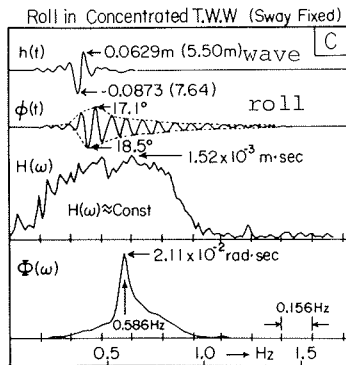


Fig.5 Roll in a concentrated T.W. Wave

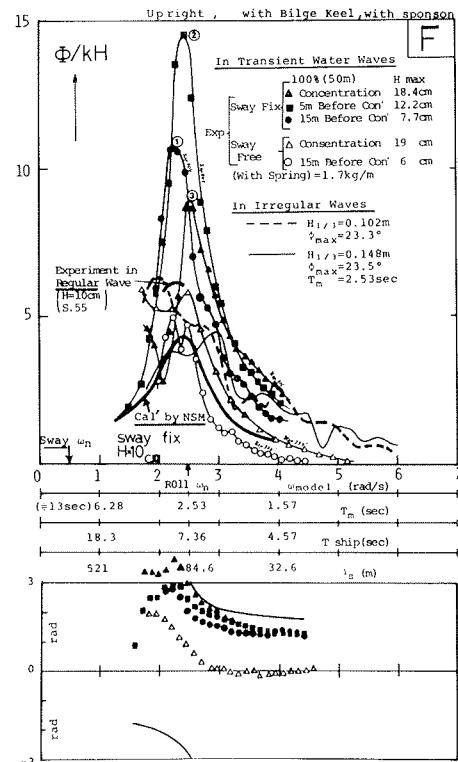


Fig.7(a) Response function of roll (Fishing Boat)

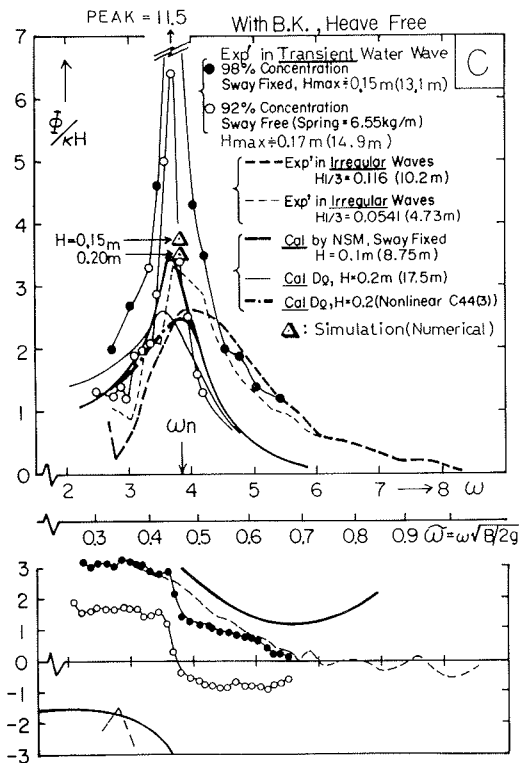


Fig.6 Response function of roll (Container Ship)

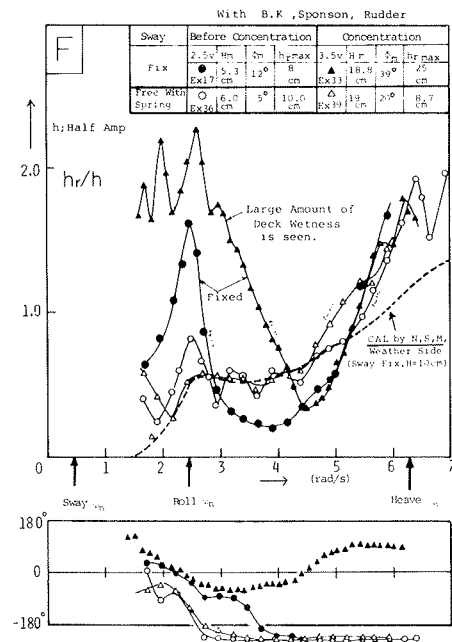


Fig.7(b) Response function of relative wave height

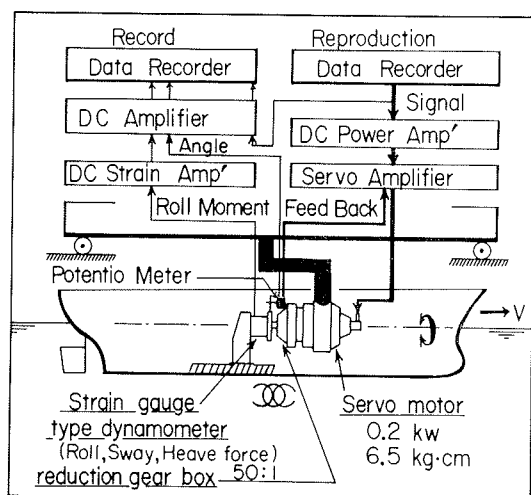


Fig.8 Rough sketch of forced roll experiment

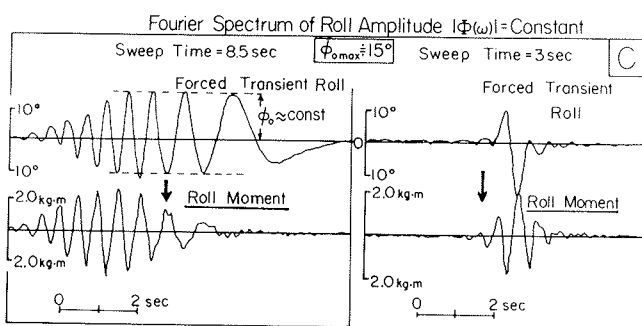


Fig.9 Example of forced transient roll

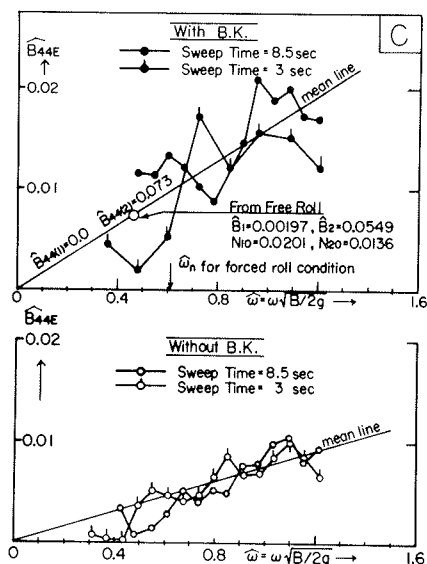


Fig.10 Obtained damping coefficients, forced transient roll ( $F_n=0$ )

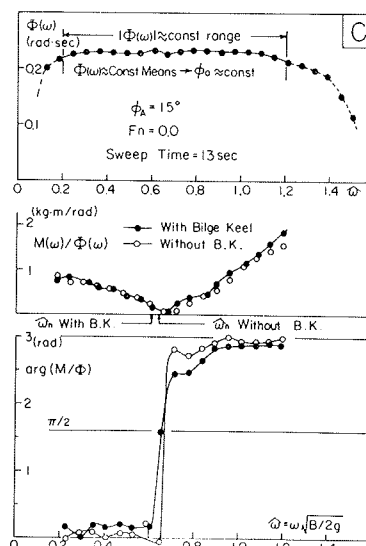


Fig.11 Response function of forced transient roll

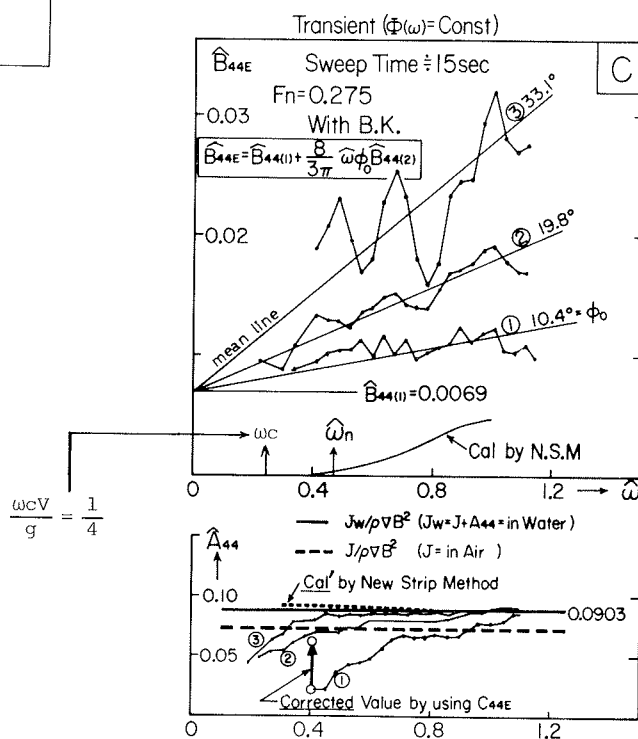


Fig.12 Obtained damping coefficients from forced transient roll ( $F_n=0.275$ )

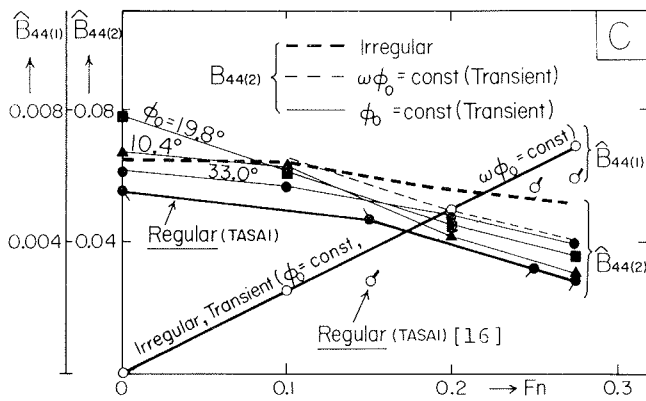


Fig.13 Obtained damping coefficients from forced transient, irregular and regular roll.

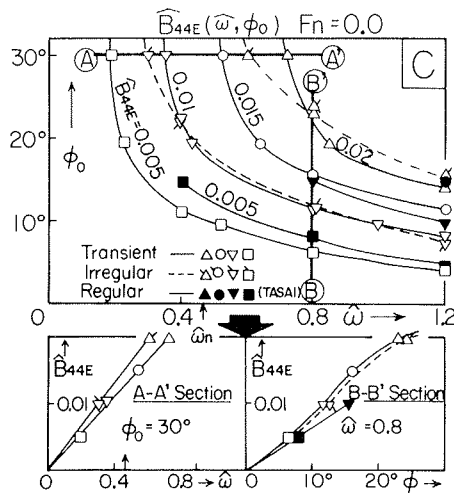


Fig.14 Iso- $B_{44E}$  ( $\omega, \phi_0$ ) curve at  $F_n=0$  and its sectional form

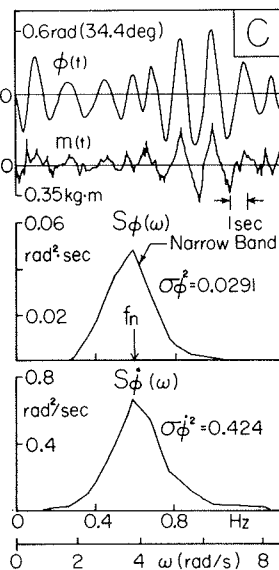


Fig.15 Example of forced irregular roll.

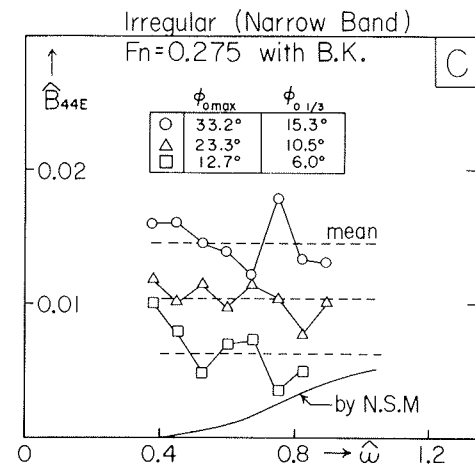


Fig.16 Obtained damping coefficient from forced irregular roll.

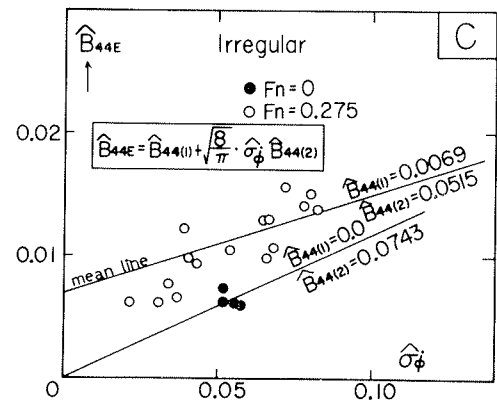


Fig.17 Relation between obtained  $B_{44E}$  and  $\sigma_{\phi}$

#### Roll Exciting Moment in High Transient Water Wave

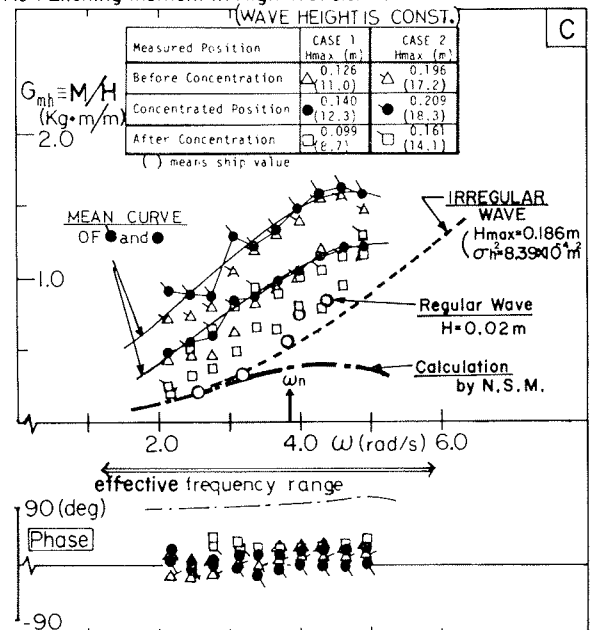
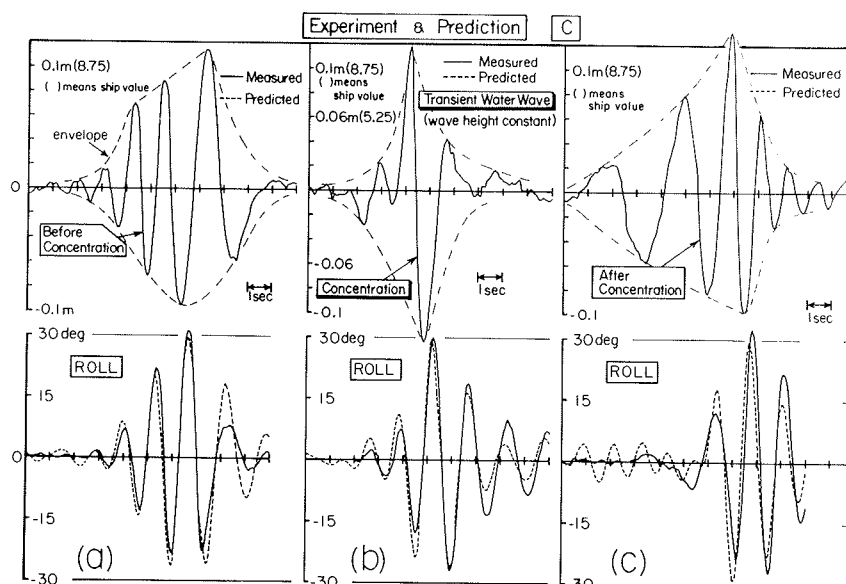
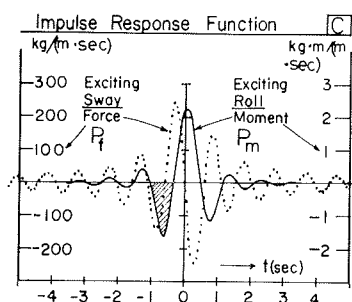
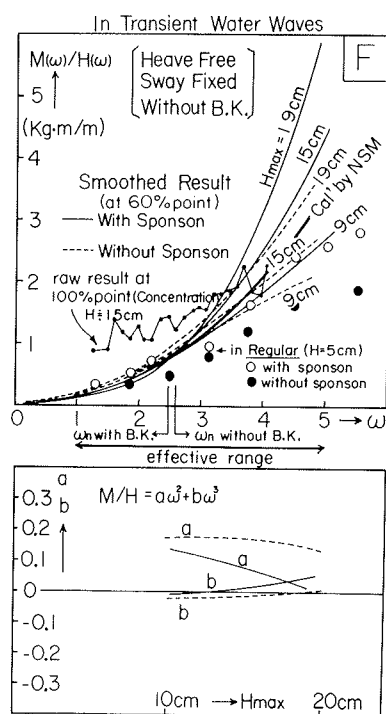
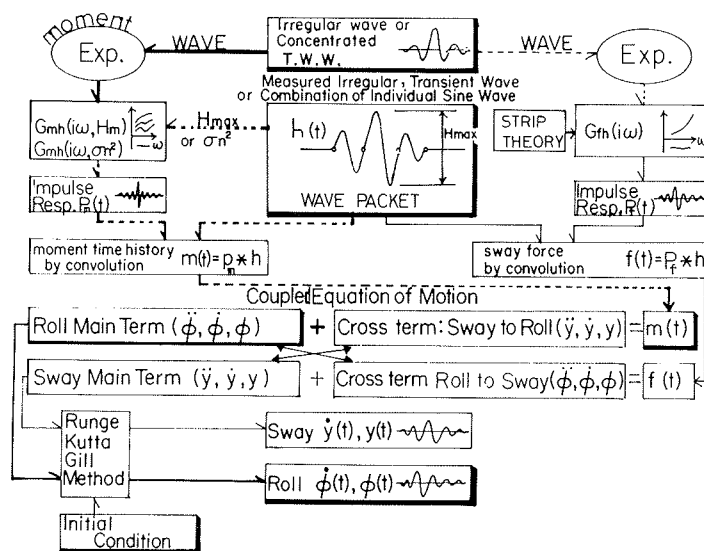
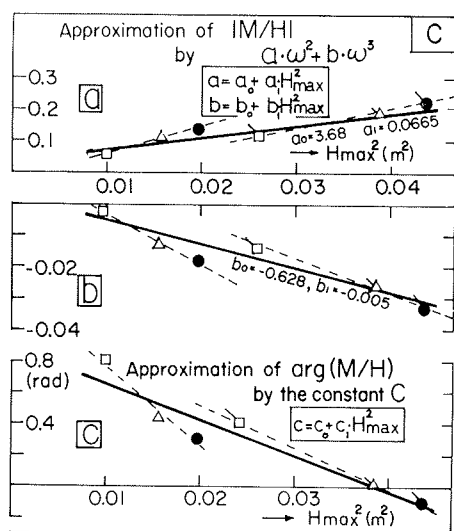


Fig.18 Wave exciting roll moment in high Transient Water Wave.



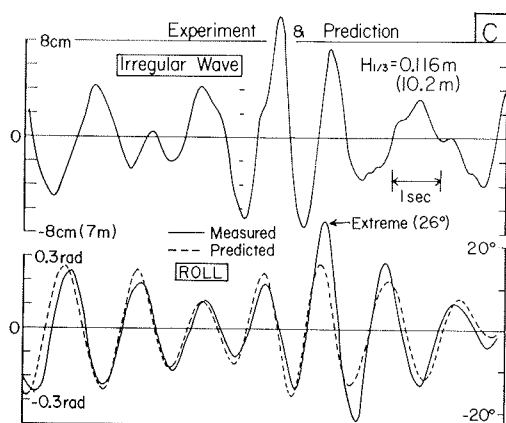


Fig.24 Prediction in high irregular waves using the results of irregular experiment.

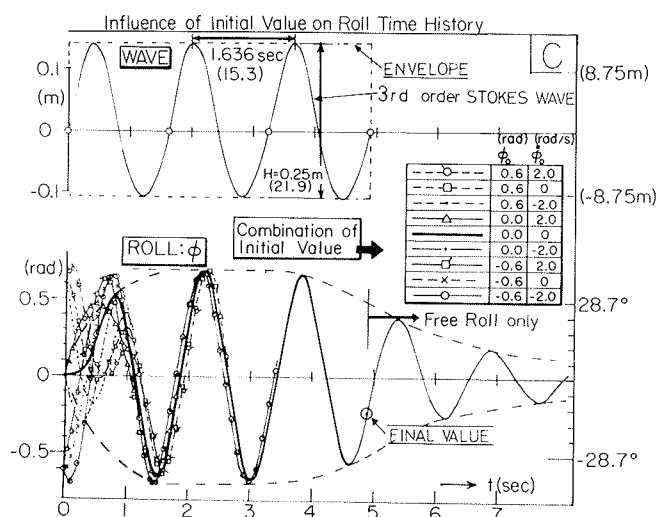


Fig.26 Numerical simulation of initial value influence on max' roll angle.

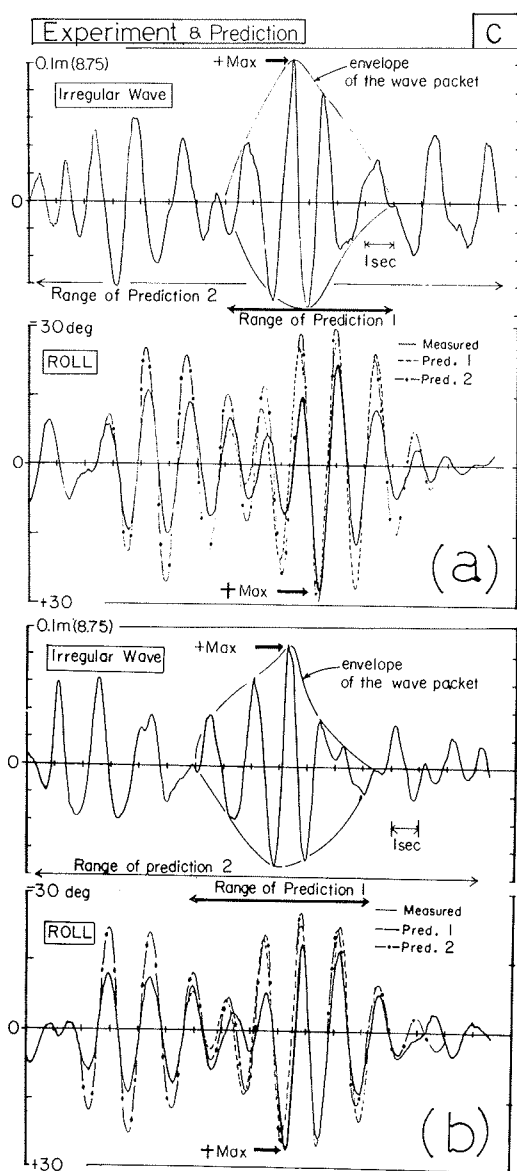


Fig.25(a),(b) Prediction of roll in high irregular waves using the results of transient experiment.

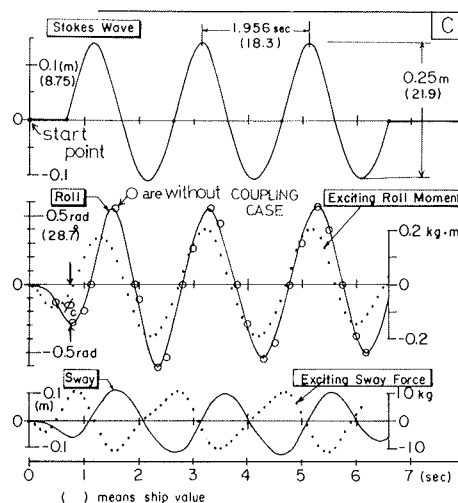


Fig.27 Numerical simulation of roll-sway coupling transient motion.

## Discussion

N. Toki (Nagasaki Experimental Tank, MHI, Japan)

The authors should be congratulated for the presentation of this interesting study. I would like to ask one question about the authors' conclusion No.12. Although the meaning of this conclusion is not clear to me, I understand it as follows: Given the value of maximum wave height of an individual wave to which ships have to expect to encounter during a long time of operation, the authors intend to use TWW with the maximum wave height to obtain non-linear rolling response. And the authors claim that the obtained response would be a good prediction of extreme rolling.

If my understanding of their conclusion is correct, I have to express a controversial opinion about it. As shown in Fig. 3 of the present paper, time histories of irregular wave in individual wave packets are quite different each other. And a response to a wave packet with higher elevation is not necessarily larger than that with lower elevation, as observed in comparison of wave packets A and D in Fig. 3. Therefore, we have to study more in detail about the variety of extreme wave shapes contained within a long record of irregular wave.

Last year, I presented a short paper to JTTC meeting concerning to the numerical model to get the variety of extreme wave shapes. The result of further study will be published as a University of Michigan report.\* In the paper, I created a large number of samples of individual waves with 1/10000 highest mean wave elevation contained within a very long record of irregular wave. Fig. A shows the histogram of crest front steepness of individual waves obtained by the examination of the samples. This figure clearly shows how various extreme wave shapes are. And, accordingly, we can imagine how various ship responses are. It is clear that we cannot define extreme wave condition by using only the value of individual wave height.

Any comments by the authors would be highly appreciated.

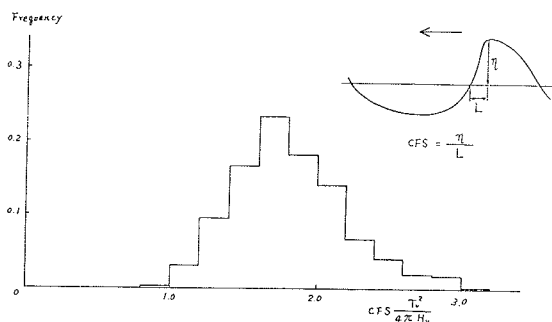


Fig. A Histogram of Crest Front Steepness for Individual Waves with 1/10000 Highest Mean Elevation

\* "On the Generation of Irregular Waves to be Used for the Design Studies of Floating Structure" The Univ. of Michigan Report

### Author's Reply

The author would thank Mr. N. Toki for his essential discussion. He pointed out that we must take in not only the wave height but also the variety of individual wave period of the wave packet.

Of course it is true in general sense, so we used the word "severe wave" and not "high wave" carefully. Because the word "severe" contains the meanings of the effect of period or frequency.

But about the rolling motion, it is well known generally, that rolling is a narrow banded phenomena, and so, it is sensitive only to its natural frequency and not to the local shape of wave. So it is proper considering the extreme roll occurs in high waves with the frequency near resonance. This fact is seen in the fig. 3.

From that reason, we used irregular waves which peak frequency of the spectrum is designed to coincide with roll natural frequency, as mentioned in our paper. (see Fig. 4). And as for Transient Water wave, the central frequency of the Fourier spectrum is designed to coincide with roll natural frequency, as shown in Fig. 5. Because in such a T.W.W., the apparent frequency of concentration becomes roll natural frequency. This central frequency represents the so called carrier wave.

As mentioned above, we only notice on the roll natural frequency, so the frequency or the period is not included as a parameter.

But when we notice the phenomena like shipping water, the shorter waves than that corresponds to roll natural frequency become more important (see Fig. 7 (b)), or local wave shape like wave breaking becomes more important, so the notice of the discussor should be considered.

V. Ankudinov (Hydronautics, Inc., USA)

It seems to be the complicated transient roll motion cannot be described by the simplistic equations with the constant coefficients.

Disagreement of the calculations with the transient test data seems to confirm that the equations should be written in the convolution integral form especially for the exciting forces.

It should be realized that the equation of motion in the convolution form will lose the traditional interpretation of their coefficients (added mass, damping, etc.), and physical interpretation of the equation will become somewhat difficult.

#### Author's Reply

Thank you for Dr. V. Ankudinov for his important discussion.

The linear equation of motion with coefficients containing frequency  $\omega$  as parameter, is generally expressed as an integro-differential equation.

But here, as mentioned in section 4.1, we approximated nonlinear damping by the polynomials of  $\omega$  like equation (7), so its coefficients are expressed as constant apparently. But  $\omega$  based changes are evaluated as a whole. And this point is further described in appendix 1.

We think that above mentioned approximation resulted the capability of estimating the large amplitude transient rolling by using such a simplified equation of motion.

It is noticed that, after calculating exiting roll moment time history  $m(t)$  by the convolution integrals, the nonlinear equation of motion is solved by so called Runge-Kutta-Gill method.

But between  $m(t)$  and  $h(t)$ , there exists nonlinearity according to wave height, so we include  $H_{\max}$  as a parameter in the transfer function  $M(\omega)/H(\omega)$ . So it should be noticed that the impulse response function of  $m(t)$  contains  $H_{\max}$  as a parameter, and these are no longer linear expressions in precise meanings.

This procedure is of course for the purpose of convenience for nonlinear phenom-

ena, but judging from the estimated result, this method is suitable concerning the extreme maximum roll amplitude.

S. Kastner (College of Engineering Bremen, FRG)

Let me congratulate you for this extremely good paper on combining experimental work with simulation studies.

Are you considering to extend your method to other headings than beam seas, i.e. following and quartering seas in particular?

#### Author's Reply

The author would like to thank for Prof. S. Kastner's comment. The rolling motion becomes large not only in beam seas but in quartering seas, so, as the discussor pointed out, the consistent examination to other headings than beam seas is also very important.

So, we want to make such examinations, but the experiment seems difficult in our towing tank even in zero forward case, because of wave reflection exist from the side wall.

As urgent problem, we consider the examination of nonlinear sway-roll coupling motion which estimation was inadequate in this paper (section 5.3), because of nonlinearity of sway-roll coupling term.

# STABILITY EXPERIMENTS IN THE FOLLOWING SEA WITH SHIP SPEED - AN UTILIZATION OF CIRCULATING WATER CHANNEL -

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## ABSTRACT

Making use of the stationary waves in a circulating water channel, the stability experiments in the following sea with ship speed were planned to perform.

In the former part of this report, the stationary waves generated by the wave-making board and their characteristics are investigated. Applying these results, the desired waves can be generated by setting the wave making parameters adequately.

In the later part of this report, the stability experiments of a trawler are discussed. They are carried out in the four cases, that is the cases with - and without ship speed in still water and the cases that the ship is in the wave crest and in the wave trough. The stability moments were measured to the extent of large angle of inclination in each case. Analysing the experimental data, the GZ-curves are drawn in Fig. 14 and Fig.15 and it is seen from the figure the ship in the wave crest has the lower GZ-curve among others.

In the appendixes, formulas of the linear wave theory and their usage are mentioned briefly.

Through the experiments it is verified that the stationary waves in a circulating water channel are very useful and they have many possibilities to apply other fields of research.

## NOMENCLATURE

B	Maximum molded breadth of load water line
BM	Transverse metacentric radius

C	Stream velocity
D	Molded depth
$d_a$	Draft at AP
$d_f$	Draft at FP
$d_{\Sigma}$	Draft of amidships
$Fn$	Froude number = $C/\sqrt{gh}$ or $C/\sqrt{g\lambda}$
GM	Transverse metacentric height
GZ	Righting arm
g	Acceleration due to gravity
$H_1$	Wave height measured from first crest to second trough
$H_2$	Wave height measured from second crest to third trough
h	Water depth
L	Length between perpendiculars
lw	Wetted chord length of board
P	Integral disturbance pressure applied to the surface of stream (Theory)
t	Trim
W	Volume of displacement
x,y	Horizontal and vertical ordinates
$\eta(x,y)$	Vertical displacement of free surface measured from $y = 0$



$\theta$	Incident angle of board
$\kappa$	Wave number = $g/C^2$
$\lambda$	Wave length measured from first crest to second crest
$\rho$	Density of water
$\phi(x,y)$	Velocity potential
$\phi(x,y)$	Velocity potential which corresponds to the disturbance caused by the application of the pressure

## 1. INTRODUCTION

The stationary waves generated in a circulating water channel provides a powerful tool for the experimental ship-hydrodynamics.

The first purpose of this research is to present data of such waves generated by a wavemaking board in a small circulating water channel.

The method to generate waves, scratching the water surface by a board was developed by N. Hogben [1], who named it "wavedozer" after bulldozer because of the similar operations. Since that time, the wavedozer has been used by many investigators in the towing tanks. The wavedozers used by such researchers were not the same and the differences seemed to be mainly in the incident angle of the wavemaking board and in the under water length of the board at zero speed. For example, the wavemaking board used by O. Grim [2] and M. Hamamoto [3] were set upright to the water surface and had a small gap between the board and the water surface before running.

In this study a wavemaking board was also tried to use and it was similar to the wavedozer but the working condition was different from that of the original.

In section two, the wavemaking experiments are reported, in which the relations between the parameters of wavemaking conditions and the generated wave profiles are described in detail. The experimental results are also compared with those of theoretical calculations.

The second purpose of this research is to carry out the stability experiments in the above mentioned waves without worrying about the limitation of measuring time as the case on a towing carriage.

In the stability experiments an ordinary pantograph type of stability balance was used to measure restoring moments, side forces, trim and sinkage. The model ship used was a trawler and her stabilities were investigated in the four cases; in still water with - and without forward speed, in the wave crest and wave trough amidship with forward speed. The detail of the experiments and the results are shown in section three together with the calculated results in various ways.

## 2. WAVEMAKING IN A CIRCULATING WATER CHANNEL

In the present section the stationary waves generated in a circulating water channel are discussed. The waves obtained thus will be available widely, because model ship experiments in the waves require a very long time of measurement.

There may be several ideas to make stationary waves in a streaming water; e.g. blowing compressed air directly on stream surface or setting a suitable obstacle in stream. The former method is preferable to keep the stream velocity undisturbed. However, it was found rather difficult to make large amplitude of waves without spreading water sprays like a running air cushion vehicle.

Having tried several ideas, a simple wavemaking board on the stream was adopted. The board was a wooden flat plate having sharp lower edge and it was set with a certain angle of inclination to the flow of circulating water channel as shown in Fig. 1. This is a similar device to the "wavedozer" originated by N. Hogben who used in the towing tank.

### 2.1 Circulating Water Channel and Wave-making Board

The series of experiments were performed in the circulating water channel of Hiroshima Mercantile Marine College. The length of experimental section of the water channel is 3.0 m, its width is 1.2 m, depth is 1.24 m and the maximum stream velocity is about 2 m/sec. The water depth is 0.8 m and it is kept constant automatically. The stream velocities fluctuate a little according to time and position and the fluctuation is about  $\pm 4\%$  of the reference velocity. The reference velocity was measured by a pitot tube in the middle portion of experimental section and the pitot tube was set at 0.3 m below the water surface. The reference velocity did not always represent the mean velocity of

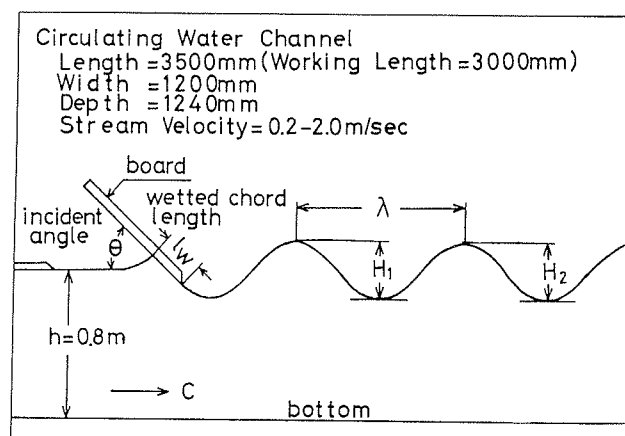


Fig. 1 Circulating water channel and generated waves

the stream.

The wavemaking board had the size of 1.2 m length, 0.3 m width and 27mm thickness and the board length was the same size as the width of the water channel so as to keep no clearance between side wall of the channel and the board. The lower edge of the board had a sharp edge of  $45^\circ$  angle. The board was set up at the 0.15 m down stream from the starting point of experimental section and its incident angle ( $\theta$ ), its wetted chord length ( $lw$ ) could be adjusted arbitrarily. The definition of  $\theta$  and  $lw$  are also shown in Fig. 1 together with the wave height  $H_1$ ,  $H_2$  and the wave length  $\lambda$ .

## 2.2 Wavemaking Experiments

The wavemaking experiments were mainly carried out in the range of wave profiles being normal, not distorted and not broken. The size of waves which could generate in the circulating water channel were approximately from 0.1 m to 3 m in length and from a few millimeters to 0.2 m in height. The wave profiles were measured by the scales in the side windows of water channel and at the same times photo records were taken.

The generated wave profile were generally very good and stable as shown in Fig. 2, but they became unstable when wave crest broke. Then the profiles changed its features considerably according to fluctuation of either stream velocity or a slight change of water depth which is automatically adjusted.

According to the linear theory of waves, the wave profiles made by wavemaking board depend on the depth of water  $h$ , the stream velocity  $C$ , the board incident angle  $\theta$  and the board wetted chord length  $lw$ . In the above four parameters,  $h$  was 0.8 m and kept constant through all the experiments. As the lengths of almost waves for practical use in the water channel, were less than twice of the water depth  $h$ , they were sufficient to regard as the waves of infinite water depth.

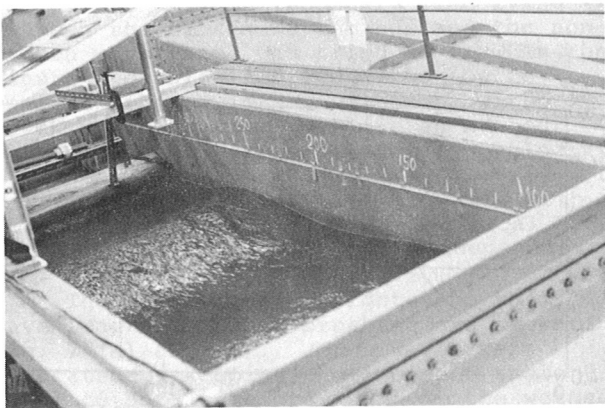


Fig. 2 Wavemaking board and generated waves

The inner structure of the waves was roughly surveyed by measuring the x-component of orbital velocities.

All of the experimental results were compared with the calculated results according to the two-dimensional linear theory of wave which is described in Appendix(A) of this paper. In the process of above comparison, two difficulties were arisen. The one was how to establish the relation between the disturbance pressure

$P$  applied to the surface of stream in the theory and the experimental wave parameters  $C$ ,  $\theta$  and  $lw$ . After several trials, a very simple relation was established as explained in Appendix(B) of this paper. The other difficulty was to determine the uniform stream velocity  $C$  accurately from the measurement. However, in the analysis of experimental results, the reference velocity was used instead of the true uniform velocity for convenience' sake.

In the following, the three cases of wavemaking experiments are described in detail.

### a) The Case Varying Wetted Chord Length $lw$

In the first case, the wavemaking effects by varying the wetted chord length  $lw$  were investigated, though the velocity of stream  $C$  (1.27 m/sec) and the incident angle  $\theta$  ( $38^\circ$ ) were kept constant. The experimental results are shown in Fig. 3. The wave heights and the wave lengths versus the wetted chord lengths are plotted in Fig. 3 (a) and (b) respectively. Fig. 3 (c) shows a typical wave profile in the present cases.

As the wetted chord length  $lw$  fluctuated violently, the correct measurement was very difficult and spent fairly a long time.

The breaking of the first wave crest was observed when  $lw$  was larger than 6.3 cm in this case. As seen in Fig.3 (c), the first wave height  $H_1$  is larger than the second one  $H_2$  in general. In Fig. 3 (b) the wave length decreases according that the wetted chord length increases.

It may be explained as follows; the increase of wetted chord length inevitably accompanies the decrease of stream velocity owing to the severe scratching of stream surface, and the decrease of stream velocity turns to the decrease of wave length according to the theoretical relation  $\lambda = 2\pi C^2/g$ .

Comparing the results of theoretical calculation with the experimental results in Fig. 3, both are in close agreement generally, except the slight difference in wave lengths as shown in Fig. 3 (b). The discrepancies seem to come from the pressure model, described in Appendix(B), in which the effects of wavemaking parameters  $lw$ ,  $\theta$ ,  $C$  to the pressure  $P$  may not be taken into account adequately.

As no remarkable differences of the calculated results are obtained between the

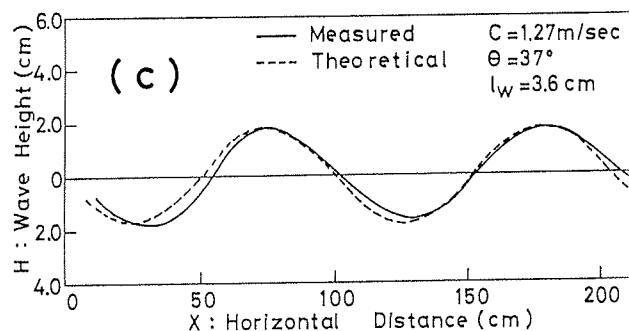
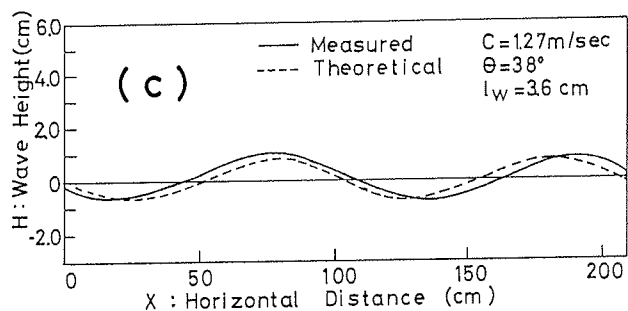
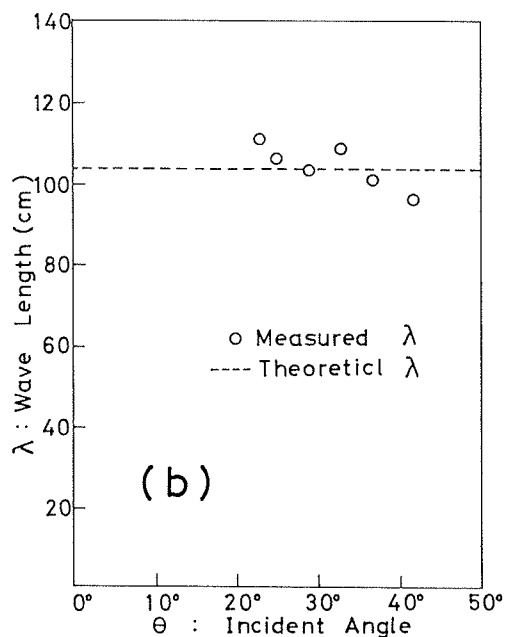
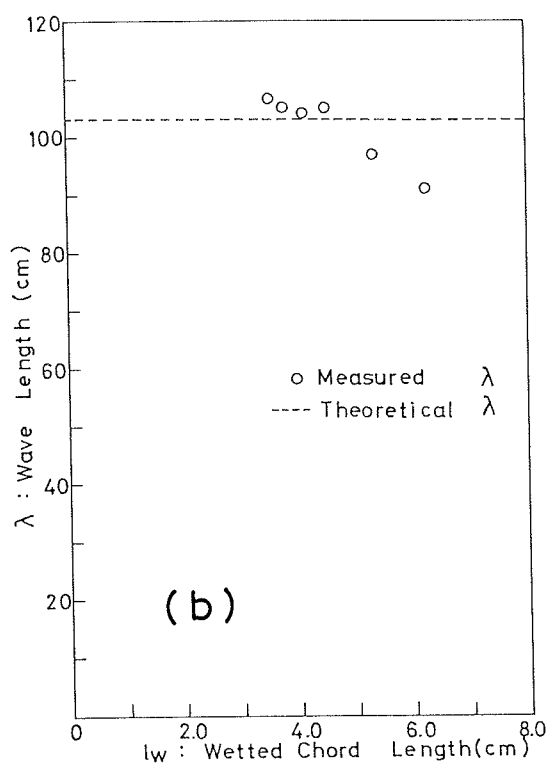
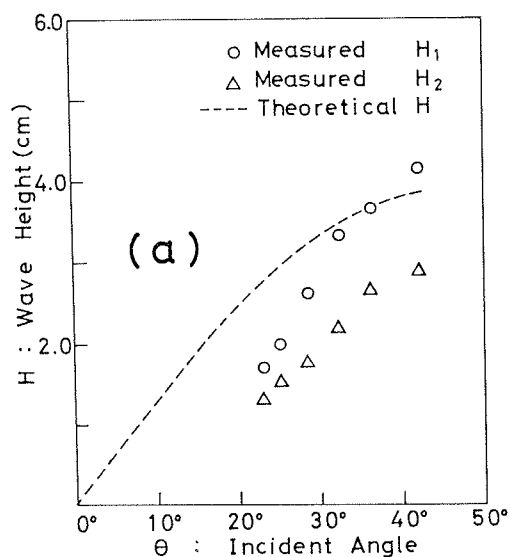
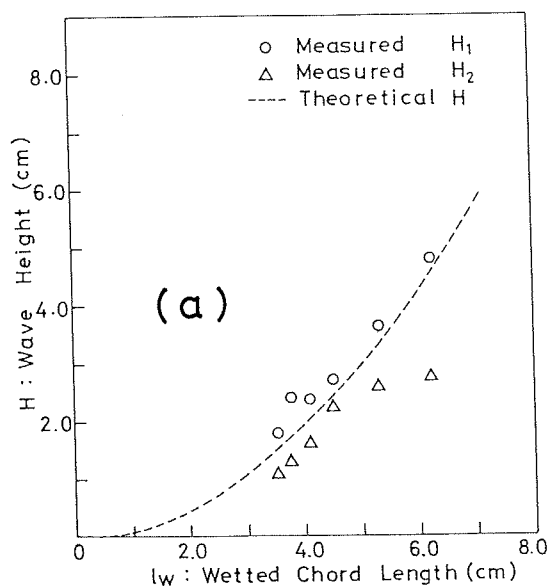


Fig. 3 The case varying wetted chord length  $l_w$

Fig. 4 The case varying incident angle  $\theta$

case of infinite water depth and the case finite, only the results of infinite case will be shown afterwards.

#### b) The Case Varying Incident Angle $\theta$

In the second case, the effects of incident angle  $\theta$  on the wavemaking were inquired under the condition of constant velocity  $C$  (1.27 m/sec) and constant wetted chord length  $l_w$  (5.6 cm). The wave heights and the wave lengths obtained are plotted in Fig 4. (a) and (b) versus the incident angles. The general tendencies are almost the same as the case of varying the wetted chord length.

The first wave crest began to break at  $44^\circ$  in the incident angle. An example of wave profile in this case is shown in Fig. 4 (c).

#### c) The Case Varying Stream Velocity $C$

In the third case, the stream velocity  $C$  was varied and its effects on wave profiles were investigated. Being the incident angle  $\theta$  kept constant at  $35^\circ$  and varying the stream velocity  $C$ , the wetted chord length  $l_w$  is naturally changed by  $C$  and it is shown in Fig. 5 (b) by square marks. From the figure, a linear relation between  $C$  and  $l_w$  is found and the relation is applied to theoretical calculations.

The wave heights, wave lengths and an example of wave profiles in this case are shown in Fig. 5 (a), (b) and (c) respectively.

Comparatively large waves, having non-breaking crest, were often used in the model ship experiments, and to generate such waves, the critical relations between wavemaking parameters were investigated. All the experimental data in various conditions were plotted in Fig. 6 and the discrimination of broken wave cases from non-broken ones were made by the solid marks in the figure. The stream velocities in the experiments were within the range of  $0.8 \sim 1.7$  m/sec and the drafts along the wavemaking board at zero stream velocity were  $1 \sim 8$  cm. The boundary line lying between the broken wave range and the non-broken wave range was tried to draw in the figure and this line suggested how to generate the maximum wave height without breaking. The boundary line as stated above do not depend on the stream velocity  $C$ . The reason is that the wetted chord length  $l_w$  and the wave length  $\lambda$  are both proportional to  $C^2$ , then the critical values of  $l_w/\lambda$  is apparently independent on stream velocity.

The maximum wave heights and their wave lengths without breaking are shown in Fig. 7 (a) and (b). These waves were generated by keeping the incident angle  $\theta$  equal to  $35^\circ$  as before, however the wetted chord length  $l_w$  were carefully adjusted in order to get the highest wave at every stream velocity.

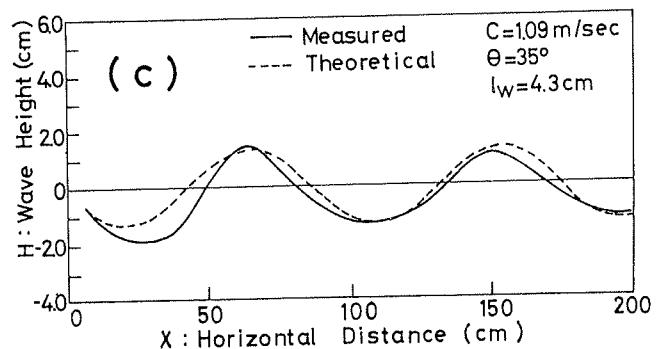
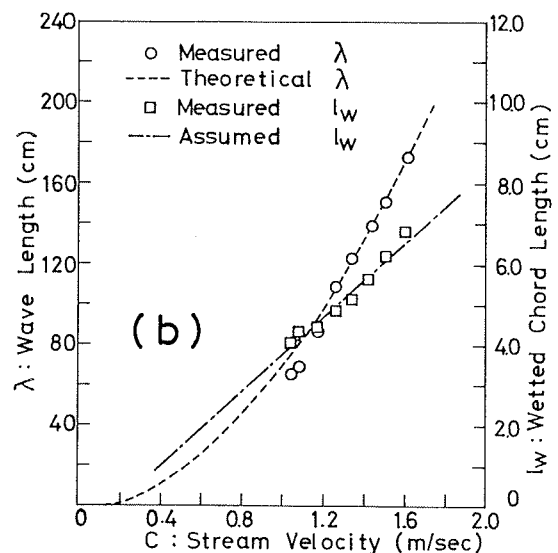
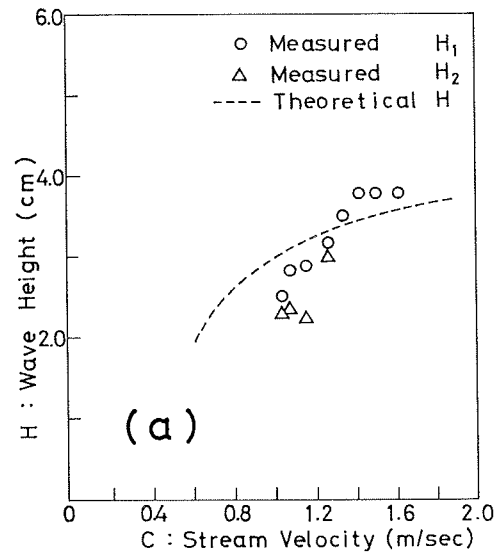


Fig. 5 The case varying stream velocity  $C$  ( $\theta = 35^\circ$  const.  $l_w$ : changed naturally)

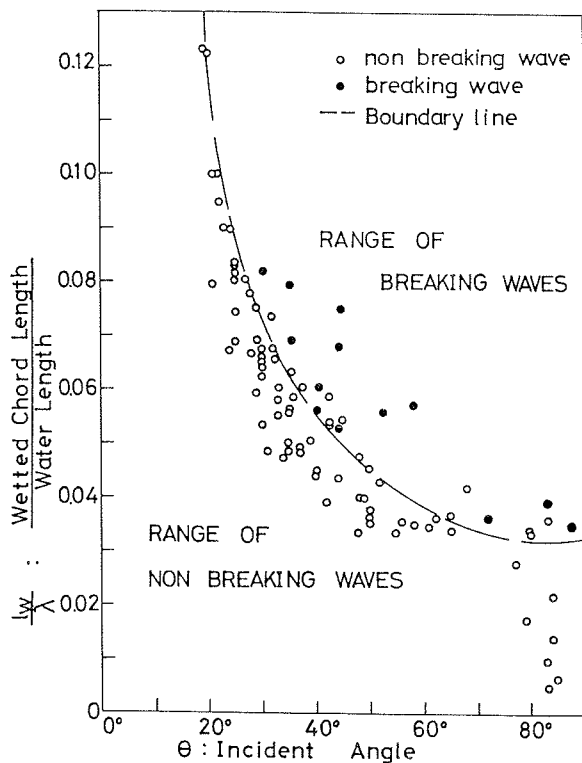


Fig. 6 Range of wave breaking

Let us give attention to the differences between Fig. 5 and Fig. 7. In the case of Fig. 5 the wetted chord length  $lw$  was changed naturally according to the stream velocity  $C$ . On the other hand in the case of Fig. 7 the  $lw$  was adjusted to produce the highest wave at each stream velocity  $C$ , though the other conditions were the same. In Fig. 7 the symbol of double circles  $\odot$  indicate the waves used in the stability experiments to be mentioned later.

### 2.3 Velocity Measurements in the Waves

The distribution of  $x$  - component orbital velocities were measured and the inner structures of waves were roughly compared with the theoretical one. The velocities were measured by a pitot tube in the center vertical plane of the water channel.

In Fig. 8 and Fig. 9, the measured and the calculated distributions of orbital velocities are shown respectively. The wave measured now was generated in the conditions of  $C = 1.27$  m/sec,  $\theta = 41^\circ$  and  $lw = 5.4$  cm and it was 3.6 cm in height and 1.02 m in length.

The orbital velocity at a certain point was obtained by subtracting the stream velocity without wave from the velocity in the wave at the same point and not at the same time.

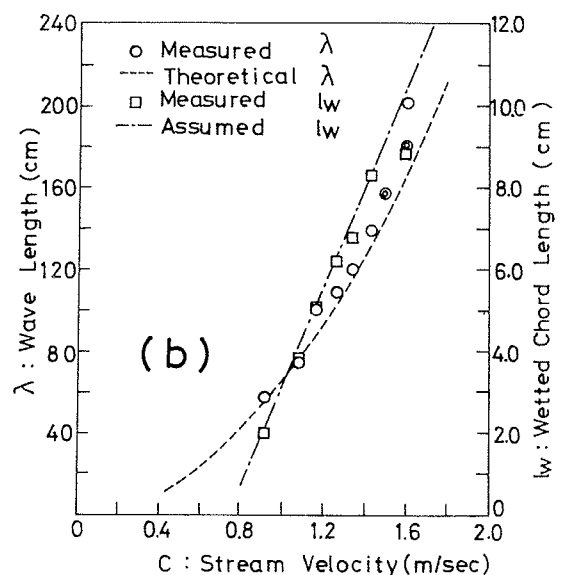
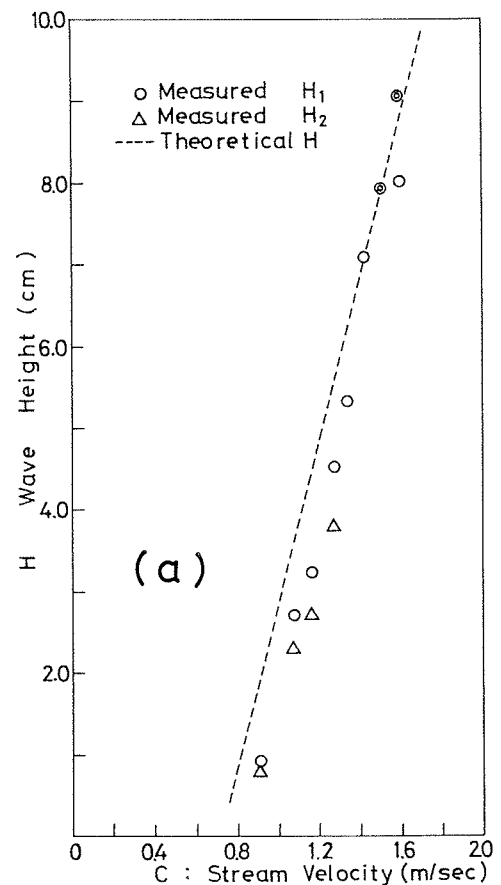


Fig. 7 The case varying stream velocity  $C$  ( $\theta = 35^\circ$  const.  $lw$  : adjusted)

Comparing the results of experiments in Fig. 8 and those of theoretical calculations in Fig. 9, the generated waves have almost the same inner structure as that of the theoretical waves except in the close vicinity of wavemaking board or the points of deviated velocities from the reference

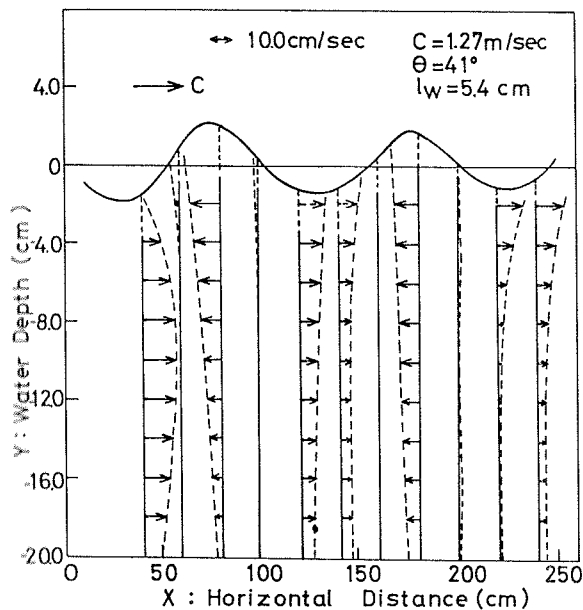


Fig. 8 Distribution of stream velocity in waves (experiment)

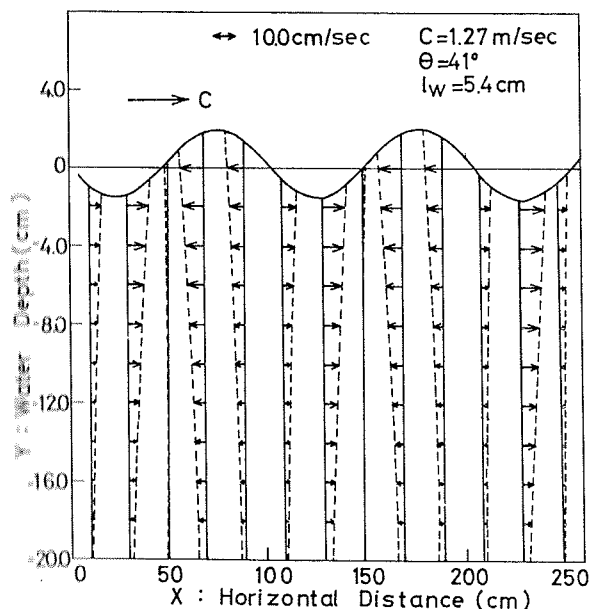


Fig. 9 Distribution of stream velocity in waves (theory)

velocity and it was concluded that the generated waves were able enough to use in a stability experiments in the following sea.

### 3. STABILITY EXPERIMENTS IN THE FOLLOWING SEA WITH SHIP SPEED

#### 3.1 Model Ship and Instruments

The model ship is a trawler and her reduced scale is one twelfth. Her body plan and the principal particulars are shown in Fig. 10 and Table 1 respectively. The model has a remarkable large cruiser stern which contributes to her stability considerable. The water-tight deck was planked on the bulwark top, for the convenience of measuring stability at a large angle of inclination in the running condition. This enabled to reduce the

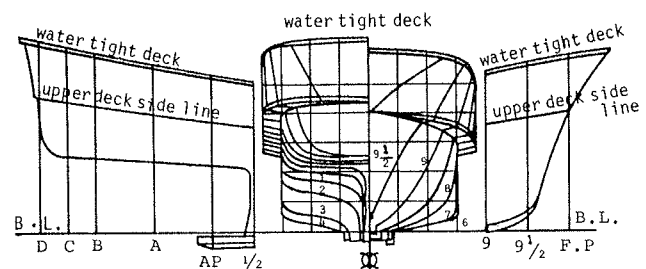


Fig. 10 Body plan of trawler model

Principal Particulars

	SHIP	MODEL
L <sub>OA</sub>	19.15	159.58
L <sub>pp</sub>	14.40	120.00
B <sub>pp</sub>	3.05	25.44
D	1.38	11.50
Δ	28.02 (t)	15.82 (kg)
d <sub>f</sub>	0.41	3.43
d <sub>a</sub>	1.38	11.50
d <sub>trim</sub>	0.90	7.47
GM		8.07
KM		3.28
KG		14.96
KG		11.68
KG		9.31 (Aft)
Scale	12	: 1
Length Unit	( m )	( cm )

Table 1. Principal Particulars of trawler model

unfavorable phenomena that the immersed part of bulwark produced complex side force and moment.

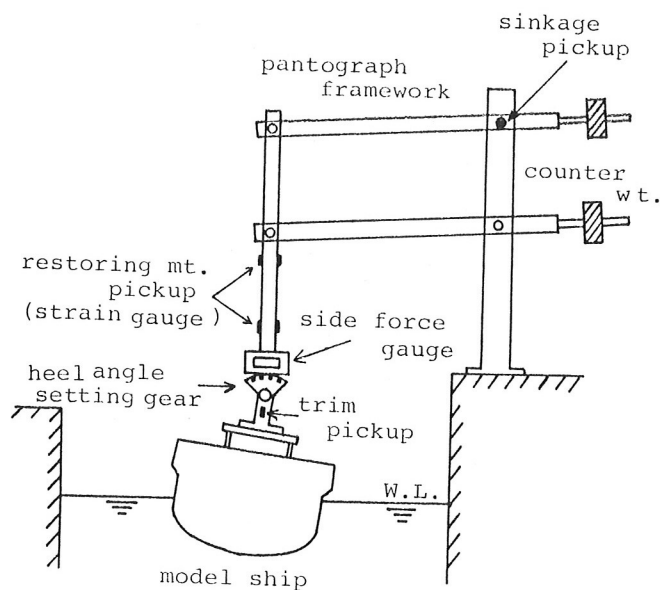


Fig. 11 Measuring system of stability experiment

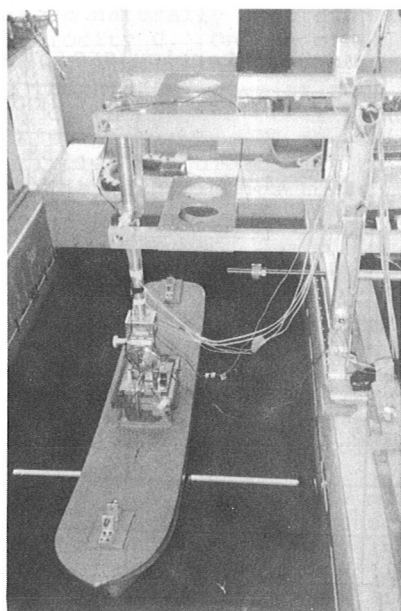


Fig. 12 Trawler model and measuring system in circulating water channel

The model ship was connected to the stability balance which was a framework of pantograph type as seen in Fig. 11 and Fig. 12 and it worked as a balancing holder of model in the stability experiments. Setting a heel angle, the model ship was permitted to trim, to heave or sink and a little to sway, even though it was not permitted to yaw and to surge.

The restoring moment was measured by the strain gauges attached to the vertical rod of framework and the side force was picked up by the differential transformer built in the square block gauge. (see Fig. 13) The trim angle in the center vertical plane of hull and the heave or sink were measured by the potentiometers and all the measured values were recorded by a pen-oscillograph simultaneously.

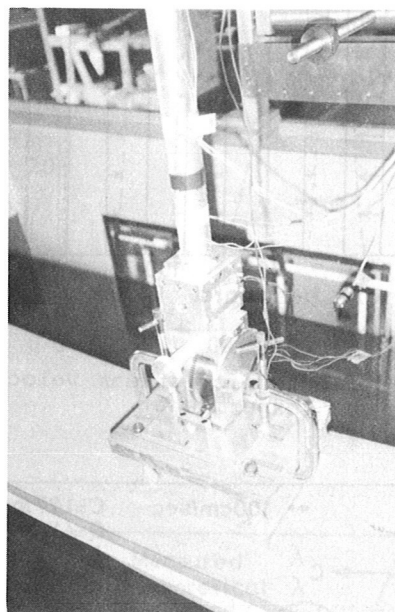


Fig. 13 Main part of measuring system

### 3.2 Stability Experiments and Results

Repeating the inclining tests, the GM of model ship was determined to be 3.28 cm.

The stability experiments of the model ship were carried out in the following cases;

- (1) zero speed in still water,
- (2) running in still water ( $F_n = 0.45$ )
- (3) running in following sea, wave crest is amidship ( $F_n = 0.57$ ,  $H = 0.09$  m,  $\lambda = 1.80$  m,  $H/\lambda = 0.050$ ,  $\lambda/L = 1.50$ )
- (4) running in following sea, wave trough is amidship ( $F_n = 0.53$ ,  $H = 0.078$  m,  $\lambda = 1.59$  m,  $H/\lambda = 0.049$ ,  $\lambda/L = 1.33$ )

In all cases, measurement were made in the inclinations to both sides (port and starboard), and took an average of measured values. The experimental conditions



differed a little in each case because to adjust the relative positions of waves and ship were difficult in the experimental section of water channel, but it brought no essential difficulties.

The stability curves are indicated in Fig. 14 and Fig. 15. In the cases of following sea with forward speed, the ship in wave trough has the largest GZ, to the contrary, in wave crest has the smallest one and the case of without wave is between them. This mutual relations in GZ curves are generally valid until the hull inclinations reach the angle of maximum GZ. The wave used and the stability experiment in the wave crest are shown in Fig. 2 and Fig. 16. In Fig. 16, the complex superposing of oncoming waves, bow and stern waves can be seen. In the running conditions, every GZ curve goes down to zero quickly after passing its maximum and when the model ship loses her stability, she seems to be merely in drifting. The GZ curve of running condition is slightly higher than that of stopped and it may be caused by the ship-generated waves.

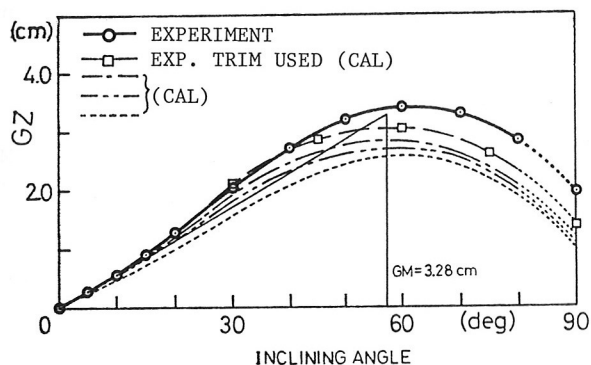


Fig. 14 Stabilities in still water (exp. & cal.)

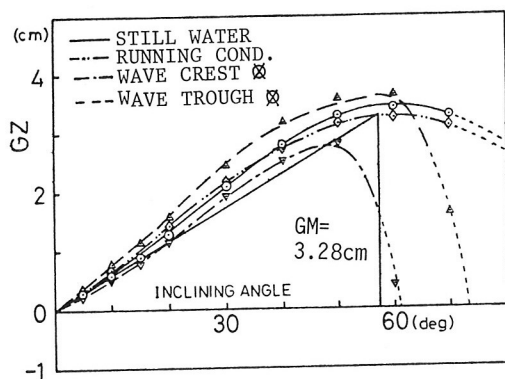


Fig. 15 Stabilities in waves (exp.)

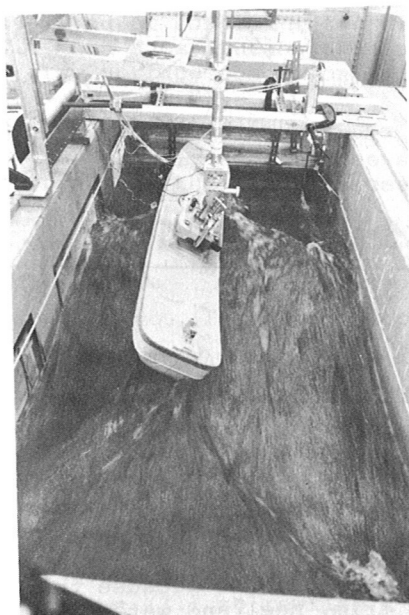


Fig. 16 Stability experiment, (wave crest amidship)

In running conditions of the model ship, it is considered that some of the experimental results are not realistic in the range, over some large angle of inclination. This is also seen in the records of trims (Fig. 17) and side forces (Fig. 18) and in the cases of real ship, the phenomena would be quite different.

As the stream of circulating water channel is not strictly parallel to its side walls, the model ship should be set at zero incident angle hydrodynamically. If not, the restoring moments, the trim angles and side forces measured, would be quite different in port and starboard side.

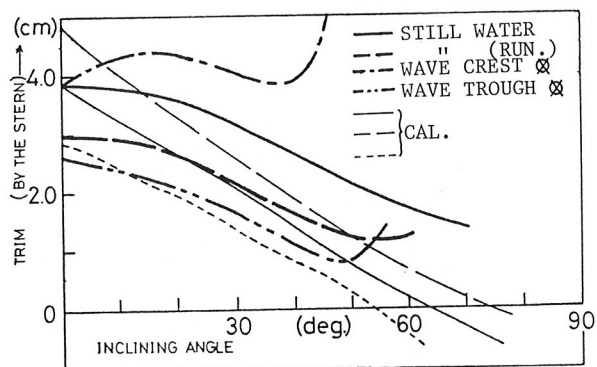


Fig. 17 Trims at heeling condition (exp. & cal.)



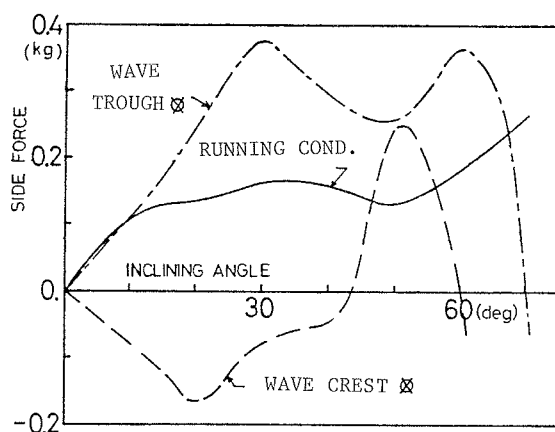


Fig. 18 Side forces at heeling conditons (exp.)

The bodily sinkage of the model ship could not be obtained because the corrections of trim, heel and water level are complex and the amount of corrections was not so small.

The calculated stabilities in still water are added in Fig. 14 for comparison with experimental results. The calculations were tried in several ways; by integrator, and by computer, and it was found that the discrepancies between each results were unexpectedly large, especially in the trim free conditions.

After some examinations, a computer calculating method was adopted in which Simpson rule was applied to the twenty-six-divided sections of ship length.

At first the calculation was carried out in the trim free condition and comparing the result with that of experiment, the coincidence was not so satisfactory (Fig. 14). As this disagreement was considered coming from the difference of trim angle (Fig. 17), the calculation was tried to carry out with the experimental trim angle and a little better result was obtained, however the discrepancy was still remaining, and the reason could not be found by now. In Fig. 14 and Fig. 17, the effects of varying initial trim on the stability and on the proceeding trim calculated, are shown respectively.

Through these calculations, it was found that the effect of a large cruiser stern on the stability was considerably large and its effect also depended on the trimmed condition.

#### 4. CONCLUSION

The stationary waves in a ciuculating water channel are generated by a wavemaking board and the characteristic of wave profiles are found to depend on the board incident angle  $\theta$ , wetted chord length  $lw$  and the stream velocity  $C$ .

The relations between the wave profiles and the wavemaking parameters  $\theta$ ,  $lw$ ,  $C$  were investigated and shown in Fig. 3 ~ Fig. 7.

The experimental waves were compared with the theoretical waves and both were in good coincidence.

In Appendix (B), the evaluation of pressure  $P$  which was applied on the surface of stream in theoretical calculations was discussed.

Making use of the stationary waves as mentioned above, the stability experiments of a trawler model were performed and the main results were shown in Fig. 14 and Fig. 15 together with the calculated results.

From the figures, it was found that:

- i) The ship in wave crest has the less stability than others and in wave trough has the more.
- ii) The ship had almost the same stability in still water and in running conditions
- iii) Owing to the large cruiser stern, the effect of trim on the stability is considerable.

Through these investigations, the experimental techniques in the circulating water channel were developed a little more towards problems in ship stability.

#### ACKNOWLEDGEMENT

The authors are deeply grateful to Mr. H. Hinata and Mr. I. Nagao who assisted us and carried out the experiments. The authors also express their heartfelt thank to Prof. M. Hamamoto of Osaka University and to Prof. Y. Himeno of Osaka Prefectural University giving us the valuable data.

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# APPENDIX (A) FORMULAS IN LINEAR THEORY OF WAVES

Formulas of two-dimensional stationary waves in a uniform stream are introduced after A.S. Peters [4]. Take the co-ordinate system as in Fig. 19, and a constant pressure  $P$  is applied at the origin 0 on the free surface of stream, with a uniform velocity  $C$ . The water depth is  $h$ , and the elevation of free surface is denoted  $y = \eta(x)$ . The forces applied to the fluid are the surface force  $P$  and the gravity, and then the fluid motion can be assumed irrotational.

A velocity potential  $\phi(x, y)$  in steady state takes the form of

$$\phi(x, y) = \phi(x, y) - Cx, \quad (1)$$

where  $\phi(x, y)$  is a velocity potential caused by the disturbance pressure  $P$ . As  $\phi(x, y)$  is a harmonic function, Laplace equation is satisfied.

$$\phi_{xx} + \phi_{yy} = 0 \quad (2)$$

The free surface conditions take the forms of

$$\frac{P}{\rho} + g\eta - C\phi_x = 0 \quad \text{on } y = 0, \quad (3)$$

$$\phi_y + C\eta_x = 0 \quad " \quad (4)$$

where  $\rho$  is the density of water,  $g$  is the acceleration due to gravity.

To solve  $\phi$ , the surface elevation  $\eta$  in equation (4) may be assumed, so

$$\eta = -\frac{1}{C} \int_{-\infty}^x \phi_y(x, 0) dx \quad \text{on } y = 0, \quad (5)$$

Equation (5) is substituted into (3) and introduce a new function defined by the following equation is introduced;

$$\chi(x, y) = C\phi_x(x, y) + \frac{g}{C} \int_{-\infty}^x \phi_y(x, y) dx = \frac{P}{\rho}, \quad (6)$$

where  $\chi(x, y)$  satisfies Laplace equation.

The pressure  $P$  represented by Fourier integral is

$$\frac{P}{\pi} \int_0^{\infty} e^{uy} \cos ux \, du, \quad (7)$$

where  $P$  is concentrated in an infinitely narrow band of the surface at origin.

Differentiating  $\chi(x, y)$  with respect to  $x$  yields

$$\begin{aligned} \chi_x(x, y) &= C\phi_{xx}(x, y) + \frac{g}{C} \phi_y(x, y) \\ &= \frac{P}{\pi\rho} \int_0^{\infty} u e^{uy} \sin ux \, du. \end{aligned} \quad (8)$$

Rewriting (8) with (2), the following differential equation is obtained:

$$\phi_{yy} - \kappa\phi_y = \frac{P}{C\rho\pi} \int_0^{\infty} u e^{uy} \sin ux \, du, \quad (9)$$

where  $\kappa$  is  $g/C^2$ . This equation can be solved by using of complex integral. Solving it, and substituting the solution into the free surface condition,

$$\eta(x) = \frac{C}{g} \phi_x(x, 0), \quad (10)$$

we obtain the profile of the free surface in the down stream of the disturbance.

$$\frac{\pi g \rho}{\kappa P} y = -2\pi \sin \kappa x + \int_0^{\infty} \frac{u e^{ux}}{u^2 + \kappa^2} du \quad \text{on } x > 0, \quad (11)$$

The integral in eq. (11) can be represented as follows,

$$-C\kappa x \cos \kappa x + \left(\frac{1}{2}\pi - \text{Si}\kappa x\right) \sin \kappa x, \quad (12)$$

where

$$Ciu = -\int_u^{\infty} \frac{\cos u}{u} du, \quad \text{Si}u = \int_u^{\infty} \frac{\sin u}{u} du. \quad (13)$$

For the case of distributing a uniform pressure  $P$  on a finite band  $(a, b)$ , it is necessary in eq. (11) to rewrite  $x-\alpha$  for  $x$ , to replace  $P$  by  $P_0 \delta\alpha$ .

Integrating  $\eta(x)$  with respect to  $\alpha$  in the range of  $(a, b)$ , the wave profile is obtained in this case:

$$\begin{aligned} \eta(x) &= \frac{\kappa P_0}{\pi g \rho} \left[ \int_a^b \{-2\pi \sin \kappa(x-\alpha) \right. \\ &\quad \left. + \int_a^b \frac{u e^{u(x-u)}}{u^2 + \kappa^2} du\} d\alpha \right]. \end{aligned} \quad (14)$$

Lamb [5] discussed the same problem in a finite depth by introducing a dissipative force and formulated the wave profile in the form of

$$\eta(x) = \frac{P}{\pi P C^2} \left\{ \int_0^{\infty} \frac{\cos(xu/h)}{u \coth u - \frac{gh}{C^2}} du \right\}, \quad (15)$$

where  $h$  is a finite depth of water.

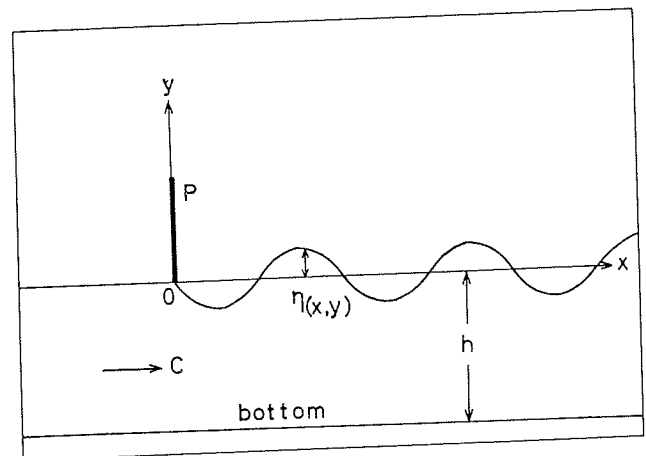


Fig. 19 Co-ordinate system of theoretical calculations

## APPENDIX(B) TENTATIVE MODEL OF PRESSURE P IN THEORETICAL CALCULATIONS

The action of wavemaking board on the surface of stream may be considered as a transducer, by which the stream velocity  $C$  is transformed to a fictitious disturbance pressure  $P$ , even if the pressure  $P$  and the velocity  $C$  has no relation physically.

However in the calculation of wave profile by linear theory, it is necessary to evaluate the pressure  $P$ . For the purpose of evaluating the pressure, a tentative model of pressure  $P$  was considered as shown in Fig. 20. The tentative model of pressure consisted of its intensity  $P_0 = \rho g l_w \sin \theta$  and its exerted interval  $l_w \cos \theta$ . The characteristics of pressure model are that the calculated wave heights would be the same if the amount of pressure  $P$  ( $\rho g l_w \sin \theta \times l_w \cos \theta$ ) is the same, and the fact do not depend on the form of pressure distribution. However, as seen in section 2, the calculated results by this pressure model are in good coincidence with experimental results and this suggests the tenta-

tive model is practically useful, however a more reasonable pressure model should be investigated in future.

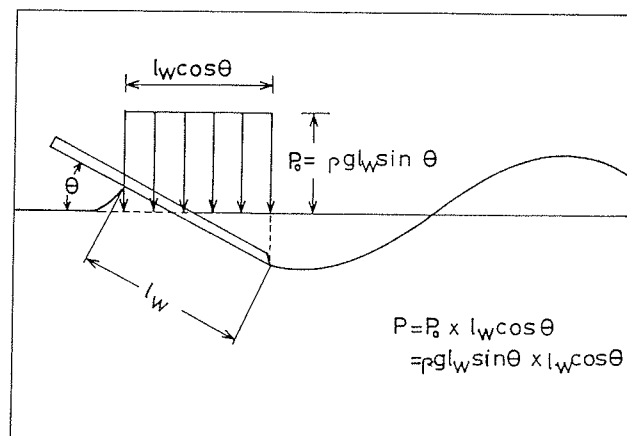


Fig. 20 Tentative model of pressure

## Discussion

M.R. Renilson (University of Glasgow, UK)

I have two quick points. The first one is concerning the waves from the wavedozer. I have also used a wavedozer in a CWC and am pleased to see that the present author has spent a lot of time measuring the wave as I did not have time. I also found that the first wave was higher than the rest and so I carried out my experiments 3 or 4 wavelengths from the wavedozer. (ref. A)

My other question is relating to the model used. Is it the same as that used by Professor Nomoto in an earlier paper to this conference? If so, would it be possible to compose the results presented in this paper with his results?

ref-(A) Renilson M.R., Driscoll, A.  
"Broaching - An Investigation into the loss of Directional Control in Severe Following Seas" RINA Spring Meetings 1981

S. Kastner (College of Engineering Bremen, FRG)

I refer to Fig. 15 in your papers: Did you compare your measured righting arms in wave crest and wave trough with hydrostatic calculations?

### Author's Reply

I would like to answer to Dr. Renilson for his kind comment and discussions.

(1) In the present cases, the first waves were measured higher than the second ones. The reasons can be considered as follows;

- i) In the case of near critical condition of wave breaking, the first wave is higher than the second, however, in the case of far from critical condition, the two waves have almost the same heights.
  - ii) There was a slight difference in the stream velocities between the position of the first wave and that of the second.
- (2) As you have pointed out, the model ship used was a "geosim" of Prof. Nomoto's and the "geosims" were widely used in Japan. Of course we compared our experimental results with those of other institute and it was found that after correcting GM values, they were in fairly good coincidence.

I would like to answer to Prof. Kastner.

We tried to calculate the righting arms in the wave crest and the wave trough by several different methods but the results were not satisfactory. It was found that in the present model, a little change of trim gave the severe effects to the calculation of righting arms, especially in a wave and some special scheme of calculation might be needed.

Lastly, I would like to express much thanks to Dr. Fujii (Mitsubishi Heavy Industries, Ltd., Japan) for his kind important comment after the session.

He pointed out that in Fig. 5(b), the author found a linear relation between  $l_w$  and  $c$  experimentally, however, in the explanation of Fig. 6 the author stated  $l_w$  and  $\lambda$  were both proportional to  $c^2$ .

According to Dr. Fujii's comment, the author must insert a word "partially" to the explanation of Fig. 5(b), that is "a partially linear relation between  $l_w$  and  $c$  was found" is the correct explanation.

*Session VIIIa*

## Shipping Water

*Chairmen*

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U.S. Coast Guard  
U.S.A.

Prof. Michio Nakato  
Hiroshima University  
Japan

## THE EFFECTS OF FREE WATER ON DECK ON THE MOTIONS AND STABILITY OF VESSELS

BRUCE H. ADEE\* AND IBRAHIM CAGLAYAN

\*University of Washington

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### ABSTRACT

A survey of the literature concerning the stability and motion response of vessels with low freeboard indicates that free water on the deck of these vessels may play a significant role in vessel capsizing.

To deal with this problem, existing mathematical models were extended to include additional nonlinear terms in the two-dimensional equations of motion. Shallow water equations are used to model the water on deck, and the forces and moments resulting from the free water enter the equations as external excitations. The equations are formulated in the time domain.

Extensive experiments are described and compared with the theoretical prediction. A rectangular tank was oscillated and water movement recorded as water depth, amplitude and frequency of oscillation were varied.

### INTRODUCTION

Vessel casualty statistics gathered in the United States over many years indicate that fishing and offshore supply vessels experience a large number of accidents. These vessels, which generally have low freeboard and high bulwarks, routinely experience conditions where water is trapped on deck. The entrapped water poses a stability problem and may have contributed to the capsizing of a number of these smaller vessels.

One carefully documented casualty investigation [1] into the capsizing of the Fishing Vessel PATTI-B at Ocean City Inlet, Maryland, U.S. on 9 May 1978 indicates the role water on deck may have played in the

loss of this vessel.

...the probable cause of accident was the capsizing of the PATTI-B due to the combined effects of water trapped on its deck, its being temporarily poised on a wave, and the overturning moment of the anchor line.

Figure 1 shows the static stability curves for the PATTI-B at the time of the accident and the role various factors may have played in the capsizing of this vessel.

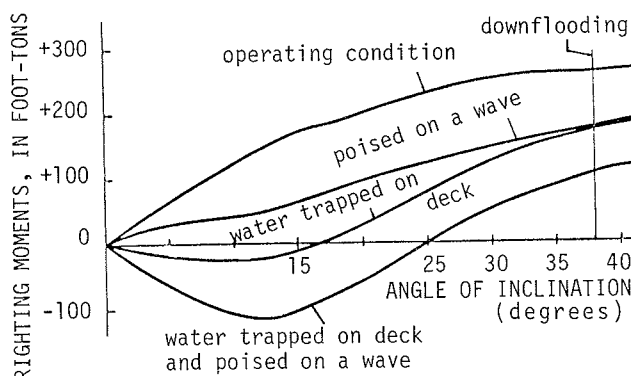


Fig. 1. Stat. stability curves for PATTI-B

A recent survey of literature related to the problem of water on deck [2] indicates that research emphasis has been placed on liquid sloshing in LNG tanks. The approaches used in analyzing the water on deck problem may be grouped into four categories: static, dynamic, probabilistic and hydrodynamic.

In this report the hydrodynamic approach first proposed by Dillingham [3]

for computing the motion of water on deck has been adopted. Experiments have been performed to verify the accuracy of this solution. The equation for roll motion has been written to include the effects of water on deck as well as nonlinear damping and restoring moments. A time domain solution is proposed, using Laplace techniques, which leads to a phase plane stability analysis.

#### MOTION OF WATER ON DECK

The motions of water on deck can be analyzed using the conservation of mass and momentum equations formulated by Stoker [4] for shallow water depth.

$$\frac{\partial v}{\partial t} + v \frac{\partial v}{\partial y} = -g \frac{\partial h}{\partial y} \quad (1a)$$

$$\frac{\partial h}{\partial t} + v \frac{\partial h}{\partial y} = -h \frac{\partial v}{\partial y} \quad (1b)$$

In these equations,  $h$  represents water depth at any point along the  $y$  axis. This axis is directed parallel to the undisturbed free surface, with  $v$  representing velocity parallel to the  $y$  axis and  $g$  the acceleration of gravity.

Dillingham [3] has modeled the motions of water on deck by writing Equations 1a and 1b in a moving reference frame.

$$\frac{\partial v}{\partial t} + v \frac{\partial v}{\partial y} = -a_z \frac{\partial h}{\partial y} + f \quad (2a)$$

$$\frac{\partial h}{\partial t} + v \frac{\partial h}{\partial y} = -h \frac{\partial v}{\partial y} \quad (2b)$$

where

$$f = -\ddot{\eta}_2 \cos \theta - \ddot{\eta}_3 \sin \theta + \omega^2 y + \dot{\omega} h_{fb} - g \sin \theta. \quad (2c)$$

Figure 2 illustrates the coordinate systems and terms used to formulate the equations. It should be noted that  $a_z$  denotes the component of acceleration perpendicular to the deck. Using the operator splitting technique described by Sod [4], Equations 2a and 2b are expressed as follows:

$$\frac{\partial v}{\partial t} + v \frac{\partial v}{\partial y} + a_z \frac{\partial h}{\partial y} = 0, \quad (3a)$$

$$\frac{\partial h}{\partial t} + v \frac{\partial h}{\partial y} + h \frac{\partial v}{\partial y} = 0, \quad (3b)$$

$$\frac{\partial v}{\partial t} = f. \quad (3c)$$

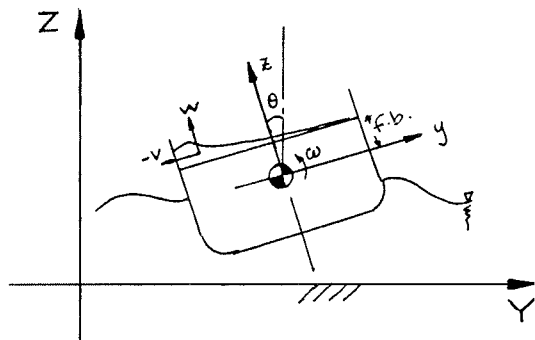


Figure 2. Coordinate systems.

An approximate solution which is described later is used to find "f" by solving Equations 3a and 3b. The final solution is reached by solving Equation 3c. Equations 3a and 3b constitute the Riemann or Dam Breaking problem, which was treated by Dillingham [3]. The solution of the Dam Breaking problem is advanced in time steps using the Random Choice Method.

The deck is divided into a grid of cells  $\Delta y$  wide; similarly, time is divided into increments  $\Delta t$  long. Equations 2a and 2b are subject to the following conditions:

$$\begin{aligned} v &= v_i^n \quad \left\{ \begin{array}{l} (i - 1/2)\Delta y < y < (i + 1/2)\Delta y \\ t = n\Delta t \end{array} \right. \quad (4) \\ h &= h_i^n \end{aligned}$$

At each time step the Riemann problem must be solved for the set of equations 3a and 3b after ensuring that the disturbance created in one cell does not propagate any farther than its boundaries. This is called the Courant-Friedrichs-Lewy condition, which requires

$$\frac{\Delta y}{2\Delta t} > |v| + c. \quad (5)$$

Here,  $c$  is the local celerity and  $v$  is the local velocity. By combining the solutions of the Riemann problem a solution at time  $(n + 1)\Delta t$  is obtained. This is an exact solution. From this exact solution, a piecewise solution is constructed. A random sampling procedure is used for this. If "r" is a random number between -0.5 and +0.5 and if

$$v(y, (n+1)\Delta t), h(y, (n+1)\Delta t) \quad (6)$$

is the exact solution to the Riemann problem, then the piecewise constant solution can be constructed as follows:

$$\begin{aligned} v_{i+1/2}^{n+1} &= v[(i+r)\Delta y, (n+1)\Delta t] \\ h_{i+1/2}^{n+1} &= h[(i+r)\Delta y, (n+1)\Delta t] \end{aligned} \quad (7)$$

The solution is evaluated alternately at grid cells  $i\Delta y$  and  $(i + 0.5)\Delta y$  where  $i = 0, \pm 1, \pm 2$ .

The solutions of Equations 2a and 2b are used to find the forces and moments created as a result of the sloshing motion.

To determine the validity of the theory and the method of solution a series of experiments was conducted. A special apparatus was designed and built, consisting of a driving mechanism which provided a sinusoidal oscillation to a tank of water. The apparatus is shown in Figure 3. A 1 HP variable speed AC motor was connected to a Scotch Yoke mechanism through a 1:15 gear reduction unit. The Scotch Yoke ensured sinusoidal motion. By adjusting the length of the input link of the yoke, the amplitude of the output motion could be changed.

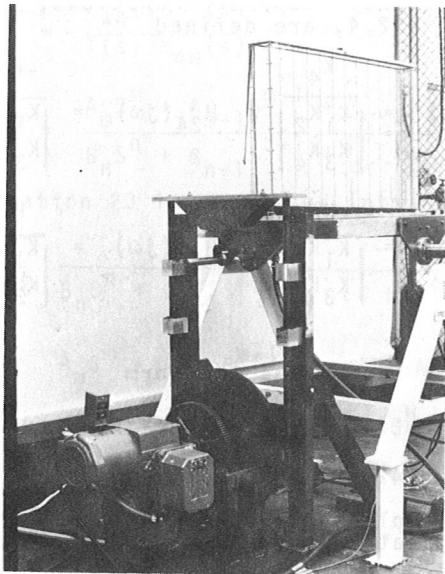


Figure 3. Apparatus.

A tank built of 0.5 inch thick plexiglass was mounted on a frame connected to the yoke. The tank dimensions were 3 inches wide by 24 inches deep by 36 inches long internally. The tank was covered to ensure that the quantity of water in it remained constant. The switch activating the motor was also connected to a time lapse movie camera capable of taking one frame every half second. The motion of the oscillating frame and the tank were recorded on a strip chart recorder. The recordings indicated that the motions were sinusoidal with no detectable transient period.

The matrix of experiments consisted of close to 200 runs with variations in the amplitude and frequency of oscillation and the depth of water in the tank. Amplitudes tested included 3.0, 5.0, 7.5, 9.5, 11.5 and 14.0 degrees. The frame was oscillated at 15, 20, 25, 30, 35, 40 and 45 rpm's, and the water depths for the tests were 0.5, 1.0, 2.0 and 3.0 inches.

Figure 4 shows a typical experiment.

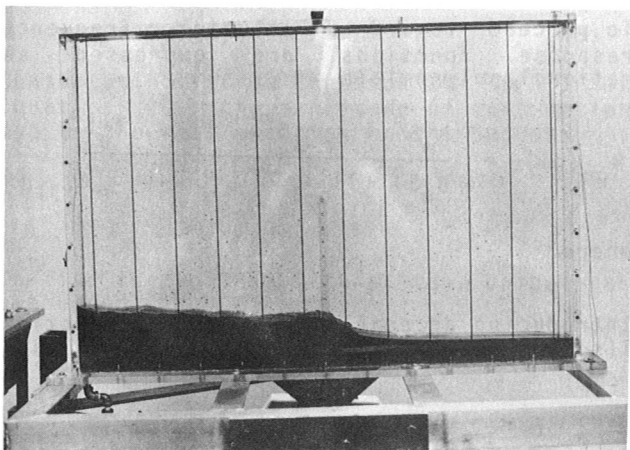


Figure 4. A Typical Experiment.

Each experiment was run for a duration of approximately one minute, providing 120 frames of film per run. After the films were developed, they were mounted on a special projector which projects a single frame at a time. The image was projected onto a digitizing table. About 12 points were digitized in each frame. These data points were transferred to a large computer and plotted along with the theoretical predictions. A selected number of these experiments and a discussion of the results is provided at the end of this paper.

#### FORCES AND MOMENTS CAUSED BY WATER ON DECK

The sloshing of water on deck creates forces and moments affecting the motions and stability of a vessel. The force resulting from pressure exerted against the bulwarks is

$$\int_0^{h_b} P(\pm \frac{B}{2}, z) dz, \quad (8)$$

where the pressure is

$$P(y, z) = \rho a_z(y)z. \quad (9)$$

The moment due to the mass of water on deck is expressed as:

$$\int_{-B/2}^{B/2} P(y, 0)y dy, \quad (10)$$

where  $B/2$  denotes the half beam.

#### MOTIONS OF A VESSEL WITH WATER ON DECK

This study focuses on the roll response of the vessel, with a goal of predicting instability and capsizing. The analysis is restricted to motions in the transverse plane of the vessel. The heave (subscript 3), sway (subscript 2), and roll (subscript 4) equations of motion [5] are written:

$$\begin{aligned} (M + A_{22})\ddot{\eta}_2 + B_{22}\dot{\eta}_2 + A_{24}\ddot{\eta}_4 + B_{24}\dot{\eta}_4 &= F_2, \\ (M + A_{33})\ddot{\eta}_3 + B_{33}\dot{\eta}_3 + C_{33}\eta_3 &= F_3, \\ (I + A_{44})\ddot{\eta}_4 + B_{44}\dot{\eta}_4 + C_{44}\eta_4 + A_{42}\ddot{\eta}_2 \\ + B_{42}\dot{\eta}_2 &= M_4. \end{aligned} \quad (11)$$

The external forces  $F_i$  and moment  $M_i$  can be separated into components resulting from the waves and the free water on deck. These are

$$\begin{aligned} F_2 &= F_{2w} + F_{2wod}, \\ F_3 &= F_{3w} + F_{3wod}, \\ M_4 &= M_{4w} + M_{4wod}. \end{aligned} \quad (12)$$

In these equations the subscript "w" represents wave excitation and "wod" represents excitation due to water on deck. After adding nonlinear viscous damping and higher order nonlinear restoring terms, the equations become

$$\begin{aligned} & (M + A_{22})\ddot{\eta}_2 + B_{22}\dot{\eta}_2 + A_{24}\ddot{\eta}_4 + B_{24}\dot{\eta}_4 \\ & = F_{2w} + F_{2wod} - B'_{22}\dot{\eta}_2^2, \\ & (M + A_{33})\ddot{\eta}_3 + B_{33}\dot{\eta}_3 + C_{33}\eta_3 = F_{3w} + F_{3wod} \\ & - B'_{33}\eta_3^2, \\ & (I + A_{44})\ddot{\eta}_4 + B_{44}\dot{\eta}_4 + C_{44}\eta_4 + A_{42}\ddot{\eta}_2 + B_{42}\dot{\eta}_2 \\ & = M_{3w} + M_{3wod} - B'_{44}\dot{\eta}_4^2 - C_{4ho}, \end{aligned} \quad (13)$$

where

$$C_{4ho} = C'_{11}\eta^2 + C'_{22}\eta^3.$$

A time domain analysis of the equations of motion is performed by using Laplace techniques suggested by Livingston [5]. The analysis begins by assuming linear coupled sway and roll equations

$$\begin{aligned} & (M + A_{22})\ddot{\eta}_2 + B_{22}\dot{\eta}_2 + A_{24}\ddot{\eta}_4 + B_{24}\dot{\eta}_4 = F_{2w} \\ & = \bar{F}_{2w}e^{-j\omega t}, \\ & (I + A_{44})\ddot{\eta}_4 + B_{44}\dot{\eta}_4 + C_{44}\eta_4 + A_{42}\ddot{\eta}_2 + B_{42}\dot{\eta}_2 \\ & = M_{4w} = \bar{M}_{4w}e^{-j\omega t}. \end{aligned} \quad (14)$$

Because the equations are linear, the solutions  $\eta_2$  and  $\eta_4$  have the form

$$\begin{aligned} \eta_2(t) &= \bar{\eta}_2 e^{-j\omega t}, \\ \eta_4(t) &= \bar{\eta}_4 e^{-j\omega t}, \end{aligned} \quad (15)$$

where  $\eta_2$  and  $\eta_4$  represent amplitudes of sway and roll, respectively. Substituting Equation 15 into 14, the amplitudes become

$$\begin{aligned} \bar{\eta}_2 &= \frac{K_4}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}} F_{2w} - \frac{K_2}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}} M_{4w} \\ \bar{\eta}_4 &= \frac{K_1}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}} M_{4w} - \frac{K_3}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}} F_{2w} \end{aligned} \quad (16)$$

where,

$$\begin{aligned} K_1 &= -\omega^2(M + A_{22}) - j\omega B_{22}, \\ K_2 &= -\omega^2 A_{24} - j\omega B_{24}, \\ K_3 &= -\omega^2 A_{42} - j\omega B_{42}, \\ K_4 &= -\omega^2(I + A_{44}) - j\omega B_{44} + C_{44} \end{aligned} \quad (17)$$

The Frequency Response Functions,  $H_{pq}(j\omega)$  for  $p, q = 2, 4$ , are defined

$$\begin{aligned} H_{22}(j\omega) &= \frac{K_4}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}}, & H_{24}(j\omega) &= \frac{-K_2}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}} \\ H_{42}(j\omega) &= \frac{-K_3}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}}, & H_{44}(j\omega) &= \frac{K_1}{\begin{vmatrix} K_1 & K_2 \\ K_3 & K_4 \end{vmatrix}}. \end{aligned} \quad (18)$$

Equation 16 now has the form

$$\begin{aligned} \bar{\eta}_2 &= H_{22}(j\omega)F_{2w} + H_{24}(j\omega)M_{4w}, \\ \bar{\eta}_4 &= H_{42}(j\omega)F_{2w} + H_{44}(j\omega)M_{4w}. \end{aligned} \quad (19)$$

If the amplitudes of roll and sway motions and excitations are assumed to be complex,

$$\begin{aligned} \bar{\eta}_{22} &= |\bar{\eta}_{22}|e^{-j\phi_1(\omega)}, \\ \bar{\eta}_{24} &= |\bar{\eta}_{24}|e^{-j\phi_2(\omega)}, \\ \bar{\eta}_{42} &= |\bar{\eta}_{42}|e^{-j\phi_3(\omega)}, \\ \bar{\eta}_{44} &= |\bar{\eta}_{44}|e^{-j\phi_4(\omega)}, \\ \bar{F}_{2w} &= |F_{2w}|e^{-j\phi_5(\omega)}, \\ \bar{M}_{4w} &= |M_{4w}|e^{-j\phi_6(\omega)}. \end{aligned} \quad (20)$$

then the Frequency Response Functions are

$$H_{22}(j\omega) = \frac{|\bar{\eta}_{22}|}{|F_{2w}|} e^{-j(\phi_1(\omega) + \phi_5(\omega))}, \quad (21a)$$

$$H_{24}(j\omega) = \frac{|\bar{\eta}_{24}|}{|M_{4w}|} e^{-j(\phi_2(\omega) + \phi_6(\omega))}, \quad (21b)$$

$$H_{42}(j\omega) = \frac{|\bar{\eta}_{42}|}{|F_{2w}|} e^{-j(\phi_3(\omega) + \phi_5(\omega))}, \quad (21c)$$

$$H_{44}(j\omega) = \frac{|\bar{\eta}_{44}|}{|M_{4w}|} e^{-j(\phi_4(\omega) + \phi_6(\omega))}, \quad (21d)$$

To proceed toward a solution, frequency response functions are expressed as rational polynomials

$$H_{pq}(j\omega) = \frac{A_m S^m + A_{m-1} S^{m-1} + \dots + A_0 S^0}{B_n S^n + B_{n-1} S^{n-1} + \dots + B_0 S^0} \quad (22)$$

where

$$m \leq n-2.$$

Introducing an auxiliary temporary function  $T(s)$ , Equation 21d becomes



$$H_{44}(j\omega) = \frac{\eta_{44}(s) T(s)}{T(s) M_{4w}(s)} \quad (23)$$

$$= \frac{A_m S^m + A_{m-1} S^{m-1} + \dots + A_0 S^0}{B_n S^n + B_{n-1} S^{n-1} + \dots + B_0 S^0}.$$

Then Equation 23 is separated into

$$\frac{T(s)}{M_{4w}(s)} = \frac{1}{B_n S^n + B_{n-1} S^{n-1} + \dots + B_0 S^0}, \quad (24a)$$

$$\frac{\eta_{44}(s)}{T(s)} = A_m S^m + A_{m-1} S^{m-1} + \dots + A_0 S^0 \quad (24b)$$

Converting Equation 24a to the time domain one obtains

$$B_n T^n(t) + B_{n-1} T^{n-1}(t) + \dots + B_0 T^0(t) = M_{4w}(t). \quad (25)$$

When the state derivatives are written, this equation is reduced to a series of first order differential equations

$$\begin{aligned} \dot{Y}(0) &= Y(1) = \dot{T}(t) \\ \dot{Y}(1) &= Y(2) = \ddot{T}(t) \\ \dot{Y}(n-2) &= Y(n-1) = T^{(n-1)}(t) \\ \dot{Y}(n-1) &= \frac{1}{B_n} (M_{4w}(t) - B_{n-1} Y(n-1) \\ &\quad - \dots - B_0 Y(0)) \end{aligned} \quad (26)$$

Additional forces, such as the force and moment due to water on deck and viscous damping force, are added to  $M_{4w}(t)$ . After differential Equation 26 is solved, Equation 24b is converted to the time domain

$$\eta_{44}(t) = A_m T^m(t) + A_{m-1} T^{m-1}(t) + \dots + A_0 T^0(t), \quad (27)$$

and choosing  $m = n-2$ , it becomes

$$\eta_{44}(t) = A_{n-2} T^{n-2}(t) + A_{n-3} T^{n-3}(t) + \dots + A_0 T(t). \quad (28)$$

Substituting  $Y$  from Equation 26 for  $T$  in Equation 28, the amplitude of roll motion due to the roll exciting moment becomes

$$\eta_{44}(t) = A_{n-2} Y(n-2) + A_{n-3} Y(n-3) + \dots + A_0 Y(0) \quad (29)$$

To find the roll amplitude, the components are added

$$\begin{aligned} \eta_2(t) &= \eta_{22}(t) + \eta_{24}(t), \\ \eta_4(t) &= \eta_{42}(t) + \eta_{44}(t). \end{aligned} \quad (30)$$

## STABILITY OF MOTIONS

The stability of the motions resulting from excitation of waves and the effect of the pressure of the water on deck can be analyzed by plotting the phase plot of the system, as suggested by Kuo and Odabaşı [7] in a comprehensive treatment of the subject. In this case, roll velocity will be plotted versus roll angle.

## CONCLUSION AND DISCUSSION

Glimm's method applied to the Dam Breaking Problem as formulated by Dillingham for the sloshing of water on deck shows very good agreement with experimental measurement of the water motion over a wide range of water depths, amplitudes of oscillation and frequencies. Considering the often apparently turbulent nature of the sloshing water the agreement with theory is remarkable.

Figures 5, 6, 7 and 8 show a comparison of the theoretically predicted and experimentally measured water motion in an oscillating tank for water of varying depth. The experimental results are identified at discrete points by the asterisk, and the solid line indicates the theoretical prediction. The results indicate that the theory is not sensitive to the depth of water even under conditions where there is no water in a large portion of the tank.

Figure 9 shows the predicted and measured motion of the water in the tank when the amplitude of oscillation was 14.0 degrees. Comparing Figure 9 with Figure 5 reveals that the theory predicts the water motion well at least up to the amplitude of 14.0 degrees.

Over a wide range of frequencies the theory provides good correlation with experimental measurements. The only range where the theory fails is near the frequency of tank resonance. Under these conditions, the water motion becomes extreme. Figures 10, 11 and 12 show the experiments for deeper water as the natural frequency of oscillation is approached. Note the deterioration in the correlation between theory and experiment. Figures 13, 14 and 15 illustrate the case where there are experiments on either side of the resonant frequency. While the theory does not predict the results well near resonance, it improves as the test frequency diverges from resonance.

Unfortunately, scaling the model experiments to full scale indicates that for most vessels, resonant conditions are quite possible at sea.

In applying Dillingham's theoretical predictions over a wide range of conditions an interesting observation concerning the volume of water on deck was made. In the theoretical predictions the volume seems to oscillate around the initial volume. When the Courant-Friedericks-Lewy condition (Equation 5) is violated, the volume

changes rapidly and the calculations become unstable. This problem may be eliminated by altering the time step to insure satisfaction of the condition. However, it also appears that if the condition is too strongly satisfied the calculation procedure also becomes unstable.

At present, the investigation is at the point where computer code for the time domain solution is almost complete. Following the present theoretical phase of the project, further experiments are being designed for a wave tank in which a two-dimensional model with water on deck will be subjected to the incident wave excitation.

#### ACKNOWLEDGEMENTS

The authors wish to thank Dr. J. T. Dillingham for his assistance in conducting this investigation. By granting permission to use his computer codes, he has greatly enhanced our efforts. We also gratefully acknowledge Mr. W. Livingston's kind cooperation. Special credit also should be extended to Ms. C. J. McCutcheon for her assistance in preparing the manuscript of this paper.

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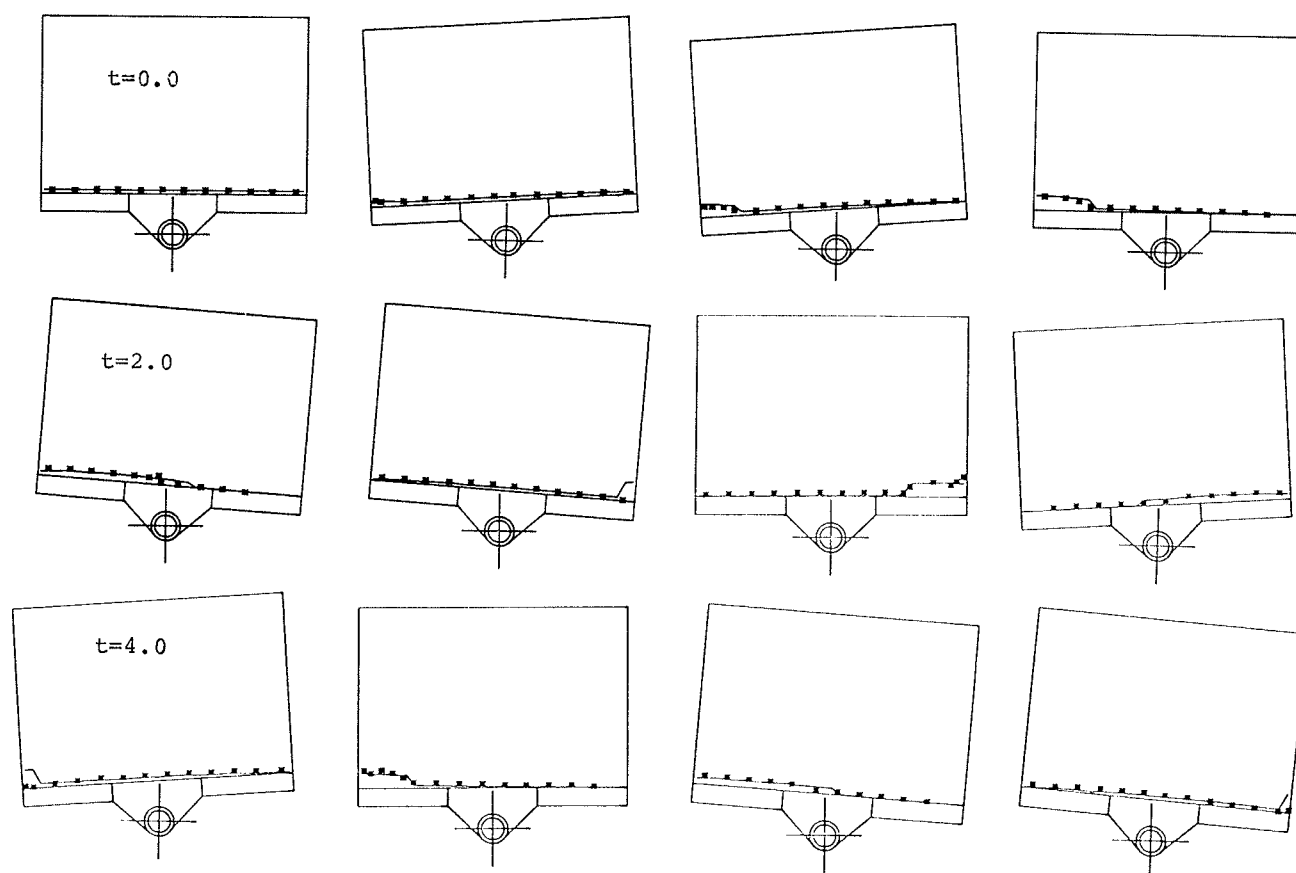


Figure 5. Time history of water sloshing experiment, water depth = 0.5 in, amplitude = 5 degrees, frequency = 0.33 Hz, time increment = 0.5 sec.

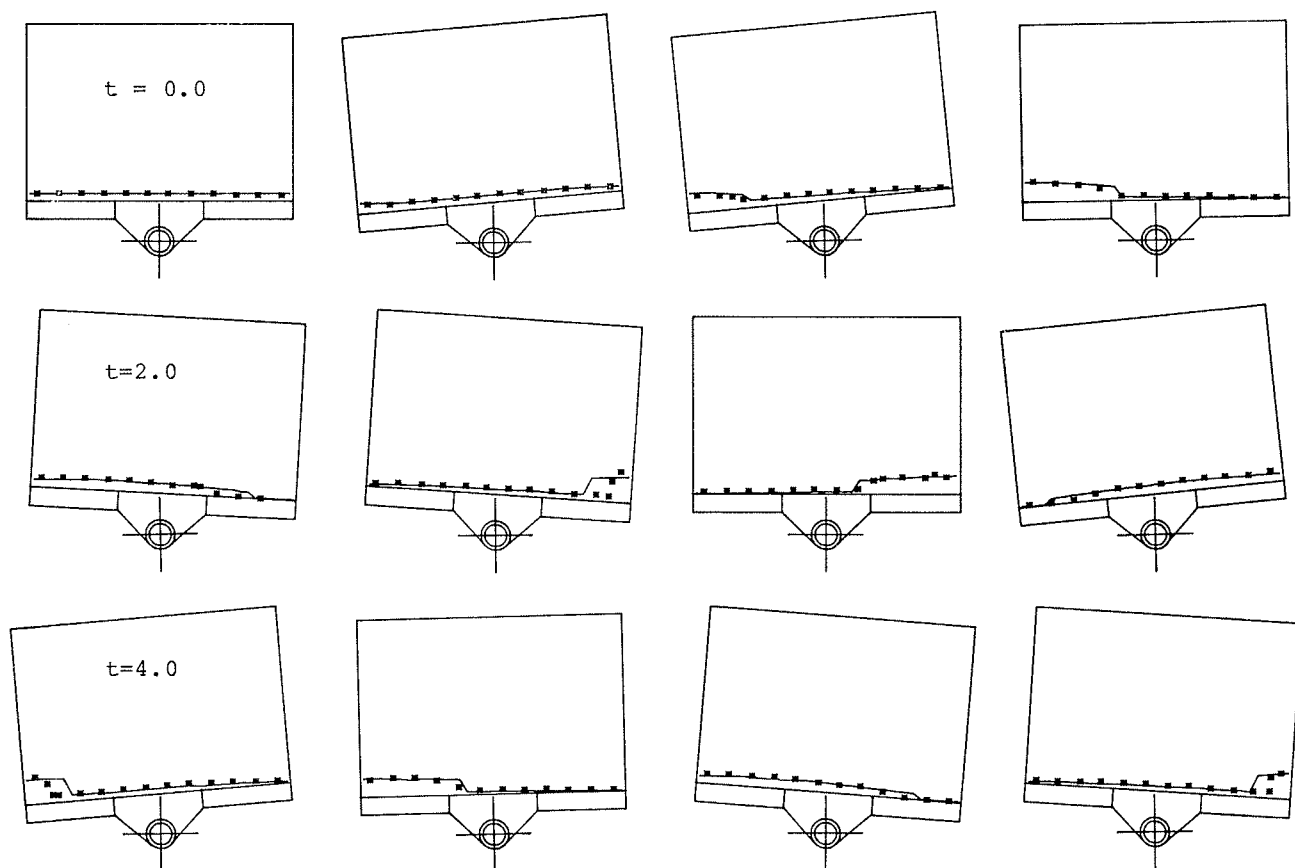
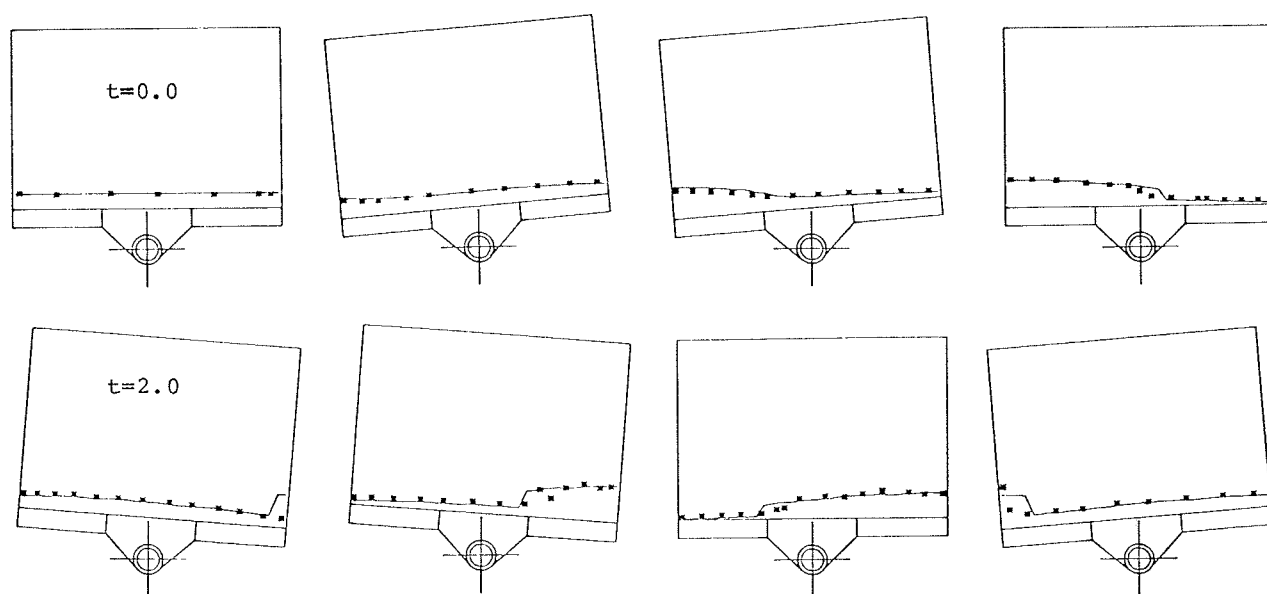


Figure 6. Time history of water sloshing experiment, water depth = 1.0 in, amplitude = 5.0 degrees, frequency = 0.33 Hz, time increment = 0.5 sec.



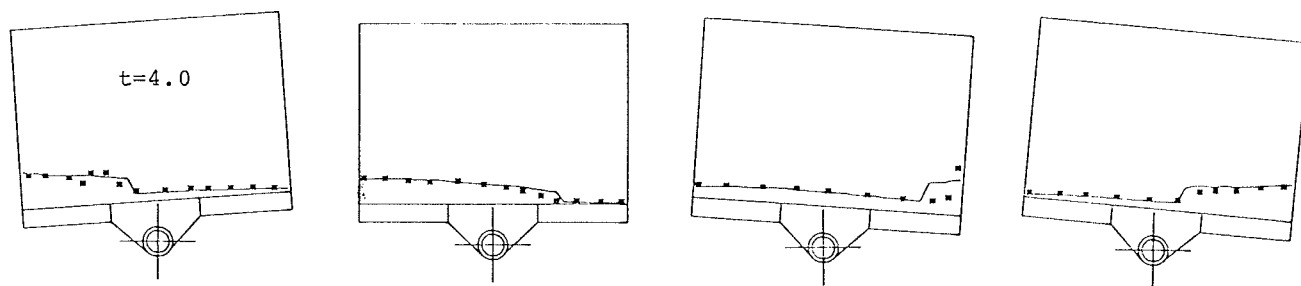


Figure 7. Time history of water sloshing experiment, water depth = 2.0 in, amplitude = 5.0 degrees, frequency = 0.33 Hz, time increment = 0.5 sec.

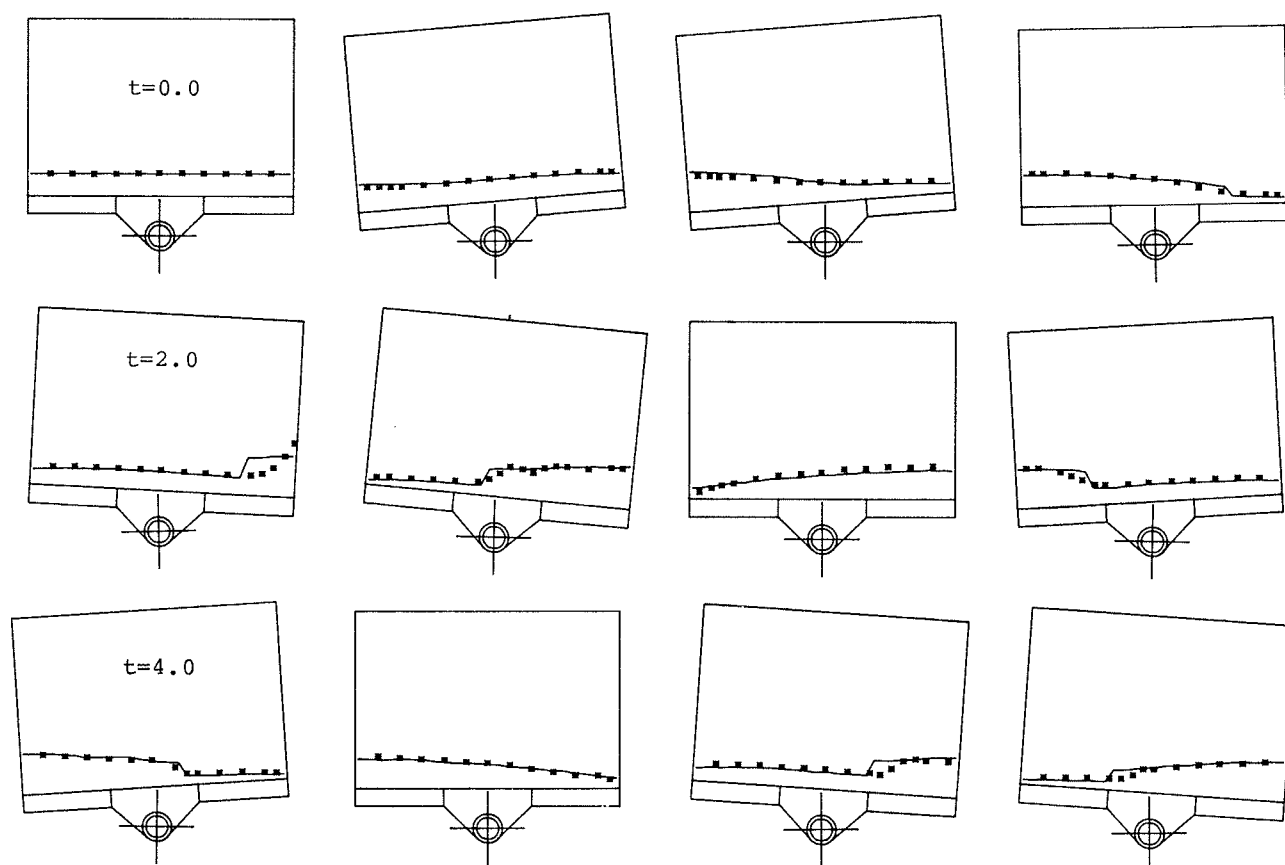


Figure 8. Time history of water sloshing experiment, water depth = 3.0 in, amplitude = 5 degrees, frequency = 0.42 Hz, time increment = 0.5 sec.

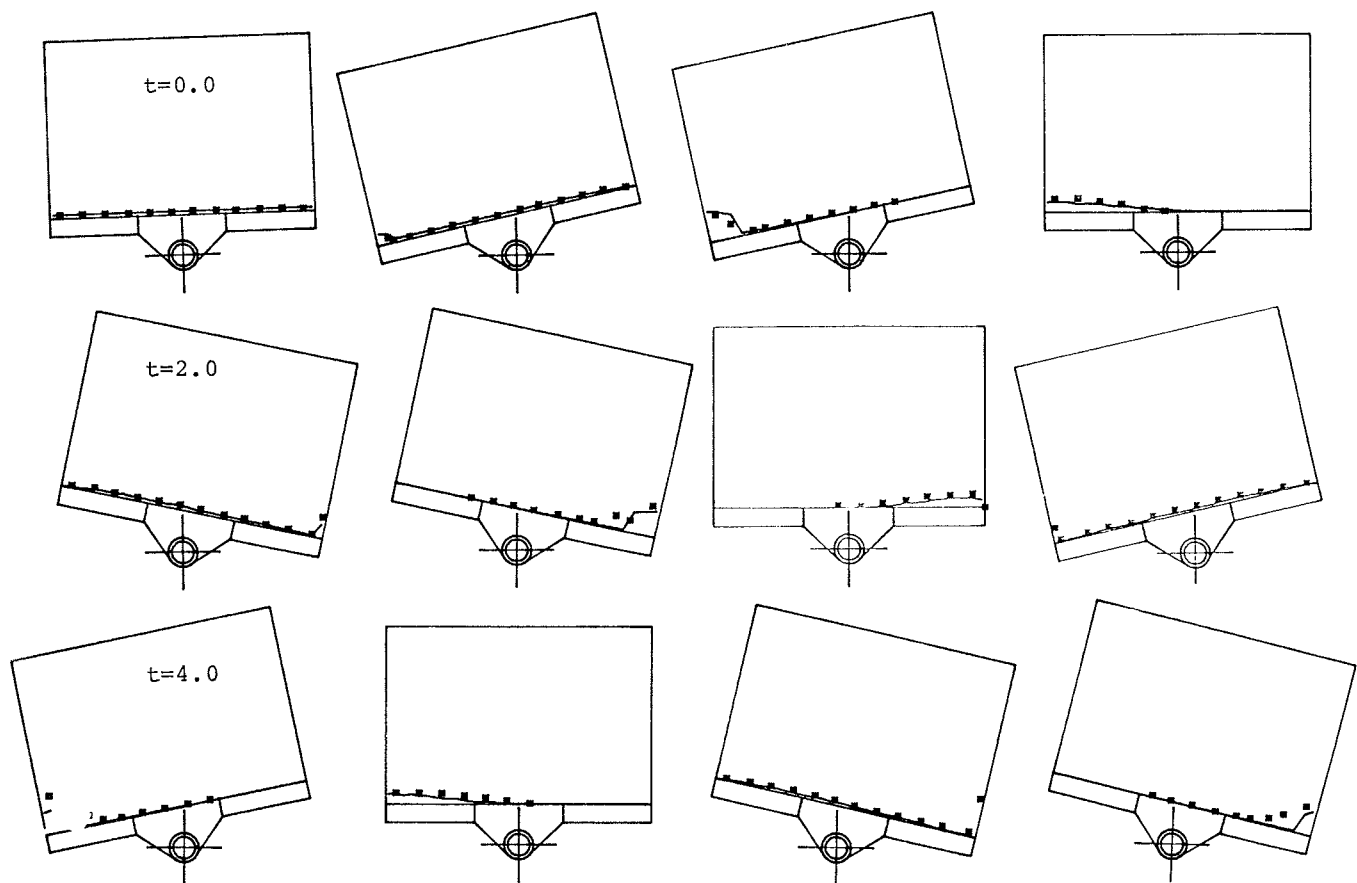
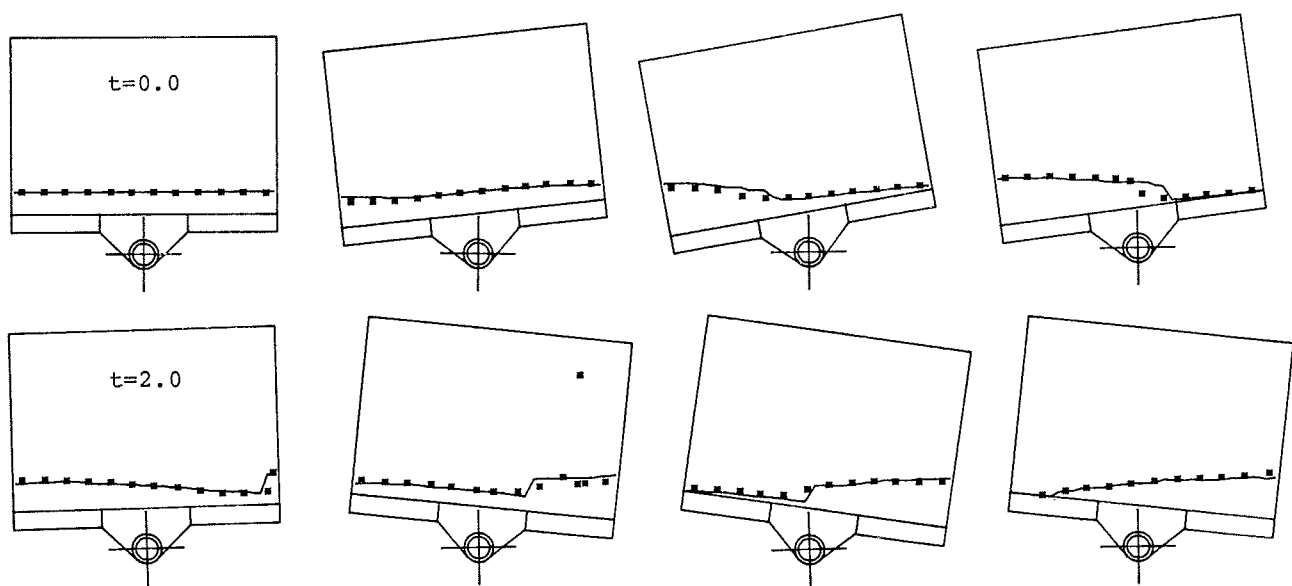


Figure 9. Time history of water sloshing experiment, water depth = 0.5 in, amplitude = 14.0 degrees, frequency = 0.42 Hz, time increment = 0.5 sec.



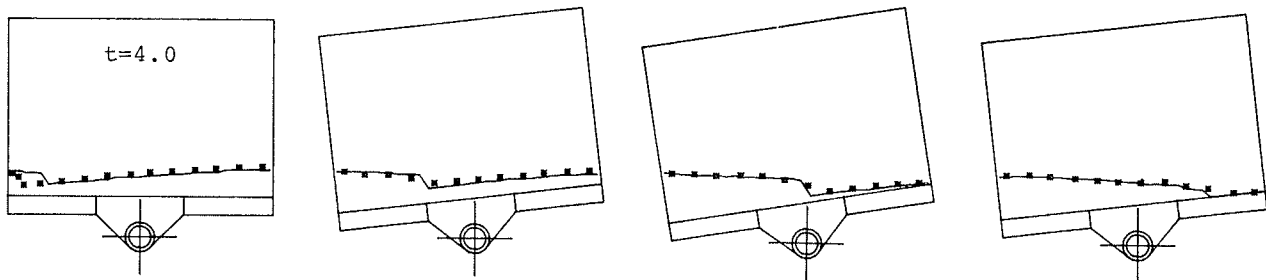


Figure 10. Time history of water sloshing experiment, water depth = 3.0 in, amplitude = 9.5 degrees, frequency = 0.25 Hz, time increment = 0.5 sec. (a point in the upper right hand corner of the tank indicates wave breaking during experiment).

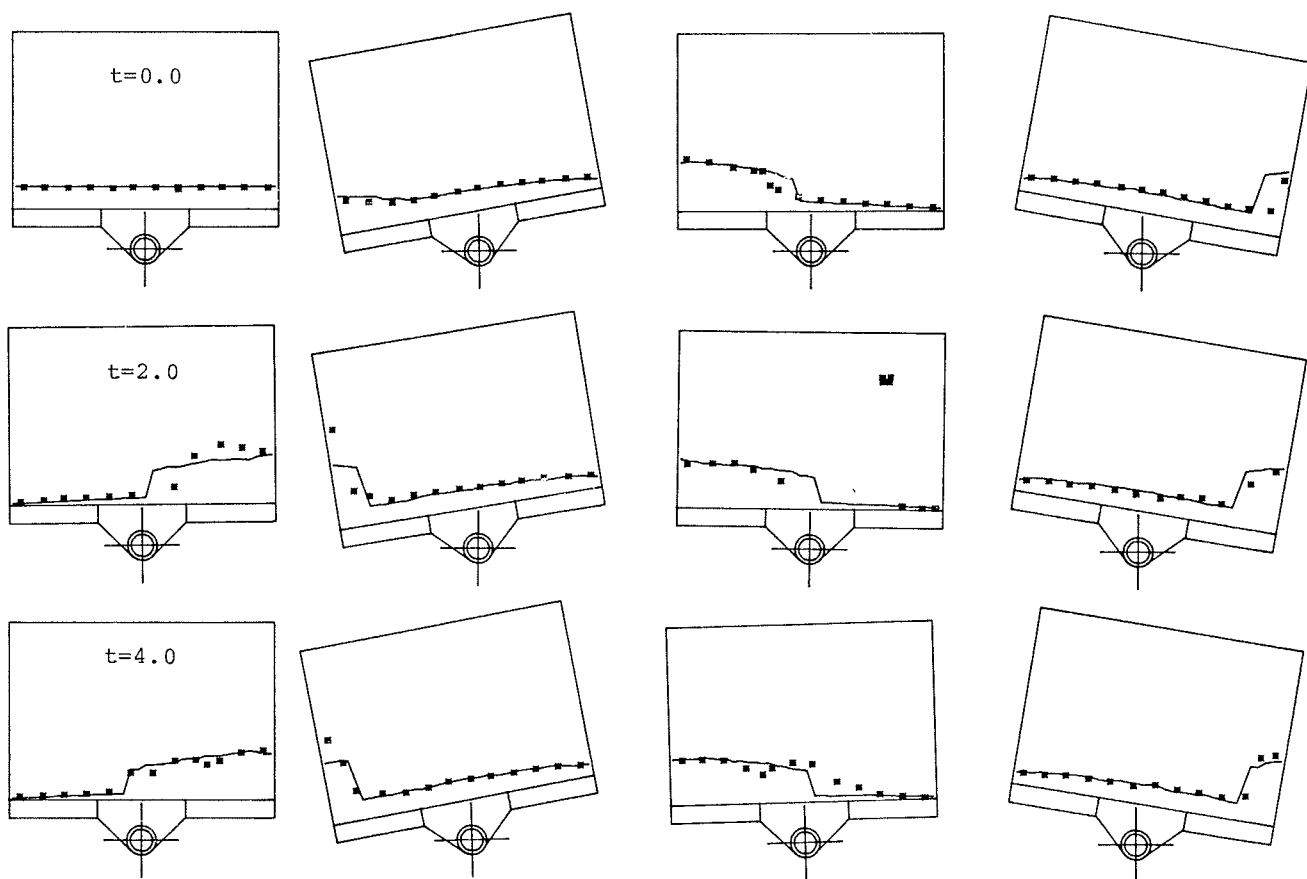


Figure 11. Time history of water sloshing experiment, water depth = 3.0 in, amplitude = 9.5 degrees, frequency = 0.50 Hz, time increment = 0.5 sec (multiple points in upper right hand corner of tank indicate severe wave breaking during experiment).

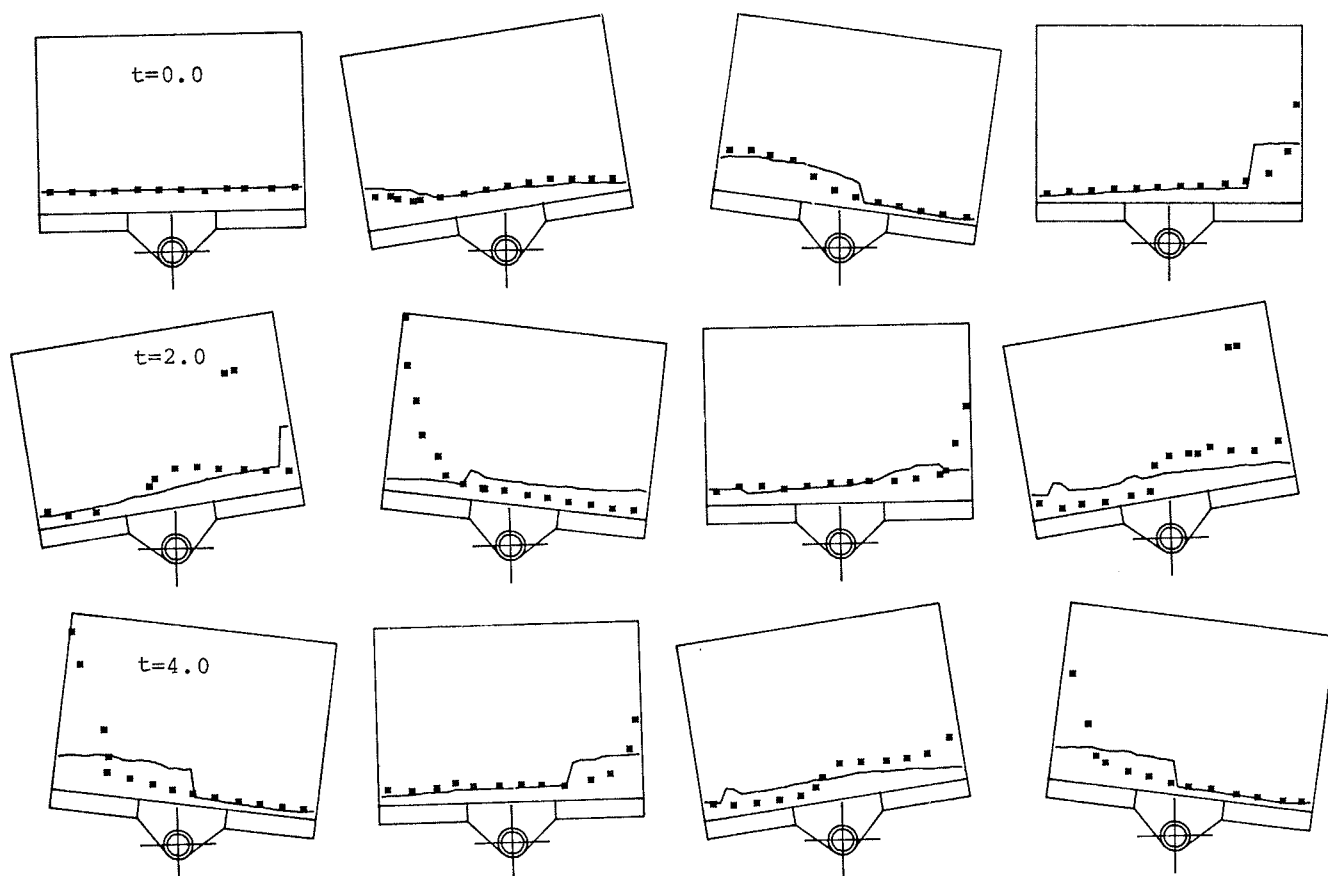
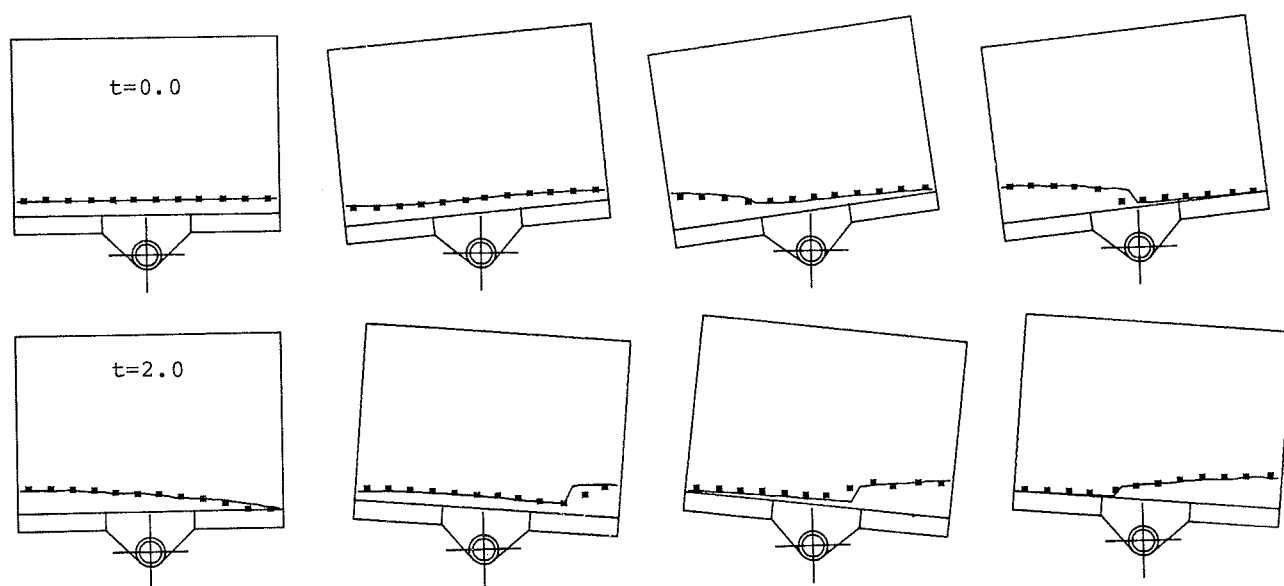


Figure 12. Time history of water sloshing experiment, water depth = 3.0 in, amplitude = 9.5 degrees, frequency = 0.67 Hz, time increment = 0.5 sec (points in upper right hand corner of tank indicate wave breaking during experiment).



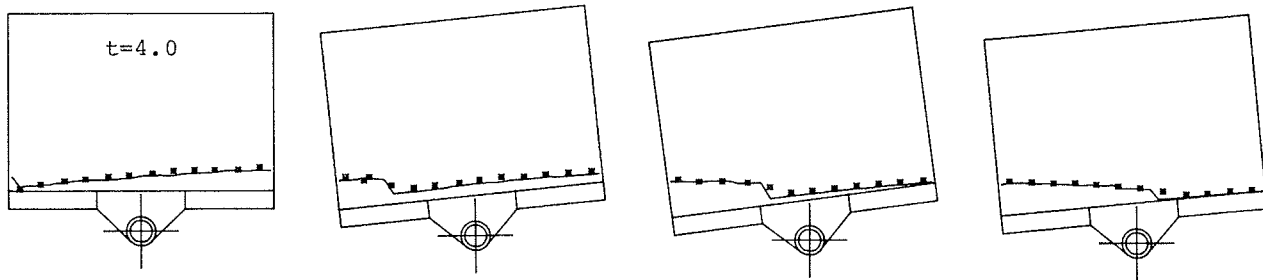


Figure 13. Time history of water sloshing experiment, water depth = 2.0 in, amplitude = 7.5 degrees, frequency = 0.25 Hz, time increment = 0.5 sec.

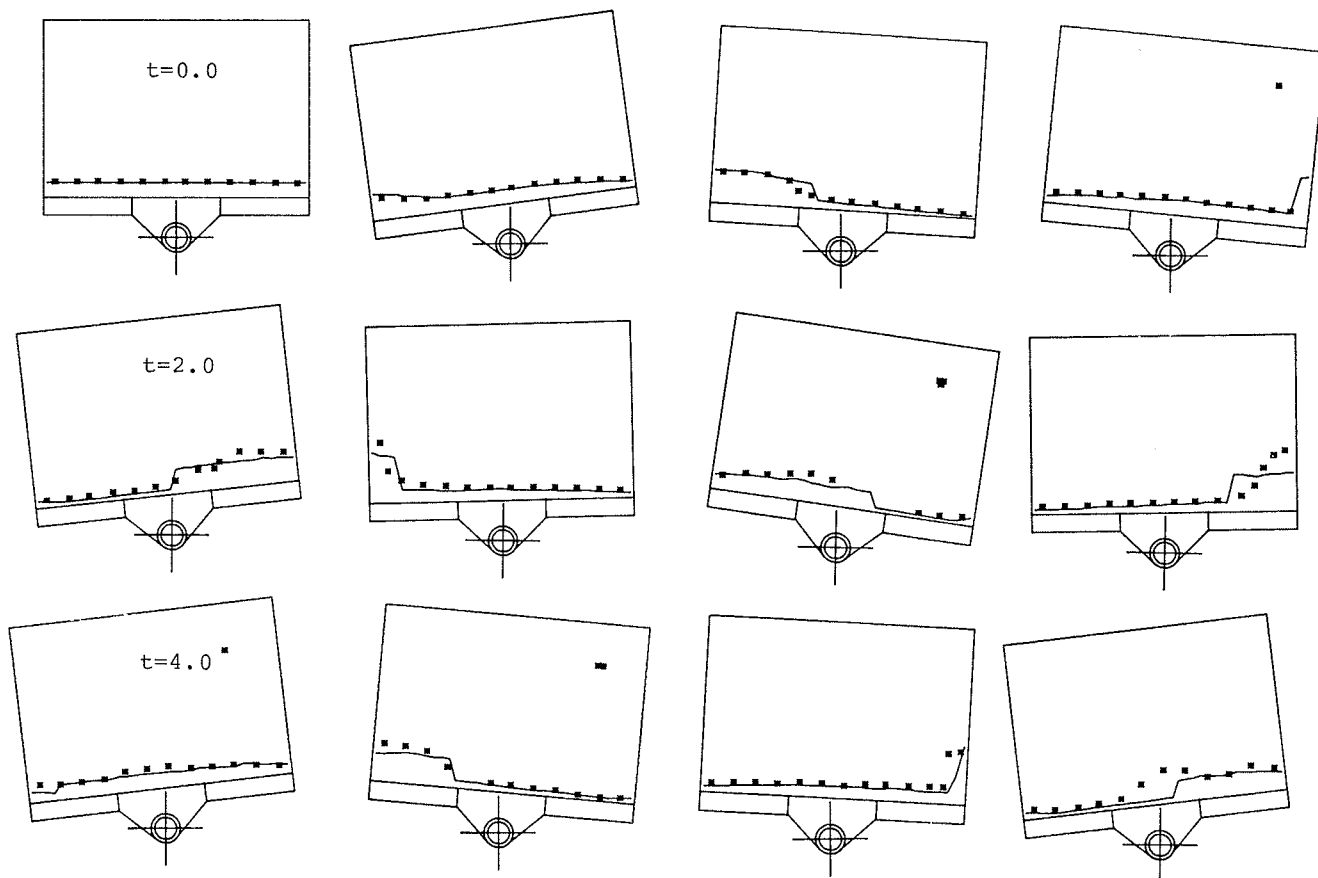


Figure 14. Time history of water sloshing experiment, water depth = 2.0 in, amplitude = 7.5 degrees, frequency = 0.58 Hz, time increment = 0.5 sec (number of points in upper right hand corner of tank indicate severity of wave breaking during experiment).



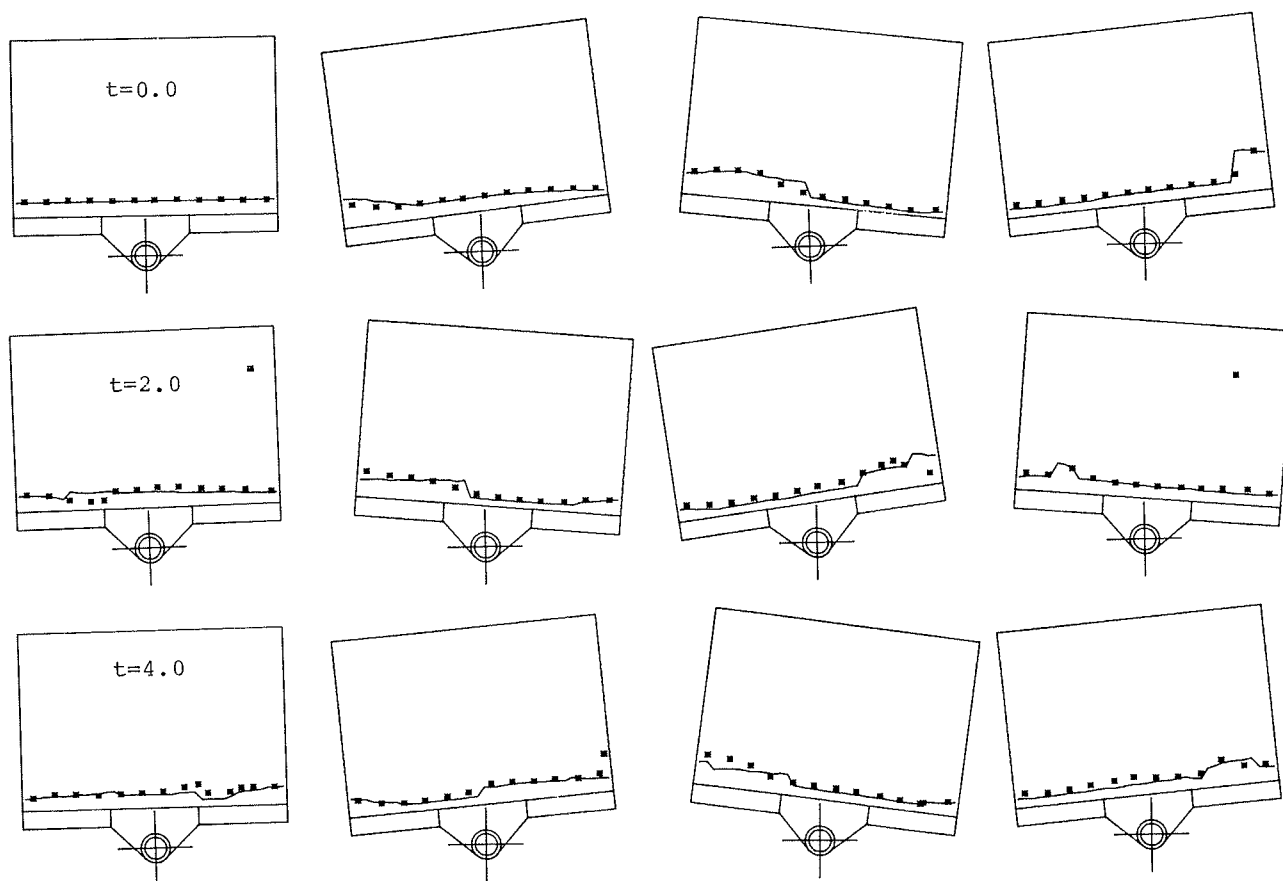


Figure 15. Time history of water sloshing experiment, water depth = 2.0 in, amplitude = 7.5 degrees, frequency = 0.75 Hz, time increment = 0.5 sec (point in upper right hand corner of tank indicates wave breaking during experiment).

## Discussion

R. Barr (Hydronautics, Inc., USA)

The authors are to be congratulated on what I consider to be a very important result, the initial validation of a method which can be used to predict the dynamic behavior of water on deck. We believe that water on deck plays a crucial role in many cases of capsizing, and suitable analytical methods for predicting the dynamic behavior of on-deck water area a crucial part of any analytical method for predicting capsizing in these cases. While this method is at present applicable only to two-dimensional problems, we feel it offers a potentially valuable approach for dealing with realistic three-dimensional problems of vessels.

Some years ago we observed at hydronautics capsizing of a supply vessel model in large regular head waves. Photographs indicate that capsizing resulted from a combination of:

- ° shipping of large quantities of water

on deck and the highly dynamic behavior of this water

- ° loss of stability as the vessel "lept" from the wave crest and water-plane area significantly decreased.

The highly dynamic behavior of on-deck water was observed in the large quantity of water flowing over the top of the bulwark and in the "jetting" nature of flow out of the freeing ports, indicating high hydrostatic pressures at the bulwarks.

D. Vassalos (Univ. of Strathclyde, UK)

I would like to congratulate the authors in undertaking to validate an analytical procedure. Sometimes it is not appreciated that this may be more strenuous than developing the procedure in the first place.

I just want to make a point regarding freeing ports and ask the authors whether they have considered them at all or plan to do so.

Y. Yamakoshi (National Research Institute  
of Fisheries Engineering,  
Japan)

Thank you for your useful presentation.  
I have a simple question.

1) The transverse motions in waves are  
considered as a coupling of Roll-Sway-Yaw.

The authors treated only a effect of  
rolling on the trapped water on deck.

Would you give us some comments about  
the effect of sway motion on the trapped  
water on deck.

F. Ursell (Manchester University, UK)

In the discussion to the following  
paper by Dr. Pinch I shall describe some  
preliminary results obtained by Hulme and  
Jones. These are closely related to the  
work of Dr. Adey, and a comparison of their  
results and his should prove most illumi-  
nating.

#### Author's Reply

The authors would like to express their  
appreciation for the kind sentiments ex-  
pressed by the discussers.

The tests conducted to date should be  
considered initial validation of Dillingham's  
theory. We strove to construct the most  
fundamental test for this purpose with the  
thought of extending the tests to include  
additional factors if these first tests were  
successful. On a real vessel the inflow  
and outflow of water will be important  
factors. We would like to extend our tests

to include the changes in the amount of  
water on deck. However, we also want to  
perform experiments of practical signifi-  
cance for realistic seaway conditions and  
are concerned about the scale relationships  
between a small model and real vessels with  
water flowing on and off the deck.

The authors are concerned about dupli-  
cating work undertaken at other institutions.  
We wish to express our thanks to Prof. Ursell  
and others for providing information on  
previous studies and current investigations.

Mr. Yamakoshi raises the point that  
dealing with all six degrees of freedom  
and including the critical coupling terms  
is very important. Certainly, this is the  
ultimate goal. The authors believe this  
goal will be achieved by taking small and  
careful steps. The theory which was used  
has only been developed for the two-dimen-  
sional case and before attempting to extend  
it to three dimensions, we felt verifica-  
tion of the fundamental precepts was re-  
quired. As a result we conducted the  
simplest experiment possible.

On the basis of our experimental re-  
sults, the authors believe the theory which  
describes the water motion on deck will  
work for any motion or combination of coupl-  
ed motions so long as the frequency is not  
near the natural frequency of the tank.  
Since the calculations are performed in  
the time domain one must only prescribe  
the motions of the deck as a function of  
time in order to calculate the water  
motion.

# WATER ON DECK - A THEORETICAL STUDY

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United Kingdom

## ABSTRACT

The effect on stability of the presence on deck of an asymmetrically placed weight of significant size is examined. A two-dimensional model is used and a criterion derived that is equivalent to but more convenient than the classical result. Numerical results are presented for a ship with an elliptic transverse cross-section.

## NOMENCLATURE

W	weight of ship
H	weight of water on deck
U	buoyancy force
$\rho$	density of salt water
g	acceleration due to gravity
2a	breadth of ship
h	depth of ship
l	length of ship
Oxy	axes fixed relative to the ship
O'x'y'	axes fixed relative to the mean free surface
G	position of the centre of mass of the ship
k	depth of G below deck level
M	mass of ship
$I_G$	moment of inertia of ship about the longitudinal axis through G
M'	mass of ship including added mass
$I'_G$	moment of inertia of ship including added moment of inertia
B	the centre of buoyancy and (later) a coefficient in equation (6)
(X,Y)	the coordinates of B in Oxy coordinates
A	the point on deck at which H acts and (later) a coefficient in equation (6)
d	distance of A from the centre line

$x_1$ and $x_2$	the x-coordinates of the points where the water-line intersects the hull
K	the point on the evolute of the curve of buoyancy that corresponds to the equilibrium position of B
$\theta$	angle of heel
$\theta_0$	equilibrium angle of heel
$\phi, z$	variables representing small deviations from equilibrium
$y'_G$	vertical coordinate of G in O'x'y' coordinates
$\bar{y}'_G$	the equilibrium value of $y'_G$
$L_1, L_2, \epsilon_1, \epsilon_2$	arbitrary constants
$L_H$	heeling moment
$L_R$	righting moment.

## 1. INTRODUCTION

In this study of the effect of water on deck a very simple model is used. It is assumed that a significant amount of water has been trapped on one side of the deck and that the centre of mass of this water does not move much relative to the ship. If this is the case then one can pose the question "How is stability affected by the presence of a large weight placed asymmetrically on deck?" Throughout this investigation it is assumed that the ship has a uniform transverse cross-section described by the equation  $y = f(x)$  in the Oxy coordinate system and that the free surface of the sea (assumed calm) is  $y = mx + c$  where  $m = -\tan \theta$  and  $c$  is dependent on the loading and orientation of the ship. We assume that  $f(x)$  is such that there are, at most, two points at which the free surface intersects the hull. The results presented here are valid for a convex hull with a finite number of corners.

Before we can discuss stability we must

first establish the equilibrium conditions. If we are concerned only with static equilibrium then we can do all our calculations in  $(x,y)$  coordinates. If we then wish to use classical hydrostatics to derive a condition for statical stability the same coordinate system is appropriate. The classical stability argument, involving the curve of buoyancy and its evolute, is not the only way of examining statical stability. Indeed, if we want to discover stability conditions for asymmetrical deck loads of various sizes, this approach is not very convenient for reasons that will be discussed later (see 2.2). John [1] showed that the classical static stability conditions are sufficient but not necessary for ship stability. He examines the problem of a rigid body partially immersed in a heavy liquid and deals with the linearized problem in which the non-linear boundary conditions that must hold on the time-dependent boundaries are replaced by linear conditions on fixed boundaries. The approach adopted here is rather different; nevertheless the static stability criterion found is equivalent to that given by John [1] page 34. Here we examine the equations of small oscillations about equilibrium and deduce from them a static stability criterion that tells us, for each value of  $H$  (the deck load) what positions of  $G$  give a stable equilibrium position.

## 2.1 Equilibrium Conditions

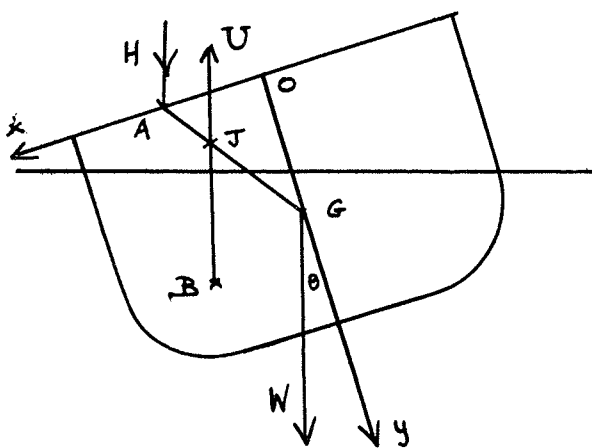


Figure 1

For static equilibrium we require that

$$H + W = U \quad (1)$$

and that the total moment of these three vertical forces is zero everywhere. If we take moments about  $G$  we have

$$Hd \cos \theta + Hk \sin \theta = UX \cos \theta + U(k-Y) \sin \theta \quad (2)$$

where

$$U = \rho g l \int_{x_1}^{x_2} (f(x) - mx - c) dx \quad (3)$$

$$UX = \rho g l \int_{x_1}^{x_2} x(f(x) - mx - c) dx \quad (4)$$

and

$$UY = \rho g l \int_{x_1}^{x_2} \frac{1}{2} ((f(x))^2 - (mx + c)^2) dx. \quad (5)$$

The equations (1) and (2) determine the equilibrium values of  $\theta$  and  $c$ , if they exist. The left-hand side of (2) is the heeling moment  $L_H$  and the right-hand side the righting moment  $L_R$ .

## 2.2 A Classical Stability Argument

Condition (2) can be rewritten in a number of ways. Suppose we take moments about  $J$ , the point where the line of action of  $U$  cuts  $AG$ , then

$$H(AJ) = W(JG).$$

From this we can deduce the  $(x,y)$  coordinates of  $J$  for equilibrium namely  $\left\{ \frac{Hd}{H+W}, \frac{Wk}{H+W} \right\}$ .

$J$  must lie on the normal to the curve of buoyancy that passes through  $B$ . The classical hydrostatic argument is that if  $J$  lies below  $K$  on the normal through  $B$  then there is stability and that if  $J$  lies on the other side of  $K$  there is instability, where  $K$  is the centre of curvature of the curve of buoyancy at  $B$ . This argument gives us, in principle, an inequality that tells us for any  $H$  how high  $G$  can be in the ship before static instability occurs. It relies on the observation that in an infinitesimal disturbance from equilibrium the instantaneous centre of rotation is the centre of curvature of the curve of buoyancy at the equilibrium position of  $B$  (i.e. the corresponding point on the evolute). For  $H = 0$  our ship floats with its axis of symmetry vertical so if we are interested in static stability for small disturbances from equilibrium we need only take into account the local behaviour of the curve of buoyancy - in effect we only need to know the position of  $G$  relative to the "initial metacentre". If  $H$  is non-zero, so that the ship floats asymmetrically, the condition for stability is analogous namely  $J$  must lie below  $K$ . But if we wish to examine stability for a range of values of  $H$  then we need to know the curve of buoyancy and its evolute for a finite range of  $\theta$  and for varying ship displacement (since the value of  $H$  must affect displacement). This is not a convenient approach.

We can avoid any argument that involves the curve of metacentres by finding the equilibrium position for each  $H$  and then heeling the ship through a small angle  $\Delta\theta$  and displacing it through a small vertical distance. We can then examine the forces brought into play by such a disturbance and

set up the equations of small oscillations. To do this we need to work with axes that are fixed in space. Suitable axes are shown in Figure 2, where  $O'x'$  is in the mean free surface and  $O'y'$  is vertically down. The position of the ship is then completely specified by  $y'_G$  and the angle  $OG$  makes with  $O'y'$ .

### 2.3 Small Oscillations About Equilibrium

Here we examine our ship with its deck load behaves when displaced from its equilibrium position.

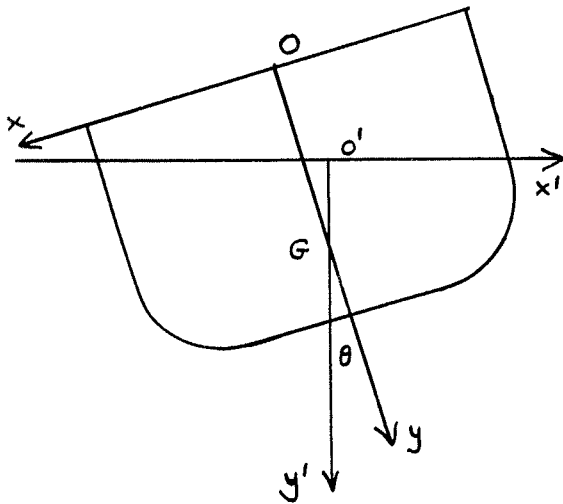


Figure 2

We do not include sway so the  $x'$  coordinate of  $G$  is always zero. In the  $Oxy$  system the equilibrium position is characterised by  $(\theta_0, c_0)$  i.e. the values of  $\theta$  and  $c$  that satisfy (1) and (2). In the  $O'x'y'$  system the equilibrium is  $(\theta_0, \bar{y}'_G)$  where  $\bar{y}'_G = (k - c_0) \cos \theta_0$ .

We now displace the ship vertically through a small distance and simultaneously heel it through a small angle so that the position of the ship is now described by  $\theta = \theta_0 + \Delta\theta$ ,  $y' = \bar{y}'_G + \Delta y'$  we can then calculate the corresponding changes in  $U$ ,  $UX$  and  $UY$ . To do this we return to (3), (4) and (5) to calculate  $\Delta U$ ,  $\Delta(UX)$  and  $\Delta(UY)$  in terms of  $\Delta\theta$  and  $\Delta c$  where

$$c = -\Delta y' \sec \theta_0 - (\bar{y}'_G \sec \theta_0 \tan \theta_0) \Delta\theta.$$

It should be noted that the integrands and the limits of the integrals are functions of  $\theta$  and  $c$  (or, alternatively,  $\theta$  and  $y'_G$ ). We then use the principle of linear momentum in the vertical direction and the principle of angular momentum about  $G$  to obtain the equations of motion. We then linearize so as to obtain the equations of small oscillations of the system about its

equilibrium position and obtain a pair of coupled second order equations. It is convenient at this stage to write  $\phi = \Delta\theta$   $z = y'$ , so that the required equations are:

$$A\ddot{z} + Bz + C\phi = 0 \quad (6)$$

$$D\ddot{\phi} + Cz + F\phi = 0 \quad (7)$$

where

$$B = \xi \sec \theta_0, \quad C = \sec \theta_0 \left\{ \frac{\xi_2}{2} \sec \theta_0 + \bar{y}'_G \tan \theta_0 \right\},$$

$$F = \left[ (kW - UY) \sec \theta_0 + \frac{\xi_3}{3} \sec^3 \theta_0 + \xi_2 \bar{y}'_G \tan^3 \theta_0 + \xi (\bar{y}'_G)^2 \sec \theta_0 \tan^2 \theta_0 \right]$$

where  $UY$  is evaluated at the equilibrium position and

$$\xi = \rho g l (x_2 - x_1),$$

$$\xi_2 = \rho g l (x_2^2 - x_1^2),$$

$$\xi_3 = \rho g l (x_2^3 - x_1^3)$$

where  $x_2$  and  $x_1$  are the roots of  $f(x) = -x \tan \theta_0 + c_0$ . In simplifying the coefficient  $F$ , use has been made of conditions (1) and (2).

As is well-known the coefficients  $A$  and  $D$  pose serious difficulties. When a ship moves, even in water that is initially at rest, it generates waves so it would be unrealistic to take  $A$  to be  $M$  the mass of the ship and  $B$  to  $I_G$  its moment of inertia about  $G$ . We can however try to estimate the "added mass" and "added moment of inertia". This aspect of the problem is not discussed here but is under active consideration. If however we are interested in static stability the actual values of  $A$  and  $D$  are irrelevant, for the following reason. If the equilibrium is stable we would expect the solution for  $\phi$  to be of the form  $\phi = L_1 \cos(\omega_1 t + \epsilon_1) + L_2 \cos(\omega_2 t + \epsilon_2)$  for real  $\omega_1, \omega_2$ . For this to happen we need  $F > C^2/B$  and this condition does not involve  $A$  or  $D$ . We can then deduce the following condition for the stability of the position  $(\theta_0, \bar{y}'_G)$

$$Wk - UY > \rho g l \cos \theta_0 \left[ (x_2^2 - x_1^2) (k - c_0) \sin \theta_0 - \frac{1}{12} (x_2 - x_1)^3 \sec^3 \theta_0 \right] \quad (8)$$

This gives us for each  $H$ , which is of course the weight of water on deck, an inequality for  $k$  the depth of  $G$  below deck level.

### 2.4 Numerical Results

The numerical calculations are based on the data given for ship 'A' in a paper concerned with the capsizing of small trawlers by A. Morrall [2] although, for comparison purposes, some calculations use

the KG of ship 'B' (see [2]). This was done because it is clear from [2] that 'B' is more stable than 'A' despite the fact that their other data are broadly similar.

Ship 'A' has breadth 6.86 metres, depth 3.35 metres, length 22.09 metres and displacement 167.6 tonnes. Its KG is 3.153 metres. The KG of 'B' is 2.58 metres. Thus in our notation  $a = 3.43$ ,  $h = 3.35$ ,  $l = 22.09$ ,  $d = 1.715$  and these values are fixed throughout.

In performing the calculations it is convenient to divide both  $W$  and  $H$  by  $\rho g l$  so for 'A' with a force displacement of 167.6 tonnes,  $W$  is taken to be 7.4 and the value of  $H$  starts at zero and is increased in steps of length 0.1. It is useful to note here that  $H = 0.35$  is roughly equivalent to the case when the deck is flooded to a depth of one foot, over half its breadth and one third of its length.

The object of the calculations is to illustrate the effect on stability of both the size of  $H$  and the value of KG. The variable that indicates the value of KG is  $k = \text{depth of ship} - \text{KG}$ . In our first calculation we took the KG of ship 'B' ( $k = 0.77$ ) and decreased  $k$  (raised  $G$ ) in subsequent runs.

Before any calculations could be done a suitable shape for the transverse cross-section had to be chosen. A rectangular shape proved to be too stable to be interesting; a parabolic shape gave results that seemed much too unstable even for small  $H$  and small KG. The results here are for a half-ellipse with semi-axes  $a$  and  $h$ . The computer programme allows the deck edge to become immersed. The results clearly show the influence of  $H$  and KG on stability. In Table 1 some typical results for KG = 2.58 are given for various values of  $H$ .  $\theta_1$  and  $\theta_2$  are the approximate positions of equilibrium. In each case  $\theta_1$  proved stable and  $\theta_2$  unstable.

Table 1

H	$\theta_1^\circ$	$\theta_2^\circ$
0	0	67
.2	3	62
.4	6	59
.6	9	57

Once the stable position is reached the righting moment  $L_R$  shows a healthy domina-

nance over the heeling moment for a wide range of  $\theta$ . It seems clear that for ship 'B' the static effect of an asymmetrical weight on deck is not a significant factor in causing capsizing. The situation for ship 'A' with KG = 3.153 is rather different as is shown in Table 2.

Table 2

H	$\theta_1^\circ$	$\theta_2^\circ$
0	0	42.5
.2	8.5	40.5
.4	16.5	36.5
.6	24.5	32.5

There are no equilibrium positions at all for larger  $H$ , the heeling moment dominates over the whole range. Let us examine the case  $H = .4$  more closely. The equilibrium positions are  $16.5^\circ$  and  $36.5^\circ$ .  $L_R$  peaks at about  $30^\circ$  and thereafter drops off rapidly, vanishing at about  $41^\circ$ .  $L_R$  only dominates for about  $20^\circ$  of the range and even at its peak is only about 1.6 times the corresponding  $L_H$ . Thus it would seem that ship 'A' is worse affected by an asymmetrical deck load than 'B' and that this is because of its high KG which is dangerously close to the critical value of KG for which the purely statical effect of a moderate amount of water on deck is big enough to cause capsizing. One could therefore conclude that for a ship with such a high KG statical effects play a significant role in the bringing about of capsizing.

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## Discussion

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In some circumstances a roll motion tends to build up slowly, and the ship's master may then be able to take remedial

action; in other circumstances the roll motion may build up quickly. In particular this may happen when the falling parts are ineffective and water is skipped on deck. Static forces may then be calculated as in

Dr. Pinch's paper, but if the ship then rolls what is the magnitude of the dynamic force? For the case when the whole deck is covered some relevant calculations have recently been carried out at Manchester University by Dr. A. Hulme and Dr. A.F. Jones, two of Dr. Pinch's colleagues. They assume that a tank of rectangular section containing water is given a simple harmonic small oscillation about a horizontal axis, and that two-dimensional waves are thus set up in the tank. It was found experimentally by Van den Bosch and Vugts in 1964 that the nature of the motion varies greatly, de-

pending on the frequency. Hulme and Jones have developed a shallow-water theory involving 4 independent dimensionless parameters, some of which are usually small. The equations are hyperbolic and have been treated both analytically and numerically. It has been found that shock fronts (bores) occur in certain circumstances, and that the resonant case requires separate treatment. Preliminary results largely explain the observations of Van den Bosch and Vugts and also show that the dynamic forces may considerably exceed the static forces.

# THE PREDICTION OF DECK WETTING IN BEAM SEAS IN THE LIGHT OF RESULTS OF MODEL TESTS

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## ABSTRACT

The theoretical model of shipping water on deck in beam seas based on the linear theory and the main statistical parameters which may describe the intensity of this phenomenon have been presented.

The influence of wind and drift on the characteristics of water level motions at the ship's side and on the statistical parameters of exceeding freeboard have been taken into account.

The comprehensive model tests carried out both in a model tank and on natural wind waves on a lake have been described. A method of analysis of the experimental results and the main effects of this analysis have been presented. They have been compared with results of numerical calculations based on the presented theoretical model.

The statistical parameters for exceeding bulwark edge obtained both experimentally and theoretically are in sufficient agreement.

## NOMENCLATURE

$A_w$  - waterplane area  
 $\bar{A}_i$  - relative amplitude of reflected wave  
 $D_u$  - variance of random process "u"  
 $D_{\dot{u}}$  - variance of rate of change of process "u"  
 $F$  - height of ship's side measured from still waterline to upper bulwark edge  
 $F_o$  - freeboard height  
 $F_N$  - bulwark height  
 $F_w$  - change in draught of side due to heel  $\theta_s$   
 $\bar{g}$  - acceleration due to gravity  
 $GM_o$  - initial metacentric height  
 $\bar{h}$  - mean value of water oscillations on ship's side  
 $h(t)$  - instantaneous position of the water level on the ship's side, resulting from ship motions on the wave  
 $h_w(t)$  - relative motions on ship's side caused by

the action of the varying wind moment  
 $h_T(t)$  - resultant instantaneous water level height on ship's side  
 $\delta h_i$  - deformation of wave profile caused by the component of orbital motion in "i" direction  
 $J_{xx}$  - mass moment of inertia of ship about G-X axis  
 $K_u(\tau)$  - auto-correlation function of process "u"  
 $k$  - wave number  
 $L$  - ship length  
 $M_F(t)$  - hydrodynamic wave heeling moment  
 $M_w(t)$  - variable wind heeling moment  
 $\bar{M}_w$  - steady wind heeling moment  
 $m_{ij}$  - added mass in the direction "i" caused by motion in direction "j"  
 $m_{ij}^D$  - diffractive added mass - " -  
 $N_{ij}$  - damping coefficient - " -  
 $N_{ij}^D$  - diffractive damping coefficient - " -  
 $"N"$  - index for weather side  
 $\bar{N}_F$  - mean number of exceedances in time  $T^*$   
 $\bar{n}_F$  - mean number of exceedances in 1 second  
 $\bar{OM}$  - distance of the metacentre from the origin of the moving axes  
 $OG$  - distance of the c.g. from the origin of the moving axes  
 $P$  - probability  
 $\bar{p}$  - mean wind pressure  
 $p_o$  - wind pressure fluctuation amplitude  
 $\delta p(t)$  - pressure fluctuations about the mean value  
 $r_o$  - wave amplitude  
 $S$  - area presented to wind  
 $S_u(\omega)$  - spectral density function of process "u"  
 $S_{\zeta w}(\omega)$  - wave spectrum  
 $T$  - ship's draught  
 $T_h$  - mean period of ship's side relative motions  
 $t$  - time  
 $V_{dr}$  - drift velocity of ship  
 $\bar{V}_w$  - mean wind velocity  
 $\delta V_w(t)$  - wind velocity oscillations about the mean value  
 $"z"$  - index for lee side



$Z_F(t)$  - vertical component of the hydrodynamic wave excitation force  
 $Z_O$  - centre of drift resistance  
 $Z_S$  - centre of area presented to wind  
 $Y_F(t)$  - horizontal component of the hydrodynamic wave excitation force  
 $Y_W(t)$  - variable horizontal wind-exciting force  
 $\bar{Y}_W$  - steady horizontal wind force  
 $Y_{dr}$  - side resistance of drift  
 $y$  - half-width of transverse hull section under consideration  
 $\alpha$  - wave slope  
 $\beta_i$  - phase shift due to "i" motion relative to wave  
 $\gamma$  - specific gravity of water  
 $\Delta$  - ship displacement  
 $\delta_i(\omega)$  - response amplitude operator of motion "i" (R.A.O.)  
 $\delta_h(\omega)$  - R.A.O. of relative motions of water surface at ship's side caused by waves  
 $\delta_{hw}(\omega)$  - R.A.O. of relative water surface motions caused by wind rolling  
 $\delta_{rw}(\omega)$  - R.A.O. for wind rolling  
 $\Phi$  - undisturbed wave potential  
 $\Phi_h(i\omega)$  - transfer function for relative motions on ship's side caused by wave induced hull motions  
 $\Phi_w(i\omega)$  - transfer function for relative motions on ship's side caused by wind  
 $z_a, z_e$  - co-ordinates of the ship's centre of gravity in the stationary co-ordinate system  
 $z_o, z_e, z_p$  - hull motions amplitudes: sway, heave, roll  
 $z_w, z_w$  - co-ordinates of a wave particle undergoing orbital motions, relative to the stationary co-ordinate system  
 $\varphi$  - variable heel angle of ship  
 $\theta_s$  - static ship heel  
 $x_i$  - wave exciting force or moment coefficient taking into account the effect of the ratios of ship hull dimensions to the wave length  
 $\omega$  - angular frequency  
 $\omega_r$  - natural frequency of rolling  
 $\omega_h$  - " " of heaving  
 $\bar{\omega}_h$  - mean angular frequency of ship's side relative motions  
 $\rho$  - specific density of water  
 $\rho_a$  - " " of air  
 $\bar{T}$  - mean period of duration of single exceedance  
 $\Delta h$  - mean height of single exceedance  
 $\zeta_w(\theta)$  - undisturbed wave profile

## 1. INTRODUCTION

Opinions on the effect of the shipping of green water on ship safety are divided. However the opinion predominates that deck wetting presents a real danger to small, low freeboard ships, especially fishing vessels. Deck wetting endangers the crew working on the deck and hinders ship operations such as fishing. Moreover intensive deck wetting may lead to the accumulation of water on the deck, the weight of which may be significant relative to the ship's displacement. In such a case there may occur a significant reduction in stability followed by a capsize in waves.

These problems have been reflected in some of the paragraphs in the "Torremolinos '77" Convention on Fishing Vessel Safety.

As far as large vessels are concerned, opinions on the effect of deck wetting are various. The mass of water on deck is insignificant in relation to the displacement and loss of stability is minimal. In fact water on deck may even act as a roll stabilizer (e.g. [1]). So, although deck wetting may constitute a danger as far as structural safety is concerned (e.g. in large tankers [3]), it may be concluded that stability safety is unaffected.

However, capsize experiments recently carried out at the Hamburg Ship Model Basin indicate the necessity of verifying these conclusions. A detailed analysis of films taken during the experiments shows the significance of the submergence of the bulwark.

Bulwark and deck edge submergence may radically alter the rolling characteristic by preventing the ship from rolling back to the vertical position. This results in a phase shift between the roll and wave motions and hence alters the initial conditions prior to the arrival of the next wave group. As a result of a small restoring moment the ship may take such a long time to return to its upright position that the next large wave may capsize it.

For large ships, this effect is probably attributed to due to the hydrodynamic resistance presented by the bulwark motion rather than to the load of the water on deck.

The phenomenon is characterised by the longitudinal axis of roll shifting from the c.g. to the vicinity of deck edge about which the ship pivots. The heeling moment due to the wave is further enhanced by a moment resulting from the hydrodynamic force causing heaving (fig.1a).

A particularly dangerous condition arises when the bulwark submerges in quartering seas. The impact of a steep quartering wave on the stern causes a lateral hull movement combined with a rotation about a vertical axis (sway and yaw) coinciding with a leeward heel. If as a result of this movement the bulwark submerges further lateral motion causes the deck to "plough under" the water (fig.1b). These results an increase in the hydrodynamic resistance to the lateral motion which causes an increase in the heeling moment. If at the same time a large, steep wave moving along the hull from the stern towards midships, causes a large heaving force directed upwards, then there arises an additional heeling moment, as described above. Since the lateral stability one the wave crest is considerably reduced, the described event usually results in a capsize. Therefore, for very dynamic motions in quartering seas the submerged bulwark and deck edge can act as a pivot and severely increase the probability of capsize. Analogous events in which the vessel heeled to large angles passed safely if the bulwark and deck edge did not become submerged.

It may be seen then that in distinction from deck wetting of small ships, where loss of safety may arise from the accumulation of water on deck, large ships may encounter dangerous situations resulting from a radical

change in the motion caused by deck edge immersion. In this case it would be more correct to talk of deck-edge submergence rather than deck wetting. In both cases however, the occurrence of the phenomenon is dependent on the relative motion between bulwark edge and wave.

In the light of the observations made previously, it may be stated that the safety of every vessel may be endangered by the bulwark becoming submerged.

The submergence of the upper edge of the bulwark could then be accepted as one of the ship safety criteria\*. In order to do this it would be necessary to theoretically predict the occurrence of the phenomenon for given weather conditions, and to estimate its intensity.

A theoretical model for deck wetting in beam seas based on relative motion between wave and deck edge has in the past been prepared by the author ([4], [5]). This theoretical model was verified by correlation with model tests in regular tank waves as well as in natural waves in a lake.

This paper compares the model test results with theoretical predictions.

## 2. A MATHEMATICAL MODEL OF RELATIVE MOTIONS OF WATER SURFACE AT THE SHIP'S SIDE

The derivation of the expressions describing relative motions on the ship's side and the analysis of the problem are to be found in [5]. Here only the main steps leading to the final expressions are given.

The following assumptions were made initially:

- Wind and waves act upon the vessel simultaneously. The vessel lies beam on to the incoming waves and drifts freely. The forward velocity of the vessel is zero;
- the two-dimensional wave motion is a stochastic, stationary and ergodic process having a normal probability density distribution;
- for the considered interval of time, the mean wind velocity is constant and the wind velocity oscillations about this value are a stochastic process which is uncorrelated with the wave motions;
- the flow around the hull is potential and two-dimensional. Viscous damping forces are only significant for roll damping;
- the hull executes motions which may be described by a set of linear differential equations. Non-linear damping and restoring moments should be accounted for by a linearization technique;
- the amount of water on deck does not basically influence the motions in waves.

The axes fixed in space on the water surface  $O\xi\eta\zeta$  and ship's body axes  $OXYZ$  were chosen as shown in fig.2.

The following ship's motions equations

\* It should be noted that the bulwark-edge immersion as discussed here should not be confused with deck-edge or bulwark-edge immersion determined for static calm water conditions, hitherto in use.

were formulated in accordance with the assumptions given above:

sway:

$$\left( \frac{\Delta}{g} + m_{yy} \right) \ddot{\eta}_g + N_{yy} \dot{\eta}_g + m_{y\varphi} \ddot{\varphi} + N_{y\varphi} \dot{\varphi} = Y_F(t) + Y_w(t)$$

roll:

$$\left( 1 + m_{\varphi\varphi} \right) \ddot{\varphi} + N_{\varphi\varphi} \dot{\varphi} + m_{\varphi y} \ddot{\eta}_g + N_{\varphi y} \dot{\eta}_g + \Delta \bar{G} \varphi = M_F(t) + M_w(t) \quad (1)$$

heave:

$$\left( \frac{\Delta}{g} + m_{zz} \right) \ddot{\zeta}_g + N_{zz} \dot{\zeta}_g + \gamma A_w \zeta_g = Z_F(t)$$

The assumed linearity of the dynamics and the assumed independence of waves on the instantaneous wind speed make possible the application of superposition and the separation of motions and deck wetting caused by wave and by wind.

### 2.1 The Oscillations of the Water Level on Ship's Side, Caused by Regular Beam Waves

The hydrodynamic exciting forces and moments in equation (1) are given as follows:

$$Y_F(t) = \chi_{\eta} \frac{\Delta}{g} \ddot{\eta}_w + m_{yy} \ddot{\eta}_w + N_{yy} \dot{\eta}_w + m_{y\varphi} \ddot{\alpha} + N_{y\varphi} \dot{\alpha}$$

$$M_F(t) = \chi_{\varphi} \Delta \bar{G} \alpha + m_{\varphi\varphi} \ddot{\alpha} + N_{\varphi\varphi} \dot{\alpha} + m_{\varphi y} \ddot{\eta}_w + N_{\varphi y} \dot{\eta}_w + \bar{O} \bar{G} Y_F(t) \quad (2)$$

$$Z_F(t) = \chi_{\zeta} \gamma A_w \zeta_w + m_{zz} \ddot{\zeta}_w + N_{zz} \dot{\zeta}_w$$

The wave velocity and acceleration components appearing in the above equation were derived from the well known relations:

$$\zeta_w = \frac{1}{g} \frac{\partial \Phi^*}{\partial t} \Big|_{z=0}; \quad \dot{\zeta}_w = \frac{\partial \Phi^*}{\partial \eta} \Big|_{z=0}; \quad \alpha = \frac{\partial \zeta_w}{\partial \eta} \quad (3)$$

where the velocity potential  $\Phi^*$  of the undisturbed flow is given by the following form:

$$\Phi^* = \frac{i \tau g}{\omega} \exp[-k\zeta + i(k\eta - \omega t)] \quad (4)$$

Assuming that the added mass and damping coefficients appearing in equations (1) and the corresponding diffractive added mass and damping coefficients in (2) are known, it is possible to solve (1).

Since the steady-state motions are of interest, particular integrals of the following form are sought:

$$\varphi = \varphi_0 \cos(\omega t + \beta_\varphi) \\ \eta_g = \eta_0 \cos(\omega t + \beta_\eta) \quad \zeta_g = \zeta_0 \cos(\omega t + \beta_\zeta) \quad (5)$$

The solution of (1) gives the amplitudes  $\varphi_0$ ,  $\eta_0$ ,  $\zeta_0$ , and the phases  $\beta_\varphi$ ,  $\beta_\eta$ ,  $\beta_\zeta$ .

The instantaneous water level  $h(t)$  on the ship's side is the result of the hull motions, the wave motion and the distortion of the wave on the ship's side. The method for determining the water level on the ship's side is shown on fig.2. Let us consider for example, the lee side.

The water level height  $h_z = ZZ'$  relative to the calm water waterline (point Z) in the position "1" is given by

$$h_z(t) = \frac{h_z^*(t)}{\cos \varphi} \quad \text{where} \quad h_z^*(t) = \zeta_w(t) \Big|_{\eta_z} - \zeta_z(t) \quad (6)$$

The vertical ordinate for Z is:

$$\zeta_z = \zeta_g + y \sin \varphi - \bar{O} \bar{G} \cos \varphi = \bar{O} \bar{G} + \zeta_g + y \sin \varphi - \bar{O} \bar{G} \cos \varphi$$

and the abscissa of  $Z'$  is :

$$\eta_z = \eta_z - Z'Z \sin \varphi = \eta_0 + y \cos \varphi + \overline{OG} \sin \varphi - Z'Z \sin \varphi$$

Simplifying in accordance with the linear theory gives:

$$\eta_z \approx \eta_0 + y \quad \quad \quad \zeta_z \approx \zeta_0 + y \varphi \quad (7)$$

$$h_z(t) \approx h_z^*(t)$$

By taking into account the distortion of the wave profile  $\delta \zeta_w$  at the position having abscissa  $\eta_z$ , the water level on the lee side may be expressed by :

$$h_z(t) = \zeta_w(t, \eta_z) + \delta \zeta_w(t, \eta_z) - \zeta_0(t) - y \varphi(t) \quad (8)$$

where the functions  $\zeta_0(t)$  and  $\varphi(t)$  are given by (5).

The time dependent co-ordinate  $\eta_z$  shows the direct influence of sway on the ship's side water level height, resulting from the hull's change of position relative to the wave profile.

The resulting wave profile may be expressed in terms of the body axes by:

$$\zeta_w(t) = r_0 \cos[k\eta_0 \cos(\omega t + \beta_2) + ky - \omega t] \quad (9)$$

As can be seen the regular wave ceases to be harmonic in the moving co-ordinate system, since the argument of the cosine function is another time dependent harmonic function.

Further analysis showed however that the difference between the wave profile in the stationary co-ordinate system and the apparently distorted profile relative to body axes, is very small for waves having a high wave length to wave height ratio and may therefore be ignored in computations. Therefore, the direct effect of sway may be ignored and it may be assumed that  $\eta_z \approx y$ .

The determination of wave profile distortion in a form which may be applied analytically is difficult. In order to determine the resulting deformed wave profile on the ship's side it is necessary to determine the resultant velocity potential  $\Phi_v$  which is the sum of the radiation potential  $\Phi_r$ , the undisturbed wave potential  $\Phi^*$  and the diffraction potential  $\Phi_d$ . It is then possible to find the amplitude and phase of the deformed wave profile from the free surface boundary condition :

$$(\zeta_w + \delta \zeta_w) = \frac{1}{g} \left. \frac{\partial \Phi_v(z, y)}{\partial t} \right|_{z=0} \quad (10)$$

Hitherto the resultant potential  $\Phi_v$  for a hull rolling in waves could only be determined numerically. (Computations of wave profile distortion based on (10) using the multipole potential method has later carried out Dudziak [2]). However, it seems that if the calm water added mass and damping coefficients and diffraction coefficients are known, then it would be possible to formulate an analytical expression for the wave profile deformation, based on existing theoretical papers.

A hull oscillating on the calm water surface generates a system of progressive waves travelling to infinity (damping), and local disturbances taking the form of standing waves (added mass) [8].

The well-known relationship between the

amplitude of the generated progressive wave and the motion damping (per unit length) is following :

$$N_{ii}' = \frac{\rho g^2}{\omega^3} \bar{A}_{ii}^2 \quad (11)$$

If the reverse situation is now considered in which the progressive wave meets a stationary hull then as a result of the impact and deformation of the wave, a similar phenomenon to that described previously, occurs i.e. local disturbance and a reflected progressive wave. The quantitative relations between the parameters of these systems of waves and the respective diffractive elements of the wave exciting force must be the same as before, since the physical nature of the phenomenon is the same ([7], [9]).

This paper has described that part of the deformation which is considered to dominate i.e. the deflection and reflection of the incoming exciting wave. The diffractive damping of the wave exciting force is related to this phenomenon.

Resolving the orbital motion of the particles of the exciting waves into two components in the directions of the axis  $O-\eta$  and  $O-\zeta$ , and making use of the dependence (11), the relative amplitudes of the reflective waves may be defined as :

$$\bar{A}_1 = \frac{\delta h_1}{r_0} = \sqrt{\chi \frac{N_{11}' \omega^3}{g^2 \rho}} \quad ; \quad \bar{A}_2 = \frac{\delta h_2}{r_0} = \sqrt{\chi \frac{N_{22}' \omega^3}{g^2 \rho}} \quad (12)$$

where according to [10] is taken :

$$N_{ii}^D = \chi N_{ii} \quad ; \quad \chi = e^{-kT} \frac{\sin \frac{2\pi y}{\lambda}}{\frac{2\pi y}{\lambda}} \quad (13)$$

The generation of reflected waves is directly related to the wave particle velocity. It has been therefore assumed that the maximum values  $\delta h_1$  and  $\delta h_2$  occur at the moment when they attain the maxima of  $\zeta_w$  and  $\eta_w$ . Taking into account in addition, that in view of the reflection phenomenon, the sign of the deformation should be opposite to that of the component velocities in the undisturbed wave, the phase and general form of the reflected wave is taken to be as follows :

$$\delta h_1 = r_0 \bar{A}_1 \cos[(k\eta - \omega t) + \frac{\pi}{2}]$$

$$\delta h_2 = \begin{cases} r_0 \bar{A}_2 \cos(k\eta - \omega t) & \text{for the weather side} \\ -r_0 \bar{A}_2 \cos(k\eta - \omega t) & \text{for the lee side} \end{cases} \quad (14)$$

The height of water level on the lee side may now be written as follows :

$$h_z(t) = r_0 \cos(ky - \omega t) + r_0 \bar{A}_1 \cos(ky - \omega t + \frac{\pi}{2}) + -r_0 \bar{A}_2 \cos(ky - \omega t) - \zeta_0 \cos(\omega t + \beta_1) - y \varphi \cos(\omega t + \beta_0) \quad (15)$$

The above function called next as "the function of water oscillations on the lee side" is the sum of harmonic functions having the same frequency  $\omega$  and may therefore be written as

$$h_z(t) = r_0 \delta_{hz} \cos(\omega t + \beta_z) \quad (16)$$

where the amplification factor  $\delta_{hz}$  and phase  $\beta_z$  are determined from :

$$\delta_{hz}(\omega) = [\gamma^2 \delta_\phi^2 + \delta_\xi^2 + \bar{A}_\xi^2 + (1 - \bar{A}_\eta)^2 + 2\gamma \delta_\phi \delta_\xi \cos(\beta_\phi - \beta_\xi) + 2\gamma \delta_\phi \bar{A}_\xi \sin(\beta_\phi + k\gamma) - 2\gamma \delta_\phi (1 - \bar{A}_\eta) \cos(\beta_\phi + k\gamma) + 2\delta_\xi \bar{A}_\xi \sin(\beta_\xi + k\gamma) - 2\delta_\xi (1 - \bar{A}_\eta) \cos(\beta_\xi + k\gamma)]^{\frac{1}{2}} \quad (17)$$

$$\operatorname{tg} \beta_z = \frac{-\gamma \delta_\phi \sin \beta_\phi - \delta_\xi \sin \beta_\xi - \bar{A}_\xi \cos k\gamma - (1 - \bar{A}_\eta) \sin k\gamma}{-\gamma \delta_\phi \cos \beta_\phi - \delta_\xi \cos \beta_\xi - \bar{A}_\xi \sin k\gamma + (1 - \bar{A}_\eta) \cos k\gamma} \quad (18)$$

Similarly, the function describing water oscillations on the weather side has been derived:

$$h_N(t) = r_0 \delta_{hN} \cos(\omega t + \beta_N) \quad (19)$$

$$\delta_{hN}(\omega) = [\gamma^2 \delta_\phi^2 + \delta_\xi^2 + \bar{A}_\xi^2 + (1 + \bar{A}_\eta)^2 - 2\gamma \delta_\phi \delta_\xi \cos(\beta_\phi - \beta_\xi) - 2\gamma \delta_\phi \bar{A}_\xi \sin(\beta_\phi - k\gamma) + 2\gamma \delta_\phi (1 + \bar{A}_\eta) \cos(\beta_\phi - k\gamma) + 2\delta_\xi \bar{A}_\xi \sin(\beta_\xi - k\gamma) - 2\delta_\xi (1 + \bar{A}_\eta) \cos(\beta_\xi - k\gamma)]^{\frac{1}{2}} \quad (20)$$

$$\operatorname{tg} \beta_N(\omega) = \frac{\gamma \delta_\phi \sin \beta_\phi - \delta_\xi \sin \beta_\xi - \bar{A}_\xi \cos k\gamma + (1 + \bar{A}_\eta) \sin k\gamma}{\gamma \delta_\phi \cos \beta_\phi - \delta_\xi \cos \beta_\xi - \bar{A}_\xi \sin k\gamma + (1 + \bar{A}_\eta) \cos k\gamma} \quad (21)$$

## 2.2 The Influence of Wind on the Oscillations of Water Level on the Ship's Side

The influence of the wind is characterized by changes in the ship's response.

In accordance with the assumptions, the instantaneous wind velocity may be considered to be the sum of a constant mean velocity  $\bar{V}_w$  and the velocity pulsations  $\delta V_w(t)$  about the mean.

The mean pressure  $\bar{p}$  (corresponding to  $\bar{V}_w$ ) results in a constant horizontal force  $\bar{Y}_w$  causing drift and a constant heeling moment  $\bar{M}_w$ .

$$\bar{Y}_w = k_w \bar{p} S = k_w S \frac{\rho_p \bar{V}_w^2}{2} \quad (22)$$

$$\bar{M}_w = k_w \bar{p} S (z_s - z_0) \quad (23)$$

The drift is resisted by a side force  $Y_{dr}$ :

$$Y_{dr} = k_s L T \frac{\rho V_{dr}^2}{2} \quad (24)$$

The drift velocity may be determined by equating (22) and (24):

$$V_{dr} = \bar{V}_w \sqrt{\frac{\rho_p k_w S}{\rho k_s L T}} \quad (25)$$

The non-dimensional coefficients  $k_w$  and  $k_s$  are usually determined experimentally.

The influence of drift on rolling and consequently on the characteristics of water oscillations on the ship's side, appears as a change in the excitation frequency:

$$\omega_e = \omega - k V_{dr}$$

The moment  $\bar{M}_w$  causes a static heel  $\Theta_s$  towards the leeward side,

$$\Theta_s = \frac{\bar{M}_w}{\Delta g \bar{M}_s} \quad (26)$$

about which the ship rolls.

Consequently the mean draught on each side changes by:

$$F_w = y \Theta_s \quad (27)$$

i.e. decreases on the weather side and increases on the lee side.

The pressure pulsations  $\delta p(t)$  (corresponding to  $\delta V_w(t)$ ) causes a time varying aerodynamic horizontal force:

$$\delta Y_w(t) = k_w S \delta p(t) \quad (28)$$

and moment:

$$\delta M_w(t) = k_w S (z_s - \bar{KG}) \delta p(t) \quad (29)$$

The varying moment  $\delta M_w(t)$  causes rolling which is similar in theory to that caused by waves. Assuming that the pressure pulsations about the mean value are a random continuous stationary process, it is possible to express them as a harmonic series. The rolling characteristic may then be determined by solving the equations of motion (1) and expressing the exciting forces and moments by (28) and (29) while assuming the wave excitation to be zero.

As a result, the amplification factor for wind excited rolling  $\delta_{rw}$  and the amplification factor  $\delta_{zw}$  for horizontal oscillations are obtained.

In view of the great inertia of the vessel as well as the large side force  $Y_{dr}$  relative to the small values and short duration of the force  $\delta Y_w(t)$ , the sway caused by wind fluctuations is negligible.

The wind rolling will cause symmetric water level oscillations on the ship's side, given by:

$$h_w(t) = y \rho_s \delta_{rw}(\omega) \sin(\omega t + \epsilon_\rho) \quad (30)$$

where the R.A.O. for these oscillations per amplitude of exciting wind pressure, is given by:

$$\delta_{rw}(\omega) = y \delta_{rw}(\omega) \quad (31)$$

## 2.3 Water Oscillations on the Ship's Side in Natural Conditions

In natural conditions, the wind and waves act on the ship simultaneously. The resultant water level on the ship's side  $h_r(t)$  is the sum of wind oscillations and wave oscillations:

$$h_{rN}(t) = h_z(t) + h_w(t) \pm y \Theta_s \quad (32)$$

where the index "N" - indicates the weather side and "Z" - the lee side.

Since the waves and wind are continuous random stationary processes and the ship motions are described by a set of linear differential equations, then the function describing water level oscillation on the ship's side, is also a continuous random, stationary function.

For known statistical characteristics for waves  $\zeta_w(t)$  and wind  $\delta p(t)$  it is possible to determine the statistical characteristics of the ship's side water oscillations on condition that their transfer function is known.

The water level oscillation can be described in the following complex form:

$$H_{rN}(t) = \Phi_{hN}(i\omega) \zeta_w(t) + \Phi_w(i\omega) \delta p(t) \pm y \Theta_s \quad (33)$$

where the wave induced water level oscillation transfer function has the form:

$$\Phi_h(i\omega) = \delta_h(\omega) e^{-i\beta} \quad (34)$$

and the wind induced transfer function is given by :

$$\Phi_w(i\omega) = \delta_{hw}(\omega) e^{-i\varepsilon_\varphi} \quad (35)$$

The response amplitude operator (R.A.O.) for wave-induced oscillation  $\delta_h(\omega)$  is determined from (17) for the lee side and from (20) for the weather side. The respective phase characteristics are given by (18) and (21). The R.A.O. for wind induced oscillations is given by (31).

The variances of the water level oscillations ( $D_T$ ) and of the velocity of oscillations ( $D_{\dot{T}}$ ) are given by :

$$D_{T_z} = \int_0^\infty \delta_{h_z}(\omega) d\omega + \int_0^\infty \delta_{hw}(\omega) d\omega = D_{h_z} + D_{hw} \quad (36)$$

$$D_{\dot{T}_z} = \int_0^\infty \omega^2 \delta_{h_z}(\omega) d\omega + \int_0^\infty \omega^2 \delta_{hw}(\omega) d\omega = D_{\dot{h}_z} + D_{\dot{h}_w} \quad (37)$$

where the wave induced oscillation spectrum is given by :

$$\delta_{h_z}(\omega) = [\delta_{h_z}(\omega)]^2 S_z(\omega) \quad (38)$$

and the wind induced spectrum :

$$\delta_{hw}(\omega) = [\delta_{hw}(\omega)]^2 S_p(\omega) \quad (39)$$

In reality, the varying wind induced heeling moment is very small compared to the moment of inertia of the ship and wave excitation moment. Consequently, the amplitudes of wind rolling are very small and their effect on the ship's side water oscillations and their variance are negligible. It can therefore be neglected in practical calculations and only wave induced oscillation variance be considered :

$$D_T \approx D_h = \int_0^\infty S_h(\omega) d\omega ; D_{\dot{T}} \approx D_{\dot{h}} = \int_0^\infty \omega^2 S_h(\omega) d\omega \quad (40)$$

As in the case of waves, the probability density of instantaneous values of the oscillation function  $h(t)$  is normally distributed whereas the amplitudes of the oscillations are a Rayleigh distribution. The probabilistic properties of the function  $h(t)$  are shown on fig.3 .

By determining the variances  $D_h$  and  $D_{\dot{h}}$  for each side it is possible to obtain all the statistical characteristics of the relative motions on ship's sides.

### 3. STATISTICAL CHARACTERISTICS OF DECK

#### WETTING

It has been assumed that deck wetting occurs when the water level rises above the upper edge of the bulwark. On the basis of this definition the evaluating of shipping water on deck resolves itself to the determination of the statistical characteristics of the exceedances of a level  $F$  by a random function  $h(t)$  (see fig.4) :

$$h(t) > F$$

The derivation of the parameters characterizing the intensity of wetting and a discussion of them may be found in [6]. The basic characteristics are following :

#### The probability of deck wetting

- a) assuming that deck wetting occurs when the value of function  $h(t)$  exceeds the height of the freeboard (from normal distribution) :

$$P_\tau = P(h > F_0 \pm F_w) = \frac{1}{2} - \Phi\left(\frac{F_0 \pm F_w}{\sqrt{D_T}}\right) \quad (41)$$

where:  $\Phi(x) = \frac{1}{\sqrt{2\pi}} \int_0^x e^{-\frac{t^2}{2}} dt$  - Laplace integral function,

variance  $D_T$  - calculated by (36) or (40) respectively for the weather or lee side,

$F_w$  - is taken with "+" for the weather side and "-" for the lee side.

The quantity  $P_\tau$  can be interpreted as the ratio of the total duration of the exceedances to the total duration of the process (fig.4) :

$$P_\tau = \frac{\sum_{i=1}^N \tau_i}{T^*} \quad (42)$$

- b) assuming that the deck wetting takes place when the amplitude  $h_0$  of the water level oscillations exceeds the height of the freeboard (from Rayleigh's distribution) :

$$P_F = P(h_0 > F_0 \pm F_w) = \exp\left[-\frac{(F_0 \pm F_w)^2}{2D_T}\right] \quad (43)$$

The probability may be interpreted as the ratio of the number of exceedances to the total number of amplitudes  $h_0$ , during the time interval  $T^*$  :

$$P_F = \frac{n}{N} \quad (44)$$

Mean exceedance height

$$\bar{\Delta h} = \sqrt{\frac{\pi}{2} D_T} \left[1 - 2\Phi\left(\frac{F_0 \pm F_w}{\sqrt{D_T}}\right)\right] \exp\left(\frac{(F_0 \pm F_w)^2}{2D_T}\right) \quad (45)$$

Mean number of exceedances in time  $T^*$

$$\bar{N}_F = \frac{\bar{\omega}_h T^*}{2\pi} \exp\left[-\frac{(F_0 \pm F_w)^2}{2D_T}\right] \quad (46)$$

where

$$\bar{\omega}_h = \sqrt{\frac{D_{\dot{h}}}{D_T}} \quad \text{- mean frequency of water oscillations on the ship's side}$$

The probability, that  $n$  exceedances occur in time  $T^*$  :

$$P_n = P(n|T^*) = \frac{\bar{N}_F^n}{n!} e^{-\bar{N}_F} \quad (47)$$

where  $\bar{N}_F$  is determined from (46).

Mean duration of a single deck wetting event :

$$\bar{\tau} = \frac{\pi}{\bar{\omega}_h} \left[1 - 2\Phi\left(\frac{F_0 \pm F_w}{\sqrt{D_T}}\right)\right] \exp\left(\frac{(F_0 \pm F_w)^2}{2D_T}\right) \quad (48)$$

The probability that the deck wetting period is greater than  $t^*$  :

$$P_t = P(\tau > t^*) = \exp\left(-\frac{t^*}{\bar{\tau}}\right) \quad (49)$$

where the mean duration of deck wetting  $\bar{\tau}$  is given by (48).

All the discussed statistical exceedance parameters are functions of the freeboard height  $F$ , as well as being dependent on the dynamic characteristics of the ship ( $\delta_h(\omega)$ ) and on the wave and wind spectra  $S_{\eta}(\omega)$ ,  $S_p(\omega)$  (which is expressed by the variances  $D_{\tau}$  and  $D_{\ddot{\tau}}$ ).

For a particular ship and for a particular sea state the exceedance parameters are directly related to each other.

The relations between three main ones ( $\bar{\Delta}h$ ,  $\bar{\tau}$ ,  $\bar{N}_F$ ) are as follows:

$$\bar{\Delta}h = \frac{P_z}{P_F} \sqrt{2\pi D_{\tau}} \quad (50) \quad \bar{N}_F = \frac{\bar{\tau}}{T_h} P_F \quad (51)$$

$$\bar{\tau} = \frac{P_z}{P_F} \bar{\tau}_h \quad (52) \quad \bar{\Delta}h = \sqrt{\frac{D_{\ddot{\tau}}}{2\pi}} \bar{\tau} \quad (53)$$

$$\text{where } \bar{\tau}_h = \frac{2\pi}{\omega_h}$$

These relations confirm the intuitive understanding of this phenomenon.

The most interesting relation is the common dependency of the mean duration  $\bar{\tau}$  of a single exceedance and the mean exceedance height  $\bar{\Delta}h$ , since the danger of the bulwark going under water and water on deck is governed by these two parameters.

As can be seen from (53), the mean exceedance height is directly proportional to the mean period of wetting. The magnitude of the coefficient of proportionality is governed by the variance of the velocity of water oscillations on the ship's side  $D_{\ddot{\tau}}$ . This is affected by the character of the oscillation's transfer function  $\delta_h(\omega)$  as well as by the wave spectrum  $S_{\eta}(\omega)$  and by their relative positions.

The more the maxima of  $\delta_h(\omega)$  and  $S_{\eta}(\omega)$  are shifted to the higher frequencies, the greater  $D_{\ddot{\tau}}$ . This means that if more energy is carried by the high frequency waves then, for a certain mean duration of a single deck wetting, the exceedance heights are greater.

For a given spectrum, the exceedances will be the higher, the greater the ship's tendency to execute high frequency motions i.e. the greater the ship's "stiffness".

#### 4. MODEL TESTS

Model tests on ship motions and deck wetting were carried out at the Ship Research Institute of the Technical University Gdańsk, both on regular waves in towing tank, and in natural wind-wave conditions on a lake.

A typical low-freeboard side trawler, type B-14, which is used by the Polish fishing fleet, was selected. The main parameters for the 1:25 model were:

$\Delta = 76.0$ kg	$C_B = 0.56$	$\overline{KG} = 15.0$ ;	15.42 cm
$I_{pp} = 215.4$ cm	$C_p = 0.63$	$\overline{GM}_O = 2.68$ ;	2.26 "
$B = 36.0$ "	$C_{wp} = 0.80$	$J_{xx} = 12.2$ ;	13.5 kgcmsec <sup>2</sup>
$T = 17.3$ "	$C_M = 0.88$	$\overline{\omega_{xx}}/B = 0.348$ ;	0.365
$H = 19.8$ "	$\overline{FM}_O = 7.5$ cm	$\overline{\omega_{\ddot{\tau}}}_B = 3.9$ ;	3.43 sec <sup>-1</sup>
$F_N = 4.4$ "	$\overline{KM}_O = 17.68$ "	$T_{\ddot{\tau}} = 1.61$ ;	1.84 sec
$F_O = 2.5$ "	$S = 3010$ cm <sup>2</sup>		
$F_1 = 3.5$ "	$Z_S = 22.9$ cm		
$F_2 = 4.5$ "	$\omega_{\ddot{\tau}} = 6.23$ sec <sup>-1</sup>		

Those parts of the model above the water were made according to the full scale form of the hull with the forecastle, poop, deckhouse and the bulwarks with freeing ports (photo 1).

In order to investigate the effect of the freeboard  $F$  on the deck wetting characteristics, the freeboard could be altered by 40 % and 80 %, relative to the built height  $F_O$ , without changing the displacement, position of the centre of gravity  $\overline{KG}$  or the mass moment of inertia  $J_{xx}$ .

#### 4.1 Regular Wave Tests

The waves generated in the tank were regular, cylindrical and having a more or less sinusoidal profile. The waves travelled parallel to the towing tank's sides.

The model was positioned beam-on to the incoming waves and was free to oscillate and drift.

The following values were continuously measured and recorded:

- wave profile  $\zeta_N(t)$  in fixed axes, in front of the model,
- roll  $\varphi(t)$  by means of a gyroscope positioned inside the model,
- vertical  $\ddot{z}(t)$  and horizontal  $\ddot{y}(t)$  accelerations of the model's centre of gravity, by means of accelerometers fixed inside the hull, near to the c.g.,
- water oscillations on the sides at midships ( $h_N(t)$  and  $h_Z(t)$ ), by means of a probe mounted on each side,
- mean drift speed.

The test program included measurements for both values of  $\overline{KG}$  and for all three freeboard heights  $F_O$ ,  $F_1$  and  $F_2$ .

The wave parameters included both the roll resonance frequency as well as the heave resonance frequency, in the range of steepnesses  $h/\lambda = 1/10 \div 1/60$ .

The ratio of wave amplitudes to the freeboard height was from  $r_O/F = 0.8 \div 2.8$  for  $F_O$  to  $r_O/F = 0.5 \div 1.5$  for  $F_2$ . With the bulwark included this was:  $r_O/F = 0.3 \div 1.1$  for  $(F_O + F_N)$  and  $r_O/F = 0.25 \div 0.8$  for  $(F_2 + F_N)$ .

Therefore, the program practically included all the parameters which govern deck wetting.

The R.A.O.'s determined from these measurements for roll, heave and water oscillations on the ship's side were plotted as points on fig. 8 ÷ 13.

#### 4.2 Model Tests in Natural Irregular Waves and Wind

The tests were carried out at the research station by the lake Jeziorak.

The same model was used with the same instrumentation as in the tank tests.

The measurements were taken from self-propelled catamaran, specially adapted for seakeeping experiments.

The electrical generators, recorders and experimenters were located on the catamaran.

The model was able to drift freely beam-on to the main direction of incoming waves,

in an undisturbed region of waves. The model could move in waves in six degrees of freedom and the signals from the measuring devices were carried to the catamaran by loosely hanging cables (photo 1). The model motions and water level oscillations on its sides were measured identically as in the regular wave tests.

Moreover, the mean  $\bar{V}_w$  and instantaneous  $V_w(t)$  wind velocities were measured at a height  $\sim 1.2$  m above the water level.

Measurements were carried out for various intensities of irregular waves motion such that the ratio of the significant wave amplitude to the freeboard height lay in the range:  $\bar{h}_s/2F = 1.12 \div 3.28$ , or including the bulwark height:  $\bar{h}_s/2(F+F_N) = 0.49 \div 1.19$ .

The duration of each individual measurement was long enough to be able to determine the statistical characteristics of the processes being measured (duration of measurement approx. 12 minutes, on average over 400 rolls of the model). However, the measurements were short enough for the recorded random functions to be stationary.

An example of the recorded signals is given on fig. 5.

The spectral analysis and statistical analysis of the random processes was carried out by the analogue machine ISAC-NORATOM. As a result the following characteristics were obtained for every measured process "U"

- auto-correlation function  $K_u(\tau)$ ,
- spectral density function  $S_u(n)$ , where  $n = \omega/2\pi$ ,
- distribution function of the instantaneous values  $F(u)$ .

An example of the results obtained from the analog machine is shown in fig. 6.

On the basis of these results the following was obtained:

- the variance of the process, as the value of the auto-correlation function for  $\tau = 0$ :  $D_u = K_u(0)$ ,
- the variance of the rate of change of the process, from the spectral density distribution:  $D_{\dot{u}} = \int_0^\infty \omega^2 S_u(\omega) d\omega$ ,
- mean values and the probabilities of exceeding certain values of a process - directly from the distribution of instantaneous values.

On the basis of the obtained spectral densities, the R.A.O.'s was determined from the relation:

$$\delta_u(\omega) = \sqrt{\frac{S_u(\omega)}{S_{\dot{u}}(\omega)}} \quad (54)$$

The roll and heave R.A.O.'s obtained by this method are shown as thin lines on fig. 8, 9, and the ship's side water level oscillations for both  $\bar{K}G$  values - on figs. 10 + 13.

Moreover fig. 7 shows the spectral densities of the measured quantities for the purpose of comparing the common relationship between them.

## 5. RESULTS OF COMPUTATIONS AND OF MODEL TESTS

An algorithm and program was prepared on the basis of the theoretical relations given in sections 2 and 3. This program computes the ship motions on beam seas,

relative water motions on the ship's side and the variance and statistical exceedance characteristics for an input wave spectrum. The wind effect is taken into account by calculating the drift velocity  $V_{dr}$  and the static heel  $\theta_s$ .

The hydrodynamic mass coefficients  $m_{ij}$  and  $N_{ij}$  were calculated by using SCORES [10]. The non-linear roll damping was taken into account by adding a viscous damping factor corresponding to the mean resonance amplitudes.

This program was used to compute transfer functions and relative motions for the model used in model tests. The computations were carried out for both positions of centre of gravity ( $\bar{K}G = 15.0$  cm and  $\bar{K}G = 15.42$  cm); with and without drift; with and without wave deformations; with all geometrical and loading parameters the same as for the model.

The computed R.A.O.'s for heave and roll for  $\bar{K}G = 15.0$  cm are shown in figs. 8, 9. Plotted over these are the R.A.O.'s obtained from both regular and irregular wave tests (for  $\bar{K}G = 15.42$  the shape of the curves and agreement of results are similar).

Generally it can be said that the formulated equations describe the hull motions in beam seas accurately. This gives a basis for checking the computations of water surface relative motions.

The relative motion R.A.O.'s for model tests and numerical computations are given on figs. 10 + 13 for both  $\bar{K}G$  positions.

The following may be noted:

- the comparison between results obtained from the regular wave tests and those obtained by means of spectral analysis from tests in natural irregular waves shows remarkably good agreement.
- The theoretically computed R.A.O.'s also agree with the measured results. The conformity of these results is improved, particularly for the lee side, by taking into account wave deformations.
- The R.A.O. for relative motions on the weather side has two distinct maxima: one in the roll resonance region, the other in the heave resonance region;
- for the lee side the R.A.O. has only one maximum in the roll resonance region. The magnitude of this maximum is close to that for the weather side. There is no second maximum in the short wave region. This is caused by phase shifts of particular hull motions components for the lee side and by the damping out of short waves on the leeward side.

The difference between the characteristics for the weather and lee side is distinctly shown on fig. 7, where the wind, model motion and relative water motion spectra for each side were plotted for the measurement 12. This is typical for results obtained from the tests.

As can be seen the greater deck wetting danger is decidedly on the weather side. This is also confirmed by the fact that the waves move towards the hull on the windward side, thereby favouring the conditions for wetting resulting from exceedance. This is not the



case on the leeward side, on which the waves move away from the hull.

The described in section 4 program was used to compute the variance of relative motions as well as the statistical parameters of freeboard and bulwark exceedance, for all measurements taken during the wind wave tests. Each time the measured real wave spectrum and real wind speed were input to the program. The effect of wind pressure fluctuations was not taken into account because these would have been low frequency oscillating lying outside the frequencies significant for the model (see fig. 7).

The results of the computations with the wave deformation included ( $A_{23} \neq 0$ ) and without its inclusion ( $A_{23} = 0$ ), are shown with respective results obtained from measurements, on table 1.

As can be seen, the variances  $D_h$  and  $D_B$  are in good agreement with the real variances obtained from the measurements. Larger differences occur for some of the statistical parameters of bulwark edge exceedance. This results from the values of these parameters being small (the differences are much smaller for low values of  $F$  e.g. for a freeboard without a bulwark). In spite of this the results obtained are comparable with the results of the measurements and for all practical purposes the calculation method may be recognized as sufficiently accurate.

## 6. CONCLUSIONS

The theoretical considerations, numerical computations and the results of model tests carried out for medium to heavy sea states (with respect to the ship type and scale), make possible the following conclusions :

- 1) The relative water motions on the sides of a ship rolling in beam seas can be described sufficiently well by linear theory in cases where the deck wetting is not too intensive.  
With regard to rolling, non-linear effects can be accounted for by linearization techniques.
- 2) The good correlation between model test results in deterministic i.e. regular wave conditions and irregular wave test results obtained by spectral analysis, indicates that the identification methods of linear dynamic systems may be used for determining seakeeping qualities with respect to relative water motions on the ship's sides.  
In other words, the transfer functions obtained for deterministic conditions (either experimentally in regular waves or numerically) give a basis for predicting relative water motions in irregular waves.
- 3) The application of theoretically obtained relative motion transfer functions to the calculation of statistical exceedance parameters, sometimes gives results differing from those obtained directly from irregular wave model tests. The results are however comparable and the tendencies

shown are as those shown by the majority of measured values. Taking into account the stochastic nature of the investigated process and the prognostic character of the computed exceedance parameters, the theoretical results can be accepted as being sufficiently correct for practical purposes.

- 4) The relative water surface motions on the weather side have a decidedly more intensive character. The magnitude of the variance of this process is governed by the heave and roll characteristics (and, of course, by the wave motion intensity).

The variance for the lee side is considerably smaller because in this case only the region of roll resonance is the decisive factor.

Therefore the statistical characteristics of the exceedances for the weather side can be considered as a measure of the intensity of deck wetting. The lee side, on the other hand, is more relevant for investigations into the danger caused by the bulwark and deck edge submergence. The corresponding statistical characteristics for this side constitute a measure of the degree of immersion of the bulwark edge.

Considering that :

- the statistical characteristics of the exceedances, such as the probability of exceedance  $P_F$ , mean exceedance height  $\Delta h$  and mean exceedance period  $\bar{\tau}$ , which are in agreement with an intuitive physical interpretation, can be used to estimate the seakeeping qualities of a ship with respect to deck wetting and deck edge immersion of the leeward side;
  - the previous conclusions confirm the correctness of the calculation of these parameters, for a given sea state, by theoretically obtained transfer functions;
  - it is possible to develop the theoretical model presented herein for the case of a ship moving at any angle to the waves;
  - there undoubtedly exists a connection between deck wetting and deck edge immersion, and the danger of stability loss;
- the statistical characteristics of the water level exceeding the bulwark edge could be taken as one of the ship safety criteria.

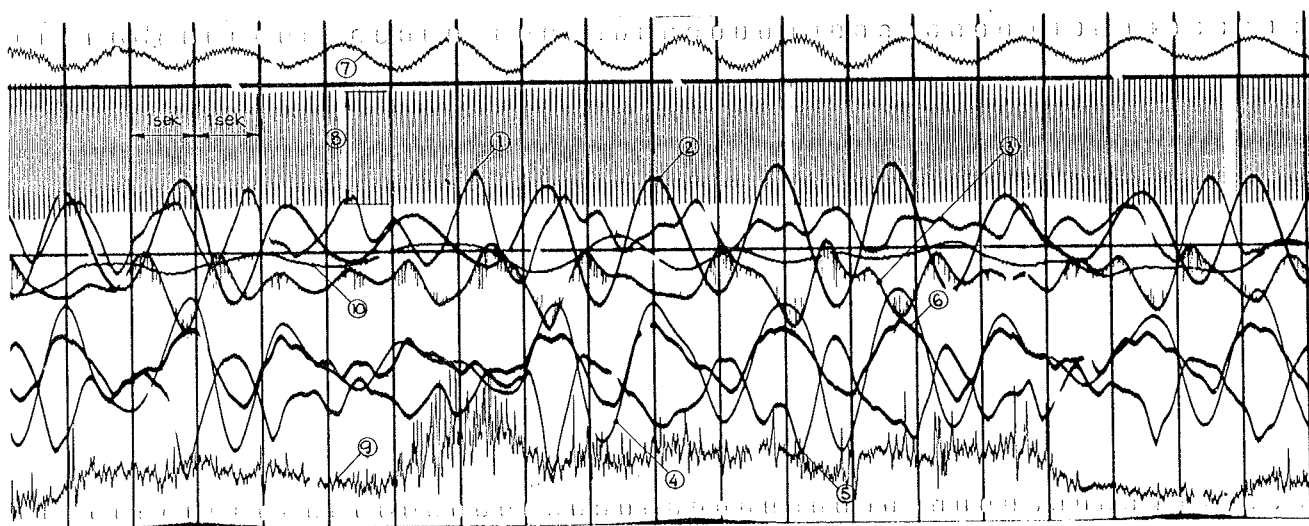
The exceedance of certain maximum values of these parameters for certain weather conditions, would then be considered as endangering the stability of the ship.

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|--|---|
| ① — Waves in the mobile system ( $L_z(t)$ )                                | ⑦ — Vertical acceleration of wave sounder ( $\ddot{z}(t)$ ) |
| ② — Rolling ( $\varphi(t)$ )   | ⑧ — Mean wind velocity ( $\bar{V}_w$ )                      |
| ③ — Heave acceleration in the mobile co-ordinates system ( $\ddot{z}(t)$ ) | ⑨ — Instantaneous wind velocity ( $V_w(t)$ )                |
| ④ — Sway acceleration in the mobile co-ordinates system ( $\ddot{y}(t)$ )  | ⑩ — Direction of the wind                                   |
| ⑤ — Water surface oscillations at the weather side ( $h_w(t)$ )            |   |
| ⑥ — Water surface oscillations at the lee side ( $h_z(t)$ )                |   |

Fig. 5. Fragment of the record tape of the experiments in natural wind beam waves.

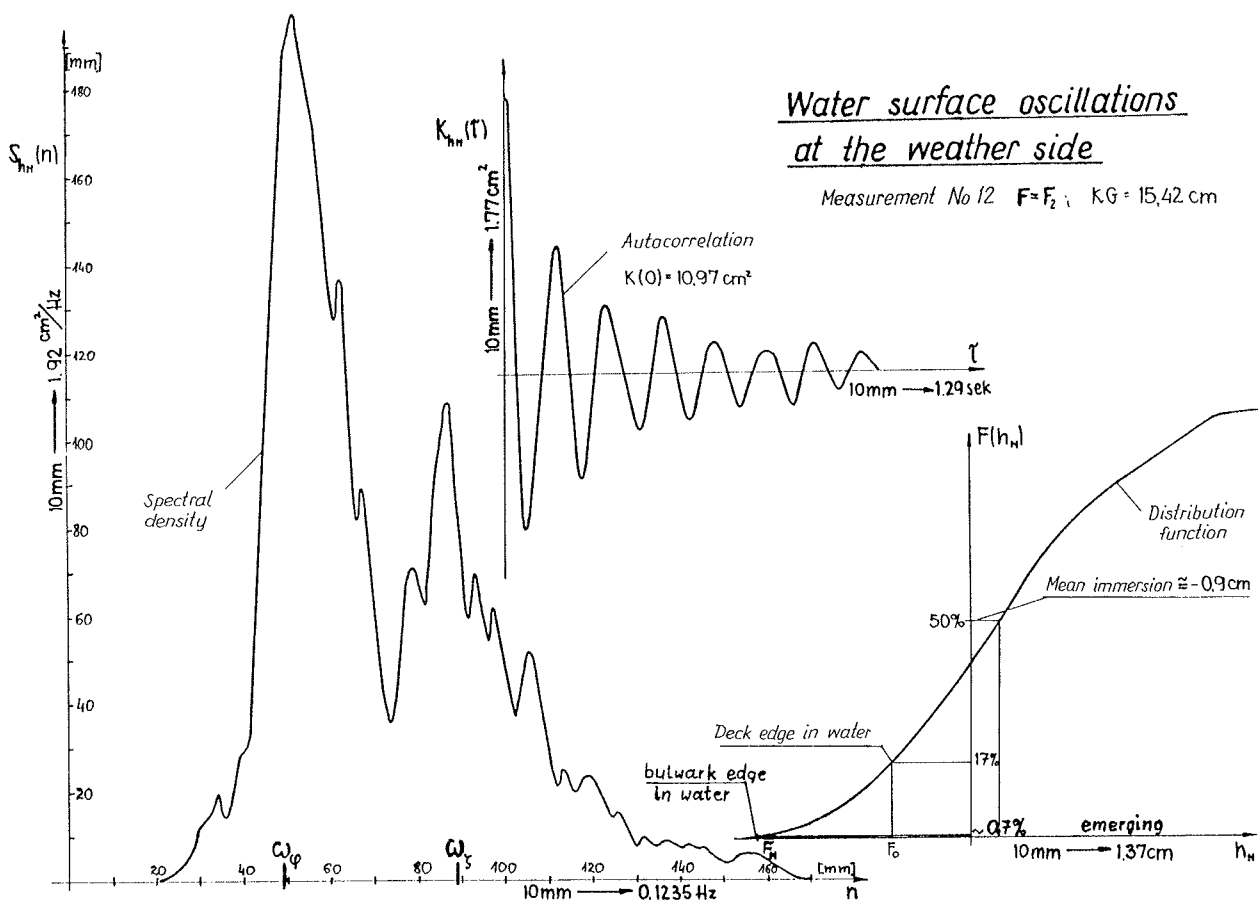
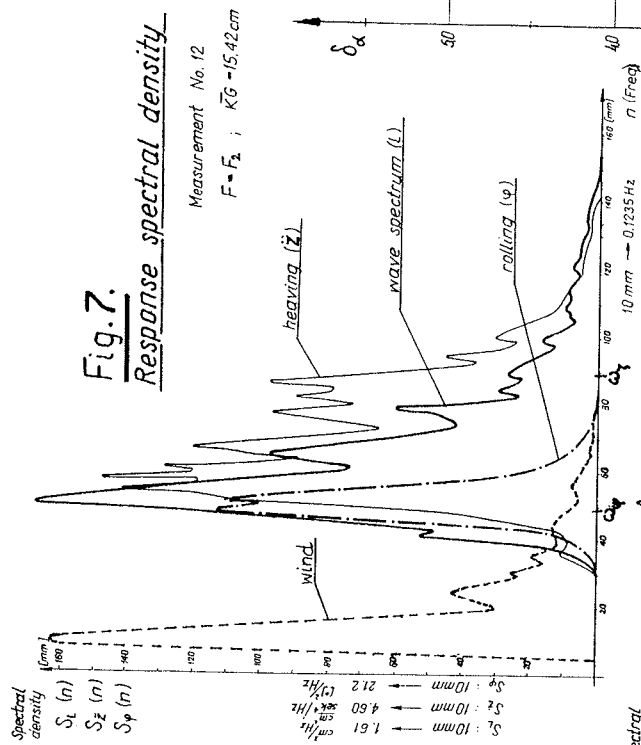
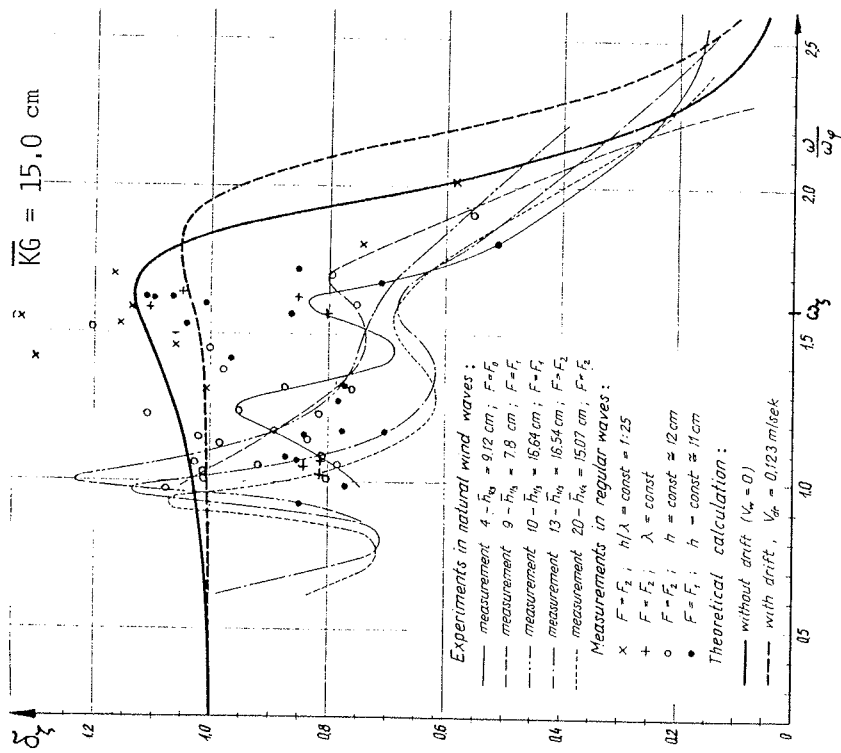
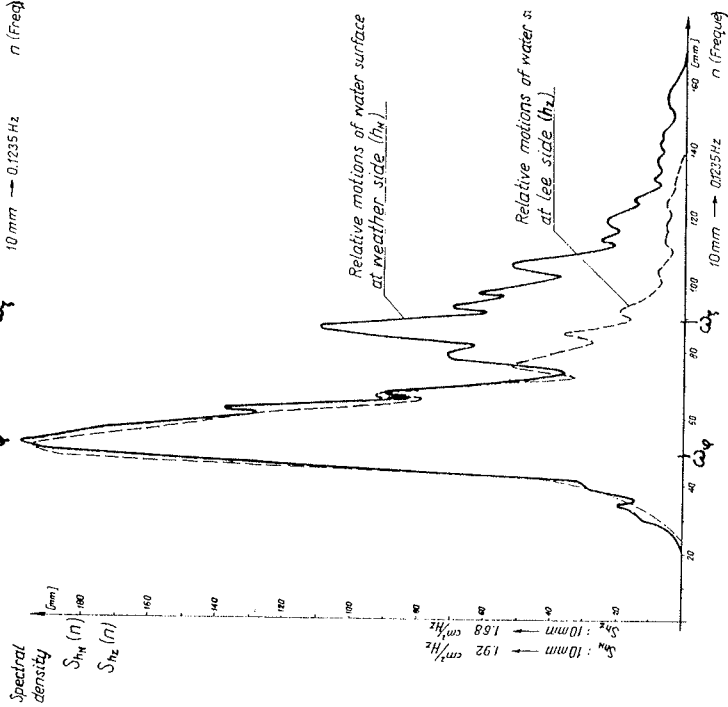


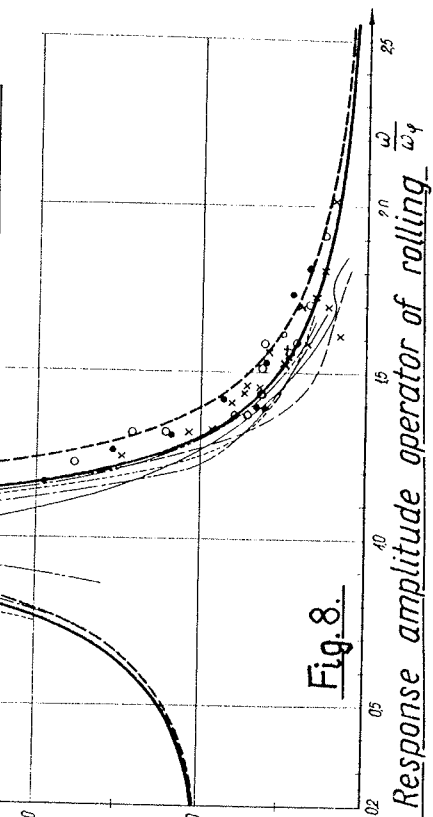
Fig. 6.



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**FIG. 9** RESPONSE AMPLITUDE OPERATOR OF HEAVING



**Fig. 8.**

Response amplitude operator of rolling

# Response amplitude operator of relative motions of water surface at weather side

$\bar{KG} = 15.0 \text{ cm}$

Theoretical calculations:

— without deformation of wave profile;  $V_{dr} = 0.123 \text{ m/sec}$   
 - - - with deformation of wave profile;  $V_{dr} = 0.123 \text{ m/sec}$

Experiments in natural wind waves:

— measurement 4 -  $\bar{h}_{1/3} = 9.12 \text{ cm}$ ;  $F = F_0$   
 - - - measurement 9 -  $\bar{h}_{1/3} = 7.80 \text{ cm}$ ;  $F = F_1$   
 - - - measurement 10 -  $\bar{h}_{1/3} = 16.64 \text{ cm}$ ;  $F = F_1$   
 - - - measurement 13 -  $\bar{h}_{1/3} = 16.54 \text{ cm}$ ;  $F = F_2$   
 - - - measurement 20 -  $\bar{h}_{1/3} = 15.07 \text{ cm}$ ;  $F = F_2$

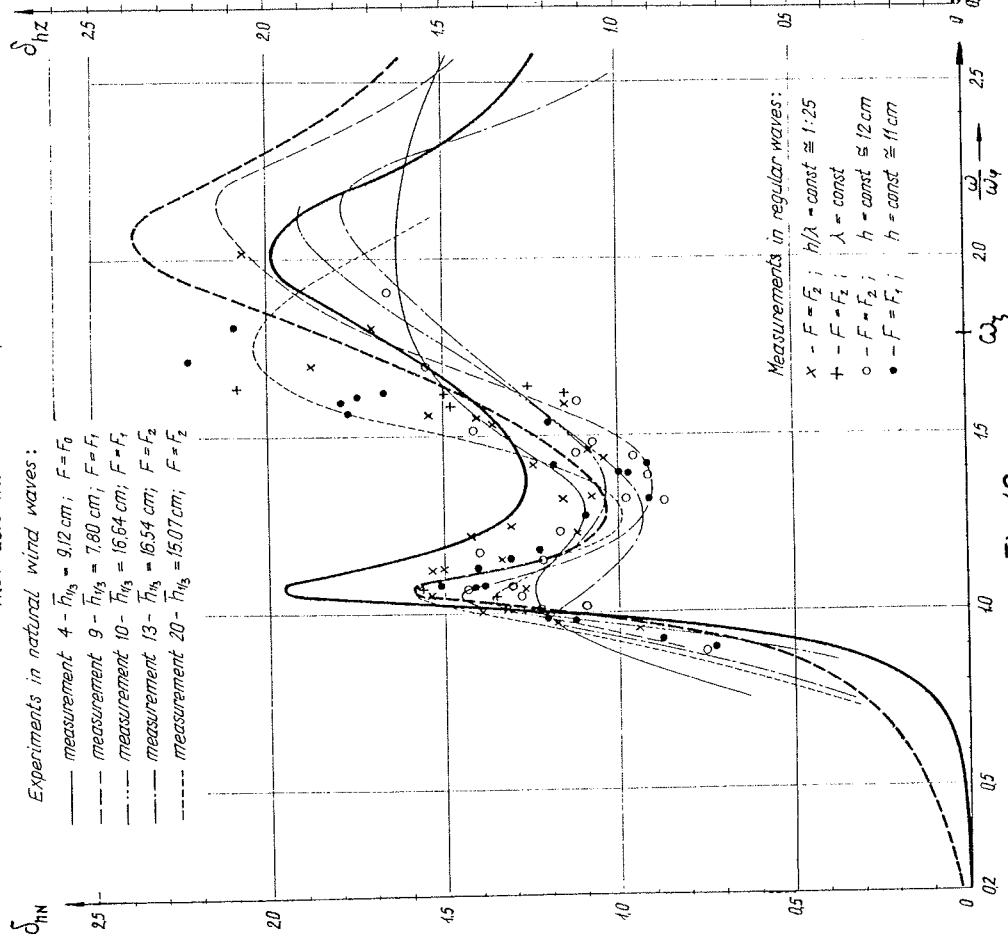


Fig. 10.

# Response amplitude operator of relative motions of water surface at lee side

$\bar{KG} = 15.0 \text{ cm}$

Theoretical calculations:

— without deformation of wave profile;  $V_{dr} = 0.123 \text{ m/sec}$   
 - - - with deformation of wave profile;  $V_{dr} = 0.123 \text{ m/sec}$

Experiments in natural wind waves:

— measurement 4 -  $\bar{h}_{1/3} = 9.12 \text{ cm}$ ;  $F = F_0$   
 - - - measurement 9 -  $\bar{h}_{1/3} = 7.8 \text{ cm}$ ;  $F = F_1$   
 - - - measurement 10 -  $\bar{h}_{1/3} = 16.64 \text{ cm}$ ;  $F = F_1$   
 - - - measurement 13 -  $\bar{h}_{1/3} = 16.54 \text{ cm}$ ;  $F = F_2$   
 - - - measurement 20 -  $\bar{h}_{1/3} = 15.07 \text{ cm}$ ;  $F = F_2$

Measurements in regular waves:

x  $F = F_2$ ;  $h/\lambda = \text{const} \approx 1:25$   
 +  $F = F_2$ ;  $\lambda = \text{const}$   
 o  $F = F_2$ ;  $h = \text{const} \approx 12 \text{ cm}$   
 •  $F = F_1$ ;  $h = \text{const} \approx 11 \text{ cm}$

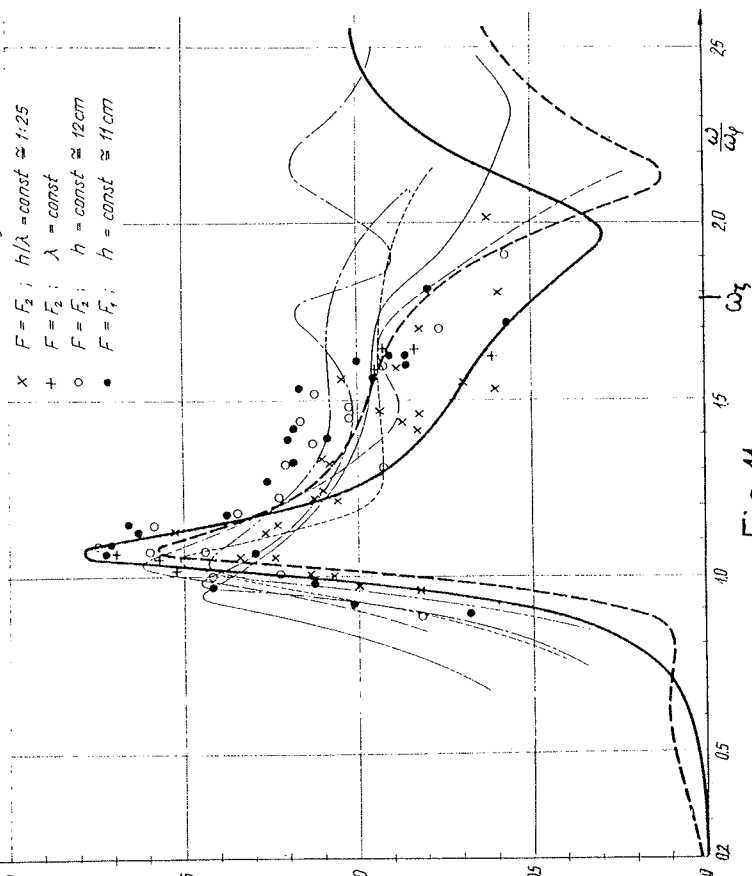
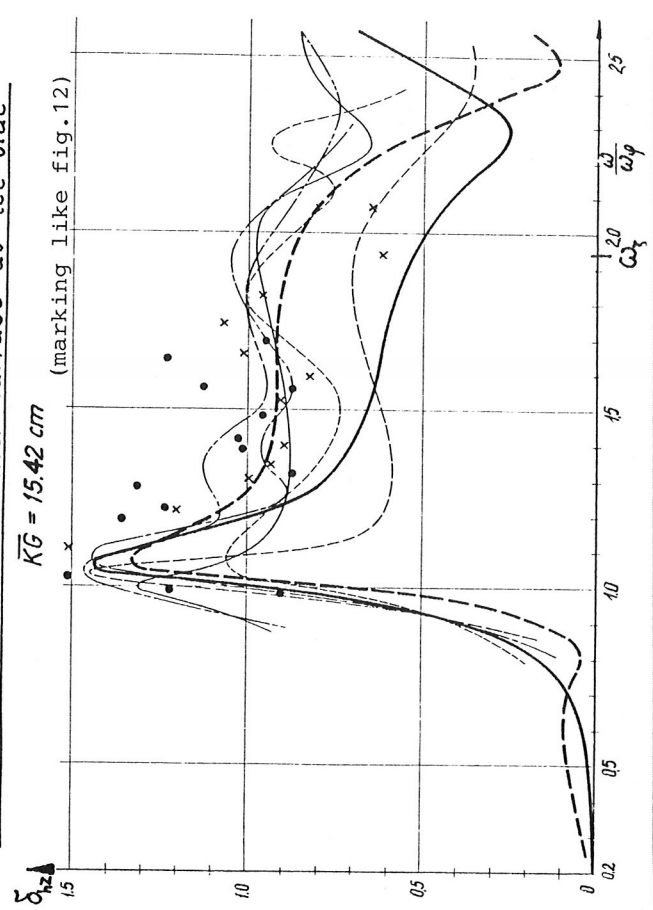
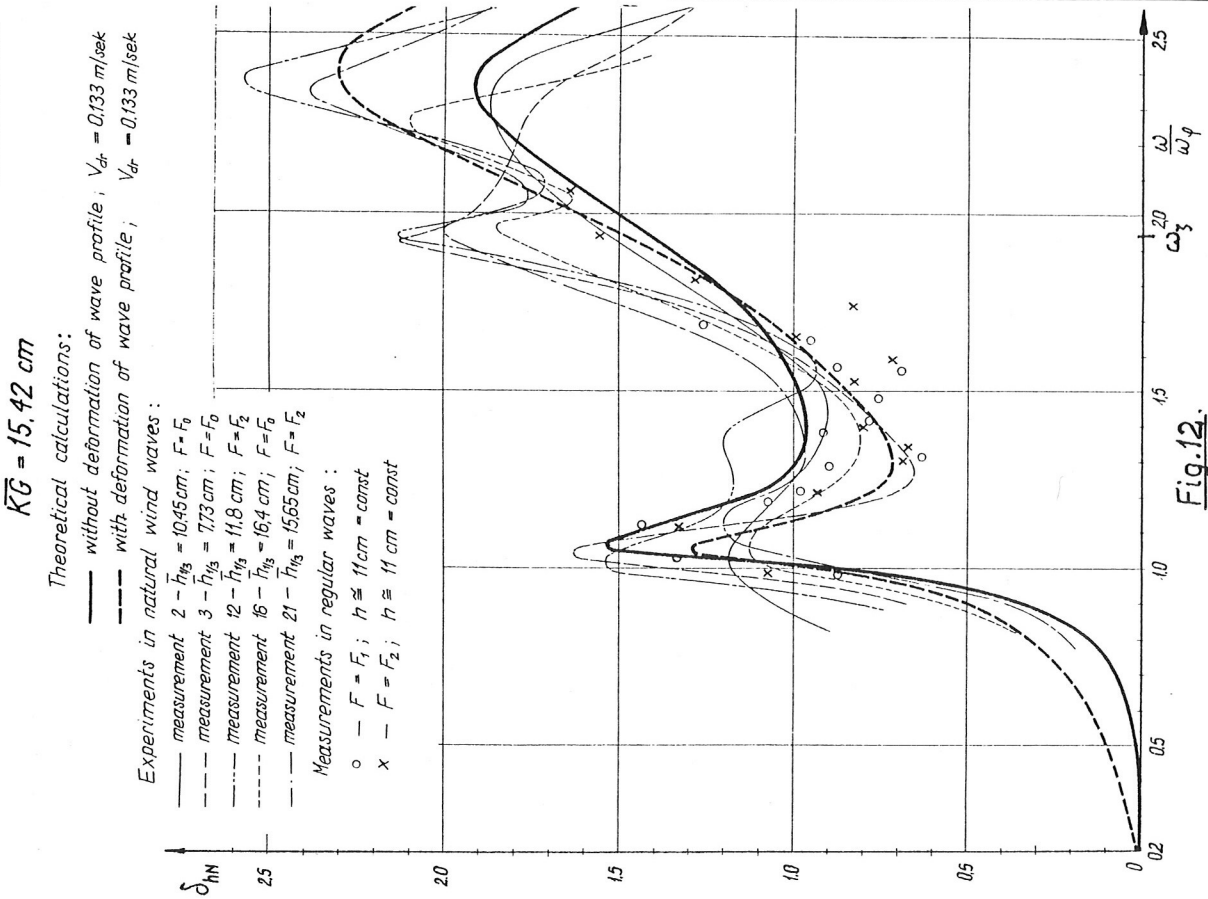


Fig. 11.

**Fig.13.** *Response amplitude operator of relative motions of water surface at lee side*



*Response amplitude operator of relative motions of water surface at weather side*



**Fig.12.**



**PHOT. 1** THE MODEL IN IRREGULAR WAVES ON THE LAKE

TAB. 1 THE STATISTICAL CHARACTERISTICS OF THE BULWARK EDGE EXCEEDANCE BY THE WATER LEVEL AT THE MODEL SIDE

$\bar{K}G = 15.0 \text{ cm}$		Weather side								Lee side							
		$D_h$ cm <sup>2</sup>	$D_k$ cm <sup>2</sup> sec <sup>-2</sup>	$P_F$ %	$\bar{n}_F$ sec <sup>-1</sup>	$P_T$ %	$\bar{\tau}$ sec	$\bar{\Delta h}$ cm	$D_h$ cm <sup>2</sup>	$D_k$ cm <sup>2</sup> sec <sup>-2</sup>	$P_F$ %	$\bar{n}_F$ sec <sup>-1</sup>	$P_T$ %	$\bar{\tau}$ sec	$\bar{\Delta h}$ cm		
4	Calcu- lated	9.502	275.8	6.1	0.052	0.9	0.172	1.141	4.514	100.7	0.9	0.007	0.1	0.159	0.636		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	8.427	287.7	4.2	0.039	0.6	0.151	1.024	4.764	109.1	1.2	0.009	0.1	0.161	0.669		
9	Experiment	8.78	314.9	4.9	0.047	0.4	0.149	1.06	4.78	124.4	1.2	0.009	0.1	0.156	0.67		
	Calcu- lated	6.556	209.5	2.0	0.018	0.3	0.144	0.829	2.440	63.6	0.0	0.000	0.0	0.105	0.335		
10	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	6.024	224.3	1.4	0.014	0.2	0.128	0.769	3.198	81.2	0.1	0.001	0.0	0.124	0.447		
	Experiment	5.61	228.2	1.1	0.011	0.2	0.118	0.72	3.20	77.6	0.1	0.001	0.0	0.05	0.45		
20	Calcu- lated	21.458	447.1	27.0	0.196	5.3	0.269	2.272	15.121	257.4	26.8	0.176	5.2	0.297	1.903		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	17.726	390.6	20.5	0.153	3.8	0.245	1.932	12.199	229.5	19.6	0.135	3.5	0.262	1.586		
13	Experiment	16.24	335.4	18.0	0.134	2.0	0.239	1.79	16.2	328.4	29.0	0.208	5.0	0.279	2.01		
	Calcu- lated	15.595	338.4	17.4	0.129	3.1	0.238	1.749	10.539	183.5	14.1	0.094	2.4	0.255	1.379		
3	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	12.955	302.6	12.2	0.094	2.0	0.214	1.489	8.837	169.0	9.7	0.068	1.5	0.228	1.182		
	Experiment	13.6	329.3	13.7	0.107	2.0	0.215	1.55	11.8	226.4	17.3	0.120	4.0	0.258	1.52		
16	Calcu- lated	24.581	535.7	32.4	0.241	6.7	0.277	2.555	16.932	298.5	30.4	0.203	6.1	0.302	2.082		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	20.013	485.2	25.0	0.196	4.8	0.245	2.151	13.930	262.7	23.5	0.162	4.4	0.273	1.767		
12	Experiment	17.75	474.7	21.1	0.174	2.5	0.224	1.94	15.2	272.6	26.3	0.177	5.5	0.288	1.90		
	$\bar{K}G = 15.42 \text{ cm}$																
21	Calcu- lated	4.820	187.8	0.5	0.005	0.1	0.113	0.619	0.987	35.4	0.0	0.000	0.0	0.000	0.000		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	5.460	237.1	0.9	0.009	0.1	0.113	0.695	2.075	61.1	0.0	0.000	0.0	0.056	0.176		
2	Experiment	5.6	241.0	1.0	0.010	0.1	0.120	0.72	1.39	46.7	0.0	0.000	0.0	0.000	0.000		
	Calcu- lated	8.636	255.3	5.0	0.043	0.7	0.166	1.060	3.535	75.2	0.2	0.002	0.0	0.145	0.500		
16	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	7.783	286.0	3.6	0.035	0.5	0.143	0.965	5.229	111.7	1.5	0.011	0.2	0.170	0.719		
	Experiment	8.5	259.3	4.9	0.043	0.7	0.163	1.05	5.06	122.3	1.3	0.012	0.1	0.137	0.70		
12	Calcu- lated	15.257	277.9	13.7	0.093	2.3	0.248	1.651	10.669	152.2	18.3	0.110	3.3	0.297	1.462		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	12.244	243.4	8.4	0.060	1.3	0.219	1.360	10.643	173.2	18.2	0.117	3.3	0.278	1.459		
21	Experiment	14.1	355.0	11.8	0.094	2.0	0.207	1.55	9.3	145.7	14.1	0.110	3.0	0.218	1.300		
	Calcu- lated	9.834	241.9	5.5	0.044	0.8	0.185	1.145	5.212	92.0	2.4	0.016	0.3	0.196	0.751		
12	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	8.227	246.5	3.1	0.027	0.4	0.156	0.975	6.436	121.9	4.8	0.033	0.7	0.206	0.909		
	Experiment	10.97	297.5	7.6	0.056	0.6	0.207	1.27	7.9	133.2	8.2	0.052	1.8	0.250	1.09		
2	Calcu- lated	14.095	237.6	12.5	0.082	2.1	0.253	1.559	10.263	142.3	15.9	0.094	2.8	0.293	1.393		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$	11.063	194.8	7.1	0.047	1.1	0.226	1.258	10.051	160.3	15.3	0.097	2.6	0.271	1.368		
2	Experiment	15.9	347.1	16.1	0.119	2.0	0.235	1.74	12.0	190.0	20.4	0.129	3.0	0.286	1.58		
	$\frac{\bar{A}_{2,1}}{A_{2,1}} \neq 0$																

## Discussion

A.Y. Odabasi (The British Ship Research Association, UK)

Dr. Grochowalski presented us a very interesting and informative paper on deck wetting in beam seas. I would like him to clarify the following points.

(1) When a ship has a steady heeling angle longitudinal and lateral modes of oscillations became coupled due to the loss of part starboard symmetry, and Ref. [D1] indicated that this coupling may be important if the steady heel angle is larger than a few degree. Since the present approach does not include this affect, what is the author's opinion on the subject?

(2) Deck edge immersion produces vortices which depending on the relative motion between the ship and the wave system may have a destabilising affect and is distinct from the viscous effects considered within the classical roll damping considerations. I would therefore like to know whether the author considered the effect of this additional vortex generation during the determination of system parameters from tests?

### Reference:

[D1] LEE, C.M. and KIM, K-H, "Prediction of Motion of Ships in Damaged Condition in Waves", Proc. Second Int. Conf. on Stability of Ships and Ocean Vehicles, Tokyo, 1982.

### Author's Reply

Answering Dr. Odabasi with respect to his first point:

The purpose of the presented work is to examine whether water surface motions on the ship's sides can be described sufficiently well by linear theory and whether the statistical characteristics of the upper edge of bulwark exceedences by the water

surface can be predicted sufficiently correct for practical purposes. Therefore, in the theoretical model of the relative water motions the typical set of the equations describing the hull motion in beam waves with coupling between roll and sway has been adopted. This naturally does not include the coupling effects due to steady heeling angle. These problems, as yet, are not sufficiently investigated.

I think, the coupling forces and moments due to steady heeling angle can be significant in the case of a ship moving in head or oblique seas. In that case, the asymmetry of the immersed hull causes additional coupling effects.

If the ship lies beam on to waves (in the 2-dimensional approach) the above effect are not very significant in comparison with the other moments. Particularly in the case of small ships and fishing vessels, which can develop a considerable angle of heel due to water on deck, the effects of steady heel on the rolling moment are not significant because of the oval form of the cross sections.

In the presented experiments, the steady heeling angles about which the model rolled were small, about 4-6 degrees, and the asymmetry of the cross-sections was insignificant.

To the second point:

The vorticial effects on the water surface have not been considered; only the effect of reflection of waves on the ship's side has been taken into account.

If the exceedences of the upper edge of bulwark are rare then the deformations caused by vorticial effects are small in comparison with the others.

However, the roll damping coefficients have been corrected on the basis of roll experiments in calm water.

*Session VIIIb*

## Stability of Special Ocean Crafts (Part II)

*Chairmen*

Prof. Luis Mazarredo  
ASINAVE-Escuela T.S. de Ingenieros Navales  
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Prof. Michael Schmiechen  
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# OPERATING PERFORMANCE OF A SAIL EQUIPPED TANKER IN WAVES AND WIND

NORIHIRO MATSUMOTO, MORIO INOUE AND MASANOBU SUDO

Nippon Kokan K.K.

## ABSTRACT

A new type of sail ship advanced mainly by an engine and in the auxiliary by sails was developed and is operated at present as a general merchant ship. Concerning this ship, stability and effect of the sail on rolling have been studied. It was confirmed that "the Rule on Stability" in Japan was applicable for estimating heel angle induced by steady wind and dynamic stability under gusts at the development stage.

Effect of a sail on rolling can be dealt with by a quasi-static approach in conjunction with results of wind tunnel tests. Adding sail damping terms to the equations of motion, the ship motion in waves is calculated by the strip theory. From the results of calculation, it was evident that rolling amplitudes of a ship with sail decreased by 20% compared to ordinary ships without sails at the resonant frequency.

However, the test results on roll damping of the full scale ship in the actual sea led to a conclusion which was contrary to the results of the prediction method by reason of excessively low wave height. If forced rolling induced by waves is considerably large, there rests the possibility that sail damping may be expected to decrease rolling angle.

## 1. INTRODUCTION

Having gradually been replaced by steamers from the latter half of the 19th century to the early part of the 20th century, sailing ships had completely ceased to exist as a general merchant ship. But, in order to cope with the rise in oil prices, wind energy is being considered for use in thrusting a ship as one of the methods for energy conservation.

"Shin-Aitoku-Maru" was developed against this background as a new type of sail ship which does not need any additional handling compared with ordinary motor ships.[1,2] The design criteria of ship stability and damping effect of sail on rolling motion which were examined during the development stage, are presented in the report.

## 2. SHIP PLAN

### 2.1 Hull

The ship is a 699 gross ton type small scale tanker. The main particulars of this ship are given in Table 1. A view of the ship in a ballast condition is shown in Photo 1. In order to keep rolling motion in seas in check, the area of bilge keels is designed as twice that of ordinary ships; that is, 28.2 m in length and 0.4 m in width.

Table 1 Principal dimensions of the ship and the sail

Hull		Propeller	
Length between perpendiculars	m 66.0	Diameter	m 2.65
Breadth moulded	m 10.6	Pitch ratio	0.68
Depth moulded	m 5.2	Number of blades	4
Mean draft moulded	m 4.17	Type	CPP
Trim	m 1.25	Rudder	
C <sub>b</sub>	0.672	Area	m <sup>2</sup> 7.245
G <sub>0</sub> M	m 0.78	Aspect ratio	1.37
T <sub>p</sub>	sec 9.0	Sail	
		Height	m 12.15
		Breadth	m 8.0
		Area	m <sup>2</sup> 97.2 × 2

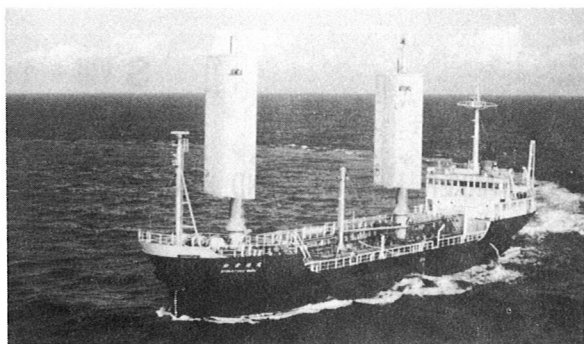


Photo 1 View of the sailing ship in ballast condition

## 2.2 Sail Equipment

After the shape of the sail was chosen from results of wind tunnel tests to be of a rectangular outline with laminar sections constructed of hard material, general performance was confirmed by the large model ship "Daioh" in the actual sea. The mast of the sails are positioned at square station 4.4 and square station 8.7. A section of the sail is shown in Fig.1. Aiming for a long term saving of energy of 10%, the dimensions of the sail were decided. The principal dimensions of the sails are given in Table 1. When the sails are stretched, the attack angle against the wind is controlled to give maximum thrust by sail. On the other hand, when the sails are folded, the attack angle is controlled to give minimum sail resistance induced by the wind.

Relative wind direction and velocity used to calculate the attack angle of the sails for automatic control are measured by a wind meter on top of the rear mast. Maximum wind velocity under which the ship can be operated with stretched sails is decided to be 20 m/sec which may be either relative or true wind velocity. Automatic control pattern of sail operation against wind direction are shown in Fig. 2.

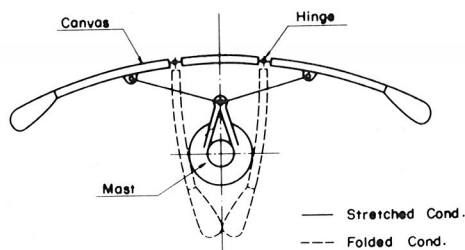


Fig. 1 Section of the sail

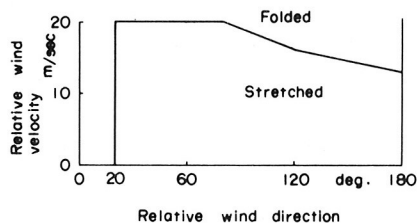


Fig. 2 Automatic control of sail operating system

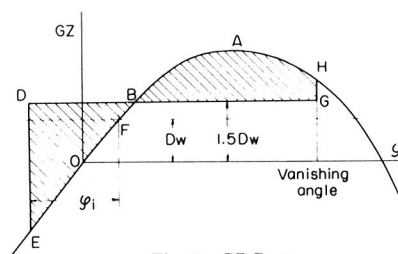


Fig. 3 GZ Curve

## 3. STABILITY

### 3.1 Design Standards

This ship was planned according to "the Rules on Stability" in Japan. Wind induced heel angle and evaluation of dynamic stability are derived from a righting-lever curve as follows. [3]

(1) Increase of inclining lever by gusts

Gusts are determined to be  $\sqrt{1.5}$  times the velocity of steady wind by statistical analysis of ocean weather. The inclining lever  $D_w$  for steady wind velocity is given by formula (1).

$$D_w = k A h / d \quad (\text{m}) \quad (1)$$

where  $A$  : Longitudinal sectional area of a ship above water line ( $\text{m}^2$ )

$h$  : Height of C.G. of  $A$  above C.G. of longitudinal sectional area below the water line. (m)

$d$  : Displacement of ship (ton)

$k$  : from Table 2

(2) Dynamic stability

From Fig.3, if  $c = \text{area ABGH} / \text{area BDE} > 1.0$  is maintained, a ship is stable dynamically. The resonant rolling angle  $\phi_i$  of a ship in irregular waves under standard wind velocity can be derived as follows. When the maximum rolling amplitude out of 20 to 50 rollings is considered, taking into account the fact that its phase rarely coincides with that of a gust, it becomes approximately 0.7 times the rolling amplitude in regular waves.

$$\phi_i = \sqrt{138.5 \text{ rs} / N} \quad (\text{deg.}) \quad (2)$$

where  $r = 0.73 + 0.6 \text{ OG} / d$

OG = Vertical distance from the water line to C.G. of a ship.

$$s = p - qT$$

$T$  : Resonant rolling period

$$= 2.01 \kappa_{xx} / \sqrt{GM}$$

$\kappa_{xx}$  : Radius of gyration of moment of inertia around the x-axis.

$d$  : Draft

$p$  and  $q$  : from Table 2

Table 2 k, p and q coefficient

Ship	Standard Wind Velocity	k	p	q
Ocean going	26 m/s	0.0514	0.151	0.0072
Coasting I	19	0.0274	0.158	0.0100
Coasting II	15	0.0171	0.155	0.0180

Consequently, charts showing the relation between relative wind velocity and direction in any load condition can be presented for example as in Fig. 4, where the sail's angle of attack is controlled by the afore mentioned method such that  $c$  is equal to 1.0. The hull and the arrangement of the sails were considered at the development stage such that the value of  $c > 1.0$  when relative or true wind velocity was 20 m/sec.

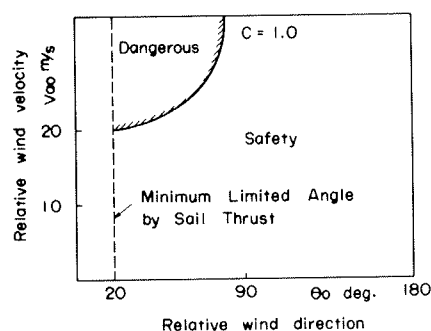


Fig. 4 Limit of wind condition for operating sails

### 3.2 Confirmation of Stability by Tank Tests

#### (1) Outline of tests

Seakeeping tests were carried out to confirm the design standard at the seakeeping tank where waves and the wind were generated simultaneously. Tests carried out in the most severe environmental conditions were desired but this was not possible. So, in the first step, having selected conditions as severe as possible, tests were carried out to confirm the accuracy of the prediction method for inclining angle and motion in waves under wind conditions. In the second step, safety of the design standard was confirmed by this prediction method.

The scale of the model is 1/22, that is 3 m in length and its load condition is full. The kinds of tests performed are free rolling tests under wind and still air conditions, and sea keeping tests under standing and sailing conditions. Test conditions are given in Table 3. The metacentric height in upright condition  $G_0M$  and resonant rolling period  $T_\phi$  were determined from inclining tests and free rolling tests, and the values are  $G_0M = 0.036m$  and  $T_\phi = 2.14sec$ .

#### (2) Heel angle induced by steady wind.

Presupposing the most dangerous situations, the beam direction of the sails were made coincident with the center-line of the ship at zero ship speed only; that is, sail angle  $\phi = 0$ .

Table 3 Wind and wave conditions

Item of tests	Wind velocity m/s	Regular waves		Irregular waves		Incident wave angle deg
		Height m	Period sec	Significant height m	Mean period sec	
Free rolling test	0, 4.3	—	—	—	—	—
Stability test at standing	4.3	0.25	2.1	0.25	2.1	90
Ship motion test at propelling	2.1	0.07	1.3	0.07	1.3	120, 90
	4.3	0.16	2.1	0.16	2.1	30

\*) Wave proceeding direction is coincident with true wind direction.

Moment induced by steady wind is presented by formula (3), which requires wind force coefficients obtained from wind tunnel tests of a sail.

$$M_W = \frac{1}{2} \rho_a V_{a0}^2 (Y'_s A_s h_s + Y'_H A_H h_H) = D_W \cdot d \quad (3)$$

$V_{a0}$ : Relative wind velocity (m/sec)

$Y'$ : Lateral wind force coefficient

subscript,  $S$ : sail

$H$ : hull

As inclining angle due to waves is negligible, the mean heel angle for tests listed in Table 3 were calculated by formula (3) with wind force only. The results are shown in Fig. 5. Predicted values are coincident with the test results.

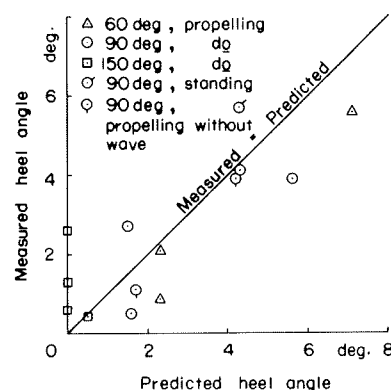


Fig. 5 Heel angle under wind and wave

#### (3) Extinction coefficient obtained by free rolling tests.

The area of the bilge keels of the ship are larger than that of ordinary ships. Bertin's coefficients  $N$  were derived from free rolling tests by varying ship speed and wind while keeping the sail angle at zero. The results of tests are shown in Table 4. A quasi-static approach gives the damping coefficient of the sail as formula (4) under the condition of zero ship velocity. Furthermore, the damping coefficient is converted to the extinction coefficient  $N$  by the formula (5).

Since the increment of the  $N$  coefficient of the sail at a wind velocity 4.3 m/sec. is calculated as 0.01 which approximates the test result of 0.008, the predicting method can be judged applicable.

$$B_s = \frac{1}{3} \rho_a Y_s' b V_{a0} (H_T^3 - H_B^3) \quad (4)$$

$$N = B_s \pi / (2 J_{xx} \omega_x \varphi_m) \quad (5)$$

- $b$  : Chord length of sail (m)  
 $H_B$  : Height of sail bottom above C.G. of ship.  
 $H_T$  : Height of sail top above C.G. of ship.  
 $J_{xx}$  : Moment of inertia around the x-axis.  
 $\omega_x$  : Resonant circular frequency.  
 $\varphi_m$  : Roll angle

Table 4 Bertin's extinction coefficient

Ship velocity	Wind		$N_{10}$
	Velocity	Direction	
0	0	—	0.022
0	4.3 m/s	90 deg	0.030
1.2 m/s	0	—	0.032

#### (4) Roll angle in waves

The rolling response in resonant regular waves was calculated by the strip theory named STF method (Salvessen-Tuck-Faltinsen) using the previously determined  $N$  coefficient and the roll damping coefficient of the sail as mentioned later.

The roll amplitudes determined by calculation are 10% larger than the amplitudes found from experiments. Assuming that the  $N$  coefficient in waves varies from the value from free rolling tests, the  $N$ -coefficient is made larger until the amplitudes coincide with the results of experiments. As a result,  $N_{10} = 0.027$  at zero velocity of the ship and  $N_{10} = 0.039$  at 1.2 m/sec velocity are adopted in later analysis. Amplitudes of roll motion in regular waves compared with the predicted amplitudes are shown in Fig. 6.

Standard deviations of roll amplitudes in irregular waves analyzed from the results of experiments compared with the predicted standard deviations are shown in Fig. 7. As in the relation of amplitudes in regular waves, the two roll amplitudes in irregular waves are nearly equal.

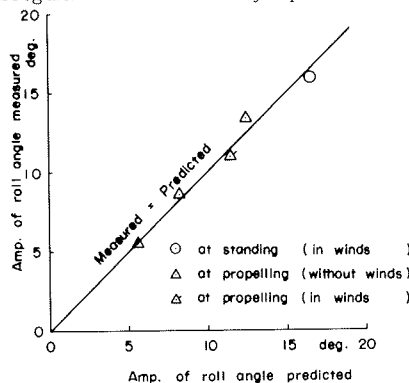


Fig. 6 Roll angle at resonance in regular beam waves and wind

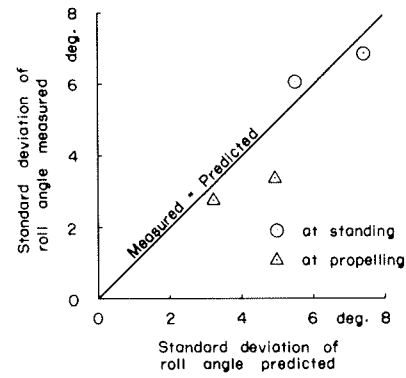


Fig. 7 Standard deviation of roll angle in irregular beam waves and wind

Consequently, it was confirmed that the prediction method based on the strip theory could be applied to ships with sails in the range of 15 degrees of roll angle.

#### (5) Comparison between design standards and predicted values

Significant wave height corresponding to a wind velocity of 20 m/sec. was derived by the ITTC method and were found to be 7.7 m. [4] The resonant wave period ; 8.55 sec was adopted as a mean period. Next, 1/20~1/50 maximum expected values of roll amplitude were calculated directly from the ISSC wave spectrum characterized by the afore mentioned period and height. Besides this work, the value was compared with the value calculated according to "the Rules on Stability" with a comparison of heel angles one of which was derived from "the Rules on Stability" and the other which was calculated by the prediction method. The comparisons are shown in Table 5. The results from "the Rules on Stability" are coincident with the results from the prediction method which was already confirmed by the tank test.

Consequently "the Rules on Stability" is practicable to a sailing ship just as this ship with a small sail area.

Table 5 Comparison of stability at standing with the Rule on Stability

	The rules on Stability	The results of this prediction
Heel angle(deg)	4.5	4.0
Roll angle(deg)	18.3	18.6

## 4. ROLLING CHARACTERISTICS

### 4.1 Wind Force on a Sail

Forced oscillation tests in the horizontal direction were carried out at a small wind tunnel in order to study the influence of sail wind force on the rolling characteristics of a ship. Scale of the model is 1/50.

#### (1) Static test under steady wind

Varying the attack angle of the sail from -15 degrees

to 100 degrees, lift and drag of the sail were measured. The Reynold's number is  $2.3 \times 10^5$  at a wind velocity of 20.7 m/sec. Lift coefficient  $L'$ , drag coefficient  $D'$  and rate of change of lift coefficient measured against attack angle  $\tau_o'$  are shown in Fig. 8.

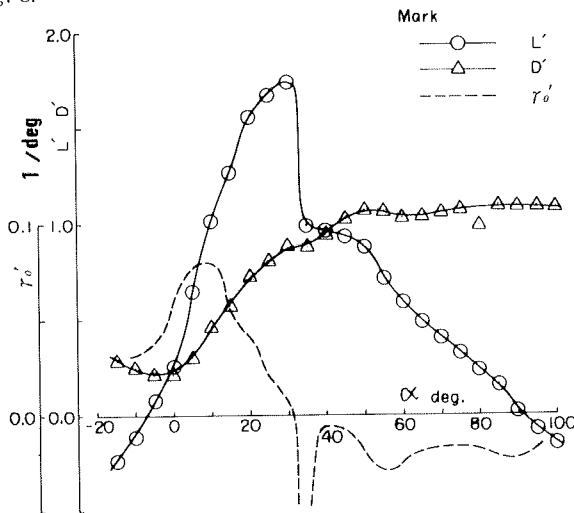


Fig. 8 Lift, drag and rate of change of lift to attack angle  
(2) forced oscillation tests

The model sail penetrated through a false water plane was connected to a dynamometer which is fixed to the oscillating unit. Forces were measured as the sail was moving harmoniously in the direction perpendicular to the wind. Forces proportional to velocity and to acceleration are obtained from the results of the tests by the harmonic analysis. Furthermore, the forced oscillation tests were carried out in still air and the same kinds of tests without sail were performed to determine the mass of the measuring equipment. Attack angles were settled as 0, 20, 30, 40, 60, 90 degrees. In these cases, the amplitude of oscillation is 20 mm~50 mm and circular frequency is 12~20 rad/sec. The forces proportional to velocity were found to be nearly zero from the results of still air tests and no sail tests. This confirmed the applicability of the measuring method.

The mass of the measuring instrument is subtracted from the force proportional to acceleration in wind and in still air conditions, which is shown plotted against the attack angle of the sail in Fig. 9. The proportional force to acceleration is nearly equal to the inertial force of the sail mass itself in still air, but this value is larger than the inertial force in the wind, and this increment is constant regardless of the attack angle.

Next, the force  $Y$  proportional to velocity  $\dot{y}$  can be presented by formula (6) with steady wind forces.

$$\left. \begin{aligned} Y &= B_s \dot{y} \\ B_s &= \frac{1}{2} \rho_a A_s (57.3 \tau_o' + D') V_{ao} \end{aligned} \right\} \quad (6)$$

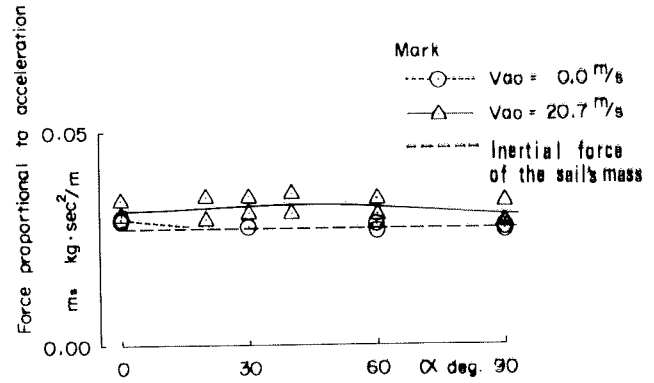


Fig. 9 Force proportional to acceleration of harmonic motion

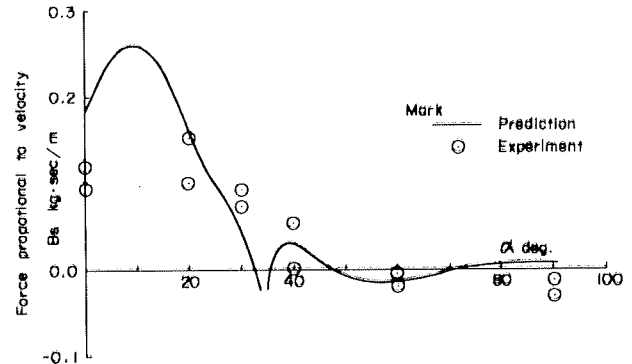


Fig. 10 Force proportional to velocity of harmonic motion

The forces proportional to velocity obtained from experiment are shown in Fig. 10 compared with the results predicted by formula (6). Note that in the figure, the force changes violently near 30 degrees where stall occurs on the sail, but in actuality the force changes continuously. Both results tend to be similar and it appears that predicted values are available for application.

Consequently, the damping force induced by a sail can be dealt with by a quasi static approach.

#### 4.2 Prediction of Damping Effect of a Sail on Rolling

##### (1) Lateral force on an oscillating sail

A prediction method for the lateral force on a ship induced by sails is considered. The right hand orthogonal system of axes moving at mean speed of a ship in the  $x$ -direction was adopted.

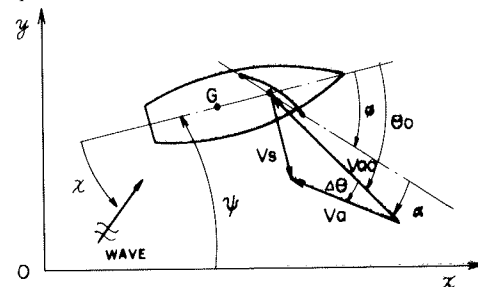


Fig. 11 Coordinate system

By considering a strip of sail whose height is  $h$  above the center of gravity (C.G.) of the ship and whose area is  $A_s$ , its sway  $y_s$  can be given by formula (7).

$$y_s = y - h\dot{\varphi} + l\dot{\psi} \quad (7)$$

$\varphi$  : Roll angle

$l$  : Distance from C.G. to strip of sail in the  $x$ -direction

$\psi$  : Yaw angle

Transverse component of the relative wind velocity  $v_s$  is given by formula (8).

$$\begin{aligned} v_s &= \dot{y}_s \\ &= \dot{y} - h\ddot{\varphi} + l\ddot{\psi} \end{aligned} \quad (8)$$

Resulting velocity  $V_a$  and relative wind direction  $\theta$  are given by formulas (9) and (10).

$$V_a^2 = V_{a0}^2 + v_s^2 - 2V_{a0}v_s \sin\theta_0 \quad (9)$$

$$\theta = \theta_0 - \Delta\theta \quad (10)$$

$$\Delta\theta = \arcsin\left(\frac{v_s}{V_a} \cos\theta_0\right)$$

Lateral force coefficient of a strip of sail  $Y_s'$  is given by the formula (11).

$$Y_s' = Y_{s0}' - 57.3\gamma'(\Delta\theta - \psi) \quad (11)$$

$$\begin{aligned} Y_{s0}' &: \text{Static lateral force coefficient} \\ &= D' \sin\theta_0 + L' \cos\theta_0 \end{aligned}$$

Assuming that higher order terms can be neglected and that  $v_s \ll V_{a0}$  and that relative wind velocity and direction are uniform and represented by values at the center of the sail, formula (12) results.

$$\begin{aligned} Y_s / \frac{1}{2} \rho_a A_s &= Y_{s0}' V_{a0}^2 - Y_c \dot{y} + Y_{c0} \dot{\varphi} \\ &\quad - Y_{cl0} \dot{\psi} + 57.3 \gamma' V_{a0}^2 \psi \end{aligned} \quad (12)$$

$$Y_c = (2Y_{s0}' \sin\theta_0 + 57.3\gamma' \cos\theta_0) V_{a0} \quad (13)$$

$A_s$  : Area of sail

$h_0$  : Distance from C.G. to the center of sail in the  $z$ -direction

$l_0$  : Distance from C.G. to the center of sail in the  $x$ -direction

While the first term of the right hand side of formula (12) is static lateral force, the second to fifth terms are lateral forces induced by motion of the ship. As  $Y_c$  changes with relative wind direction  $\theta_0$ , the lateral force coefficient  $Y_{s0}'$  and rate of change of lateral force to attack angle  $\gamma'$  are derived in the condition of 0, 20, 30, 40, 60 and 90 degrees  $\theta_0$ . These are shown in Fig. 12 and 13.

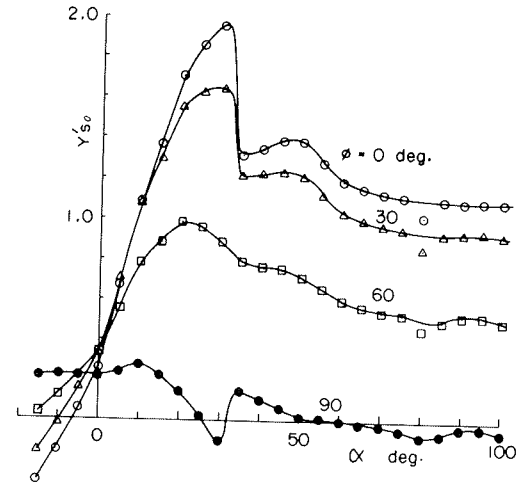


Fig. 12 Lateral force induced by sail

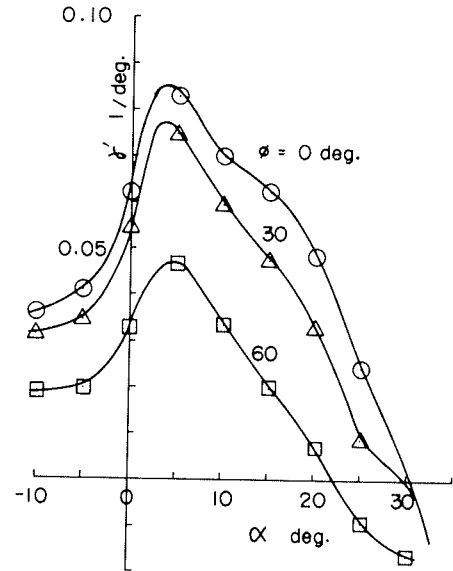


Fig. 13 Rate of change of lateral force to attack angle

## (2) Influence of lateral force of a sail on ship motion

The terms derived from the sail effect in formula (12) are added to the equations of swaying, rolling and yawing motion in waves by the strip theory. At first, part of the added terms is included in the equations and the motion response to waves are calculated for the purpose of studying the terms' contribution to the motion. Wind conditions for the calculation is as follows.

$$V_{a0} = 15 \text{ m/sec.}, \quad \theta_0 = 40^\circ$$

Wave condition is as follows.

Mean period	$T_0 = 8 \text{ sec.}$
Significant height	$H_{1/3} = 4 \text{ m}$
Incident angle	$\chi = 120 \text{ deg.}$

Sail angle is 30 degrees and attack angle is 10 degrees. Standard deviations of roll amplitudes to significant wave height is shown in Table 6. The damping effect of the force proportional

Table 6 Contribution of added terms to equations of motion

Added term	$\sigma_{\varphi}/H_{1/3}$
Proportional to $\dot{\varphi}$	0.85
Proportional to $\dot{\varphi} \& \dot{\gamma}$	0.86
Proportional to $\dot{\varphi} \& \dot{\psi}$	0.94
Whole terms	0.84
none	0.94

to rolling angular velocity is more pronounced than with other forces. Furthermore, these added terms exhibited little effect on other motions.

In the following section, the influence of certain sail conditions at a ship speed of 13 knots are presented using the irregular waves given in Table 7.

Table 7 Irregular waves for calculation

Irregular Wave No.	Wave		Wind velocity
	Mean Period	Significant height	
A	8 sec	1 m	5 m/s & 10 m/s
B	8	2	15
C	8	3	20
D	6	2	15
E	4	2	15

#### (3) Influence of attack angle on rolling

Rolling response to waves are calculated, keeping the relative wind direction at 40 degrees and varying the attack angle to 10, 20 and 30 degrees. Incident wave angle  $\chi$  is 120 degrees and relative wind velocity is 15 m/sec. The response function in regular waves is shown in Fig. 14 against  $\sqrt{\text{Ship length}/\text{wave length}}$ . The responses for attack angle of 10 and 20 degrees are smaller by 24% than those for a ship without a sail. But the damping effect decreases at an attack angle of 30 degrees.

The ratio of standard deviation  $\sigma_{\varphi}$  to significant wave height  $H_{1/3}$  (hereinafter referred to as  $\sigma_{\varphi}/H_{1/3}$ ) is shown in Fig. 15. In this figure, irregular wave B encounters a ship at a  $\chi$  of 120 degrees. In the case of irregular waves,  $\sigma_{\varphi}/H_{1/3}$  with sail is smaller by 21% than that without a sail.

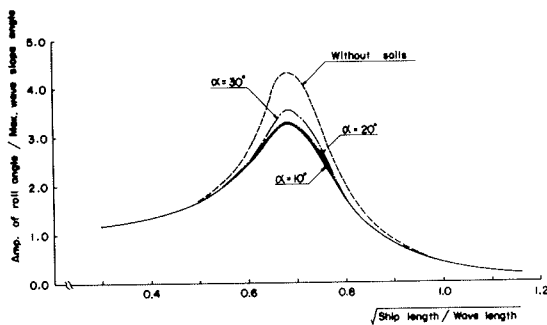


Fig. 14 Effect of attack angle of sail on rolling in regular waves

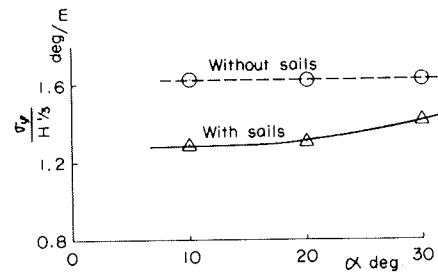


Fig. 15 Effect of attack angle of sail on rolling in irregular waves

#### (4) Influence of relative wind direction on rolling

Rolling amplitude in irregular waves were calculated, keeping the relative wind velocity at 15 m/sec and varying the relative wind direction to 10, 40 and 70 degrees. The irregular wave B encounters the ship at a  $\chi$  of 120 degrees. Attack angle is kept at 10 degrees in order to compare the effect of damping in the condition of strong force on sails. The results are shown in Fig. 16. While damping effect is large for smaller angles of relative wind direction up to 50 degrees, the effect is halved at 70 degrees.

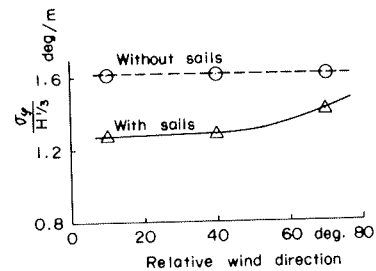


Fig. 16 Effect of relative wind direction on rolling in irregular waves

#### (5) Influence of relative wind velocity on rolling

Rolling amplitude in irregular waves are calculated, varying wind velocity and keeping all other conditions. Irregular waves A, B and C encounter the ship at a  $\chi$  of 120 degrees and relative wind velocity correspond to 20 m/sec, 15 m/sec and (10 m/sec and 5 m/sec) respectively. The results are shown in Table 8.  $\sigma_{\varphi}/H_{1/3}$  of the ship without a sail becomes small because damping force induced by viscosity becomes large in high waves under large wind velocity conditions. The ratio of decrease in  $\sigma_{\varphi}/H_{1/3}$  of a ship with sails to one without sails becomes larger with wind velocity and reaches 23% at a wind velocity at 20 m/sec.

Table 8 Effect of wind velocity on rolling in irregular waves

Relative wind velocity m/s	5	10	15	20
Significant wave height ( $H_{1/3}$ )	1.0	1.0	2.0	3.0
$\sigma_{\varphi}/H_{1/3}$ without sail (A)	1.91	1.91	1.63	1.46
$\sigma_{\varphi}/H_{1/3}$ with sail (B)	1.71	1.55	1.29	1.13
(A) - (B) / (A) %	10	19	21	23

### (6) Influence of incident wave direction on rolling

Rolling amplitudes in irregular waves are calculated, varying the incident wave direction and keeping all other conditions namely, relative wind velocity at 15 m/sec, relative wind direction at 40 degrees and attack angle at 10 degrees, constant. The irregular waves B encounter the ship at a  $\chi$  of 150 and 120 degrees, and waves D at a  $\chi$  of 90 degrees. The results are shown in Table 9. The ratio of decrease in  $\sigma_{\varphi}/H_{1/3}$  of a ship with sails to one without a sail is almost constant with respect to widely varying incident wave angles.

Table 9 Effect of incident wave angle on rolling in irregular waves

Wave angle $\chi$ deg.	150	120	90
$\sigma_{\varphi}/H_{1/3}$ without sail (A)	0.78	1.68	2.11
$\sigma_{\varphi}/H_{1/3}$ with sail (B)	0.68	1.29	1.76
(A) - (B) / (A) %	19	21	17

Table 10 Standard deviation of rolling in running condition

	With sail		Without sail
	$\alpha = 20^\circ$	$\alpha = 80^\circ$	
$\sigma_{\varphi}/H_{1/3}$	1.31	1.32	1.32
Ratio to without sail %	99	100	100

### (7) Influence of sail on rolling in running condition

Sails of small sailing boats like those of small racing yachts does not generate damping effects but generate forced oscillation effects in the running condition with small attack angle to the wind.[5] In the case of misfortune, rolling becomes larger with time and the ship may turn over. Rolling amplitude in irregular waves are calculated, setting a sail angle of 90 degrees and attack angles of 20 and 80 degrees. The irregular waves E encounter the ship at a  $\chi$  of 60 degrees under a relative wind velocity of 15 m/sec. The results are shown in Table 10. From the table, it can be seen that the sails of this ship barely contributes to the damping of rolling amplitudes in the running condition.

## 4.3 Free Rolling Tests of a Model Ship in Wind

Primary free rolling tests using Shin-Aitoku-Maru's small model ship were carried out in wind and in still air to confirm the damping effect of sails on rolling expected in the above-mentioned calculation.

In order to understand clearly the damping effect of sails on rolling, bilge keels were removed from the model ship whose scale ratio was 1/22 and whose length between perpendiculars was 3m.

The draft and the  $G_{\text{M}}$  are 3.5m without trim and 0.7m respectively and the resonant rolling period is 10.1 sec. for the actual ship. Ship speed is 0 m/s for all tests.

At first, inclining the model ship at initial heel angle of 15 degrees and setting the stretched sail at an angle of 0 degree, free rolling tests were started in wind whose velocity was about 2.4 m/s corresponding to 11 m/s for the actual ship and whose relative direction was 18 degrees. Subsequently, changing the sails' angle to 90 degrees, free rolling tests were carried out in wind whose velocity was the same as against wind and whose relative direction is 162 degrees. Lastly, free rolling tests were carried out in still air.

Damping curves are shown in Fig.17.

The states of vanishing roll angle in following wind is similar to that in still air.

This fact suggests that damping effect of sails on rolling is not recognized in following wind as in the results of the above calculation. On the other hand, rolling angle rapidly vanished in against wind.

Furthermore, free rolling tests with folded sails were carried out in still air and in against wind which were the same as those in the stretched sail condition.

Damping effect of folded sails on rolling was detected slightly, which was also shown in the damping curves, Fig.17.

Bertin's expression N at roll angle of 10 degrees was obtained from damping curves, as shown in Table 11. It is evident from this table too that the damping effect of stretched sails on rolling is notably large in against wind.

Following the above theory, the amount of increase of N in against wind compared with that in still air is predicted as 0.012 which corresponds to the experimental value of 0.016. It is evaluated that these two values are rather close.

In conclusion, the validity of the prediction method is confirmed by free rolling tests in the laboratory.

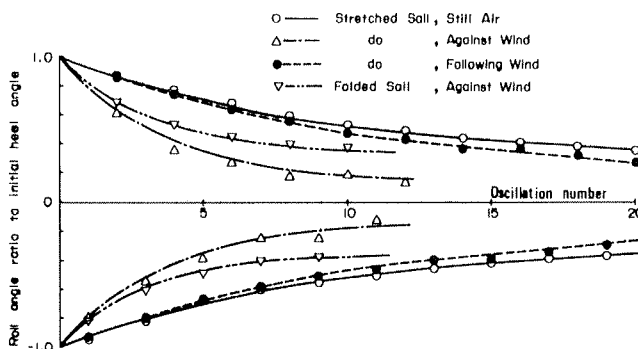


Fig. 17 Comparison of Damping Curves

Table 11  $N_{10}$  Values

Sail Cond.	Wind Cond.	$N_{10}$
Stretched	Still Air	0.007
do	Following	0.007
do	Against	0.023
Folded	Against	0.014



#### 4.4 Motion Measurements of the Full Scale Ship

Motions of the full scale ship were measured to study the influence of sails in the actual sea. As the condition without sails was impossible for "Shin-Aitoku-Maru", the condition with sails folded was adopted in place of it. For the test, motion were continuously measured under two conditions of sails for 30 minutes respectively, and the pair of data were compared with each other. Sea conditions were better than fresh

Table 12 Load condition during test

draft	$d_f$	1.92	m
	$d_m$	2.32	m
	$d_a$	4.24	m
Displacement		1,511	ton
G <sub>0</sub> M		1.80	m
$T_\varphi$		7.6	sec

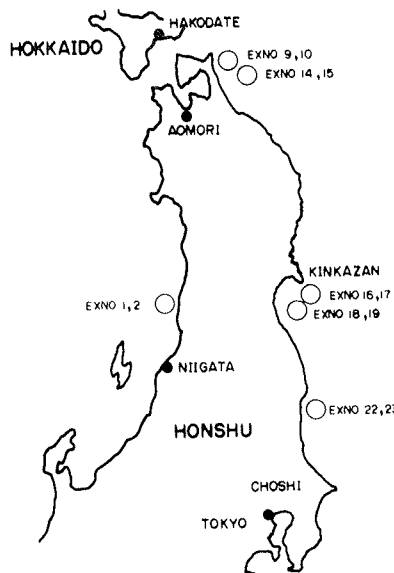


Fig. 18 Test area for the full scale ship

Table 13 Environmental condition during tests

EXNO	Wave			Ship velocity $m/s$	Sail		True Wind		Relative Wind Direction				Relative Wind Velocity			
	Incident angle	Mean period	Signif. height		Condition Open or Close	Attack angle	Direction	Velocity	$\theta_0$	$\sigma$	$T_{02}$	$T_{24}$	$V_{a0}$	$\sigma$	$T_{02}$	$T_{24}$
	deg.	sec.	m			deg.	deg.	$m/s$	deg.	deg.	sec.	sec.	$m/sec.$	$m/sec.$	sec.	sec.
19	S 10	4	1.4	6.9	0	66.8	844	10.5	S 158.9	8.3	20.1	18.4	8.7	0.7	52.7	87.7
18				6.8	C	—	846	10.0	S 169.7	10.2	17.7	11.7	8.2	0.6	49.8	27.0
22	S 87	5	2.4	6.9	0	85.4	880	11.8	S 109.2	6.0	23.6	18.7	7.5	0.9	45.5	22.1
28				6.7	C	—	824	10.8	S 95.0	5.2	29.9	17.6	7.8	0.7	89.6	24.2
14	S 49	4.5	1.2	6.8	0	81.8	289	12.9	S 91.8	5.4	29.1	15.9	10.7	0.8	25.6	16.6
15				6.4	C	—	808	11.8	S 104.2	5.4	25.8	16.6	7.8	0.8	40.9	22.4
9	S 92	5	1.0	6.4	0	27.9	249	10.1	S 54.5	2.0	31.5	14.5	12.4	0.5	88.7	24.8
10				6.4	C	—	258	9.9	S 57.4	2.1	26.9	15.1	11.7	0.6	75.5	86.2
16	S 114	8	1.8	6.2	0	28.9	820	8.9	S 40.4	2.8	35.8	21.9	12.7	0.8	31.0	19.5
17				6.4	C	—	817	8.4	S 37.8	4.0	87.8	18.2	12.5	0.8	46.0	26.6
1	P 180	6	1.8	5.8	0	27.1	816	8.8	P 82.2	3.0	49.2	25.4	12.8	0.8	49.8	21.4
2				5.8	C	—	820	7.5	P 28.0	3.8	80.0	17.9	11.8	0.8	54.5	28.8

Table 14 Results of analysis

EXNO	Rolling				Yawing				Rudder Angle				Pitching		Heaving	
	$\varphi_0$	$\sigma$	$T_{02}$	$T_{24}$	$\psi_0$	$\sigma$	$T_{02}$	$T_{24}$	$\delta_0$	$\sigma$	$T_{02}$	$T_{24}$	$\sigma$	$T_{02}$	$\sigma$	$T_{02}$
	deg.	deg.	sec.	sec.	deg.	deg.	sec.	sec.	deg.	deg.	sec.	sec.	deg.	sec.	m	sec.
19	P 0.5	1.5	8.6	8.8	178	0.7	18.4	12.3	0.8	1.7	14.5	11.6	0.8	8.4	0.15	8.4
18	0	1.5	8.1	7.9	178	0.5	17.7	18.8	1.2	1.4	14.1	9.7	0.8	7.8	0.48	8.1
22	P 1.7	1.7	8.6	8.1	184	1.1	14.8	11.9	0.4	8.1	12.4	11.5	0.6	11.8	0.18	7.4
28	P 0.6	2.0	8.4	8.1	184	1.1	14.5	11.4	0.8	2.8	11.6	10.7	0.6	10.7	0.14	8.5
14	P 2.2	2.1	8.2	7.8	165	0.5	23.6	15.8	1.7	2.0	10.1	8.2	0.4	6.8	0.19	7.1
15	P 1.0	1.7	8.8	7.9	165	0.6	18.9	10.9	1.5	1.9	11.5	9.7	0.5	6.2	0.18	8.6
9	P 2.7	1.8	8.4	8.1	168	0.8	55.8	26.6	2.7	0.6	25.0	15.2	0.8	6.1	0.14	8.2
10	P 0.1	1.0	8.2	7.9	168	0.8	82.7	19.1	1.7	0.5	28.5	14.9	0.8	6.0	0.18	7.4
16	P 2.2	1.1	9.1	8.4	280	0.6	18.7	10.6	2.6	1.8	11.0	9.4	0.5	18.1	0.21	9.8
17	P 1.8	0.7	9.7	8.6	280	0.4	16.8	11.6	2.1	1.3	18.8	11.2	0.4	14.8	0.18	9.4
1	S 2.5	1.0	7.1	6.8	8	0.6	28.0	18.8	-1.1	0.9	18.8	8.8	0.9	7.8	0.28	5.0
2	S 0.4	0.7	6.8	6.5	7	0.5	82.1	20.5	0.5	1.1	14.7	8.6	1.2	5.0	0.80	5.0

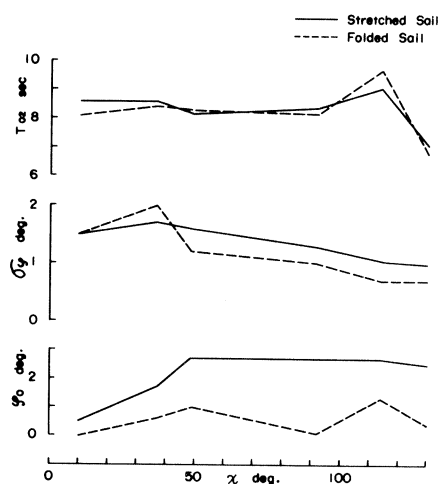


Fig. 19 Results of analysis on roll angle

EXNO	$\alpha$	Damping coefficient
19	10	1 ton · m · sec
22	37	10
14	49	37
9	92	66
16	114	73
1	130	67
*) Standard Condition	120	force by sail
		125 ton · m · sec
		force by viscous, 238 ton · m · sec

\*)  $H_{1/3} = 2 \text{ m}$ ,  $V_{ao} = 15 \text{ m/sec}$   
 $\theta_0 = 40 \text{ deg}$ ,  $\phi = 30 \text{ deg}$ ,  $\alpha = 10 \text{ deg}$   
 Roll angle at resonant  $\omega$  is smaller by 21% than one of a ship without sail.

Table 15 Damping Coefficients

Calculation Experiment Sail Condition  
 ———— ———— : Stretched  
 - - - - - - - - - - : Folded

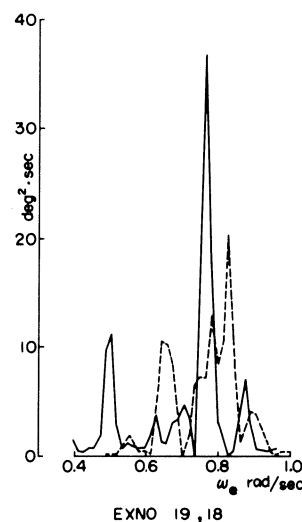
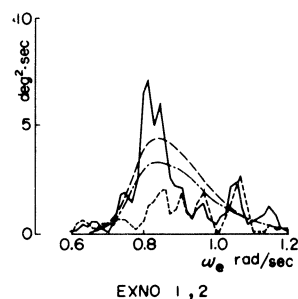


Fig. 21 Power spectrum of rolling

breeze during the testing which was carried out along the northern coast of Japan as is shown in Fig. 18. Thus, conditions were inappropriate for this kind of tests.

The load condition is ballast condition as shown in Table 12 and the environmental conditions are listed in Table 13. Incident wave direction was made to be equal to the true wind direction. Significant wave height and mean period were given by a predicting company. Relative wind velocity and direction were measured at the top of the rear mast. Results of harmonic analysis of the following phenomena were obtained; they are, rolling response, yawing response, pitching response, heaving response, rudder angle, relative wind velocity and relative wind direction. The main parts of the results are shown in Table 14. Furthermore heel angle, zero cross period and standard deviation of roll angle, zero cross period and standard deviation of roll angle against incident wave direction in Fig. 19.

Note from Fig. 19 that, while damping coefficient caused by sails in formula (12) are positive shown in Table 15 at every

incident wave direction, most of rolling angle with sails stretched is larger than one with sails folded, which is not coincident with the former conclusion. Standard deviations of relative wind velocity are about 5% of mean value and about 5 degrees for direction. Those values are rather large. Power spectrum of EXNO 1, 2 and EXNO 19, 18 chosen as typical examples of head seas and following seas are shown in Fig. 20 plotted against encountered wave frequency.

In the figure corresponding relation between the peak of the four kinds of power are indicated by chain lines. Peaks of wind velocity and direction are clearly related to peaks of yawing's and rudder's power. This fluctuation in wind velocity and wind direction seems to cause rolling at the resonant frequency. Rolling response from the same experiment is illustrated in Fig. 21. The results of calculations are presented in the figure.

In the case of following seas, calculations lead to the result that the power curve approaches infinity at  $\omega_e = g/(4V$

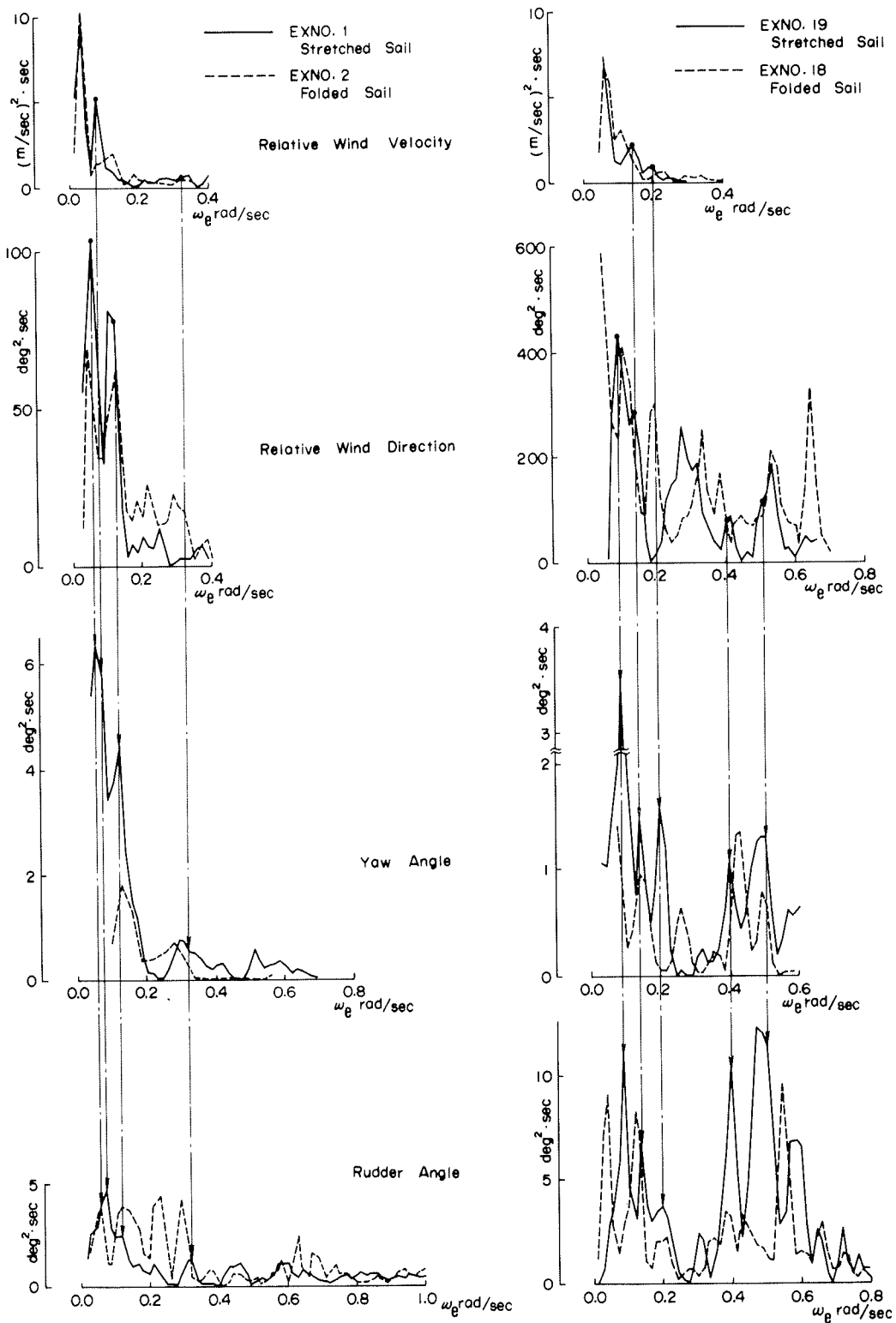


Fig. 20 Power spectrum

$\cos\chi$ ) or  $\omega=g/(2V\cos\chi)$  and beyond this frequency, power becomes very small. [6] But in the results of the experiment, a power peak exists at that frequency. The sail generates a roll damping effect in uniform wind conditions but it generates a forced oscillation effect in varying wind conditions due to the same reasons that a damping effect is generated.

Although this causes rolling at a resonant frequency, if waves are high and the forced rolling moment induced by the waves are strong, then there rests possibility that a sail may be expected to decrease rolling angle.

## 5. CONCLUSION

Stability and the effect of sails on rolling are studied for a new type of sail ship which is advanced mainly by an engine and in the auxiliary by sails. Concentrating attention on the roll damping effect caused by the sail, the conclusions of the study may be summarized as follows.

- (1) "The Rules on Stability" in Japan is applicable to a sailing ship with small sail area.
- (2) The roll damping force induced by a sail can be dealt with by a quasi-static approach.
- (3) The damping effect induced by the force proportional to rolling angular velocity is significantly larger than the effect by forces proportional to other motions.
- (4) The damping effect is notably large at an attack angle of less than 30 degrees and at a relative wind direction of less than 50 degrees.
- (5) Rolling of the ship is caused by fluctuations in wind velocity and wind direction at a resonant frequency in the actual sea. However, if forced rolling induced by waves is considerably large, there rests the possibility that sail damping may be expected to decrease rolling angle.

## ACKNOWLEDGEMENT

We are indebted to Professor K. Nomoto, Mr. Y. Endo and Mr. O. Yamamoto for the valuable suggestions, Professor K. Kijima for convenient facilities given to us during wind tunnel tests and Mr. S. Nagamatsu for excellent tank given to us during seakeeping tests.

When carrying out the wind tunnel tests and seakeeping tests both of the model ship and the full scale ship, we were supported by Mr. M. Itoh, Mr. Y. Kusakawa, Mr. A. Okuyama and Mr. K. Kaneko.

Lastly, Mr. Roy Nakagawa assisted earnestly to polish the paper.

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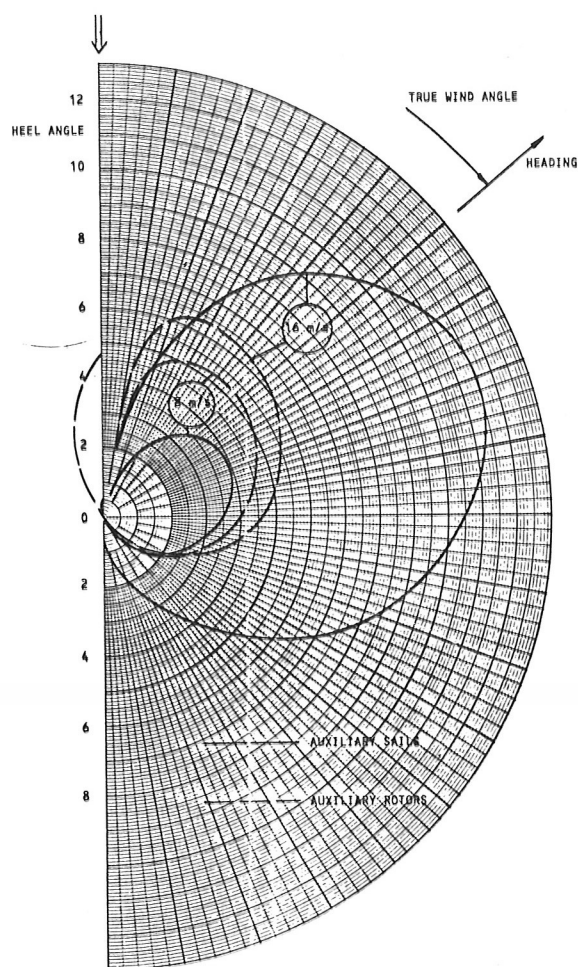
## Discussion

### A. Williams (Swedish Maritime Research Center (SSPA), Sweden)

At the Swedish Maritime Research Centre, SSPA, we have recently made comparative studies regarding conventional sails and Flettner rotors for powering of ships. Two models were made representing a coastal 900 tdw cargo vessel with these two auxiliary propulsors, wind tunnel tests were carried out and among the results gained were heel angles for different headings and

wind force. As is seen in the attached diagram the Flettner rotor alternative is much more favourable from this particular point of view being not as sensitive for large wind forces as conventional sails.

Like the "Shin-Aitoku-Maru" the rotor-equipped vessel needs no extra hands on deck which is decisive when choosing system for wind-powering of ships. Therefore my question is: Was the Flettner rotor included as an alternative in the pre-studies leading to the "Shin-Aitoku-Maru"?



I was very interested in the roll damping force induced by a sail. This is estimated by the formula (b) in the steady-state condition. But, I think, that it is necessary to consider the time-dependent lift force. For example, the lift force of the flat plate oscillated in a uniform flow is given by the following formula.

$$L = 2\pi\rho U^2 C(k) \left( \frac{h}{U} \right) a + \pi\rho h a^2 \omega$$

where  $h$ : oscillatory motion in the vertical direction

$a$ : half chord length;

$U$ : uniform flow velocity

$C(k)$ : the Theodorsen function that is the frequency-dependent force coefficient

$\omega$ : circular frequency of the oscillation

$k = \omega a / U$ : the reduced frequency

In the above equation, the first term shows the lift by the attack angle at the low frequency and the second term is the added-mass term. The ratio of the added-mass to the damping force can be expressed in the form

$$f = \frac{|\pi\rho h a^2|}{|2\pi\rho U^2 C(k) a h / U|} = \frac{k}{2|C(k)|}$$

Using these formulae,  $C(k)$  and  $f$  are calculated in some relative wind velocities  $U$ . The results are shown in the following table.

| $U$   | $k$  | $ C(k) $ | $f$  |
|-------|------|----------|------|
| 3m/s  | 0.93 | 0.54     | 0.86 |
| 10m/s | 0.28 | 0.67     | 0.21 |
| 20m/s | 0.14 | 0.79     | 0.09 |

Note: These are calculated under following condition.

$$a = 4m, \omega = 2\pi / T_\phi, T_\phi = 9s$$

From these results, I think that your method is overestimated in the roll damping calculation and also we must pay attention to the added-mass in calculating the lift force at the higher reduced frequencies. I would like to hear your opinion. Thank you.

#### Author's Reply

Thank you. I am glad to receive the discussion from my close friend Mr. Mizoguchi.

I think your opinion is completely exact. Correction of lift coefficient of the foil is shown in an Argand diagram\*. So, in order to predict damping coefficient more exactly, we are better to correct formula (13) as follows.

$$Y_c = \{2Y'_{so} \sin\theta_0 + 57.3Y' \cos\theta_0 \cdot C(k)\} \cdot V_{A0}(13)'$$

But, I think it does not affect largely on the results of prediction in the practical problem as this time. By the way, we con-

#### Author's Reply

Thank you for your information and discussion. It is well known that the first Flettner rotor was fitted to 600 gross tons ship "Baden-Baden" in 1924.

We also know superior performance of it. So, we carried out wind tunnel tests of a single rotor. Test conditions were follows.

Diameter of rotor: 200m/m, Height: 950m/m

Wind velocity: 3m/s, 5m/s, 7m/s

Maximum revolution: 1600rpm

We obtained excellent results as the same as yours. But concerning maintenance difficulties anticipated. We don't have the plan to adopt a rotor for energy conservation of a ship.

Flettner rotor is better than a conventional sail, because it gives less heeling moment.

However, in Shin-Aitoku-Maru's case, maximum heel angle at 20m/s of wind is only 6 degrees with our set-up. So, we don't think heeling force is an important factor in our case.

firmed good agreement between the results of small sail test and that of prediction in Fig.10. This owes to  $k$  value of 0.06-0.03. So, lift correction  $C(k)$  is nearly equal to 1.0.

Next, we thought the force proportional to acceleration was small enough to predict practical sail force. So, we did not include this term in the calculating program of ship motion.

\* J.N.Newman: Marine Hydrodynamics

H. Sadakane (Univ. of Osaka Prefecture, Japan)

I would like to pay my respects to that the authors have examined not only the steady heel moment of a sail but the roll damping moment of a sail.

The authors have estimated roughly the roll damping moment of a sail as the formula(4) under the condition of  $\phi=0$  and  $\alpha=90^\circ$ . But I think that the modified factor to be the form of  $C\cos\theta_1$  should be reflected on the above formula, because the wind pressure acting on the sail changes with roll angle and steady heel angle even if the flow around the sail be static, where  $C$  is constant value and  $\theta_1$  is steady heel angle.

Could the authors indicate your comments to us.

#### Author's Reply

Thank you for your discussion. If we deal large roll angle, we must consider three dimensional effect and effect of inclining sail angle at predicting stage of damping force as you recommend.

Together with this consideration, we may include the effect of inclining sail angle in the exciting wind force in the ship motion calculation.

In our paper, we adopted primary formula.

K. Fulford (British Shipbuilders, UK)

Do I understand correctly that the weather criterion referred to in 3.1 is normally only used for passenger vessels for which it was designed?

If so why was it used in this case? By choice or required by the authorities?

What would have been the conclusion if it had shown larger restrictions on sail use.

#### Author's Reply

Thank you for your discussion.

We used "the Rules on Stability" for passenger ships of Japanese government, because of the smaller lateral projection area above water line than passenger ships.

Next, our design for stability of this ship by this rule was recognized by the Authorities.

## ON THE STATIC TRANSVERSE STABILITY OF SIDEWALL HOVERCRAFTS ON CUSHION

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China

### ABSTRACT

The theoretical calculation method for predicting the transverse stability of sidewall hovercrafts on cushion are presented. The effect of the sidewall geometrical parameters (i.e. the height, width of the keel plate, the position of the sidewall hardchine etc.), cushion length-beam ratio ( $L_c/B_c$ ), cushion pressure-length ratio ( $P_c/L_c$ ), the size and position of the longitudinal stability keel of the craft, flow rate of the fans, and the distances between the lower edge of the bow/stern seals and the baseline of the sidewall, etc. on the transverse static stability of the sidewall hovercraft on cushion are also analysed.

The model and full scale craft tests for analysing the transverse stability of the sidewall hovercraft on cushion are also carried out. It is shown that the theoretical prediction agrees better with the test results.

### NOMENCLATURE

A,B,C ----characteristic parameter of the fan

Aj1, Aj2 ----leakage area under seals and sidewall without longitudinal stability keel (L.S.K.), ( $m^2$ )  
A3, A4 ----leakage area under left/right plenum chamber with L.S.K. ( $m^2$ )  
Bc ----cushion beam (m)  
B1 ----width of keel plate of the sidewall (m)  
Bs ----sidewall width at the hard chine (m)  
Bsw ----sidewall width at outer waterline on cushion (m)  
D ----displacement volume ( $m^3$ )  
GZ ----righting arm (m)  
KG ----center of gravity above the base line (m)  
Hj ----total head of the fan ( $kg/m^2$ )  
Hs ----sidewall height (m)  
Hsk ----distance between lower edge and inner draft (m)  
Hk ----sidewall depth at hard chine (m)  
h ----metacentric height (m)  
 $\bar{h}$  ----relative metacentric height ( $\frac{h}{B_c}$ )  
Lc ----cushion length (m)  
Ls ----length of the L.S.K. (m)  
Lsk ----sidewall length (m)  
Lh ----distance between lower edge of the L.S.K. and base line (m)

$l_{m1}, l_{m2}$  ---- distance between center of buoyancy ( $V_1, V_2$ ) and rolling axis, respectively (m)  
 $\bar{L}_\theta$  ---- relative righting arm ( $\frac{GZ}{Bc}$ )  
 $M_\theta$  ---- righting moment ( $M_\theta = W \cdot GZ$ ) (ton-m, kg-cm)  
 $P_c$  ---- cushion pressure (kg/m<sup>2</sup>)  
 $Q$  ---- total flow of the fan (m<sup>3</sup>/sec)  
 $\bar{Q}$  ---- relative flow coefficient  

$$(\bar{Q} = \frac{Q}{\sqrt{\frac{2 P_c}{\rho}} \cdot S_c})$$
  
 $Q_1, Q_2, Q_3, Q_4$  ---- leakage flow (m<sup>3</sup>/sec)  
 $\Delta Q$  ---- crossing flow (m<sup>3</sup>/sec)  
 $Q_b, Q_s$  ---- leakage flow under the bow/stern seal (m<sup>3</sup>/sec)  
 $Q_{sb}$  ---- total leakage flow under the bow/stern seal (m<sup>3</sup>/sec)  
 $Q_{sk}$  ---- leakage flow under the sidewall (m<sup>3</sup>/sec)  
 $S_1, S_2, S_i$  ---- duct area (m<sup>2</sup>)  
 $S_c$  ---- cushion area (m<sup>2</sup>)  
 $K$  ---- correction coefficient due to the effect of water surface  
 $\phi$  ---- flow coefficient  
 $\rho$  ---- air density (kg.sec<sup>2</sup>/m<sup>4</sup>)  
 $\theta$  ---- heeling angle (deg)  
 $\gamma$  ---- specific weight of water (kg/m<sup>3</sup>)  
 $t_i$  ---- inner draft of the sidewall (m)  
 $t_{ib}$  ---- mean-inner draft of the sidewall under heeling condition (m)  
 $t_{out}$  ---- external draft of the sidewall (m)  
 $\xi$  ---- head loss coefficient  
 $V_1, V_2$  ---- displacement volume of both sidewalls on cushion, under heeling condition respectively (m<sup>3</sup>), it is referable to  $\theta$ ,  $t_{ib}$   
 $W$  ---- all-up weight for both craft and model (ton, kg)  
 $\alpha$  ---- deadrise angle of the sidewall (deg)  
 $\beta$  ---- flare angle of the sidewall above hard chine (deg)  
 $\gamma$  ---- gap between lower edge of the bow/stern seal and base line (m)

## 1. INTRODUCTION

In the early stages of hovercraft research, the skirt was replaced on both sides of a craft by the sidewall so that to decrease the lift power of the craft. Consequently, the friction drag of the sidewall increased violently. To reduce the friction drag, some of the following measures were taken

- (1) Decrease the length-beam ratio of the craft.
- (2) Thin the sidewall.

At times, one also could find the above mentioned tendencies in The People's Republic of China. (see table 1 and Fig.1,2,3,4,5)

The design of U.S.'s XR-1 and the Chinese test craft type 3, which had the very thin sidewall were the typical instances. Both of them had the poor transverse stability. XR-1 had capsized in 1964 while turning at high speed (ref. 1) and the craft No. 3 rolled violently and periodically with low frequency. Especially when the craft weight was increased from 1.825 tons to 2.2 tons. Just as mentioned above, the similar phenomenon had also found on the craft model No.3 running on the towing tank. These phenomena denoted that the transverse stability of the craft during the early stages of the research work was unsatisfactory because of the thinner sidewall, and it could be improved with some measures. But the rationale concerned with such problems was still not quite clear.

For the purpose of evaluating the transverse stability of the craft during preliminary design some designers (ref. 2,3) have worked out a few tables and figures to determine the necessary geometrical parameters of the craft to meet the requirements for the transverse stability.

Fig.6 shows the relation between the relative thickness of sidewall and cushion length-beam ratio at various cushion pressure coefficient. According to the Fig.6, one can get the satisfactory relative metacentric height. ( $\bar{h} = 0.5$ )



Fig.7 also shows that the craft might have satisfactory transverse stability if the relation between the relative thickness of the sidewall and the cushion length-beam ratio is to be in the zone limited by the lines. The Fig 7 was drawn out by means of statistical method.

It is obvious that the Fig.6,7 mentioned above can be taken as reference at preliminary design stage, but please note that some discrepancy would emerge in case of some crafts which also have satisfactory transverse stability but are out of the zone in the Fig.7 that is because some principal parameters of the craft are quite different from what is expressed in Fig.7.

In order to investigate the transverse stability of the craft in detail. It is necessary to dedicate the theoretical method for calculating the transverse stability of the sidewall hovercraft on cushion and to study the effect of various parameters on that, so as to improve the stability by means of proper combination of various parameters. That is one of the purposes of this paper.

## 2. CALCULATION METHOD FOR PREDICTING THE STATIC TRANSVERSE STABILITY OF THE SIDEWALL HOVERCRAFT ON CUSHION

### 2.1 General Assumptions

The general assumptions for the calculation method are as follows.

- (1) The cushion pressure distribute uniformly in the plenum chamber of the craft without the longitudinal stability keel in case of either upright condition or heeling.
- (2) It may presume that the side slipping force causing by the difference of the leakage air momentums under both sidewalls of heeling craft is equal to the lateral hydrodynamic force acting on them, and the heeling moment causing by the couple of the force mentioned above

may be neglected.

- (3) The fan outlet is perpendicular to the wetdeck of the craft, and the total flow of the fan blows into plenum chamber directly.
- (4) Both the gap between the lower edge of the bow/stern seal and the base line are equal so as to make the calculation more simple.

### 2.2 Calculation of Static Transverse Stability for the Craft without L.S.K.

- (1) Energy equation
 
$$H_j = P_c + \frac{1}{2} \cdot \rho \cdot \left( \frac{Q}{S_1} \right)^2 \quad (1)$$

- (2) Flow continuous equation
 
$$Q = Q_{sb} + Q_{sk} \quad (2)$$

$$Q_{sb} = Q_s + Q_b \quad (3)$$

$$Q = Q_s + Q_b + Q_{sk} \quad (4)$$

- (3) Flow equation
 
$$Q_s = Q_b = k \cdot \phi \cdot A_{j1} \cdot \sqrt{\frac{2P_c}{\rho}} \quad (5)$$

$$Q_{sk} = k \cdot \phi \cdot A_{j2} \cdot \sqrt{\frac{2P_c}{\rho}} \quad (6)$$

From Fig.8 we have

$$A_{j2} > 0 \quad \text{when} \quad \frac{B_c}{2} \cdot \tan \theta > t_{ib} \quad (7)$$

$$A_{j2} = 0 \quad \text{when} \quad \frac{B_c}{2} \cdot \tan \theta \leq t_{ib} \quad (8)$$

- (4) Fan characteristic equation
 
$$H_j = A + BQ + CQ^2 \quad (9)$$

- (5) Weight equation
 
$$W = P_c \cdot S_c \cdot \sec \theta + \gamma \cdot (V_1 + V_2) \quad (10)$$

putting (1) ~ (8) into (9), we have

$$\left[ 1 + (2A_{j1} + A_{j2})^2 \cdot k^2 \cdot \phi^2 / S_1^2 - \frac{2C}{\rho} \cdot k^2 \cdot \phi^2 \cdot (2A_{j1} + A_{j2})^2 \right] \cdot P_c - B \cdot (2A_{j1} + A_{j2}) \cdot k \cdot \phi \cdot \sqrt{\frac{2P_c}{\rho}} - A = 0 \quad (11)$$

It is obvious that equation (11) only includes  $P_c$  and  $t_{ib}$  in case of the definite heeling angle.  $P_c$  and  $t_{ib}$  can be obtained by solving equations (10) and (11) with aid of the computer programme.

Using  $P_c$  and  $t_{ib}$ , one can get righting

moment  $M_\theta$  as follows:

$$M_\theta = (V_1 \cdot L_{m1} - V_2 \cdot L_{m2}) \cdot \gamma - W \cdot (K_G - t_{ib}) \cdot \sin \theta \quad (12)$$

### 2.3 Calculation of Static Transverse Stability for the Craft with L.S.K.

Assume the network of the Fan-duct-cushion system can be simplified as Fig.(9),  
(10)

(1) Energy equation

$$H_j = P_{c1} + \frac{1}{2} \cdot \rho \cdot g \cdot \left( \frac{Q_1}{S_1} \right)^2 = P_{c2} + \frac{1}{2} \cdot \rho \cdot g \cdot \left( \frac{Q_2}{S_2} \right)^2 \quad (13)$$

$$P_{c1} - P_{c2} = \frac{1}{2} \cdot \rho \cdot g \cdot \left( \frac{\Delta Q}{L_s \cdot h_{sk}} \right)^2 \quad (14)$$

(2) Flow continuous equation

$$Q = Q_1 + Q_2 = Q_3 + Q_4 \quad (15)$$

$$Q_1 = Q_4 + \Delta Q \quad (16)$$

$$Q_3 = Q_2 + \Delta Q \quad (17)$$

(3) Flow equation

$$Q_3 = k \cdot \phi_0 \cdot A_3 \cdot \sqrt{\frac{2P_{c2}}{\rho}} \quad (18)$$

$$Q_4 = k \cdot \phi_0 \cdot A_4 \cdot \sqrt{\frac{2P_{c1}}{\rho}} \quad (19)$$

$$\Delta Q = k \cdot \phi_0 \cdot L_s \cdot h_{sk} \cdot \sqrt{\frac{2(P_{c1} - P_{c2})}{\rho}} \quad (20)$$

(4) Fan characteristic equation

$$H_j = A + BQ + CQ^2 \quad (21)$$

From the equations (13)~(21) we have the following equations

$$\text{Let } A'_3 = k \cdot \phi_0 \cdot A_3 \\ A'_4 = k \cdot \phi_0 \cdot A_4$$

Then

$$A + B \cdot \left( A'_3 \cdot \sqrt{\frac{2P_{c2}}{\rho}} + A'_4 \cdot \sqrt{\frac{2P_{c1}}{\rho}} \right) + C \cdot \left( A'_3 \cdot \sqrt{\frac{2P_{c2}}{\rho}} + A'_4 \cdot \sqrt{\frac{2P_{c1}}{\rho}} \right)^2 = P_{c1} + \frac{\rho \cdot g}{2 \cdot S_1^2} \cdot \left( L_s \cdot h_{sk} \cdot k \cdot \phi_0 \cdot \sqrt{\frac{2(P_{c1} - P_{c2})}{\rho}} + A'_4 \cdot \sqrt{\frac{2P_{c1}}{\rho}} \right)^2 \quad (22)$$

$$A + B \cdot \left( A'_3 \cdot \sqrt{\frac{2P_{c2}}{\rho}} + A'_4 \cdot \sqrt{\frac{2P_{c1}}{\rho}} \right) + C \cdot \left( A'_3 \cdot \sqrt{\frac{2P_{c2}}{\rho}} + A'_4 \cdot \sqrt{\frac{2P_{c1}}{\rho}} \right)^2 = P_{c2} + \frac{\rho \cdot g}{2 \cdot S_2^2} \cdot \left( A'_3 \cdot \sqrt{\frac{2P_{c2}}{\rho}} - L_s \cdot h_{sk} \cdot k \cdot \phi_0 \cdot \sqrt{\frac{2(P_{c1} - P_{c2})}{\rho}} \right)^2 \quad (23)$$

(5) Weight equation

$$W = \frac{1}{2} \cdot S_c \cdot \sec \theta \cdot (P_{c1} + P_{c2}) + (V_1 + V_2) \cdot \gamma \quad (24)$$

So one can find the solutions of  $P_{c1}$ ,

$P_{c2}$  and  $t_{ib}$  from the equations (22), (23) and (24). Then the righting moment ( $M_\theta$ ) can be obtained

$$M_\theta = \frac{1}{8} \cdot S_c \cdot S_c \cdot \sec^2 \theta (P_{c1} - P_{c2}) - W \cdot (K_G - t_{ib}) \cdot \sin \theta + (V_1 \cdot L_{m1} - V_2 \cdot L_{m2}) \cdot \gamma \quad (25)$$

### 3. TEST FOR THE STATIC TRANSVERSE STABILITY OF THE SIDEWALL HOVERCRAFT MODELS ON CUSHION

#### 3.1 Models and Test Facilities:

The static transverse stability test for two wooden sidewall hovercraft models on cushion were carried out in Marine Design & Research Institute of China (MDRIC). The leading particulars for both models are shown in table (2), and the typical section of the sidewall is shown in Fig. (11).

During the tests, one can move the ballast to cause model heeling, then the heeling angle can be measured by the clinometer shown in Fig. (12).

#### 3.2 Experiments and Results

During the heeling of sidewall hovercraft, generally, the restoring moment of the craft will be got mainly due to the different buoyance of the sidewalls, and also due to the different cushion pressure between the left and right chambers in case of having the longitudinal stability keel on the craft.

The experiments include following contents:

- (1) Observe the effect of the fan revolution on static transverse stability.
- (2) Observe the effect of the longitudinal stability keel on static transverse stability

Model experiments are shown in Fig. (13) (14) and the experimental results are shown in Fig. (15) (16).

### 4. THE CALCULATED RESULTS AND THE EFFECT OF VARIOUS PARAMETERS ON THE STATIC TRANSVERSE STABILITY OF THE CRAFT

In order to consider the effect of various parameters on the transverse stability, and compare the calculation with the testing results of the full scale craft and models. We calculated the transverse stability of crafts and models with various parameters.

#### 4.1 Effect of the Geometric Configuration of Sidewalls on the Static Transverse Stability:

Take the sidewall hovercraft type 717 as an example. We calculate the static transverse stability of the craft with definite fan speed, duct type and form of the bow/stern seals, but various geometric configuration of the sidewalls in midship, in order to evaluate the effect of the geometric configuration of sidewalls on stability.

Assume the outer water line of the craft on cushion is under the hard chine of the sidewalls, then the principal parameters of the geometric configuration of the sidewall can be described in Fig. (17).

In general,  $B_1$  can be determined according to the strength and the condition of processing technology, therefore, we can keep this as a constant, tout can be determined by the cushion pressure  $P_c$ . So we can take the parameters  $\alpha, \beta, B_{sw}$  as variables. The first set of variables are shown in table (3).

The calculated results for the first set of variables are shown in Fig. (18).

In order to investigate the effect of  $B_1$  on stability, we get the second set of variables shown in table (4). We keep the basic parameters as same as that for the craft type 717 (i.e. principal dimension, cushion pressure-length ratio, flow rate coefficient, flare angle on the sidewall section above the hard chine, the gap between the lower edge of the bow/stern seals and the base line, fan characteristic and etc.) except parameters  $B_1, \alpha$ , then static transverse stability can be calculated and shown in Fig. (19).

From Fig. (18) and (19) we can see clearly that

- (1)  $\beta$  deeply effect the static transverse stability of the craft at large heeling angle, but not at small angle (i.e. initial transverse stability)
- (2) Linear relation of the righting arm with respect to the heeling angle only exists at small heeling angle (about 3-4 degrees), therefore, it is convenient to take the relative metacentric height ( $\bar{h} = h/B_c$ , where  $h$  is the metacentric height,  $B_c$  is the cushion beam) as one of the stability criteria of the craft.
- (3) The width of keel plate on the sidewall do not strongly effect the stability either at small heeling angles or at large ones. (Fig. 19)
- (4)  $\alpha$  strongly effect the stability either at small heeling angles or at large ones (Fig. 20).
- (5) According to two set of variables mentioned above, we can get the relation of the relative metacentric height ( $\bar{h}$ ) with respect to the relative thickness of the sidewall ( $B_{sw}/B_c$ ) (Fig. 21). It is found that the relation is more stable whatever the values of  $B_{sw}/B_c$  are obtained from any set of variables. Therefore, it is convenient to take the relative thickness of the sidewall ( $B_{sw}/B_c$ ) as a main parameter referring to the transverse stability at preliminary design.

#### 4.2 The Effect of the Lift Power (or the Fan Flow Rate) on the Transverse Stability

In the calculation equations we can see that the fan flow rate strongly effect the stability. In general, the static transverse stability deteriorates as the fan flow rate increases. It means that the stability of the craft on cushion is worse than off cushion, because the cushion pressure causes a negative transverse stability,

The effect of the input electric voltage

of the motor driving the lift fan on the stability is shown in Fig. (22). The line denotes the calculation and test result for the craft "717".

The effect of the relative flow coefficient ( $\bar{Q}$ ) on the relative metacentric height ( $\bar{h}$ ) is shown in Fig. (23), for example,  $\bar{h}$  decreases from 0.163 to 0.135 when  $\bar{Q}$  increases from 0.00615 to 0.00892 (i.e. fan speed increases from 1300 r.p.m. to 1600 r.p.m.).

#### 4.3 The Effect of the Cushion Pressure-Length Ratio on the Transvers Stability

The calculation results of stability with various cushion pressure-length ratio are shown in Fig.(24). It is found that the relative metacentric height  $\bar{h}$  decreases from 0.133 to 0.123 when the cushion pressure-length ratio increases from 21.47 to 23.06.

#### 4.4 The Effect of the Gap (Y) between the Lower Edge of Bow/Stern Seals and the Base Line on the Transverse Stability

For making a simple calculation of the transverse stability, we assume the gaps between the base line, and the lower edge of bow/stern seals are the same, and calculate the transverse stability of the craft type "3" with various gap (y) (i.e. various inner draft of the sidewall). The calculated results are shown in Fig. (25). It is obvious that the transverse stability increases with the inner draft of the sidewalls though the benefit is not greater than that obtained by increasing the thickness of the sidewalls. It means that adjustment of the inner draft of the sidewall is a good measure to control the transverse stability. This phenomenon can be traced back to the trial of the craft type "3" in 1969. At beginning, the craft operated quite well with satisfactory speed and transverse stability. But after some modifications, the all-up weight of the craft increased up to 2.2 tons and the cushion pressure-length ratio increased from

19 to 24, it is discovered that the transverse stability of the craft was deteriorated. The craft used to roll slowly with a rolling angle of about 2-4 degrees even in calm water.

After increasing the inner draft of the sidewall at stern from 0.24 metre to 0.28 metre, the unstable rolling disappeared. unfortunately we had not measured the metacentric height with various gaps for lack of time during the trial.

The effect of the inner draft of the craft type "717" with two different cushion pressure-length ratio on the relative metacentric height is also shown in Fig.(25).

#### 4.5 The Effect of the Cushion Length-Beam Ratio on the Transverse Stability

We have calculated the transverse stability of the craft type "717" with different cushion length-beam ratio shown in Fig.(26). One can see that either initial stability or stability with large heeling angle remain unchanged. Therefore it is not correct to say that the transverse stability will deteriorate definitely with the increasing cushion length-beam ratio, and one has to make a practical analysis of particular problems.

#### 4.6 The Effect of the Longitudinal Stability Kell (J.S.K) on the Transverse Stability

The metacentric height for the craft type "717" at various height of L.S.K. is shown Fig. (27) and table (5)

From the table (5) and Fig. (27) one can see that to increase the height of L.S.K. is an advantage to both initial stability and large heeling angle stability, especially for the later.

### 5. CONCLUSION AND FUTURE WORK

- (1) The calculated results of the static transverse stability of crafts by means

of using this method quite agree with the experimental results, therefore, it is advisable to use this method to check and analyse the static transverse stability of the designing or launched sidewall hovercraft.

- (2) The relative thickness of the sidewall might be considered as a main parameter that will strongly effect the transverse stability of crafts.
- (3) The transverse stability of launched craft can be improved by decreasing the flow rate coefficient ( $\bar{Q}$ ), cushion pressure-length ratio ( $P_c/L_c$ ) and increasing the gap under the bow/stern seals ( $y$ ). One can get the quantitative analysis by computers.
- (4) Putting the longitudinal stability keel on craft can improve the transverse stability of the craft, especially at large heeling angle, and will cause other unfavorable results at the same time (increasing drag, and hull weight etc.). Therefore that is not an effective way to improve the transverse stability.
- (5) Increasing cushion length-beam ratio occasionally might deteriorate the transverse stability, but it might not be the only result, one ought to make a practical analysis of particular problems.

#### 6. SUGGESTION AND EXPECTATION

- (1) The calculation method for dynamic transverse stability is suggested to study. The planing surface of the bow/stern seals may improve the transverse stability of the craft, but the stability might be deteriorated during taking-off. So it is necessary to investigate the dynamic transverse stability at various relative speed of the craft.
- (2) It is necessary to determine the criteria for the transverse stability of

sidewall crafts either at small or at large heeling angle (ref. 4, 5).

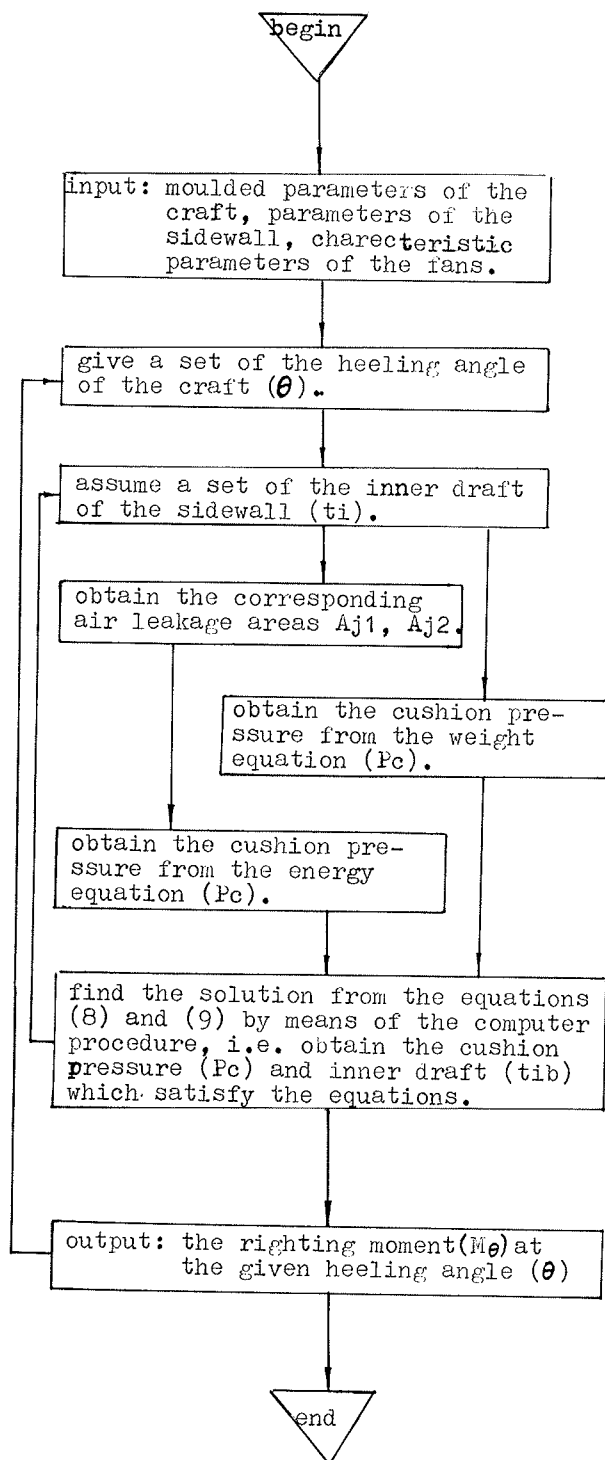
- (3) It is necessary to work out the designing standards for transverse stability of hovercrafts.

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- (2) Б.А.Колызаев, А.И.Косоруков, В.А.Литвиненко, Г.И.Попов, "Особенности проектирования судов с новыми принципами движения". Ленинград, Л., Судостроение 1974.
- (3) Б.А.Колызаев, А.И.Косоруков, В.А.Литвиненко, "Справочник по проектированию судов с динамическими принципами подержания". Ленинград, Л., Судостроение, 1980.
- (4) David, A. Lavis "The development of stability standards for dynamically supported craft, a progress report"
- (5) "Stability and control of hovercraft" Department of Industry, U.K. 1980.

## APPENDICES

1. Block diagram without L.S.K.



2. Block diagram with L.S.K.

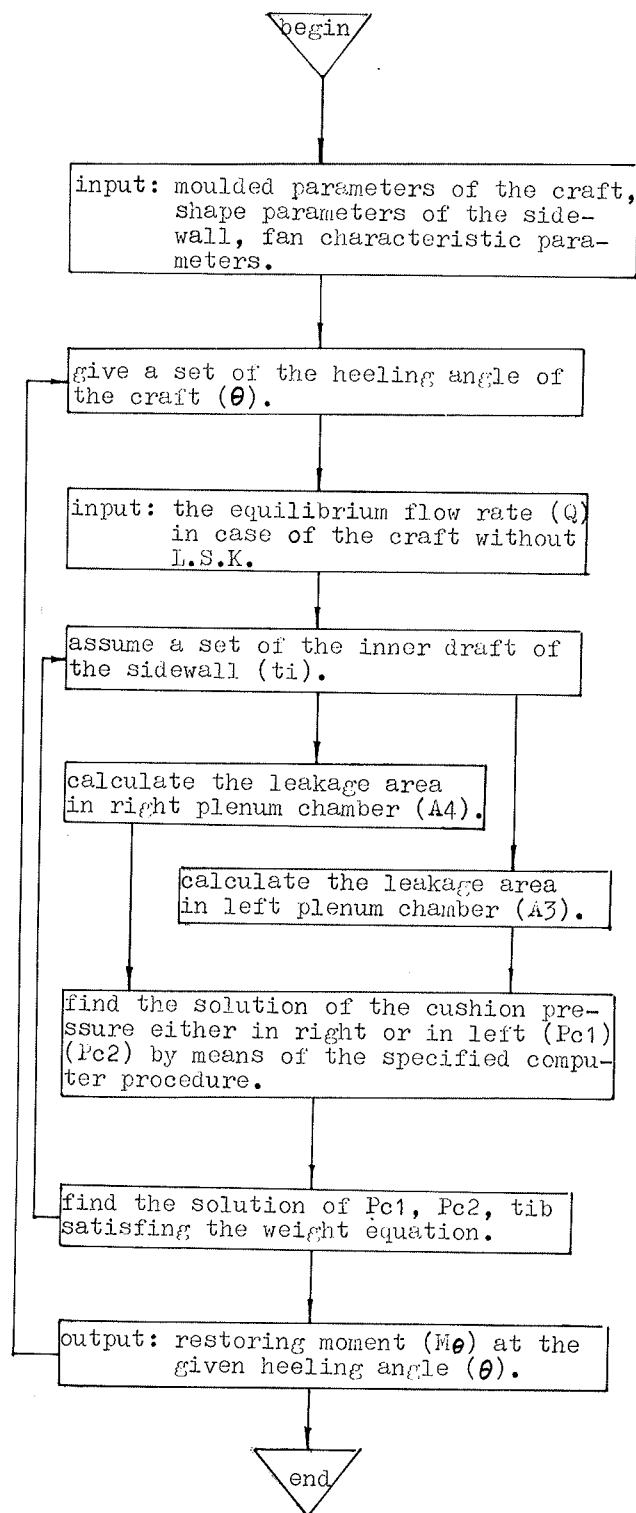


Table (1) particulars of some sidewall Hovercrafts

|                |       |       |       |       |       |       |              |              |
|----------------|-------|-------|-------|-------|-------|-------|--------------|--------------|
| code           | No.3  | 717   | 717A  | 717C  | 713   | XR-1  | HM-2 mark IV | Gorkovchanin |
| country        | China | China | China | China | China | U.S.  | U.K.         | U.S.S.R.     |
| w (tons)       | 1.825 | 13.70 | 15.95 | 18.20 | 28.0  | 10.0  | 25.0         | 14.3         |
| Lc/Bc          | 3.15  | 3.77  | 4.34  | 4.94  | 3.5   | 3.5   | 3.17         | 5.8          |
| Bsw/Bc         | 0.061 | 0.077 | 0.077 | 0.077 | 0.080 | 0.045 | 0.077        | 0.060        |
| $\alpha$ (deg) | 65    | 65    | 65    | 65    | 60    | 60    | 45           | 60           |
| Hsw/Bc         | 0.119 | 0.143 | 0.143 | 0.143 | 0.40  | 0.076 | 0.187        | 0.130        |

Table (2) Particulars of Both Models

| Description | W<br>(kg) | Lc<br>(m) | Bc<br>(m) | Hs<br>(m) | Bsw<br>(m) | $\alpha$<br>(deg) | Pc<br>(kg/m <sup>2</sup> ) | B1<br>(m) |
|-------------|-----------|-----------|-----------|-----------|------------|-------------------|----------------------------|-----------|
| Model 1     | 15.6      | 1.61      | 0.35      | 0.056     | 0.027      | 65                | ~29                        | 0.015     |
| Model 2     | 40.6      | 2.00      | 0.492     | 0.167     | 0.06       | 75                | ~37                        | 0.025     |

Table (3) The first set of variables

|                |          |          |          |          |        |
|----------------|----------|----------|----------|----------|--------|
| $\alpha$ (deg) | 40       | 45       | 50       | 55       | 60     |
| $\beta$ (deg)  | 55 60 65 | 55 60 65 | 60 65 70 | 60 65 70 | 65 75  |
| Bsw(m)         | 0.62     | 0.54     | 0.472    | 0.414    | 0.362  |
| Bsw/Bc         | 0.177    | 0.154    | 0.135    | 0.1183   | 0.1035 |

Table (4) The second set of variables

|                |             |             |             |                   |
|----------------|-------------|-------------|-------------|-------------------|
| $\alpha$ (deg) | 45          | 50          | 55          | 60                |
| B1(m)          | 0.16 0.18   | 0.16 0.18   | 0.16 0.18   | 0.16 0.18 0.20    |
| Bsw/Bc         | 0.166 0.171 | 0.146 0.152 | 0.130 0.135 | 0.115 0.121 0.128 |

Table (5) Variables of height of the L.S.K.

|                         |       |       |       |       |       |       |
|-------------------------|-------|-------|-------|-------|-------|-------|
| Lh (m)                  | 0.15  | 0.17  | 0.19  | 0.21  | 0.23  | 0.56  |
| $\frac{Lh - Y}{Hs - Y}$ | 0     | 0.049 | 0.098 | 0.146 | 0.195 | 1     |
| $\frac{h}{Bc}$          | 0.207 | 0.196 | 0.191 | 0.189 | 0.187 | 0.143 |
| h (m)                   | 0.725 | 0.685 | 0.670 | 0.660 | 0.650 | 0.500 |



Fig.1 No.3 test sidewall hovercraft



Fig.4 717C sidewall hovercraft



Fig.2 717 sidewall hovercraft

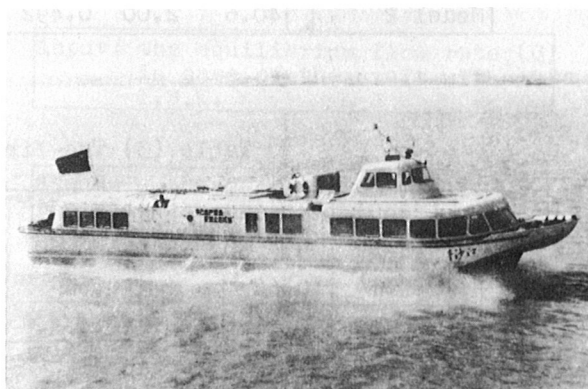


Fig.5 713 sidewall hovercraft



Fig.3 717A sidewall hovercraft

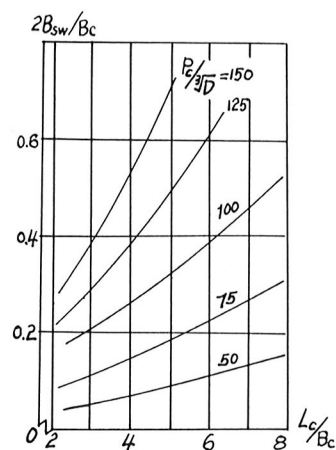


Fig.6 Relative width of sidewall versus length-beam ratio with various pressure coefficient



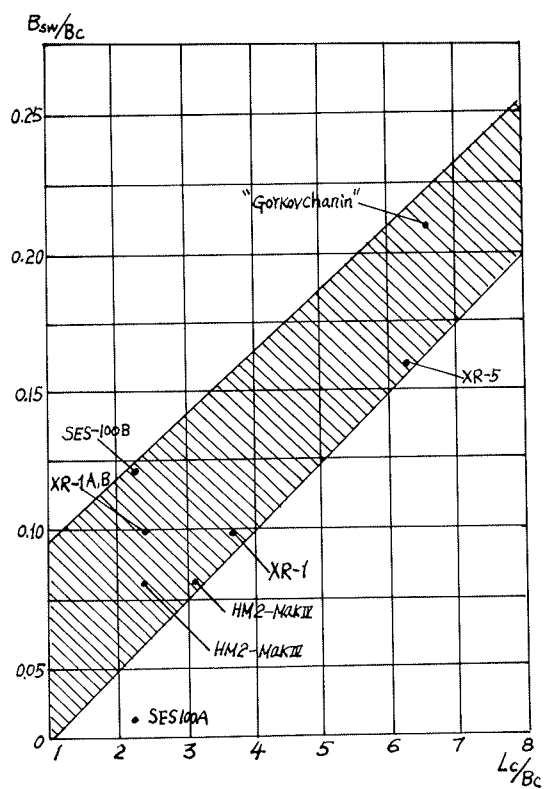


Fig.7 Relative sidewall width versus pressure length ratio

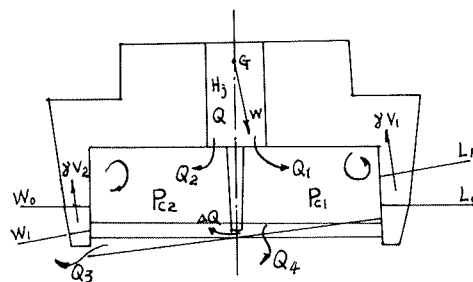


Fig.9 Typical transverse section with L.S.K.

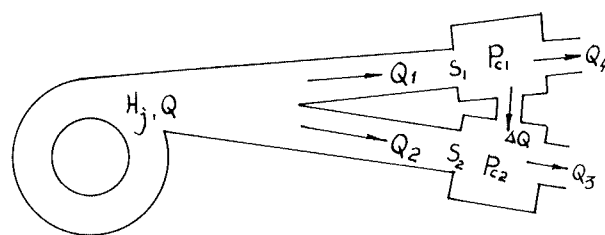


Fig.10 Network of the fan-duct-cushion system with L.S.K.

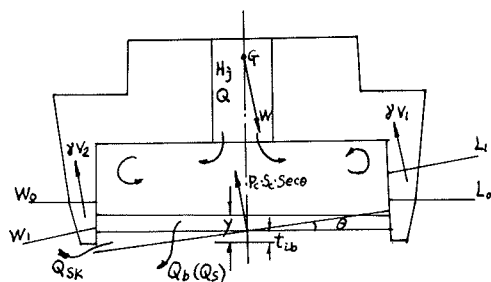
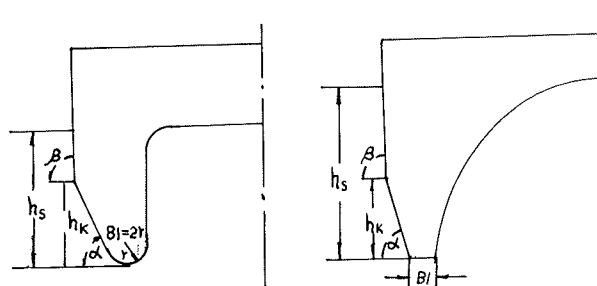


Fig.8 Typical transverse section without L.S.K.



No.1 model

No.2 Model

Fig.11 Typical transverse section for both models.

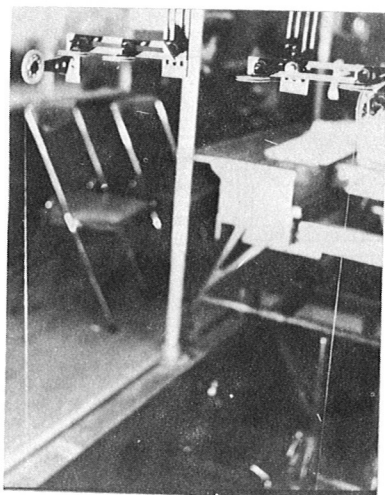


Fig.12 The clinometer

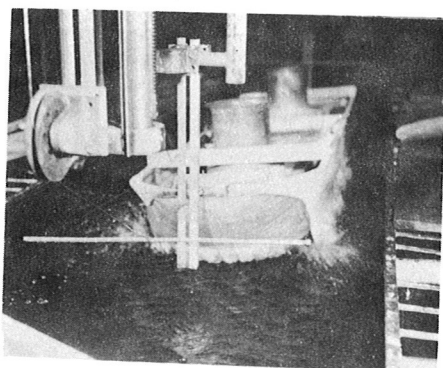


Fig.13 Model in test (1)

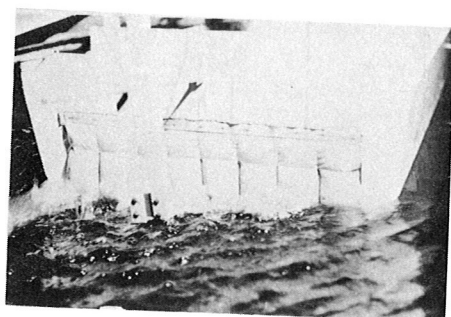


Fig.14 Model in test (2)

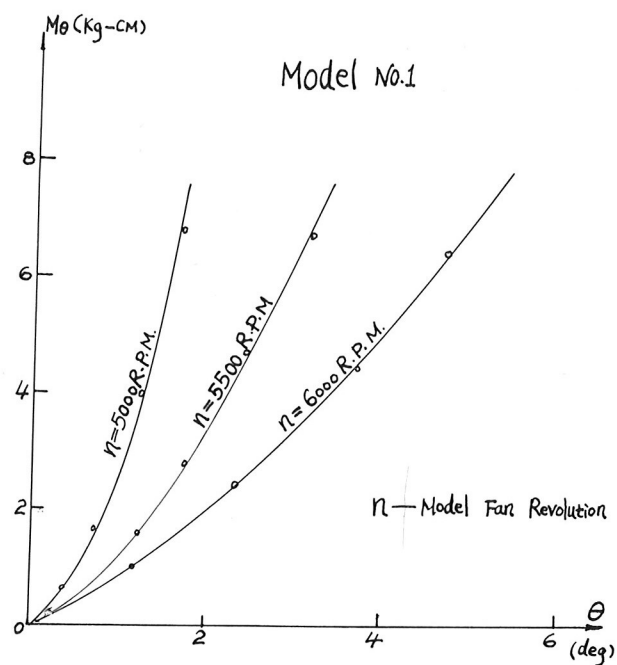


Fig.15 Righting moment versus heeling angle with various fan revolution.

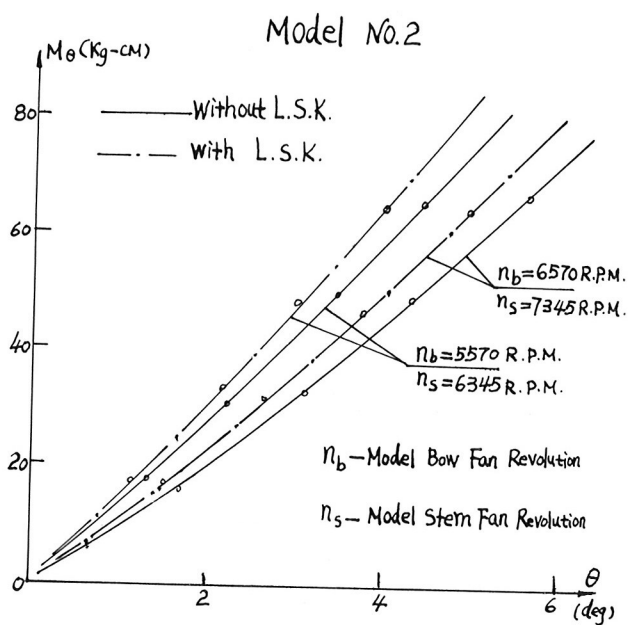


Fig.16 Righting moment versus heeling angle with various fan revolution.

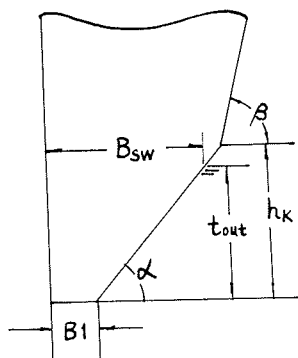


Fig.17 The geometric configuration of typical sidewall

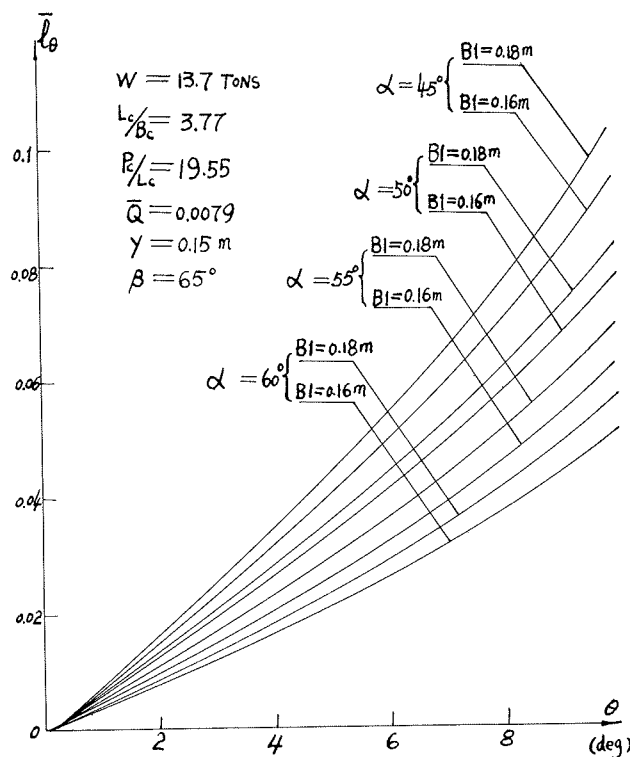


Fig.19 Relative righting arm versus heeling angle with various deadrise angle and width of keel plate of sidewall.

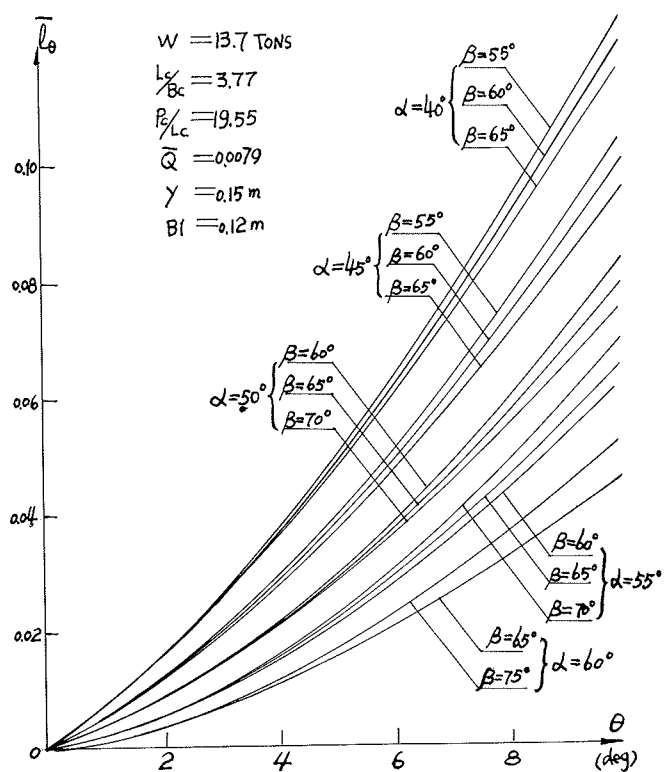


Fig.18 Relative righting arm versus heeling angle with various deadrise angle and flare angle of sidewall above hard chine.

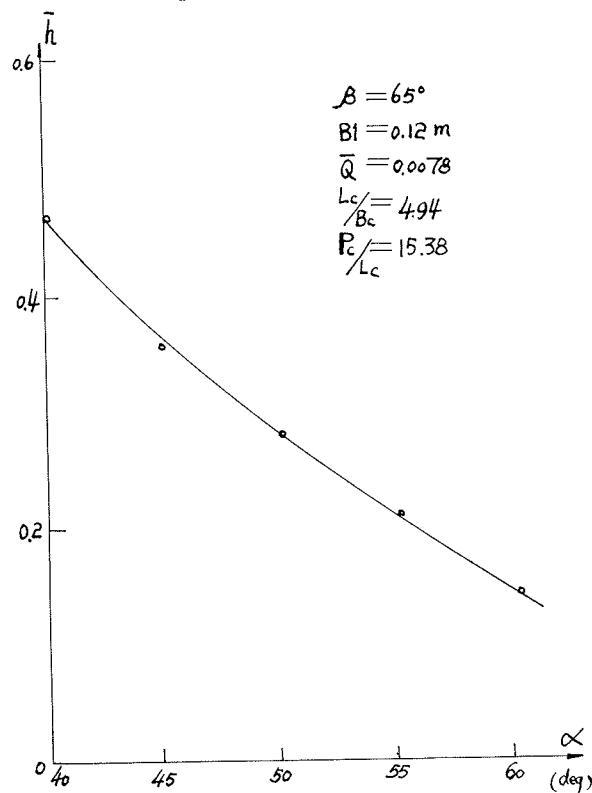


Fig.20 Relative metacentric height versus deadrise angle.

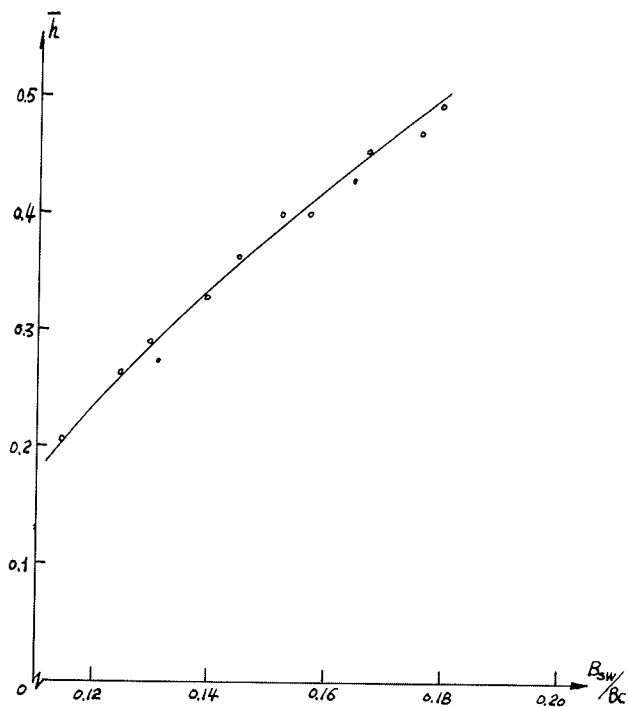


Fig. 21 Relative metacentric height versus relative sidewall width at outer water-line on cushion

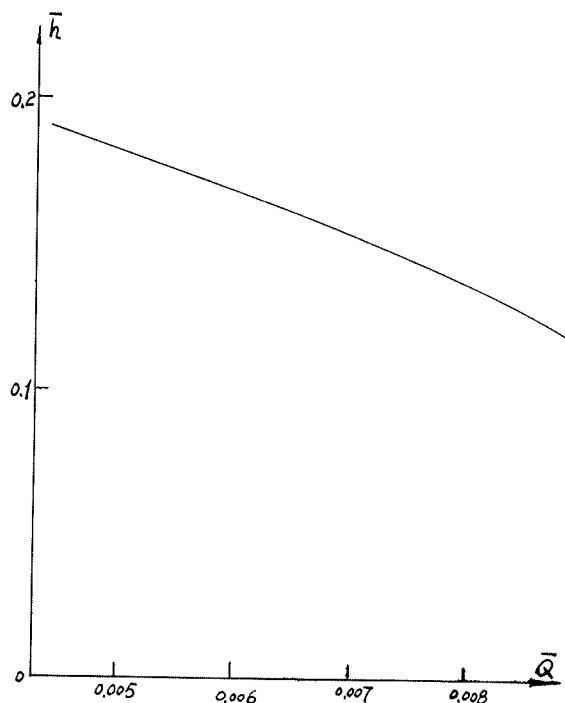


Fig. 23 Relative metacentric height versus relative flow coefficient.

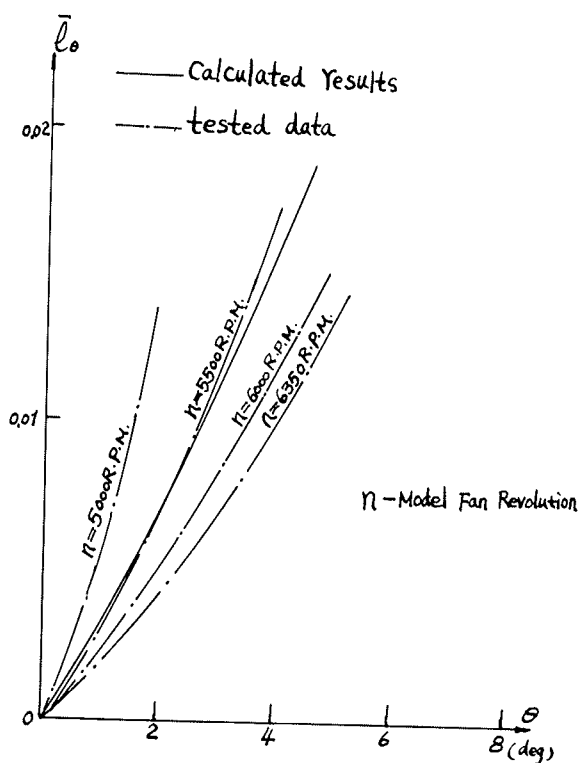


Fig. 22 Relative righting arm versus heeling angle with various fan revolution

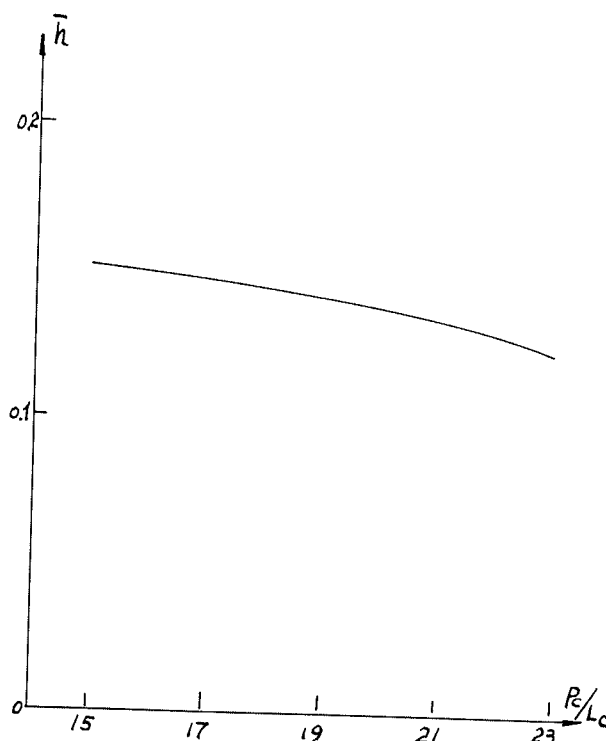


Fig. 24 Relative metacentric height versus pressure length ratio

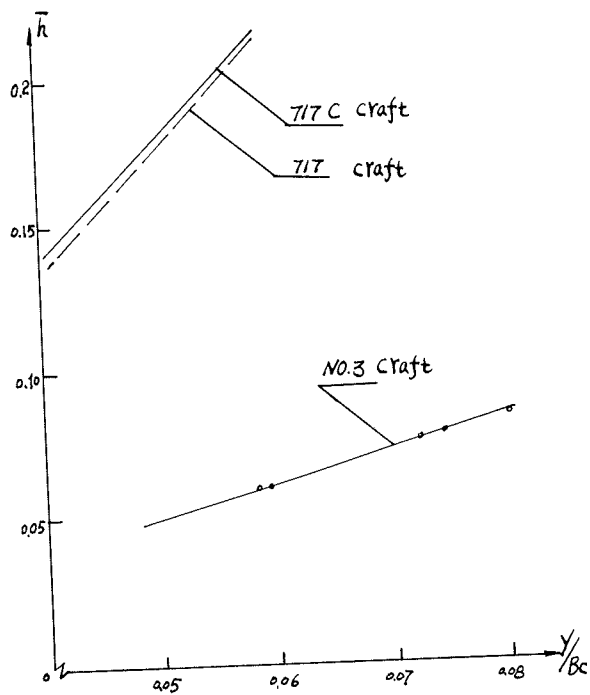


Fig. 25 Relative metacentric height versus relative gap between lower edge of the bow/stern seal and base line

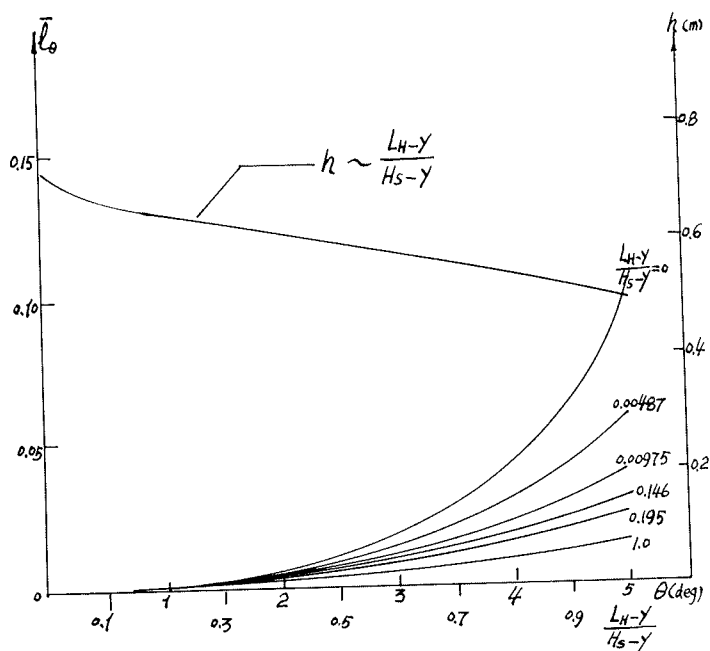


Fig. 27 Relative righting arm versus heeling angle and metacentric height versus parameter  $(lh-Y/Hs-Y)$

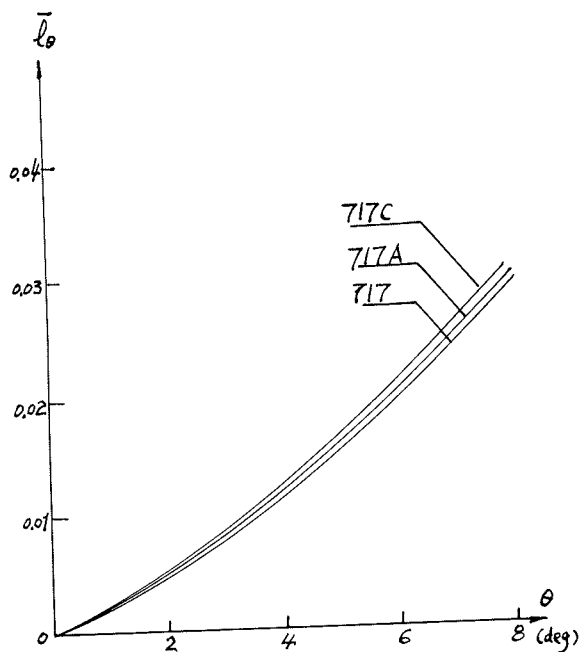


Fig. 26 Relative righting arm versus heeling angle for craft 717 and its variety.

## CAPSIZING EXPERIMENT ON A TOTALLY ENCLOSED LIFE BOAT

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1. INTRODUCTION

In connection with the amendment of the Chapter 3 of 1974 SOLAS convention, a new requirement for life boats is being developed at IMO and will be adopted by Maritime Safety Committee in May 1983.

According to the new requirement, all the life boats except for passenger ships have to be of totally enclosed type and should have self righting property.

As shown later, it will not be too difficult to design a life boat to be of self righting. However, it may happen that due to unusual loading or due to long term rise of the center of gravity, such a boat will have less stability range than 180 degrees in actual cases.

However, a boat which has a specified large range of stability and just small range for negative stability will be very difficult to capsize, and if it capsized, may easily re-right by successive waves. Such a boat will be deemed to be practically self-righting.

In this paper, the authors investigate into relationship between the range of stability of boat and easiness of capsizing and re-righting, and therefore, clarify a minimum range of stability at which a boat is deemed to be practically self-righting.

2. SHIP FORM USED FOR THE EXPERIMENT

Two ship forms, one is an existing non-self righting boat named A type and the other is self-righting named D type were used. Principal dimensions of A type is shown in Table 1, and general arrangement of A type is shown in Fig.1. Midship sections of type A and D are compared in Fig.2.

Stability curves of type A and D in full loaded condition are shown in Fig.3, and those at light condition are shown in Fig.4.

Table 1. Particulars of the tested boat (Original A type)

|                   |   |
|-------------------|---|
| Length            | 8.000 m   |
| Breadth           | 2.800 m   |
| Depth             | 1.220 m   |
| Total volume      | 18.205 m <sup>3</sup>   |
| Engine space      | 1.132 m <sup>3</sup>  |
| Net volume        | 17.073 m <sup>3</sup>   |
| No. of passengers | 45  |
| Speed             | 6 kts   |
| Main engine       | Lister Marine Diesel Engine<br>HRW 2MG/R 2.95 HP/<br>2,200RPM |
| Vol. of fuel tank | 180 ℓ   |

|                | Light loaded | Full loaded |
|----------------|--------------|-------------|
| Displacement   | 3.7977t      | 7.6727t     |
| Draft (mean)   | 0.427 m      | 0.684 m     |
| Trim           | 0.397 m      | 0.301 m     |
| TPC            | 0.133 t      | 0.153 t     |
| KB             | 0.256 m      | 0.410 m     |
| BM             | 1.338 m      | 1.018 m     |
| KM             | 1.594 m      | 1.428 m     |
| GM             | 0.774 m      | 0.479 m     |
| C <sub>b</sub> | 0.380        | 0.490       |
| C <sub>p</sub> | 0.580        | 0.620       |
| C <sub>w</sub> | 0.540        | 0.642       |
| C <sub>ø</sub> | 0.680        | 0.790       |

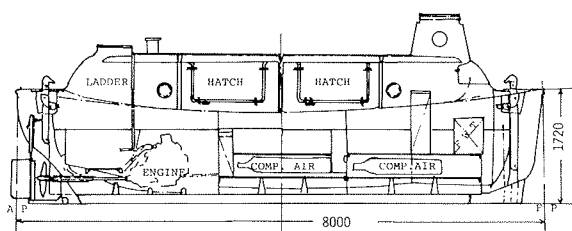


Fig. 1 General arrangement

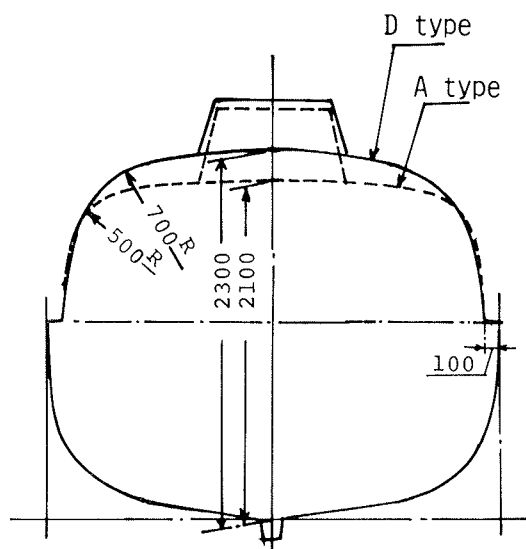


Fig. 2 Midship section (A & D)

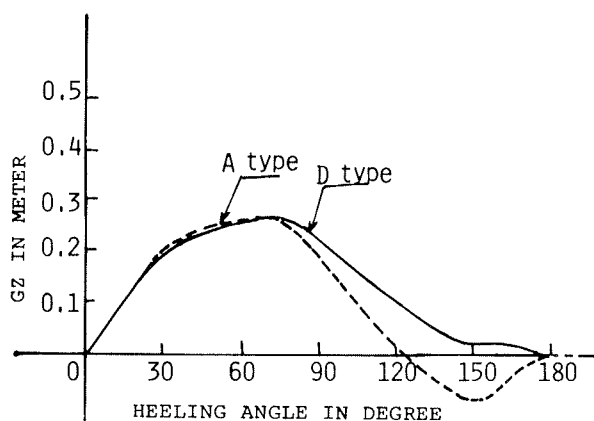


Fig. 3 Stability curve (full load, A & D)

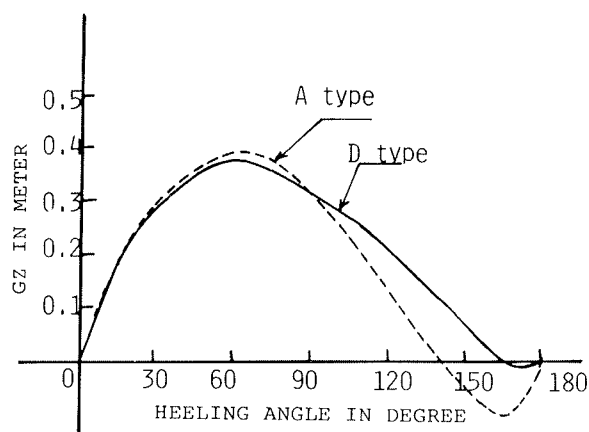


Fig. 4 Stability curve (light load, A & D)

### 3. MODEL EXPERIMENTS TO CLARIFY THE MINIMUM RANGE OF STABILITY AT WHICH A BOAT BECOMES PRACTICALLY SELF-RIGHTING

#### 3.1 Objectives

It is a matter of course that a perfect self-righting boat i.e. a boat which has 180° stability range does not capsize under any circumstance, and therefore, there is no need to test it.

In case of a boat, range of stability of which is narrower than 180° and therefore has negative stability to some extent, the following two cases will be expected to happen depending on wideness of the range of stability.

(1) Capsizes by a specified magnitude of

waves, but re-rights by smaller waves than the former.

(2) Very difficult to capsize, and if capsizes, does not stay at any upside down condition due to a residual momentum in capsizing, and re-rights again turning 360°.

In case of (2), the boat is deemed to be practically self-righting.

In case of (1), if the magnitude (for instance wave height) necessary to re-right the boat is much less than that of wave height necessary to capsize the boat, then the boat will be deemed to be practically self-righting because such magnitude of waves to be able to re-right the boat will necessarily exist in the area where waves to be capable of capsizing the boat exist.

To examine these matters, ship models with different range of stability were

tested in waves and easiness of capsizing and re-righting was investigated.

To obtain models with different range of stability, the center of gravity of Type A was adjusted so that the stability range becomes  $60^\circ$ ,  $90^\circ$ ,  $120^\circ$  (original) and  $132^\circ$ , and the center of gravity of Type D was also changed so that the stability range becomes  $90^\circ$ ,  $120^\circ$ ,  $150^\circ$ ,  $180^\circ$  (original). Therefore, 8 kinds of models with different range of stability were tested in waves.

### 3.2 Models

Principal dimensions of 1/10 scale models are shown in Table 2. The variation of center of gravity of type A is shown in Table 3, and that of type D is shown in Table 4.

Variation of stability curves due to change of the center of gravity of type A and type D is shown in Figs. 5 and 6 respectively.

Table 2

| Item       | Full scale | Model   |
|------------|------------|---------|
| Length     | 8.000 m    | 0.800 m |
| Breadth    | 2.800 m    | 0.280 m |
| Depth      | 1.220 m    | 0.122 m |
| Draft fore | 0.538 m    | 0.0538m |
| Draft aft  | 0.839 m    | 0.0839m |
| Draft mean | 0.676 m    | 0.0676m |
| Trim       | 0.301 m    | 0.0301m |

Table 3. Condition of A type models (model scale)

| Name               | A-1        | A-2        | A-0                    | A-3         |
|--------------------|------------|------------|------------------------|-------------|
| Range of stability | $60^\circ$ | $90^\circ$ | $120^\circ$ (Original) | $132^\circ$ |
| Dispt.             | 7.673 tkg  |            |                        |             |
| $d_m$              | 0.0676 m   |            |                        |             |
| KG                 | 0.1264 m   | 0.1115 m   | 0.0949 m               | 0.0851 m    |
| GM                 | 0.0107 m   | 0.0256 m   | 0.0422 m               | 0.0520 m    |
| Roll period        | 1.64 sec.  | 1.13 sec.  | 0.892 sec.             | 0.879 sec.  |

Table 4. Condition of D type models (model scale)

| Name               | D-1        | D-2         | D-3         | D-0                    |
|--------------------|------------|-------------|-------------|------------------------|
| Range of stability | $90^\circ$ | $120^\circ$ | $150^\circ$ | $180^\circ$ (Original) |
| Dispt.             | 7.969 tkg  |             |             |                        |
| $d_m$              | 0.0676 m   |             |             |                        |
| KG                 | 0.1177 m   | 0.1078 m    | 0.1004 m    | 0.0960 m               |
| GM                 | 0.0201 m   | 0.0297 m    | 0.0366 m    | 0.0408 m               |
| Roll period        |            | 1.00 sec.   | 0.281 sec.  | 0.875 sec.             |

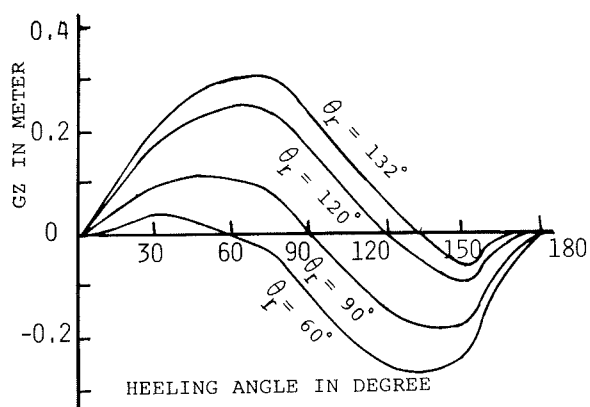


Fig. 5 Stability curves (A type)

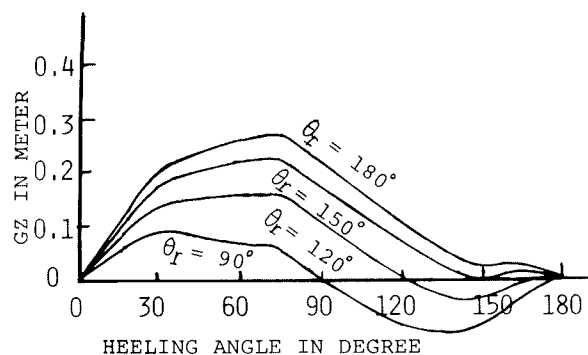


Fig. 6 Stability curves (D type)



Since the models should be water tight, it is difficult to adjust the center of gravity keeping watertightness.

Therefore, a special equipment to adjust the center of gravity by shifting a weight driven by a stepping motor was installed in each model so that the center of gravity can be adjusted by pulse signals fed from outside of the model.

### 3.3 Waves Used in the Experiment

It will be an ideal way to test a model in irregular waves which has a spectrum corresponding to a specified sea state (say sea state 6).

However it will take much time until the boat encounters a severe combination of elementary waves which is enough to let the model capsize.

In view of this matter, series of transient waves which will produce breaking wave were finally used for the experiment.

To examine easiness of capsizing and re-righting, it is necessary to test a model in transient waves with different magnitude (say height) and obtain a critical wave height at which the model will marginally capsize or re-right.

To obtain such transient waves with different wave height, wave spectra are modified so that it will produce specified wave height. The highest wave height used is 55cm in model scale, spectrum of which corresponds to sea state 6.

An example of the wave spectrum and record of the stroke of the wave generator are shown in Fig.7, and an example of special profile of generated transient wave is shown in Fig.8.

It should be noted that the wave spectra used to generate transient waves are not for fully arisen sea, but that of partially arisen sea. The reason to have used spectra for partially arisen sea is that the wave generator is not capable of generating low frequency waves contained in fully arisen sea spectra, and that only higher frequency component of wave spectra is seemed to have significant contribution to capsizing phenomena.

The transient waves used in experiment were considered to be very phenomenal case which appear at very small probability in circumstances specified by such wave spectra.

### 3.4 Capsizing Experiment

#### (1) Experiment on A type models

A-1~A-4 models were exposed in transient waves of different wave height. Result is as shown in Fig.9 where the abscissa is the range of stability and the ordinate is the wave height of transient waves. White circles in the figure show that the model did not capsize and black circles mean that the model capsized. Circles which are partially black mean that the model sometimes capsized. 360° turn

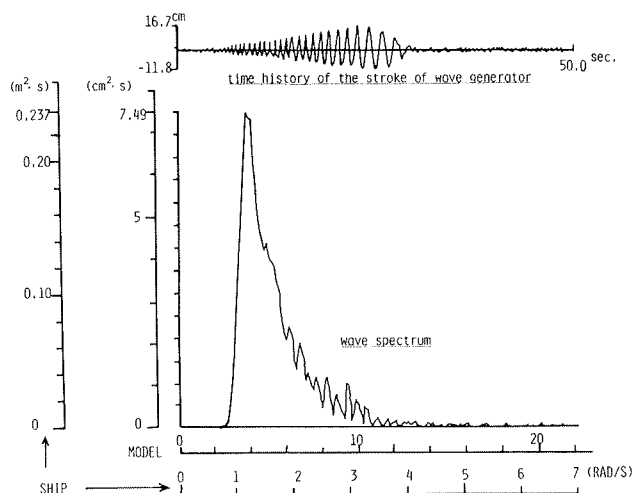


Fig. 7 Wave spectrum and time history of the stroke of wave generator

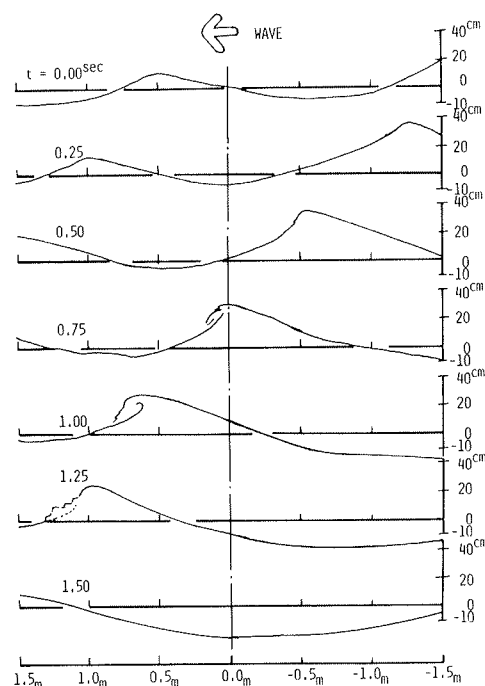


Fig. 8 An example of profile of transient wave (Sea state 6)

means that the model turned 360° and re-righted. Broken line with star marks shows a margin line for capsizing, i.e., the model does not capsize by waves lower than this line.

Re-righting experiment was also done in the same manner that models were exposed to transient waves at upside-down condition and were observed if they re-right or not.

Results are as shown in Fig.10 in the same manner as in Fig.9. Only one difference of Fig.9 and Fig.10 is that capsizing in Fig.9 is re-righting in Fig.10.

Broken line with star marks in Fig.10

means that the model did not re-right by waves lower than this line.

It will be noticed that the wave height which is necessary to let the model capsize becomes greater as the range of stability becomes wider as seen in Fig.9, and that the wave height which is necessary to let the model re-right becomes smaller as the range of stability becomes wider as seen in Fig.10.

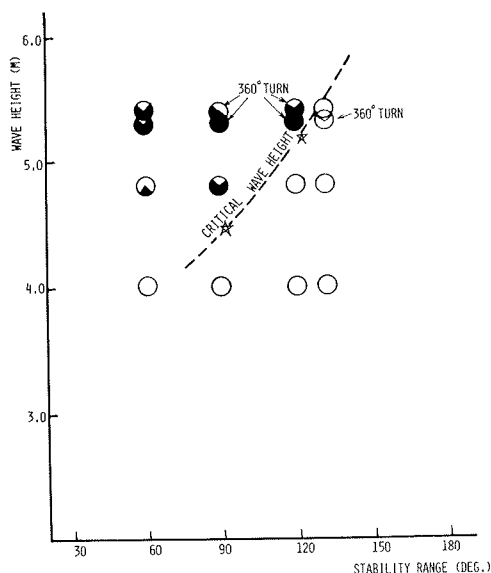


Fig. 9 Capsizing experiment on A type models

## (2) Experiment on D type models

D type models (D-1~D-4) were also tested in waves in the same manner as Type A models.

Results are as shown in Fig.11 (capsizing) and Fig.12 (re-righting) in which, way of expression is exactly the same as Figs. 9 and 10.

Since the range of stability of D type models are wider than A type models, results shown in Figs. 11 and 12 cover results for wider range of stability than Figs. 9 and 10.

It will be noticed that the margin lines for capsizing wave height in Fig.9 and Fig.11 do not exactly overlap. This seemed to be due to difference of stability curves as shown in Figs. 5 and 6.

The same trend will be observed in Figs. 10 and 12 for margin lines for re-righting wave height.

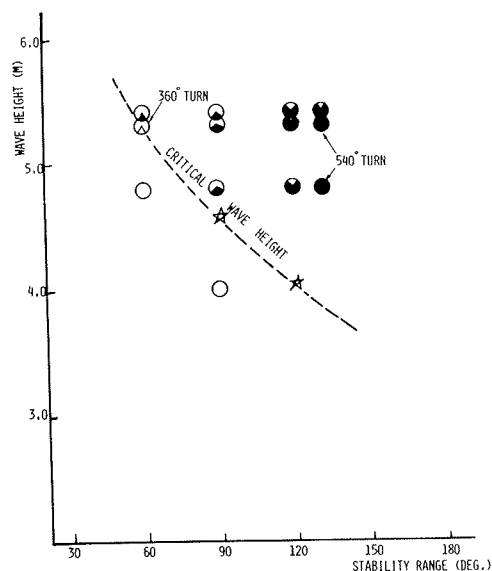


Fig. 10 Re-righting experiment on A type models

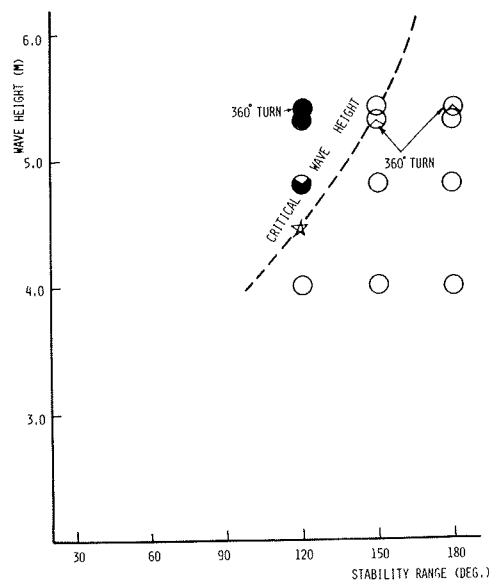


Fig. 11 Capsizing experiment on D type models

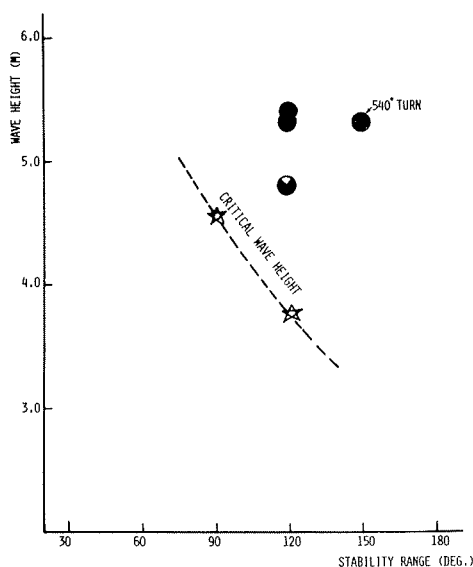


Fig. 12 Re-righting experiment on D type models

### 3.5 Analysis of the Results

- (1) Relation between the range of stability and critical wave height for capsizing and re-righting

As has been expected, critical wave height for capsizing increases as the range of stability becomes wider as seen in Fig.9 and Fig.11. And as in case of  $\theta_r$  (range of stability) is  $132^\circ$  for type A and  $150^\circ$  for type D, the model did not capsize easily and if capsized, it often re-right again turning  $360^\circ$  due to residual momentum of capsizing. In case of  $\theta_r = 180^\circ$  in D type model, the model of course did not capsize, but often turned  $360^\circ$ .

In case of re-righting experiment, critical wave height for re-righting becomes smaller as the range of stability becomes wider as has been expected. It was also observed that when the range of stability is wider than  $130^\circ$ , the model re-rights easily not only by turning  $180^\circ$  but often by turning  $540^\circ$ .

Summarizing aforementioned results, it can be concluded that when the range of stability is wider than  $130^\circ$ , a boat will be very difficult to capsize and if capsizes, it will turn  $360^\circ$  and will re-right easily, or easily re-right by successive waves much lower than the wave which let the boat capsize.

This tendency becomes so strong when the range of stability is wider than  $150^\circ$  that a boat is deemed to be practically self-righting (non-capsizing). In other words, a boat which has the range of

stability wider than  $150^\circ$  is practically impossible to force to capsize by waves.

- (2) Consideration of the procedure of capsizing

In analysing the experiment by video tapes, it was found that there are a few patterns in capsizing procedure.

In Fig.13 the procedure of development of a transient wave as it concentrates and breaks is shown in 0.2 seconds interval from the bottom to the top. In the following figures (Figs. 14-18), behavior of the model is shown superimposed with wave profiles.

(a) Fig.14 shows capsizing procedure of model A with  $\theta_r = 60^\circ$ . In Fig.14, attitudes of the model are shown for each wave profile where a model with black waterline mark show that the model capsized and a model with striped waterline mark show non-capsized case.

In this case the model was forced to drift by breaking wave and capsized just after the breaking wave passed the model.

(b) Fig.15 shows capsizing procedure of type A with  $\theta_r = 90^\circ$ .

In this case the model was wrapped into the breaking wave front and capsized at the front of the wave.

(c) Fig.16 shows capsizing procedure of type A with  $\theta_r = 120^\circ$ .

This case is similar to the former but effect of impulse of breaking wave seems dominant than the former case.

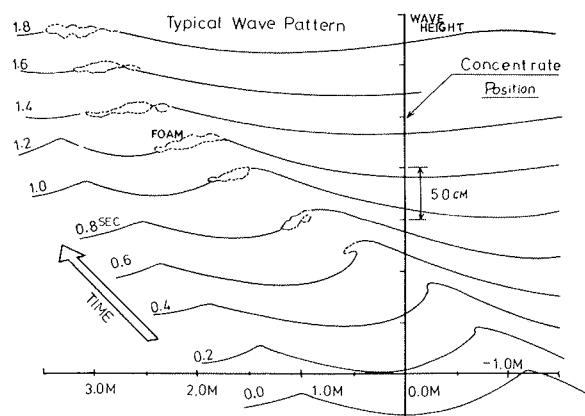


Fig. 13 Development of transient wave

- (3) Effect of position of the model relative to breaking point of the wave.

Fig.17 shows effect of position of the model.

(a) In case when the position of the model is at upstream of the breaking point of the wave, the wave breaks after passing the model, and therefore the model does not capsize.

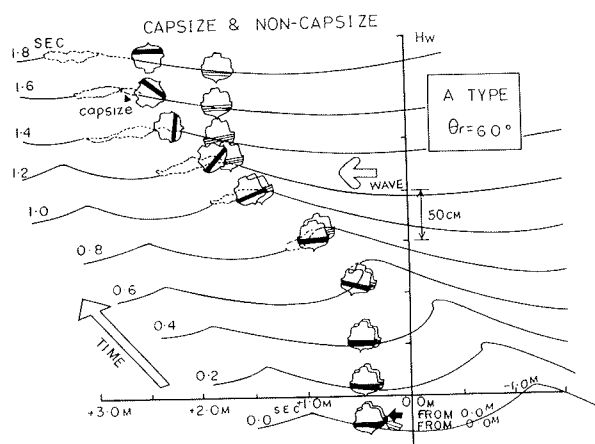


Fig.14 Procedure of capsizing of A type model with  $\theta_r = 60^\circ$

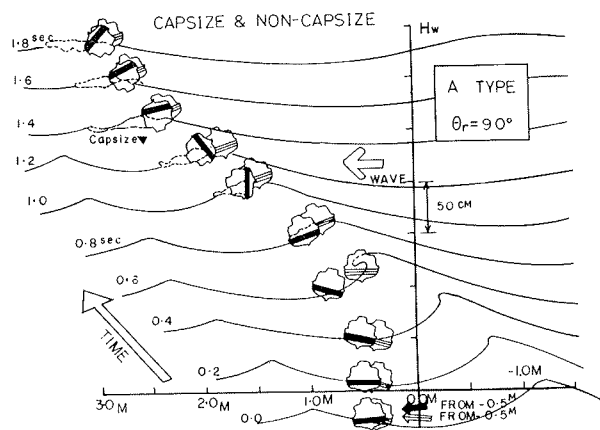


Fig.15 Procedure of capsizing of A type model with  $\theta_r = 90^\circ$

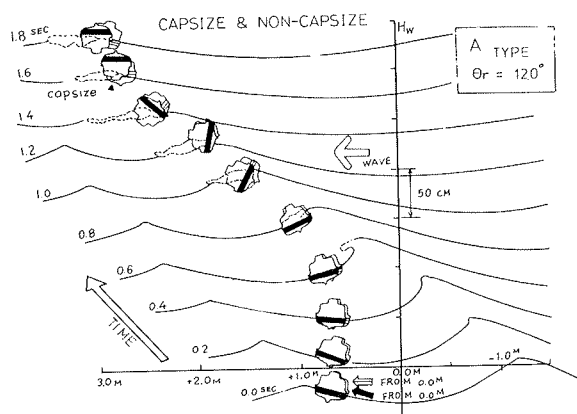


Fig.16 Procedure of capsizing of A type model with  $\theta_r = 120^\circ$

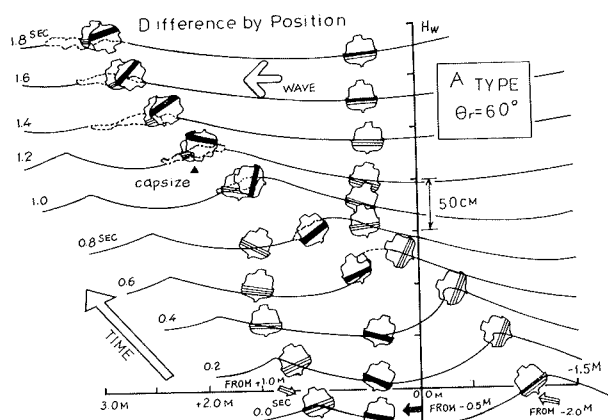


Fig.17 Effect of the model position relative to breaking point on procedure of capsizing

(b) In case when the position of the model is in the vicinity of the breaking point of the wave, the model capsizes (black waterline).

(c) When the model is situated at down stream of the breaking point, as the wave breaks before it reaches the model, resulting drift of the model without capsizing.

(4) A case of re-righting from upside down condition is shown in Fig.18.

In this case, the model is wrapped into breaking wave and re-rights while drifting with the breaker.

Though theoretical analysis has not been done this time, these observations will be useful in further theoretical analysis in the future.

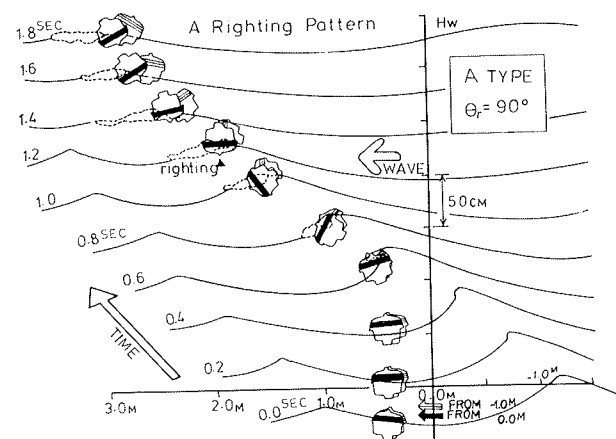


Fig.18 Re-righting at upside-down condition

#### 4. CONCLUSION

Summarizing results obtained, the followings can be concluded.

- (1) Relation between the range of stability and difficulty in capsizing and easiness of re-righting

According to capsizing experiment in breakers produced by transient waves, it is ascertained that as the range of stability of a boat becomes wider, a boat tends to be more difficult to capsize, in other words, the critical wave height at which a boat marginally capsizes increases as the range of stability of the boat increases.

When the range of stability exceeds  $130^\circ$ , a boat will hardly capsize, and if capsized, it will re-right soon turning  $360^\circ$  in most cases. This tendency becomes definite when the range of stability exceeds  $150^\circ$ .

According to re-righting experiment in which a boat was exposed to breakers at upside down condition, it is ascertained that as the range of stability of a boat becomes wider, the critical wave height at which the boat re-rights becomes lower. It was also clarified that when the range of stability exceeds  $150^\circ$ , the critical wave height to be necessary to re-right a boat is so small compared with the critical wave height to be necessary to capsize the boat that the boat never can stay at upside-down condition.

- (2) Consideration of the self-righting property

According to a strict definition of self-righting property, a boat of course should have  $180^\circ$  of the stability range.

However as far as the self-righting property in waves is concerned, a boat of which the stability range is wider than  $150^\circ$ , will re-right in any way because if it capsizes, it will re-right by successive waves even much lower than the wave which let the boat capsize or it will re-right turning  $360^\circ$  due to residual turning momentum of capsizing.

Therefore such a boat of which the range of stability is wider than  $150^\circ$  is deemed to be practically self-righting in waves.

#### ACKNOWLEDGEMENT

This investigation has been carried out as a project of the Shipbuilding Research Association of Japan forming Panel SR 181.

The authors thank all the members of Panel SR 181 and also thank staffs of Ishihara Shipyard Co. Ltd. who have cooperated in producing necessary information of the tested boats.

The authors also acknowledge great assistance given by Messrs. M. Saito and M. Sawada who have undertaken substantial part of this project as their graduate theses at the University of Tokyo.

## Discussion

E. Dahle (The Norwegian Maritime Directorate, Norway)

In the paper, the authors report investigations that have been lacking for a long time.

It is interesting to note the statements in sub-chapter 3.3., regarding the need for a deterministic test in breaking waves of varying height, in stead of carrying out endless experiments in a time-series of waves synthesized from wave spectra. This deterministic method has also been used in Norway for the same reason.

The results in the paper are of course of prime interest for life-boats. However, small boats in general can easily be made near self-rightening provided their deck houses/superstructures/bulwarks can provide buoyancy when submerged. The designer should probably carry out investigations for GZ for much larger angles than today. Further, more test for smaller fishing boats could well be carried out in the same manner as described in the paper. Regarding other aspects of the experiments, discussions to Ref(1) could be of interest.

Ref(1) Dahle, E.A. and Kjaerland, O., "The capsize of M/S HELLAND-HANSEN, Tran. RINA., Vol 122, 1980

K. Fulford (British Shipbuilders, UK)

While it is true that the classical definition of a self righting vessel implies a range of the stability of  $180^\circ$  to provide the necessary condition of being unstable in the inverted position, the paper demonstrates that a vessel which requires significantly smaller waves to right it than to capsize it may be regarded as effectively self righting without having  $180^\circ$  range. How big should this difference be?

The differences in the paper are approx.  
 $130^\circ$  range model A .72 model D .75  
 $150^\circ$  range model A .56 model D .58

and what are the implications for time spent inverted?

It is also noticeable that for a given range of stability model D is easier to capsize than model A due to differing GZ curves. Would this parameter not need to

be considered also?

There is after all little value in being self righting if this means too frequent rolling over.

Do the authors intend to investigate the trade off between resistance to capsize and greater range?

R. Barr (Hydronautics, Incorporated, USA)

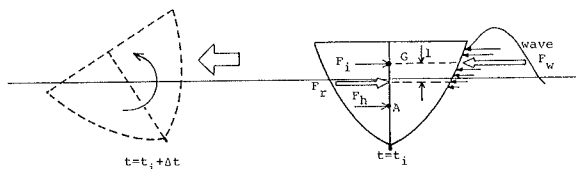
Preliminary analysis of data from capsizing tests of sailing yachts in breaking waves at the U.S. Naval Academy, as described by Prof. Johnson, has been carried out by Mr. Karl Kirkman, Head of the Hydronautics Ship Model Basin. The preliminary results are very interesting and would appear to have important implications for vessels such as lifeboats and fishing boats, as well as yachts. These results, which are illustrated in the viewgraph, are as follows:

- 1) The response of the vessel is primarily in sway rather than roll in most cases
- 2) The forces due to breaking waves appear to act primarily above the equilibrium waterline, causing a high point of force application compared with more normal waves
- 3) Capsizing, which occurred frequently, appears to occur primarily as a result of immersion of the leeward gunwale or deck edge, followed by tripping or cartwheeling about the deck edge.
- 4) Capsizing can be delayed or prevented by placing weight high in the vessel to increase roll gyradius and probably by increasing KG and thus, perhaps, actually reducing GM, of course, adequate GM must be maintained to provide stability under other operating conditions.
- 5) These results, which are still very preliminary, would seem to imply that:
  - a) weights should be placed very high to obtain a large roll inertia and gyradius
  - b) the best resistance to capsizing in beam, breaking seas will probably result with a vessel having a combination of large KG and BM, and hence a large roll moment of inertia together with an adequate GM or area under the GZ-curve.

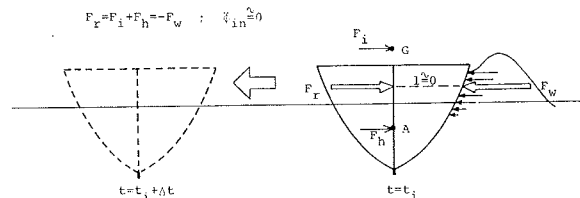
$$F_i = M\ddot{y} = M\ddot{y}, F_h = \frac{1}{2}\rho g \frac{1}{2} \ddot{y}^2 \quad (t \text{ small})$$

typically,  $F_i \approx F_h$

$$F_r = F_i + F_h = -F_w; \quad \ddot{\theta}_{in} = F_w l / (I + I_a)$$



a. design with low KG and roll moment of inertia-impact force produces large roll



b. design with high KG and large roll moment of inertia-impact force produces little roll

EFFECT OF IMPACT FORCE ON ROLL

B. Johnson (U.S. Naval Academy, USA)

I should like to congratulate the authors on a most interesting paper concerning the behavior of self-righting lifeboats in a transient breaking wave. We have run similar type tests on rescue craft and two dimensional yacht models and have also found that the model position relative to the wave breaking point is very significant as to whether the model will capsize or not. We did observe a mode of capsizing, when the model is just downstream of the jet impact point which is shown in the video tape shown of the conference. The plunging breaker jet strikes the water surface creating a secondary wave which may lift the model well above its expected position and result in rolling through 360°.

I question the use of only wave height as a criteria for capsizing, since wave slope is very important in characterizing the type of breaking wave.

B. Arndt (Schiffko GMBH, Hamburg, FRG)

Dear Professor Motora;  
The results of your investigations are of high interest to my work as consulting naval architect. When I was involved in the development of a partially enclosed self-righting lifeboat, I had problems in getting a range of more than 150 deg. of righting levers. 13 variations in height, width and thickness of the buoyancy volume of the fixed cover, which is responsible for the rerighting, were to be investigated for just one boat. Especially with smaller boats (less than 7.3m in length) this leads to a great height of the cover which is disadvantageous for manoeuvrability, due to the large lateral area. The same happened when my company was performing the stability calculations for designs of totally enclosed lifeboats in enclosed and in flooded conditions, whereas 150 deg. of righting levers were to obtain without difficulty. I always was of opinion, that this should be sufficient, as forces which would be able to capsize such a boat would reright it made easier, but up to now I was not in a position to prove it, as there were no means available for performing model or full scale tests in heavy seaway. From now on it will be easier when arguing with the administra-

tion about the definition of the self-righting capability of a lifeboat.

#### Author's Reply

To Dr. Dahle's discussion

Thank you for your encouraging remarks and valuable information. The paper you referred is very interesting for us. We surely will refer it in further work.

I agree with the opinion that GZ at much larger angle should be investigated for small vessels and fishing vessels.

To Mr. Fulford:

As for the first question, we have not checked yet. However as the ratio between critical wave height for re-righting and that for capsizing is known, we may be able to calculate the ratio of probability of appearance of such waves. From that we may get the time spent inversed.

As for the 2nd question, it is true that Model D is rather easy to capsize than Model A if the range of stability is kept the same. As Mr. Fulford pointed out, I feel that this factor should be taken into account in dealing with less stable boats. However, when the range of stability is wider than  $150^\circ$ , a boat is practically self-righting independent of the configuration of the canopy.

As for the third question, I agree that it is necessary to investigate into relation between rolling (level of comfort) and self-righting property.

To Dr. Barr:

Thank you for the interesting information.

As for the opinion that sway motion is important, I quite agree with it and also agree with the opinion that increasing of roll inertia might be of help in preventing capsize.

However, the opinion that to make the center of gravity high to reduce the heeling moment due to impulsive force of a breaking wave seems to me a little tricky.

I wonder if you can increase KG without spoiling the range of stability and the area under the GZ curve. If it is achieved, it will be an ideal way to prevent capsize of a boat.

To Prof. B. Johnson:

Thank you very much for the impressive and shocking viewgraph.

I quite agree to your opinion that a breaking wave should be precisely specified not only by the height, but by slope etc.

To Dr. Arndt:

Thank you for your discussion. I am very glad to know that you have been having the same idea as I have had, i.e. a boat is practically self-righting when its range of stability is wider than  $150^\circ$  degrees, and is quite encouraged in doing further analysis of this problem.

*Session IX*

## Stability of Offshore Structures (Part I)

*Chairmen*

Prof. Fritz Ursell  
Manchester University  
U.K.

Dr. Matao Takagi  
Hitachi Zosen Corporation  
Japan



## STABILITY ANALYSIS OF MAT TYPE JACK-UP DRILLING PLATFORM IN FLOATING CONDITION

MA CHI-LIANG, YAN ZONG-YIN AND PAN PIN

Shanghai Jiao Tong University

China

### ABSTRACT

In terms of unusual geometrical characteristics of mat type jackup drill rig, this paper discusses the following problems on its stability in floating condition: the intact stability subject to sustained wind load in arbitrary direction; the effect of separating distance between the mat and upper hull on the rig's stability in jacking process and the effect of the existence of mat on motion and stability of rig on waves. In addition, some aspects of stability criteria of such rig are involved. The first two problems are solved using computer with varying the wind load direction and the distance between mat and hull of a typical rig's simulation model.

The results may be helpful to designers for improving the stability and developing the operational guidelines of such kind of jackup rigs.

### 1. INTRODUCTION

137 rig accidents were recorded in a period of 26 years (1955-1980)[1], of which 96 mishaps were jackups'. Accidents resulting from movement (towing, moving on and off

location, jacking up and down) amounted to 49 mishaps, of which the overwhelming majority is caused by jackup movement. It implies a great importance in practice to study the stability of jackups.

A jackup drilling platform is made up of upper hull, spud legs with their jacking mechanisms, and bottom bearing structures. The later may be a singular large horizontal bottom mat to which all spud legs are connected at their lower ends, or some individual can tanks attached to the lower end of each leg, or there is no such attached structure at all but with a spherical plate at the lower end of each spud leg itself. This paper discusses the floating stability of the platform with the first type of bottom bearing structure only.

The intact stability criterion accepted for jackup drilling rigs is that commonly known as the weather criterion described in detail by C. Kuo [2]. According to this criterion, the area under the righting moment curve is not to be less than 1.4 times the area under the heeling moment curve at or before the downflooding angle or the second intercept, whichever is less. In this method, three important factors directly affecting stability criteria are: heeling moment caused

by sustained wind, righting moment of the platform and a minimum downflooding angle. The effect of the platform motion in waves on stability is only indirectly introduced in the ratio of 1.4. Such treatment for jackups stability criterion is simpler than that for ships.

Geometry and general arrangement of a mat type jackup drill rig are considerably different from those of an ordinary ship form. Generally, the mat is designed in the light of sit-on-bottom stability requirement, i.e. the ability to sustain bearing load and tipping moment. The mat form may be square, rectangular or "A" shape, with a large opening in its central part for improving rig's on-bottom stability and saving structural weight, and also a drill slot at the stern. Therefore, the length and breadth of the mat is often much greater than that of the upper hull. In addition, a number of skirts are fitted around the mat and openings, with sufficient depth under the mat bottom. All of these make the underwater part of the rig more complex than that of a surface ship. The configuration of the upper hull is very much dependent on the number of legs. It is usual that a rig with four legs forms a rectangular upper hull and a rig with three legs is quite irregular. The distribution of various wind area above water surface of a drilling rig is not uniform. It implies that the stability calculations of mat type jackups cannot be limited in transverse stability as adopted in ship design, and ought to consider the righting moment and heeling moment from any horizontal direction in order to find out the most critical stability axis. Another special and important stability problem of a mat type drill rig is the effect of mat-hull separation distance on stability in jacking process. In order to solve these two problems, a computer program based on the result by Yang Z. [3] is made up, which can be used for calculating and analysing the stability of mat type drill rigs in various floating conditions.

Although the simplified method is adopted in rules of mobile drilling units, the effect of waves on rig and the effect of motions of a rig in waves on its stability is still a very important problem. Using theoretical method to calculate the motions in waves of a mat type jackup rig is difficult due to the existence of a mat which makes a complex underwater part of the rig. Model test is a reliable research method. Whatever we take, the effect of the mat on rig's motion and the effect of motion on the stability of the rig should be clarified.

Flooding is dangerous to rig's stability. There are two categories of flooding: flooding through the under-water damage and flooding through various openings, such as hatch, vent and door, including flooding caused by unefficiency of watertight closures. Rational and careful subdivision of upper hull and mat is an important measure for designers to ensure floodability. Flooding through openings, which is different from under-water damage flooding, is closely relating to motions of the rig and boarding seas, hence is more dangerous to rig's stability than the damage flooding.

## 2. INTACT STABILITY OF JACKUP RIGS UNDER ACTION OF A WIND FORCE FROM ANY HORIZONTAL DIRECTION

Fig. 1 shows a calculation model: a four leg mat type jackup drill rig with rectangular upper hull and mat. Discussion on calculation results derived from this platform may benefit designers to make some conclusions of mat type rig intact stability. Fig. 2 shows a typical longitudinal righting arm curve (displacement 10,000 t, upper hull draft 3.2<sup>m</sup>, hull-mat separation 1.86<sup>m</sup>), it can be known from the figure that the metacentric height and max. righting arm are large, while angle of deck edge immersion and angle of max. righting arm are small. The trough of the curve corresponds to the angle of bilge emergence.

The righting arm varies with the wind

direction angle. Fig. 3 illustrates righting arm vs wind direction angle curve, with the

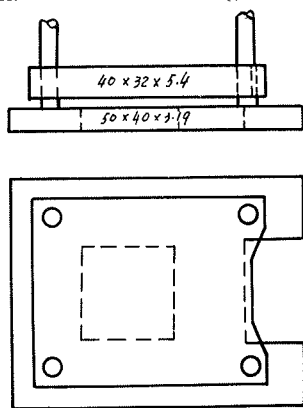


Fig. 1 Calculation Model

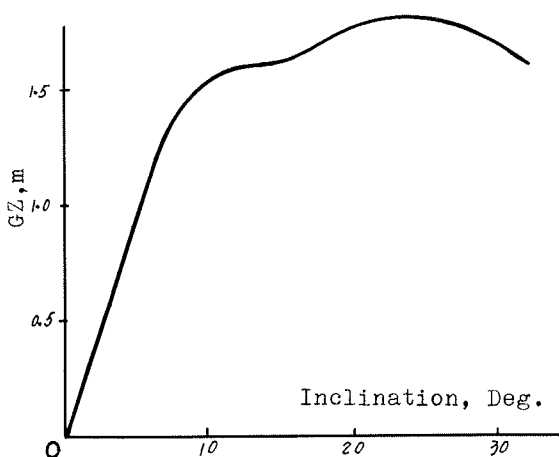


Fig. 2 Typical Righting Arm Curve

angle of inclination of the rig being constantly equal to  $5^\circ$ . It can be seen that the transverse righting lever is the smallest one and the left side of the curve is relatively higher than the right side due to asymmetry of fore and aft body of the rig.

The wind heeling moment depends on wind speed, wind direction angle, angle of inclination of the rig, etc. In calculation according to the rule, all surface area of the whole platform, including above-water and under-water parts, are divided vertically and horizontally into a great number of area elements. The wind force acting on every above-water area element exposed in wind is calculated as follows:

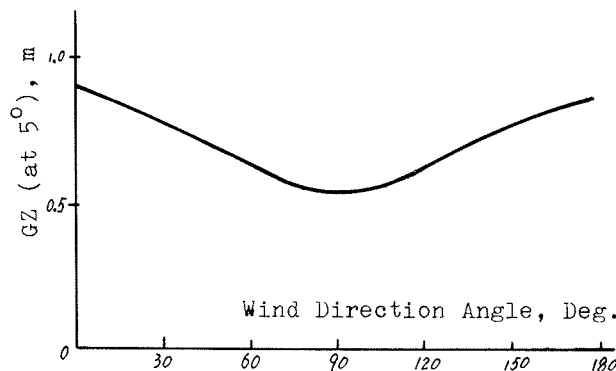


Fig. 3 Righting Arm VS Wind Direction Angle

$$\Delta F = 0.0625 C_s C_h V^2 \Delta A \quad (1)$$

Where:  $\Delta F$  = wind force on considered element, kg

$C_s$  = form coefficient of the considered structural member

$C_h$  = height coefficient of the considered structural member

$V$  = wind speed, m/s

$\Delta A$  = projected area of considered element.

Further, the wind resultant force, its center and the center of water resistance can be obtained, and then it is not difficult to calculate the wind inclining moment.

However, the vertical component of the wind force and hydrodynamic force is neglected and this will appreciably affects the accuracy of the wind moment calculation, especially for mat type drill rig with the large mat, helicopter platform, etc. E. Numata studied this problem in detail and recommended that the best way to solve the problem is to conduct model test in wind tunnel [4].

Fig. 4 shows an example of wind inclining lever curve, assuming the inclination angle of platform is zero, i.e. the platform lies in its upright position. Both two peaks of this curve correspond to the cases of oblique wind direction and not to the wind direction angle equal to  $0^\circ$ ,  $90^\circ$  or  $180^\circ$ . This implies that it is not sufficient for the stability only to check in beam wind and head wind direction and should be made in any horizontal direction.

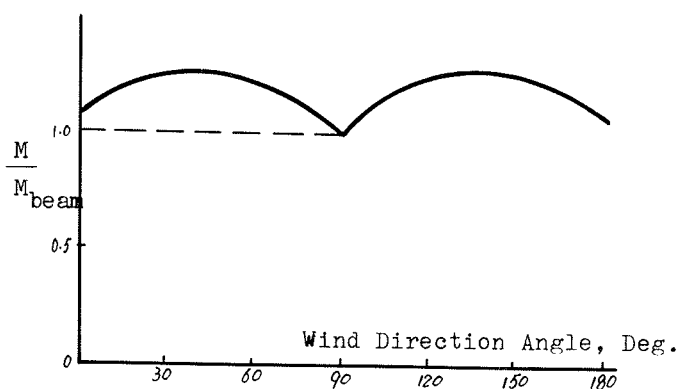


Fig. 4 Wind Inclining Moment VS Wind Direction Angle

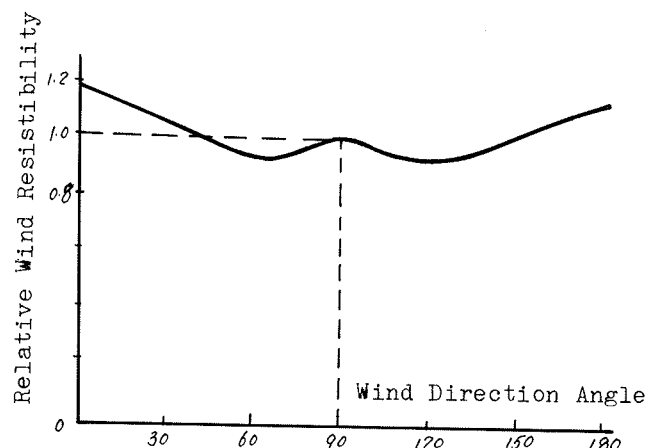


Fig. 5 Relative Wind Resistibility VS Wind Direction Angle

Minimum downflooding angle is to be calculated in stability check. The value of this angle depends on not only rig's configuration, draft and stability axis, but also the arrangement of various openings and the situation of their watertight closures, i. e. it varies largely with designer's considerations. Hence it is difficult to establish the relationship between minimum downflooding angle and wind direction angle. This problem can be simplified if assuming the min. downflooding angle does not vary with the stability axis.

Assuming the displacement of above mentioned calculation model is equal to 10,000t, wind speed 100 knots, min. downflooding angle  $12^\circ$ . The curve obtained from calculation is shown in Fig. 5, which indicates the wind resistibility of the rig varies with wind direction angle. It is obvious that beam wind is not the most dangerous one. However, the wind direction angle close to beam direction is unfavourable. The wind resistibility reaches up to the peak value when the wind direction angle is equal to  $0^\circ$  or  $180^\circ$  (head wind or following wind).

### 3. THE EFFECT OF HULL-MAT SEPARATION ON STABILITY IN JACKING PROCESS

The position of the mat varies continuously in jacking process which affects the initial and large angle stability of the rig.

This problem is of significance in practice.

Fig. 2 shows that the metacentric height is very important by which the area under the stability curve is almost decided, provided that the value of downflooding angle is not greater than  $10^\circ$  to  $15^\circ$ . Due to the weight and displacement of the rig and the metacentric radius keeping constant during jacking process, the variables are only the positions of gravity center and center of buoyancy, then the variation of metacentric height  $\Delta GM$  can be calculated as follows:

$$\Delta GM = \Delta BG = \frac{(W_m - D_m)}{D} (S_1 - S_0) = k(S_1 - S_0) \quad (2)$$

where:  $D$  = rig displacement

$W_m$  = weight of mat and legs, including ballast water in mat

$D_m$  = mat displacement

$S_0$  = min. hull-mat separation

$S_1$  = any other hull-mat separation.

Expression (2) indicates  $GM$  is directly proportional to  $(S_1 - S_0)$ , the amount of which is dependent on the coefficient  $k$ . Generally,  $W_m > D_m$ ,  $GM$  increases with the increase of hull-mat separation.

The effect of separation on large angle stability can be dealt with as follows. The righting lever consists of form stability lever and lever of stability by weights. Assume the mat of the rig is lowered from

minimum separation to another arbitrary position, then the form stability levers under these two floating conditions are the same when the rig is inclined to a same angle, if the inclined rig's mat is wholly immersed at its minimum separation position. Under such condition, the variation of righting lever is only decided by the variation of the lever of stability by weights. Otherwise, the variation of form stability lever is adverse to righting lever when the hull-mat separation increases. Generally, the righting lever decreases with the increase of separation distance.

The effect of separation on wind inclining moment is shown in Fig.6. The wind moment increases with increasing separation due to the fact that the center of water resistance rapidly shifts down, although the wind moment acting on legs decreases when the mat and legs are lowered.

The effect of separation on minimum downflooding angle: downflooding angle keeps constant if the mat is fully immersed before its lowering. Otherwise the downflooding angle will be somewhat increased after lowering.

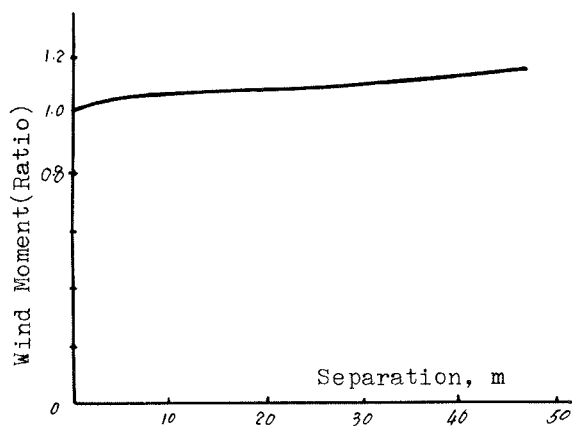


Fig. 6 Wind Moment (Ratio) VS Separation

Due to the mat lowering from minimum separation the wind resistibility of the rig varies as the curve shown in fig. 7. By comparison with the wind resistibility at minimum separation, it gradually decreases

with increasing separation, the reduction may be more than 10%. The dotted line in figure corresponds to large coefficient  $k$ , it indicates that the wind resistibility decreases, too, at the initial stage of mat lowering.

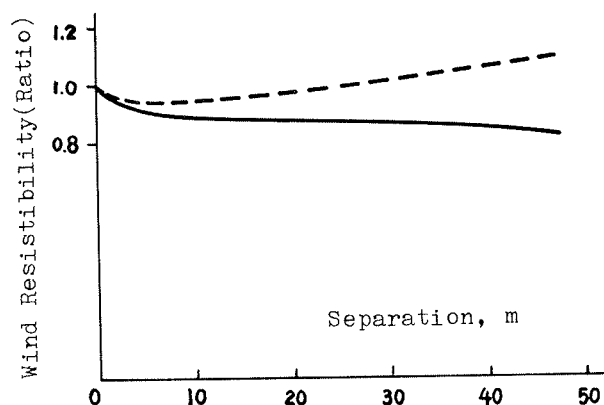


Fig. 7 Wind Resistibility (Ratio) vs Separation

#### 4. THE EFFECT OF EXISTENCE OF MAT ON MOTIONS AND STABILITY OF RIG IN WAVES

By comparison with ordinary type jackup drill rig, mat type jackup drill rig with greater displacement, mass and added mass or mass inertial moment, and smaller metacentric height due to the large added mat (including skirts under mat bottom surface) results in greater natural period and greater motion damping. In addition, the characteristics of exciting force and motion response of this type rig are somewhat similar to that of semisubmersibles due to the existence of a large mat under the upper hull.

Natural periods of ordinary type jackup drilling unit are about as follows: heave 9-11 sec., pitch 9-11 sec., and roll 11-13 sec.[5]. Where, the figures of upper limit relate to large units. Corresponding figures for mat type jackup drilling unit, which may be much greater than above-mentioned, are approximately in the following range: heave 10-13 sec. roll and pitch 15-20 sec. or even

greater than 20 sec., where roll period is somewhat greater than pitch period. These roll and pitch periods or frequencies are far away from wave energy concentration frequency range of a wave spectrum corresponding to moderate even rough seastate, hence the roll and pitch motion of the rig in waves are very small, especially in jacking process, which is generally performed under good weather. However, the heave motion is an exception due to its smaller natural period.

The natural period of a mat type rig varies with hull-mat separation. Practically, hydrodynamic interaction exists between upper hull and mat and varies with this separation. Mironer A. et al [6] studied an ocean platform, similar to mat type jackup drill rig, formed by two coaxial cylinders with a variable separation and the results show that the heave motion added mass and pitch motion added mass inertial moment increase with upper and lower cylinders separation, and approach to constant values if separation is large enough. Hence, heave natural period will approach to a constant value, and pitch natural period increases continuously due to the continuous increase of mass inertial moment of platform itself. The model test result of a mat type jackup drill rig obtained by Sadao Ando [7] indicates that the hydrodynamic coefficients vary with separation, but not increase monotonously. Added mass, added mass inertial moment, dimensionless damping coefficient and natural periods approach to minimum values when hull-mat separation is equal to 10 m. This phenomenon is probably due to the existence of large opening and slot of the mat.

Before description of heave exciting force and response of the rig, the wave excitationless ship forms presented by Seizo Motora are quoted from [8]. He found a series of wave excitationless bodies and pointed out that under specific wave frequencies heave exciting force may be equal to zero. Oo K. M. and Miller N.S. [9] studied

the heave exciting force and motion of semi-submersibles with differing geometries, and explained the zero points occurred on the heave force curve calculated by theoretical method and checked by model test. The authors of this paper were aware that the mat type jackup drill rig was to be possessed of similar to above characteristics, too. Therefore, the heave force and motion response calculations to hull-mat separation 4.5 m and 30 m conditions were made, using the rig information in reference [7]. Strip theory was adopted, but the interaction between upper hull and mat ignored. The calculated results are shown in fig. 8, 9. From fig. 8

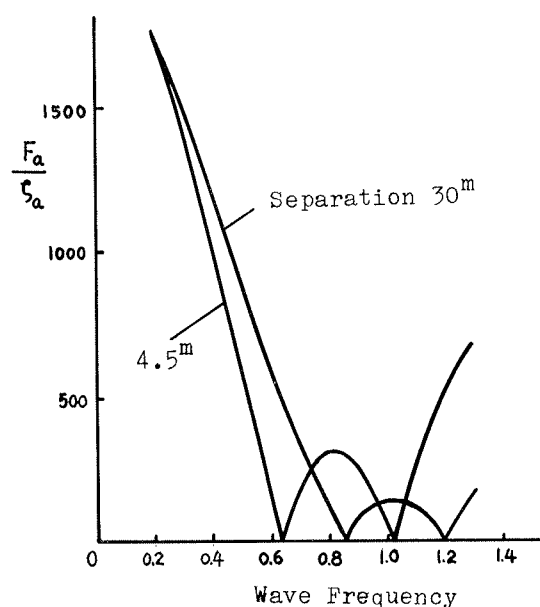


Fig. 8 Heave Exciting Force Amplitude

it can be seen that heave exciting force is reduced in an extensive range of frequency and the first zero point shifts to the lower frequency side due to the exciting force acting on mat being opposite to that acting on the upper hull. However, if the hull-mat separation is large enough, above mentioned advantageous effects will be vanished. The calculated result of heave motion response is shown in fig. 9. The linear damping coefficient values adopted in calculation are also quoted from [7]. Calculated results are in good agreement with the model test, except

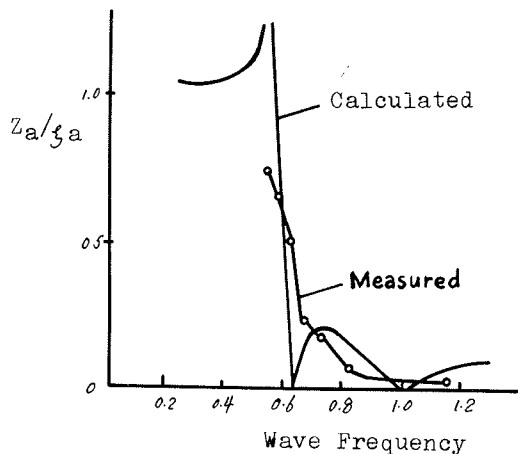


Fig. 9 Heave Response

(a) Separation = 4.5 m

the range of low frequency, where no test data can be used. In fig. 9 (a) the resonance response is slight due to the heave natural frequency being almost equal to the first zero waveless frequency. Otherwise, the resonance response will increase as shown in fig. 9 (b). The response peak values in fig. 9 are overestimated, because the actual damping coefficient is nonlinear.

Under moderate sea states, the roll and pitch amplitude are small due to the large natural periods of mat type rig. Large amplitude response can only be anticipated in very long waves or swell. However, the heave motion is not the case. Fig. 9 shows that large heave motion would occur when mean wave period is close to heave natural period. Simultaneously, due to the large relative motion between rig and wave surface, boarding seas will periodically occur. If the freeboard of the rig is unduly minor, seas will board seriously and make the deck difficult to drain. In such cases, partial flooding in various compartments is entirely possible. Moreover, if any watertight closure loses its efficiency, serious flooding occurs, the rig would lose its stability and may be capsized. In coastland, the mean period of windy waves with high steepness is readily close to rig's heave period and leads to serious heave mo-

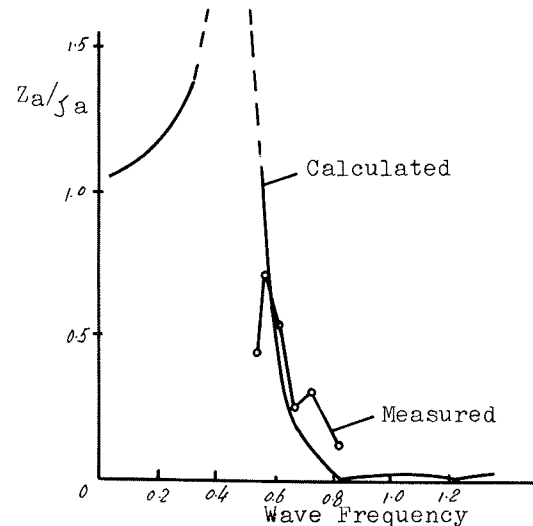


Fig. 9 Heave Response

(b) Separation = 30 m

tion accompanying boarding seas. This situation is disadvantageous to mat type jackup drill rigs.

The following points are important to ensure the operation safety of mat type drill rigs:

- (1) rational arrangement of various openings and reliability of their watertight closures, rational and careful subdivision of upper hull and mat;
- (2) laying down complete operating manual or rigorous operational guidelines, in which, from the stability and seakeeping point of view, the following items are to be included: loading condition (to control variable loads rigorously), the position of the mat at any operating condition including in worsening weather, stowage and securement of every movable item, control of free surface in liquid tank, monitoring and control of rig's motion in waves, etc., detailed statement about this problem may refer to L.C.S. Kobus and L.V. Whittington [10];
- (3) towing speed and towing direction relative to wind and waves: head sea is preferable to beam sea and following sea;
- (4) minimum freeboard should be determined on the basis of compliance with the stability requirements, and seakeeping requirements too,

the later has to be searched further.

## 5. CONCLUSIONS

Some remarks about the stability criteria of mat type jackup drill rig are presented as follows.

(1) In terms of geometrical characteristics and the characteristics of motions in waves of this type rig, the present weather criterion, which does not involve the roll amplitude, is acceptable. Stability calculations are to be performed assuming the wind force induced by a sustained wind from any horizontal direction and the mat position of the rig being in an extensive range as required by the present rules.

(2) The effect of actual freeboard, especially in towing condition, on stability and seakeeping, particularly on boarding seas, is an important problem. The minimum freeboard of the mat type rig is not able to be calculated by the normal method laid down by ICLL. Further research on minimum freeboard requirement from the seakeeping point of view is necessary.

(3) The roll and pitch are small, but the heave motion is large and obviously affects on the stability of the rig. Due to the complexity of this hydrodynamic problem, which can not be solved recently, we recommend that a few complementary requirements are to be added: to stipulate a minimum requirement value for maximum righting lever angle and minimum downflooding angle.

(4) Considering the form characteristics of mat type drill rigs, towing under following sea, quatering sea and beam sea in storm condition is unfavourable from the stability point of view.

(5) The 50 kn sustained wind speed for damage stability calculations is too small, because the effect of waves on the rig and the effect of motions on stability are not considered in damage stability calculations. Either to increase wind speed value or to present certain residual freeboard requirement may be

the proper measure of improvement.

## ACKNOWLEDGEMENTS

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# DAMAGE STABILITY AND SUBDIVISION OF SEMISUBMERSIBLE DRILLING RIGS

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## ABSTRACT

Damage stability calculations are carried out for column stabilized drilling units. Following types of rigs are considered: multi-legged rigs with circularly and rectangularly arranged columns, with and without footings, and with subdivided and non-subdivided columns.

Formulas are set up showing the influence of the geometric characteristics on the angle of inclination in the final stage of flooding. From the results conclusions are drawn as to the type of rig to be preferred from the standpoint of damage stability.

Finally, it is demonstrated that rigs with favourable characteristics can attain a probability of survival of  $P = 1$  without any subdivision of the columns. Rigs which are not primarily designed with respect to damage stability must be subdivided. It will be shown how subdivision must be in order to get also for these rigs high probability values.

## NOMENCLATURE

$a$  = half height of a compartment  
 $B$  = breadth of a rectangular rig (measured between the centers of two opposite columns)  
 $B_o$  = center of buoyancy of the remaining intact part of the rig after parallel sinkage  
 $B_\psi$  = center of buoyancy of the remaining intact part of the rig after parallel sinkage and inclination about the axis  $x'$   
 $B_\psi$  = center of buoyancy of the remaining intact part of the rig after parallel sinkage and inclination about the axis  $y'$   
 $(\overline{BM})_o$  = transverse metacenter above center of buoyancy before damage  
 $(\overline{BM})_{L_o}$  = longitudinal metacenter above center of buoyancy before damage

$(\overline{BM})_R$  = transverse metacenter of the remaining intact part of the rig above center of buoyancy after flooding  
 $(\overline{BM})_{L_R}$  = longitudinal metacenter of the remaining intact part of the rig above center of buoyancy after flooding  
 $b = B/2$  = half breadth of a rectangular rig  
 $D$  = diameter of a column  
 $e$  = half vertical extent of damage  
 $F$  = center of the remaining intact waterplane area after parallel sinkage  
 $G$  = center of gravity  
 $(\overline{GM})_o$  = transverse metacentric height before damage  
 $(\overline{GM})_{L_o}$  = longitudinal metacentric height before damage  
 $(\overline{GM})_R$  = transverse metacentric height of the remaining intact part of the rig  
 $(\overline{GM})_{L_R}$  = longitudinal metacentric height of the remaining intact part of the rig  
 $g$  = acceleration due to gravity  
 $H$  = height of a column  
 $J_{ox}$  = moment of inertia of the waterplane area of the rig before damage, related to the longitudinal principal axis  
 $J_{oy}$  = moment of inertia of the waterplane area of the rig before damage, related to the transverse principal axis  
 $J_x$  = moment of inertia of the remaining intact waterplane area after parallel sinkage, related to axis  $x$  ( $x'$ )  
 $(J_{x'})$  = moment of inertia of the remaining intact waterplane area after parallel sinkage, related to axis  $x$  ( $x'$ )  
 $J_y$  = moment of inertia of the remaining intact waterplane area after parallel sinkage, related to axis  $y$  ( $y'$ )  
 $(J_{y'})$  = moment of inertia of the remaining intact waterplane area after parallel sinkage, related to axis  $y$  ( $y'$ )  
 $J_{xy}$  = centrifugal moment of the remaining intact waterplane area after parallel sinkage, related to axes  $x$  and  $y$  ( $J_{x'y'} = 0$ )  
 $(\overline{KB})_o$  = center of buoyancy above bottom before damage  
 $(\overline{KB})_R$  = center of buoyancy of the remaining intact part of the rig above bottom after flooding  
 $\overline{KG}$  = center of gravity above bottom before and after flooding (lost buoyancy method)

$L$  = length of a rectangular rig (measured between the centers of the outer columns of a row)  
 $L$  = force of buoyancy  
 $\ell$  =  $L/2$  = half length of a rectangular rig  
 $m$  = projection of  $r$  on axis  $x'$   
 $n$  = projection of  $r$  on axis  $y'$   
 $N$  = number of columns  
 $p$  = distance of  $B_0$  from axis  $y_0$  (measured in a horizontal plane at the level of  $B_0$ )  
 $q$  = distance of  $B_0$  from axis  $x_0$  (measured in a horizontal plane at the level of  $B_0$ )  
 $R$  = distance of a column from the vertical centerline of a radial symmetric rig  
 $r$  = distance between the forces of buoyancy and weight acting on the remaining intact part of the rig after parallel sinkage ( $r^2 = p^2 + q^2 = m^2 + n^2$ )  
 $T_0$  = draft before damage  
 $\Delta T$  = parallel sinkage  
 $u$  = distance of  $F$  from axis  $y_0$  (measured in the waterplane after parallel sinkage)  
 $v$  = distance of  $F$  from axis  $x_0$  (measured in the waterplane after parallel sinkage)  
 $V_0$  = volume of displacement before and after flooding (lost buoyancy method)  
 $W$  = force of weight  
 $x_0$  = longitudinal principal axis of inertia of the waterplane area before damage  
 $x$  = axis running through  $F$  in parallel direction to axis  $x_0$   
 $x'$  = principal axis of inertia of the remaining intact waterplane area after parallel sinkage (about this axis the moment of inertia comes to a minimum value)  
 $y_0$  = transverse principal axis of inertia of the waterplane area before damage  
 $y$  = axis running through  $F$  in parallel direction to axis  $y_0$   
 $y'$  = principal axis of inertia of the remaining intact waterplane area after parallel sinkage (about this axis the moment of inertia comes to a maximum value)  
 $\alpha$  = angle between the directions of the axis  $x'$  and the longitudinal axis (applicable to rectangular rigs if  $N = 4$  and  $\eta \leq 1$  or if  $N = 6$  and  $\eta \leq \sqrt{(12-5\mu_s)/(18-6\mu_s)}$ ) or transverse axis of the rig (applicable if  $N = 4$  and  $\eta \geq 1$  or if  $N = 6$  and  $\eta \geq \sqrt{(12-5\mu_s)/(18-6\mu_s)}$ )  
 $\beta$  = angle of inclination about axis  $x'$  at which the center of the upper flat of a compartment immerses  
 $\gamma$  = angle between axis  $x_0$  and the line connecting the centers of the diagonally opposite corner columns of a rectangular rig  
 $\eta$  =  $B/L$  = breadth-length ratio  
 $\delta_0$  = angle of inclination of the platform against sea level in the final stage of flooding  
 $\mu_s$  = surface permeability  
 $\mu_v$  = volume permeability  
 $\rho_w$  = mass density of water  
 $\varphi_0$  = angle of inclination about axis  $x'$  due to loss of buoyancy ("angle of heel")  
 $\psi_0$  = angle of inclination about axis  $y'$  due to loss of buoyancy ("angle of trim")

## 1. INTRODUCTION

This paper is concerned with the influence of the geometric characteristics of semisubmersible rigs on damage stability. Only damages to the columns caused by collisions

will be considered. The intention is to find out what types of rigs can survive damages even without any watertight subdivision of the columns and what types must be subdivided in order to obtain a certain amount of survival probability.

Contrary to ships, semisubmersible offshore units have several buoyant bodies contributing to the buoyancy of the rig. In general, spacing of columns is wide enough for excluding damage to two or more adjacent columns. Hence, for a high degree of ability to survive collisions, it only must be observed that the rig can withstand the total flooding of one column. This method of providing survivability, which is only applicable to multihull units, is much more effective than the well known method of subdividing a floating body into watertight compartments. Of course, a watertight subdivision of the columns of a multi-legged rig is also possible and contributes additionally to its survivability.

From these simple considerations we may conclude that it is much more easier to attain high survival probability values for rigs than for ships. In spite of this fact there exists a number of semisubmersibles which evidently were designed without closer regard to the influence of hull configuration on damage stability. For example, three-legged rigs without a well-considered subdivision will hardly survive a damage to one of the legs.

Modern drilling rigs meet damage stability and subdivision requirements of the classification societies and of IMCO [1]. They all have a minimum amount of survival probability. But the rules are not as effective as they could be. Especially, they do not give practical instructions to the designer what type of rig should be preferred from the damage stability point of view. After the capsizing of the rig "Alexander Kielland" the Norwegian Maritime Directorate (NMD) introduced amendments to the existing national stability rules for floatable drilling units [2]. In future, semisubmersible rigs operating off the Norwegian coast must be able to withstand heeling moments even if the buoyant force of one of the legs will be lost. This requirement is an important step forward to more safety on rigs and we only can hope that IMCO will follow the Norwegian Maritime Directorate in adopting this amendment to IMCO's "MODU CODE" [1].

## 2. DAMAGE STABILITY OF DIFFERENT TYPES OF RIGS

### 2.1 Multi-legged rig of radial symmetric type with non-subdivided columns and without footings

Fig.1 of Appendix 1 illustrates a column stabilized rig without footings ( $N$  = number of columns). All columns have the same diameter  $D$ , the same distance  $R$  from the vertical centerline of the rig and are spaced equidistantly. The columns are not

subdivided by watertight bulkheads or flats. If one column is damaged the center of gravity  $G$  - according to the lost buoyancy method - is the same as before flooding. The center of buoyancy, however, changes its location because of the fact that the form of the remaining intact part of the rig differs from the original configuration. The resultant force of buoyancy  $L$  acts vertically upward through the new center of buoyancy  $B_0$ . It is equal to the resultant force of weight  $W$ , but of opposite direction. In the upright position, the distance between these two forces is  $r$ .

By reasons of symmetry  $B_0$  is located on the connecting line drawn at the level of  $B_0$  between the center of the column being flooded and the center of the rig. In this special case where no footings are arranged the center of buoyancy  $B_0$  and the center  $F$  of the remaining intact water plane area are lying on the same vertical line. The principal axis of inertia  $x'$ , about which the moment of inertia attains its minimum value, runs through  $F$  and has a normal direction to the connecting line mentioned above.

The forces of buoyancy and weight cause a list of the rig about the axis  $x'$ . As shown in Appendix 1 the angle of inclination about  $x'$  - called angle of heel  $\varphi_0$  - can be calculated from the remaining transverse metacentric height  $\overline{GM}_R$  and the leverarm  $r$  of the couple of forces. For small angles of heel the result is

$$\sin \varphi_0 = 2 \left( \frac{\frac{\mu_v}{N}}{1 - \frac{\mu_v}{N}} \right) \cdot \frac{R}{T_0} \times \quad (1)$$

$$\frac{\left( \frac{1 - 3 \frac{\mu_s}{N}}{1 - \frac{\mu_s}{N}} \right) \left( \frac{R}{T_0} \right)^2 + \frac{1}{1 - \frac{\mu_v}{N}} + \frac{1}{8} \left( 1 - \frac{\mu_s}{N} \right) \left( \frac{D}{T_0} \right)^2 - 2 \frac{\overline{KG}}{T_0}}$$

(where  $\mu_s$  = surface permeability,  $\mu_v$  = volume permeability,  $T_0$  = draft before flooding,  $\overline{KG}$  = center of gravity above base). This formula is only applicable if the final waterline is below the lower edge of the platform and above the bottoms of the columns.

The second principal axis of inertia  $y'$  of the remaining intact waterplane area (about this axis the moment of inertia attains its maximum value) intersects the axis  $x'$  in  $F$  and runs through the center of the flooded column. Because of the radial symmetry of the rig the forces of buoyancy and weight have no component moment causing an inclination about the axis  $y'$ . Thus, the inclination of the rig in this direction - designated as angle of trim  $\psi_0$  - is zero.

In a rough sea, rigs can withstand damages the more likely, the smaller the angle of inclination of the platform against sea level in the final stage of flooding is. Therefore, it can be taken as a safety standard. Generally this angle, denoted by  $\delta_0$ , must be calculated from

$\varphi_0$  and  $\psi_0$  (for details see Section 2.4). Only if flooding does not cause a trimming moment,  $\delta_0$  and  $\varphi_0$  are identical. This is true for all radial symmetric types of rigs.

From formula (1) conclusions may be drawn how the characteristics of this type of rig should be in order to minimize the angle of inclination.  $\varphi_0$  depends on six parameters:  $\varphi_0 = f(\mu_s, \mu_v, \overline{KG}/T_0, N, R/T_0, D/T_0)$ . Parameters  $N$ ,  $R/T_0$  and  $D/T_0$  are geometric factors which are of main interest because they can be freely chosen in the early design stage. Permeability and location of center of gravity, however, can only be varied within a limited range of values.

To get a comprehensive graphical representation of the results obtained for this and the following types of rigs, several sets of curves were plotted by computer. They will be published in [3]. In this paper some interesting partial results are given in tabular form. They clearly show the influence of the geometric characteristics on damage stability.

Table I presents for the multi-legged rig without footings values of  $\overline{KG}/T_0$  which must not be exceeded if after flooding of one column the angle of inclination shall be limited to  $\delta_0 = 8^\circ$ . Permeability is assumed to be  $\mu_s = \mu_v = 1$ .

TABLE I. Radial symmetric rig with non-subdivided columns and without footings:

Maximum permissible values of  $\overline{KG}/T_0$  if inclination shall be limited to  $\delta_0 = 8^\circ$  ( $\mu_s = \mu_v = 1$ ).

|                     | $R/T_0 = 2$ | $R/T_0 = 3$ | $R/T_0 = 4$ | $R/T_0 = 6$ |
|---------------------|-------------|-------------|-------------|-------------|
| N=4; $D/T_0 = 0.25$ | -           | -           | -           | -           |
| 0.50                | -           | -           | -           | -           |
| 1.00                | -           | -           | -           | -           |
| N=5; $D/T_0 = 0.25$ | -           | -           | -           | -           |
| 0.50                | -           | -           | -           | -           |
| 1.00                | -           | -           | -           | -           |
| N=6; $D/T_0 = 0.25$ | -           | -           | -           | 2.80        |
| 0.50                | -           | -           | -           | 2.80        |
| 1.00                | -           | -           | -           | 2.83        |
| N=7; $D/T_0 = 0.25$ | -           | 0           | 1.14        | > 5         |
| 0.50                | -           | 0           | 1.14        | > 5         |
| 1.00                | -           | 0           | 1.18        | > 5         |
| N=8; $D/T_0 = 0.25$ | 0           | 0.70        | 2.17        | > 5         |
| 0.50                | 0           | 0.70        | 2.18        | > 5         |
| 1.00                | 0           | 0.75        | 2.23        | > 5         |
| N=9; $D/T_0 = 0.25$ | 0.27        | 1.23        | 2.98        | > 5         |
| 0.50                | 0.28        | 1.25        | 2.99        | > 5         |
| 1.00                | 0.32        | 1.30        | 3.01        | > 5         |

In those cases where values of  $\overline{KG}/T_0$  are given,  $\delta_0 = 8^\circ$  can be attained without emersion of the bottom of any column. Values  $\overline{KG}/T_0 < 0$  and  $\overline{KG}/T_0 > 5$  are of no practical interest. The platform is assumed to be arranged at such a height that in the final stage of flooding its lower edge

does not immerge. Results which were achieved for other types of rigs will be presented in the same way.

## 2.2 Multi-legged rig of radial symmetric type with subdivided columns and without footings

Rigs that cannot withstand flooding of one column must be subdivided in order to reduce the angle of inclination to an acceptable value. The effect of subdivision is similar to that of reduction of permeability. This is especially true in the case of vertical subdivision. If the columns are subdivided horizontally the free-surface effect cannot be neglected, except the upper flat of the flooded compartment is below the final waterplane.

For this reason, separate damage stability calculations were carried out for rigs, the columns of which are subdivided by watertight flats. As can be seen in Fig. 2 of Appendix 2, the center of the flooded compartment is assumed to be located at the level of the waterplane in the initial stage of flooding. The height of the compartment ( $=2a$ ), however, can vary.

Separate calculations are necessary for two different regions: In region I the upper flat of the flooded compartment is above and in region II below the final waterplane. These two regions are approaching each other if the final waterline intersects the center of the upper flat. In that case the angle of inclination is  $\varphi_0 = \beta_I$  or  $\beta_{II}$  respectively. The difference between  $\beta_I$  and  $\beta_{II}$  follows from the different waterplane areas and the different positions of their centers. As can be seen in the lower illustration of Fig. 2, the center of the waterplane area moves back to the center of the rig as soon as the flooded compartment is fully submerged. For details of damage stability calculations see Appendix 2. Following results are obtained:

### Region I

$$\sin \varphi_0 = 2 \left( \frac{\mu_v}{1 - \frac{\mu_v}{N}} \right) \cdot \frac{a}{T_0} \cdot \frac{R}{T_0} \quad (2)$$

$$\left( \frac{1 - 3 \frac{\mu_s}{N}}{1 - \frac{\mu_s}{N}} \right) \left( \frac{R}{T_0} \right)^2 + 1 + \left( \frac{\mu_v}{1 - \frac{\mu_v}{N}} \right) \left( \frac{a}{T_0} \right)^2 + \frac{1}{8} \left( 1 - \frac{\mu_s}{N} \right) \left( \frac{D}{T_0} \right)^2 - 2 \frac{KG}{T_0}$$

In Region I the angle of inclination must satisfy the inequation  $\varphi_0 < \beta_I$  where

$$\tan \beta_I = \left( 1 - 2 \frac{\mu_v}{N} \right) \cdot \left( \frac{1 - \frac{\mu_s}{N}}{1 - \frac{\mu_v}{N}} \right) \cdot \frac{a}{R} \quad (3)$$

### Region II

$$\tan \varphi_0 = \frac{4 \frac{\mu_v}{N} \cdot \frac{a}{T_0} \cdot \frac{R}{T_0}}{\left( \frac{R}{T_0} \right)^2 + 1 + 4 \left( \frac{\mu_v}{N} \right)^2 \left( \frac{a}{T_0} \right)^2 + \frac{1}{8} \left( \frac{D}{T_0} \right)^2 - 2 \frac{KG}{T_0}} \quad (4)$$

In Region II the angle of inclination must satisfy the inequation  $\varphi_0 > \beta_{II}$  where

$$\tan \beta_{II} = \left( 1 - 2 \frac{\mu_v}{N} \right) \cdot \frac{a}{R} \quad (5)$$

Formulas (2) and (4) are applicable provided that the angles of inclination are small, the lower edge of the platform does not immerge and the bottom of a column does not emerge.

The positive effect of subdivision on damage stability becomes evident in Table (2). The limits of  $KG/T_0$  which must be observed if the angle of inclination shall not be greater than  $80^\circ$ , are considerably higher than for the rig without subdivision of the columns. The values are calculated for a homogeneous permeability of  $\mu_s = \mu_v = 1$ .

TABLE II. Radial symmetric rig with subdivided columns and without footings:

Maximum permissible values of  $KG/T_0$  if inclination shall be limited to  $\varphi_0 = 80^\circ$  ( $\mu_s = \mu_v = 1$ ).

#### a) $a/T_0 = 0.5$

|                       | $R/T_0 = 2$ | $R/T_0 = 4$ | $R/T_0 = 6$ |
|-----------------------|-------------|-------------|-------------|
| $N = 3; D/T_0 = 0.25$ | -           | -           | 4.30        |
| 0.50                  | -           | -           | 4.40        |
| 1.00                  | -           | -           | 4.40        |
| $N = 4; D/T_0 = 0.25$ | -           | 1.42        | > 5         |
| 0.50                  | -           | 1.45        | > 5         |
| 1.00                  | -           | 1.48        | > 5         |
| $N = 5; D/T_0 = 0.25$ | -           | 2.82        | > 5         |
| 0.50                  | -           | 2.83        | > 5         |
| 1.00                  | -           | 2.90        | > 5         |
| $N = 6; D/T_0 = 0.25$ | 0.29        | 3.78        | > 5         |
| 0.50                  | 0.30        | 3.80        | > 5         |
| 1.00                  | 0.33        | 3.82        | > 5         |
| $N = 7; D/T_0 = 0.25$ | 0.66        | 4.43        | > 5         |
| 0.50                  | 0.67        | 4.45        | > 5         |
| 1.00                  | 0.70        | 4.50        | > 5         |
| $N = 8; D/T_0 = 0.25$ | 0.91        | 4.95        | > 5         |
| 0.50                  | 0.93        | 4.97        | > 5         |
| 1.00                  | 0.98        | 5.01        | > 5         |
| $N = 9; D/T_0 = 0.25$ | 1.11        | > 5         | > 5         |
| 0.50                  | 1.12        | > 5         | > 5         |
| 1.00                  | 1.17        | > 5         | > 5         |

#### b) $a/T_0 = 0.25$

|                       | $R/T_0 = 2$ | $R/T_0 = 4$ | $R/T_0 = 6$ |
|-----------------------|-------------|-------------|-------------|
| $N = 3; D/T_0 = 0.25$ | 0.13        | 3.77        | > 5         |
| 0.50                  | 0.15        | 3.79        | > 5         |
| 1.00                  | 0.20        | 3.82        | > 5         |
| $N = 4; D/T_0 = 0.25$ | 0.73        | 4.96        | > 5         |
| 0.50                  | 0.74        | 4.98        | > 5         |
| 1.00                  | 0.80        | 5.01        | > 5         |
| $N = 5; D/T_0 = 0.25$ | 1.09        | > 5         | > 5         |
| 0.50                  | 1.10        | > 5         | > 5         |
| 1.00                  | 1.15        | > 5         | > 5         |

|                                |      |      |     |
|--------------------------------|------|------|-----|
| N = 6; D/T <sub>0</sub> = 0.25 | 1.33 | > 5  | > 5 |
|                                | 0.50 | 1.34 | > 5 |
|                                | 1.00 | 1.39 | > 5 |
| N = 7; D/T <sub>0</sub> = 0.25 | 1.50 | > 5  | > 5 |
|                                | 0.50 | 1.51 | > 5 |
|                                | 1.00 | 1.55 | > 5 |
| N = 8; D/T <sub>0</sub> = 0.25 | 1.61 | > 5  | > 5 |
|                                | 0.50 | 1.62 | > 5 |
|                                | 1.00 | 1.67 | > 5 |
| N = 9; D/T <sub>0</sub> = 0.25 | 1.71 | > 5  | > 5 |
|                                | 0.50 | 1.73 | > 5 |
|                                | 1.00 | 1.77 | > 5 |

Of course, the results given in Table II are only correct if flooding is limited to the compartment under consideration. Much smaller or even negative  $\overline{KG}/T_0$ -values may be obtained if location and extent of damage are of such kind that adjacent compartments are involved too. Taking this into account it can be concluded that, in reality, the effectiveness of watertight subdivision is smaller than may be assumed from Table II. In Section 3 it will be demonstrated that in the case of watertight subdivision a true judgement of the ability to survive flooding can be made by including the randomness of damage dimensions and calculating the probability of survival.

### 2.3 Multi-legged rig of radial symmetric type with non-subdivided columns and a ringlike lower hull

For reasons of minimizing motion in waves the columns of a multi-legged rig shall not be too big. Therefore, normal semisubmersibles have additional displacement bodies. They are arranged at the bottoms of the columns (e.g. one small footing at each column, one common ringlike footing for the columns of radial symmetric rigs, two longitudinal parallel footings for the columns of rectangular rigs).

From the stability standpoint, however, the additional buoyant hull should be as small as possible. Its unfavourable effect on intact stability follows from the lower height of the center of buoyancy [4]. It is to be expected that in most cases footings will also reduce damage stability. In order to get an idea to what extent the angle of inclination  $\varphi_0$  will change if footings are fitted, damage stability calculations were carried out for a radial symmetric rig with a ringlike footing (Fig.3 of Appendix 3). The dimensions of the columns are the same as in Fig.1 (Appendix 1). All columns are connected by a ring with a box beam section; breadth and height are identical with the diameter D of the columns. The bottoms of the ring and the columns are lying in the same horizontal plane.

As shown in Appendix 3 the angle of inclination  $\varphi_0$  after flooding of one column is as follows:

$$\sin \varphi_0 = \frac{2 \left( \frac{\mu_v/N}{1-\mu_v/N} \right) \cdot \frac{R}{T_0}}{\left( \frac{1-3\mu_s/N}{1-\mu_s/N} \right) \left( \frac{R}{T_0} \right)^2 + \frac{8(R/D)}{N(T_0/T_0)} + \frac{1}{1-\mu_v/N} - \frac{1}{8} \left( 7 + \frac{\mu_s}{N} \right) \left( \frac{D}{T_0} \right)^2 - 2 \left( 1 - \frac{D}{T_0} + \frac{8R}{NT_0} \right) \frac{\overline{KG}}{T_0}} \quad (6)$$

This formula can be applied if the angle of heel is small and neither the footing nor the platform will reach the surface of the water.

The values of  $\overline{KG}/T_0$  being necessary in order to obtain an angle of inclination of  $\varphi_0 = 8^\circ$  are presented in Table III. Permeability is again assumed to be  $\mu_s = \mu_v = 1$ . For conclusions which may be drawn from these results see Section 3.

TABLE III. Radial symmetric rig with non-subdivided columns and a ringlike lower hull:

|                              | R/T <sub>0</sub> = 2 | R/T <sub>0</sub> = 3 | R/T <sub>0</sub> = 4 | R/T <sub>0</sub> = 6 |
|------------------------------|----------------------|----------------------|----------------------|----------------------|
| N=4; D/T <sub>0</sub> = 0.25 | -                    | -                    | -                    | -                    |
|                              | 0.50                 | -                    | -                    | -                    |
| N=5; D/T <sub>0</sub> = 0.25 | -                    | -                    | -                    | 0                    |
|                              | 0.50                 | -                    | -                    | 0.11                 |
| N=6; D/T <sub>0</sub> = 0.25 | -                    | -                    | 0.04                 | 0.42                 |
|                              | 0.50                 | -                    | 0.14                 | 0.55                 |
| N=7; D/T <sub>0</sub> = 0.25 | -                    | 0.09                 | 0.31                 | 0.82                 |
|                              | 0.50                 | 0                    | 0.19                 | 0.95                 |
| N=8; D/T <sub>0</sub> = 0.25 | 0.06                 | 0.28                 | 0.56                 | 1.18                 |
|                              | 0.50                 | 0.13                 | 0.38                 | 0.68                 |
| N=9; D/T <sub>0</sub> = 0.25 | 0.18                 | 0.45                 | 0.78                 | 1.53                 |
|                              | 0.50                 | 0.26                 | 0.57                 | 0.92                 |

Large diameter columns with D/T<sub>0</sub> = 1 are not considered as the lower hull contributes to the displacement and, accordingly, the columns must be made more slender in comparison to a rig without footings.

### 2.4 Four-legged rig of rectangular type with non-subdivided columns and two longitudinal parallel footings

This type of rig is often preferred because of its low towing resistance and its cost-saving construction. The section of each footing is quadratic; its breadth, its height and the diameters D of the four columns are equally large. An illustration is given in Fig.4 of Appendix 4.

As compared with the radial symmetric type, rectangular rigs have an additional geometric parameter, the breadth-length ratio  $\eta = B/L$ . From Table III it may be concluded that in the special case  $\eta = 1$  a four-legged rig will hardly withstand the flooding of a column. In case the breadth-length ratio is smaller ( $\eta < 1$ ), the final angle of inclination,  $\varphi_0$ , will presumably even be greater. This assumption follows from the strong influence of breadth on intact stability.

In Appendix 4 the angles of inclination

about the two axes  $x'$  and  $y'$  are calculated. The results are applicable to rigs with  $\eta \leq 1$  and with columns of sufficient length in order that after flooding footings are still below and the platform still above water surface. Furthermore, the angles  $\varphi_0$  and  $\psi_0$  must be small. The formulas obtained are as follows:

Angle of heel:

$$\sin \varphi_0 = \frac{\pi \mu_v \frac{l}{T_0} \cdot \left[ \eta + \left( \frac{2-\mu_s}{\mu_s \eta} \right) (\xi + \eta^2 - 1) \right]}{\sqrt{2} (4-\mu_v) \cdot \sqrt{1 - \left( \frac{2-\mu_s}{\mu_s \eta} \right)^2 (1-\eta^2) (\xi + \eta^2 - 1)} \cdot \left[ 2 \frac{lD}{T_0^2} + \frac{2\pi}{4-\mu_v} + \pi \left( \frac{2-\mu_s}{4\mu_s} \right) \left( \frac{l}{T_0} \right)^2 (1+\eta^2 + \xi) - \frac{\pi}{64} (42+\mu_s) \left( \frac{D}{T_0} \right)^2 - \frac{KG}{T_0} \left( 4 \frac{l}{T_0} + \pi - \frac{\pi D}{2T_0} \right) \right]} \quad (7)$$

Angle of trim:

$$\sin \psi_0 = \frac{\pi \mu_v \frac{l}{T_0} \cdot \left[ 1 - \left( \frac{2-\mu_s}{\mu_s} \right) (\xi + \eta^2 - 1) \right]}{\sqrt{2} (4-\mu_v) \cdot \sqrt{1 - \left( \frac{2-\mu_s}{\mu_s \eta} \right)^2 (1-\eta^2) (\xi + \eta^2 - 1)} \cdot \left[ 2 \frac{lD}{T_0^2} + \frac{2\pi}{4-\mu_v} + \pi \left( \frac{2-\mu_s}{4\mu_s} \right) \left( \frac{l}{T_0} \right)^2 (1+\eta^2 + \xi) - \frac{\pi}{64} (42+\mu_s) \left( \frac{D}{T_0} \right)^2 - \frac{KG}{T_0} \left( 4 \frac{l}{T_0} + \pi - \frac{\pi D}{2T_0} \right) \right]} \quad (8)$$

New symbols are

$$l = L/2 \quad \text{and} \quad \xi = \sqrt{(1-\eta^2)^2 + \left( \frac{\mu_s \cdot \eta}{2-\mu_s} \right)^2}$$

As shown in Fig.4 the principal axes  $x'$  and  $y'$  have their origin in  $F$ ; their directions are given by  $\alpha$ . If  $\eta \leq 1$ ,  $\alpha$  is obtained as follows:

$$\tan 2\alpha = \left| \frac{2J_{xy}}{J_y - J_x} \right| = \frac{\mu_s \cdot \eta}{(2-\mu_s) \cdot (1-\eta^2)} \quad (9)$$

After parallel sinkage, heeling about axis  $x'$  and trimming about axis  $y'$ , the final angle of inclination of the platform against sea level is

$$\tan \vartheta_0 = \sqrt{\tan^2 \varphi_0 + \tan^2 \psi_0} \quad (10)$$

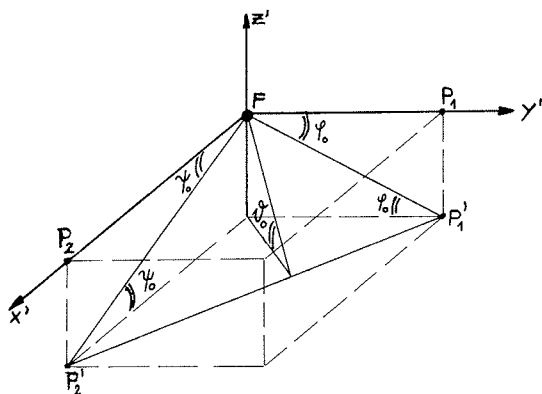


Fig.5 Position of the platform plane element  $F P_1 P_2$  after flooding

This formula can directly be derived from Fig.5. Let  $P_1$  be a point on the axis  $y'$  which will move down by heeling from  $P_1$  to  $P_1'$ , then a point  $P_2$  exists on the axis  $x'$  which will move down by trimming to the same level:  $\overline{P_2 P_2'} = \overline{P_1 P_1'}$ . After flooding, the plane marked by the points  $F$ ,  $P_1'$  and  $P_2'$  is parallel to the platform deck. The final angle of inclination,  $\vartheta_0$ , lies within a

vertical plane being perpendicular to the line  $P_1' P_2'$ .

Systematic calculations which were made for different diameter-draft ratios ( $D/T_0 = 0.25$  and  $0.50$ ), different length-draft ratios ( $l/T_0 = 4$  and  $6$ ) and different breadth-length ratios ( $\eta = 0.8, 0.9$  and  $1.0$ ) proved the correctness of the assumption that, without subdivision, the rig will hardly survive the flooding of a column. In each case a final angle of inclination of  $\vartheta_0 \leq 8^\circ$  is not attainable if permeability is  $\mu_s = \mu_v = 1$  and  $KG/T_0 \geq 0$ . In order to demonstrate the influence of  $\eta$ , additional calculations were made for  $\mu_s = \mu_v = 0.5$ . Here,  $\vartheta_0$  is considerably smaller. Table IV presents maximum values of  $KG/T_0$  that are just allowable if the final angle of inclination shall be  $\vartheta_0 = 6^\circ$ .

TABLE IV. Four-legged rectangular rig with non-subdivided columns and two longitudinal parallel footings:

Maximum permissible values of  $KG/T_0$  if inclination shall be limited to  $\vartheta_0 = 6^\circ$  ( $\mu_s = \mu_v = 0.5$ ).

|                |                | $l/T_0 = 4$ | $l/T_0 = 6$ |
|----------------|----------------|-------------|-------------|
| $\eta = 1.0$ ; | $D/T_0 = 0.25$ | 0.81        | 1.83        |
|                | 0.50           | 0.91        | *)          |
| $\eta = 0.9$ ; | $D/T_0 = 0.25$ | 0.69        | 1.59        |
|                | 0.50           | 0.80        | 1.70        |
| $\eta = 0.8$ ; | $D/T_0 = 0.25$ | 0.52        | 1.27        |
|                | 0.50           | 0.64        | 1.39        |

\*) formulas not applicable because footings become awash

In case of a breadth-length ratio of  $\eta < 1$  heeling as well as trimming occurs. Trimming can only be avoided if  $\eta = 1$ : from equation (9) follows  $\alpha = 45^\circ$

(axis  $y'$  runs through the center of the damaged column). As can be seen from Table IV,  $\eta = 1$  should be preferred because of the higher allowable center of gravity.

## 2.5 Six-legged rig of rectangular type with non-subdivided columns and two longitudinal parallel footings

A six-legged rig with a catamaran hull is a compromise between considerations of economy and of damage stability. Furthermore, columns can be made smaller in diameter and, as a consequence, excitation by waves will be less severe compared to a four-legged rig.

Damage may occur to a central column or a corner column. In the latter case the effects of flooding will be more serious. For this reason, damage stability calculations, as carried out in Appendix 5, start out from a damage to a corner column.

The footings of the six-legged rig are of the same shape as the footings of the four-legged rig in Section 2.4. The box beam section is of equal breadth and height. Each column is assumed to be floodable right down to the bottom of the hull.

The directions of the heel axis  $x'$  and the trim axis  $y'$  are obtainable from

$$\tan 2\alpha = \left| \frac{2J_{xy}}{J_y - J_x} \right| = \left| \frac{6\mu_s \eta}{12 - 5\mu_s - 6\eta^2(3 - \mu_s)} \right| \quad (11)$$

If  $\eta \leq \sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}}$ , the difference between the moments of inertia  $J_y$  and  $J_x$  is positive. The angle of direction,  $\alpha$ , must then be measured as shown in the upper illustration of Fig. 6. For values  $\eta > \sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}}$ , the difference  $J_y - J_x$  will be negative and  $\alpha$  must be established as demonstrated in the lower illustration.

Details of calculation are given in Appendix 5. It must be observed that in the case of  $\eta \leq \sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}}$  the damaged column always lies within the first quadrant of the  $x'-y'$ -system of coordinates. In the case of  $\eta > \sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}}$  the damaged column lies - with the exception of a very small range of  $\eta$ -values - within the second quadrant of the  $x'-y'$ -system. The narrow limits of the aforementioned range are as follows:

Angle of heel;  $\eta \leq \sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}}$ :

$$\sin \varphi_0 = \frac{\frac{3}{2} \pi \mu_v \frac{\ell}{T_0} \cdot \left( \eta + \frac{[12 - 5\mu_s - 6\eta^2(3 - \mu_s)] \cdot \frac{\ell}{T_0} - 1}{6\mu_s \eta} \right)}{(6 - \mu_v) \cdot \sqrt{1 + [12 - 5\mu_s - 6\eta^2(3 - \mu_s)]^2 \left( \frac{\ell}{T_0} \right)^2}} \times$$

$$\times \frac{1}{\frac{2\ell D}{T_0^2} + \frac{9\pi}{2(6 - \mu_v)} + \frac{\pi(\ell/T_0)^2}{4(6 - \mu_s)} \left( 12 - 5\mu_s + 6\eta^2(3 - \mu_s) - [12 - 5\mu_s - 6\eta^2(3 - \mu_s)] \xi \right) - \frac{\pi}{64} (26 + \mu_s) \left( \frac{D}{T_0} \right)^2 - \left( 4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \pi \frac{D}{T_0} \right) \frac{KG}{T_0}}$$

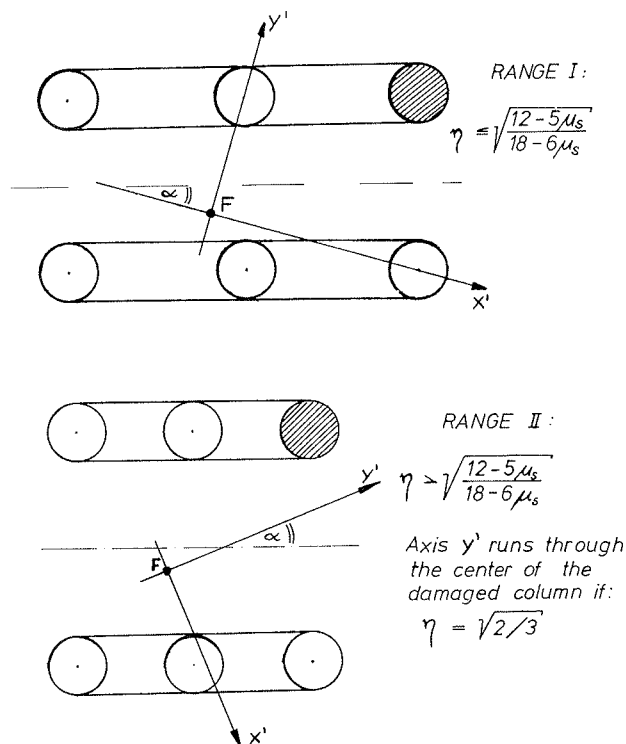


Fig. 6 Six-legged rectangular rig with two longitudinal parallel footings: Location of axes  $x'$  and  $y'$  for different breadth-length ratios  $\eta$

$$\sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}} < \eta < \sqrt{\frac{2}{3}}$$

This range is of some importance in so far as there is a change in the direction of inclination about the axis  $y'$ . Accordingly, the point of emersion of the hull is situated on the opposite side.

Depending on whether  $\eta$  is below/above  $\eta = \sqrt{\frac{12 - 5\mu_s}{18 - 6\mu_s}}$ , different formulas are obtained for  $\varphi_0$  and  $\psi_0$ :

(12)

Angle of heel;  $\eta > \sqrt{\frac{12-5\mu_s}{18-6\mu_s}}$  :

$$\sin \varphi_0 = \frac{\frac{3}{2} \pi \mu_v \frac{l}{T_0} \cdot \left( 1 + [6\eta^2(3-\mu_s) + 5\mu_s - 12] \frac{\xi-1}{6\mu_s} \right)}{(6-\mu_v) \cdot \sqrt{1 + [6\eta^2(3-\mu_s) + 5\mu_s - 12]^2 \left( \frac{\xi-1}{6\mu_s \eta} \right)^2}} \times$$

$$\times \frac{1}{\frac{2\ell D}{T_0^2} + \frac{3\pi}{2(6-\mu_v)} + \frac{\pi(\ell/T_0)^2}{4(6-\mu_s)} \left( 6\eta^2(3-\mu_s) - 5\mu_s + 12 - [6\eta^2(3-\mu_s) + 5\mu_s - 12] \xi \right) - \frac{\pi}{64} (26+\mu_s) \left( \frac{D}{T_0} \right)^2 - \left( 4\frac{\ell}{T_0} + \frac{3}{2}\pi - \pi \frac{D}{T_0} \right) \frac{KG}{T_0}}$$

(13)

Angle of trim;  $\eta \leq \sqrt{\frac{12-5\mu_s}{18-6\mu_s}}$  :

$$\sin \varphi_0 = \frac{\frac{3}{2} \pi \mu_v \frac{l}{T_0} \cdot \left( 1 - [12-5\mu_s-6\eta^2(3-\mu_s)] \frac{\xi-1}{6\mu_s} \right)}{(6-\mu_v) \cdot \sqrt{1 + [12-5\mu_s-6\eta^2(3-\mu_s)]^2 \left( \frac{\xi-1}{6\mu_s \eta} \right)^2}} \times$$

$$\times \frac{1}{\frac{2\ell D}{T_0^2} + \frac{3\pi}{2(6-\mu_v)} + \frac{\pi(\ell/T_0)^2}{4(6-\mu_s)} \left( 12-5\mu_s+6\eta^2(3-\mu_s) + [12-5\mu_s-6\eta^2(3-\mu_s)] \xi \right) - \frac{\pi}{64} (26+\mu_s) \left( \frac{D}{T_0} \right)^2 - \left( 4\frac{\ell}{T_0} + \frac{3}{2}\pi - \pi \frac{D}{T_0} \right) \frac{KG}{T_0}}$$

(14)

Angle of trim;  $\eta > \sqrt{\frac{12-5\mu_s}{18-6\mu_s}}$  :

$$\sin \varphi_0 = \frac{\frac{3}{2} \pi \mu_v \frac{l}{T_0} \cdot \left( \eta - [6\eta^2(3-\mu_s) + 5\mu_s - 12] \frac{\xi-1}{6\mu_s \eta} \right)}{(6-\mu_v) \cdot \sqrt{1 + [6\eta^2(3-\mu_s) + 5\mu_s - 12]^2 \left( \frac{\xi-1}{6\mu_s \eta} \right)^2}} \times$$

$$\times \frac{1}{\frac{2\ell D}{T_0^2} + \frac{3\pi}{2(6-\mu_v)} + \frac{\pi(\ell/T_0)^2}{4(6-\mu_s)} \left( 6\eta^2(3-\mu_s) - 5\mu_s + 12 + [6\eta^2(3-\mu_s) + 5\mu_s - 12] \xi \right) - \frac{\pi}{64} (26+\mu_s) \left( \frac{D}{T_0} \right)^2 - \left( 4\frac{\ell}{T_0} + \frac{3}{2}\pi - \pi \frac{D}{T_0} \right) \frac{KG}{T_0}}$$

(15)

Symbol  $\xi$  in equations (12) to (15) was introduced for the purpose of contracting the formulas:  $\xi = \sqrt{1 + \left[ \frac{6\mu_s \eta}{12-5\mu_s-6\eta^2(3-\mu_s)} \right]^2}$

From these formulas the final angle of inclination against sea level can be determined by using equation (10). Systematic

calculations carried out for different values of  $\mu_s$ ,  $\mu_v$ ,  $\eta$ ,  $\ell/T_0$ ,  $D/T_0$  and  $KG/T_0$  clearly show the favourable effect of the arrangement of the two additional legs. Compared to the four-legged rig the final angle of inclination,  $\varphi_0$ , is smaller. Some results are given in Table V.



TABLE V. Six-legged rectangular rig with non-subdivided columns and two longitudinal parallel footings:

a) Maximum permissible values of  $\overline{KG}/T_o$  if inclination shall be limited to  $\vartheta_o = 8^\circ$  ( $\mu_S = \mu_V = 1$ )

|                |                | $l/T_o = 4$ | $l/T_o = 6$ |
|----------------|----------------|-------------|-------------|
| $\eta = 1.0$ ; | $D/T_o = 0.25$ | < 0         | 0.52        |
|                | 0.50           | 0           | *)          |
| $\eta = 0.9$ ; | $D/T_o = 0.25$ | < 0         | 0.44        |
|                | 0.50           | < 0         | *)          |
| $\eta = 0.7$ ; | $D/T_o = 0.25$ | < 0         | 0.21        |
|                | 0.50           | < 0         | *)          |
| $\eta = 0.6$ ; | $D/T_o = 0.25$ | < 0         | 0.10        |
|                | 0.50           | < 0         | 0.20        |
| $\eta = 0.5$ ; | $D/T_o = 0.25$ | < 0         | 0           |
|                | 0.50           | < 0         | 0.09        |

\*) formulas not applicable because footings become awash

b) Maximum permissible values of  $\overline{KG}/T_o$  if inclination shall be limited to  $\vartheta_o = 6^\circ$  ( $\mu_S = \mu_V = 0.5$ )

|                |                | $l/T_o = 4$ | $l/T_o = 6$ |
|----------------|----------------|-------------|-------------|
| $\eta = 1.0$ ; | $D/T_o = 0.25$ | 1.27        | 2.47        |
|                | 0.50           | 1.41        | *)          |
| $\eta = 0.9$ ; | $D/T_o = 0.25$ | 1.20        | 2.37        |
|                | 0.50           | 1.35        | *)          |
| $\eta = 0.7$ ; | $D/T_o = 0.25$ | 0.90        | 1.80        |
|                | 0.50           | 1.04        | 1.97        |

\*) formulas not applicable because footings become awash

Comparing the results presented in Sections 2.3, 2.4 and 2.5 it can be deduced that the damage stability values of rectangular rigs with a catamaran hull and a breadth-length ratio of  $\eta = 1$  do not differ much from the damage stability values of a radial symmetric rig with a ringlike lower hull and a radial distance of columns of  $R \approx l$ . This statement is approximately true on condition that number and diameter of the columns are the same. For instance, Table III may be used for a rough assessment of damage stability of a eight-legged rectangular rig. Formulas for this type of rig are not given in this paper but can be derived in a similar way as demonstrated in Appendix 5.

### 3. ASSESSMENT OF THE ABILITY TO SURVIVE DAMAGES AND CONCLUDING REMARKS

An assessment of the ability to survive damages can be made by calculating the probability of survival. Regulations based on this principle exist for some ten years for passenger ships [5]. Contrary to ships,

rigs can attain a survival probability of  $P = 1$  under good weather conditions and of  $P = 1$  or nearly 1 under bad weather conditions. Assuming that no heeling moments are acting, a rig can be made "unsinkable" even without any watertight subdivision if following points are observed:

- high number of columns ( $N > 8$  is to be preferred)
- columns arranged as far as possible from the vertical centerline of the rig ( $R/T_o$  or  $l/T_o > 3$  is to be preferred)
- no footings
- breadth-length ratio of rectangular rigs not smaller than  $\eta = 1$
- big columns (compared to other parameters the influence of  $D/T_o$  is rather small)
- center of gravity as low as possible

Example: Nine-legged radial symmetric rig without footings ( $N=9$ ,  $R/T_o=4$ ,  $\mu_S = \mu_V = 1$ ). From equation (1) follows:

$\vartheta_o = 5.7^\circ$  if  $\overline{KG}/T_o = 1.5$  } influence of  $D/T_o$   
 $\vartheta_o = 8.0^\circ$  if  $\overline{KG}/T_o = 3.0$  } can be neglected

It is quite evident that this rig can withstand any damage to a column. Therefore, under good weather conditions, the probability of survival will be  $P = 1$ .

For "unsinkability" also under bad weather conditions, additional factors are important:

- ability to withstand large heeling moments in damaged condition ( $\vartheta_o$  should be as small as possible; large righting moments over a wide range of angles are to be aimed at)
- minimization of wind heeling moment\*) (in heeled condition the total projected area exposed to wind should be as small as possible; structural members are to be arranged and shaped with a view to a minimum wind force coefficient)
- minimization of wave heeling moment (the rig, considered to remain in its position, will be affected by a heeling moment which can be kept small if the volume of displacement is concentrated at the bottoms of the columns. Hence, footings also have a positive effect on survivability)
- avoidance of large amplitudes of wave-induced motions (small column diameters are to be preferred because the special quality of semisubmersibles, namely small amplitudes of motion, will be reached all the more the smaller the waterplane is. Another positive effect is that parametric excitation cannot occur)

If for reasons of economy and practicability the conditions stated above cannot be fulfilled, a probability of survival  $P < 1$  must be accepted. Before calculating survival probability some definitions must be made. First of all, it must be tried to define the boundary between survival and non-survival. It may be assumed that a rig

\*) The percentage of collisions occurring at severe storm conditions will be higher for rigs than for ships. Ships chiefly collide when sailing at bad visibility conditions and, according to nature, wind velocity will then be low [6].

will survive a damage if the angle of inclination caused by flooding and a resulting heeling moment does not exceed a critical value (wind heeling moment based on a wind velocity of 50 knots [1]; wave steepness according to the stability regulations for the German Navy =  $(10+0.05 \text{ m}^{-1} \lambda)^{-1}$ , where  $\lambda$  = wave length; the most unfavourable wave length may be taken). This critical angle may be determined by the location of lowest opening through which progressive flooding may take place or by an absolute value of 25 degrees, whichever is less.

In those cases in which the boundary between survival and non-survival will be almost reached, no reserve stability exists which enables the rig to survive a larger resulting heeling moment than the assumed one. This may be taken into account by introducing a factor which will reduce survival probability if the rig cannot withstand additional moments (in [5] this is done by the factor "s" which evaluates the effect of freeboard, stability and heel in the final flooded condition).

A rig which cannot withstand flooding, must be subdivided by watertight decks or bulkheads. Subdivision, however, will be only effective, if in case of collision at least some watertight decks or bulkheads remain undamaged. In order to calculate the probability that flooding will be limited to a compartment or a group of adjacent compartments it must be derived from damage statistics how location and dimensions of damage are distributed. As for rigs a sufficient quantity of damage data does not exist, realistic assumptions must be made for the frequency functions.

Collisions will mainly occur near the level of the waterplane. The vertical extent of damage will vary from very small values to large values. It can be deduced from [7] that the half vertical extent of damage,  $e$ , follows a lognormal frequency function:

$$f(e) = \frac{0.4343}{\sqrt{2\pi} \cdot \sigma \cdot e} \exp\left(-\frac{[\log e - \mu]^2}{2\sigma^2}\right) \quad (16)$$

where  $\mu$  and  $\sigma$  are parameters which must be determined from damage statistics.

In Fig.7 the lognormal function (broken line) is replaced by a linear function (solid line):

$$f(e) = f_0 \left(1 - \frac{e}{e_0}\right) \quad (17)$$

If  $e_0$  is estimated at  $e_0 = \frac{H}{6}$  ( $H$  = height of column)\*,  $f_0$  follows from

$$\int_0^{e_0} f(e) de = f_0 \int_0^{e_0} \left(1 - \frac{e}{e_0}\right) de = 1 \quad (18)$$

The solution is  $f_0 = \frac{12}{H}$  and equation (17) becomes

$$f(e) = \frac{12}{H} \left(1 - \frac{6}{H} e\right) \quad (19)$$

\* ) For comparison: in [5] a maximum total damage length of 0.24 L is assumed for passenger ships

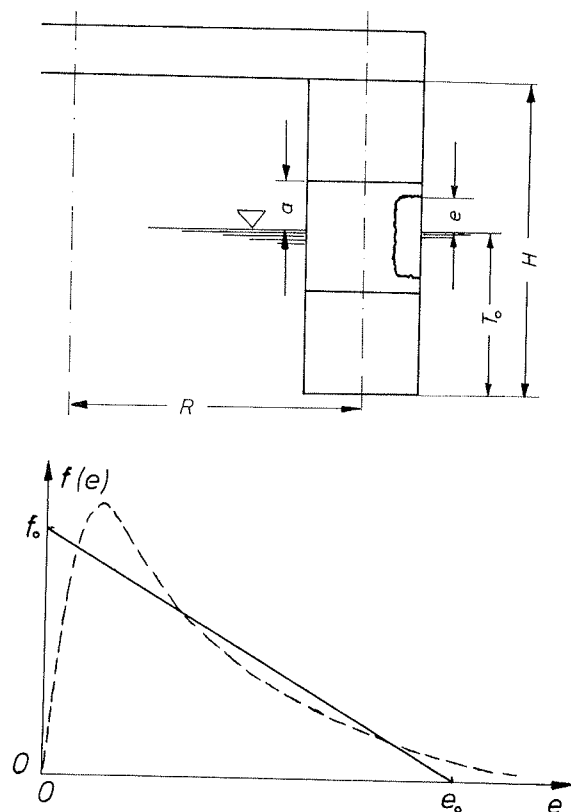


Fig.7 Horizontally subdivided damaged column and frequency function  $f(e)$  of vertical extent "e" of damage

In view of the lacking knowledge of the correct function this simplification may be accepted. Furthermore, it may be assumed that the center of damage is located at the level of  $T_0$ . Then, in case of horizontal watertight subdivision, a most simple formula can be derived for the probability  $P$  that damage extent "e" will be smaller than the distance "a" of a watertight flat from the initial waterplane (see Fig.7):

$$P\{e \leq a\} = \int_0^a f(e) de = 12 \left(\frac{a}{H}\right) - 36 \left(\frac{a}{H}\right)^2 \quad (20)$$

Of course, this formula is only applicable if  $\frac{a}{H} \leq \frac{1}{6}$ . If  $\frac{a}{H} \geq \frac{1}{6}$ , maximum damage extent  $e_0$  does not exceed flat distance "a" and thus  $P\{e \leq a\} = 1$ .

Example: Seven-legged radial symmetric rig without footings ( $N=7$ ,  $R/T_0=2$ ,  $T_0/H=0.5$ ,  $KG/T_0=1.5$ ,  $\mu_s = \mu_v = 1$ ). The influence of  $D/T_0$  on damage stability is rather small and will therefore be ignored. Heeling moments are assumed to cause capsizing if the angle of inclination due to loss of buoyancy exceeds  $\delta_0 = 10^\circ$ .

- a)  $\frac{a}{T_0} = 1$  or  $\frac{a}{H} = \frac{1}{2}$  :  $P\{e \leq a\} = 1$   $\delta_0 > 10^\circ$
- b)  $\frac{a}{T_0} = \frac{1}{2}$  or  $\frac{a}{H} = \frac{1}{4}$  :  $P\{e \leq a\} = 1$   $\delta_0 > 10^\circ$
- c)  $\frac{a}{T_0} = \frac{1}{4}$  or  $\frac{a}{H} = \frac{1}{8}$  :  $P\{e \leq a\} = 0.9375$   
 $\delta_0 = 8^\circ$  if  $e < a$  ;  
 $\delta_0 > 10^\circ$  if  $e \geq a$  .

From this example it can be seen that also rigs with non-optimum geometric characteristics (e.g. small values  $R/T_0$  as in this case) can survive damages if they are subdivided effectively. In this example, a probability of survival of  $P = 0.9375$  can be attained by the arrangement of watertight flats at a distance  $a = H/8$  from the level of  $T_0$ . As  $\delta_0 = 10^\circ$  will not be reached, the damaged rig can withstand some additional moments and, accordingly, the reduction of  $P$ , as mentioned afore, will be very small or even unnecessary.

Considering that the correct distribution of damage data may differ from the assumed one, the real probability of survival may be somewhat higher or lower than  $P = 0.9375$ . For the purpose of judging survivability, however, these differences are not problematic if  $P$ , as calculated in this paper, will be taken as a criterion.

The question of the minimum amount of  $P$  that should be required is a point of discussion and cannot be answered in this paper. From the technical point of view even a regulation prescribing  $P = 1$  for all semisubmersible rigs could be satisfied. If also values  $P < 1$  shall be permitted the minimum values can be set a good deal higher than those which are required in [5] for passenger ships.

#### ACKNOWLEDGEMENT

The author wishes to acknowledge the assistance of Mr. P. Kröger in developing computer programs for  $\delta_0$ -plots. The results given in Table I to V are taken from these plots.

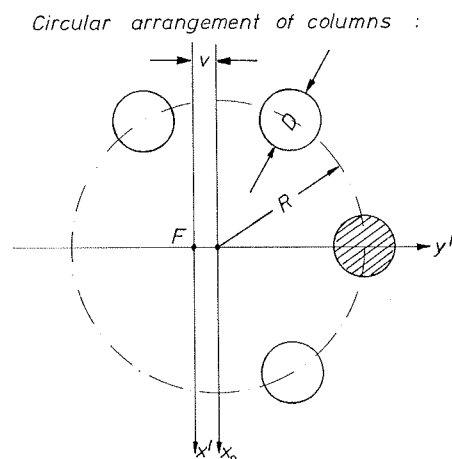
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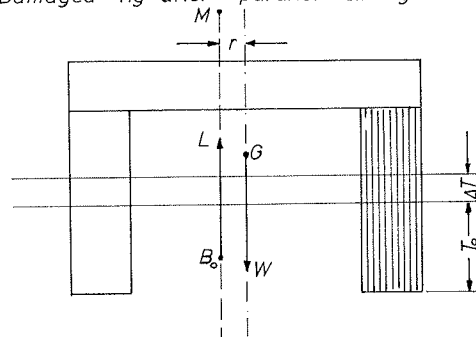
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#### APPENDIX 1 :

Calculation of damage stability of a multi-legged radial symmetric rig with non-subdivided columns and without footings



Damaged rig after parallel sinkage :



Damaged rig in final stage of flooding :

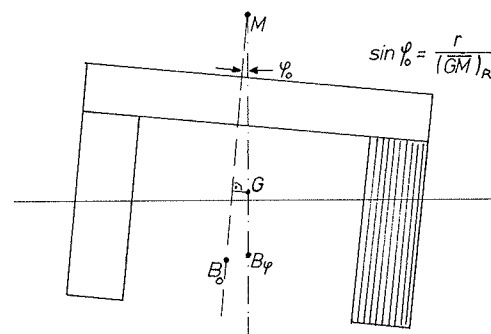


Fig.1 Radial symmetric rig without footings in damaged condition

a) Intact Stability

All symbols are listed and defined in the nomenclature.

$$\nabla_o = \frac{\pi}{4} N D^2 T_o \quad \frac{(\overline{KB})_o}{T_o} = \frac{1}{2}$$

$$\frac{(\overline{BM})_o}{T_o} = \frac{J_{ox}}{\nabla_o \cdot T_o} = \frac{1}{2} \left( \frac{R}{T_o} \right)^2 + \frac{1}{16} \left( \frac{D}{T_o} \right)^2$$

$$\frac{(\overline{GM})_o}{T_o} = \frac{1}{2} \left( \frac{R}{T_o} \right)^2 + \frac{1}{16} \left( \frac{D}{T_o} \right)^2 + \frac{1}{2} - \frac{\overline{KG}}{T_o}$$

b) Damage Stability

$$\frac{v}{R} = \frac{\mu_s/N}{1-\mu_s/N} \quad \frac{r}{R} = \frac{\mu_v/N}{1-\mu_v/N}$$

$$\frac{\Delta T}{T_o} = \frac{\mu_v/N}{1-\mu_v/N} \quad \frac{(\overline{KB})_R}{T_o} = \frac{1}{2} \left( \frac{1}{1-\mu_v/N} \right)$$

$$\frac{(\overline{BM})_R}{T_o} = \frac{J_{x'}}{\nabla_o \cdot T_o} = \frac{1}{2} \left( \frac{1-3\mu_s/N}{1-\mu_s/N} \right) \left( \frac{R}{T_o} \right)^2 + \frac{1}{16} \left( \frac{1-\mu_s/N}{1-\mu_s/N} \right) \left( \frac{D}{T_o} \right)^2$$

$$\frac{(\overline{GM})_R}{T_o} = \frac{1}{2} \left( \frac{1-3\mu_s/N}{1-\mu_s/N} \right) \left( \frac{R}{T_o} \right)^2 + \frac{1}{16} \left( \frac{1-\mu_s/N}{1-\mu_s/N} \right) \left( \frac{D}{T_o} \right)^2 + \frac{1}{2} \left( \frac{1}{1-\mu_v/N} \right) - \frac{\overline{KG}}{T_o}$$

From Fig.1 :  $\sin \varphi_o = \frac{r}{(\overline{GM})_R} = \frac{\frac{\pi}{R} \cdot \frac{R}{T_o}}{(\overline{GM})_R / T_o}$

The final formula for the angle of inclination is obtained by using the foregoing expressions for  $r/R$  and  $(\overline{GM})_R/T_o$  : see Section 2.1, equation (1) .

APPENDIX 2 :

Calculation of damage stability of a multi-legged radial symmetric rig with subdivided columns and without footings

a) Intact Stability

See Appendix 1

b) Damage Stability

Depending on whether the upper flat of the damaged compartment is located above or below final waterline, damage stability values will be different.

ity values will be different.

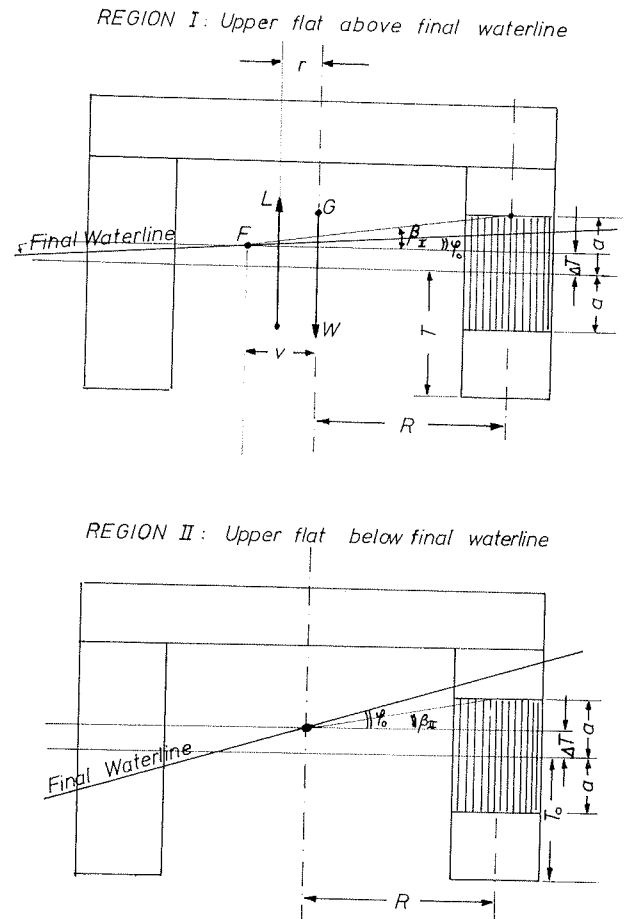


Fig.2 Radial symmetric rig with subdivided columns: upper flat of the damaged compartment above and below final waterline

Region I :  $\varphi_o < \beta_I$

$$\frac{v}{R} = \frac{\mu_s/N}{1-\mu_s/N} \quad \frac{r}{R} = \left( \frac{\mu_v/N}{1-\mu_v/N} \right) \frac{a}{T_o}$$

$$\frac{\Delta T}{T_o} = \left( \frac{\mu_v/N}{1-\mu_v/N} \right) \frac{a}{T_o}$$

$$\frac{(\overline{KB})_R}{T_o} = \frac{1}{2} \left[ 1 + \left( \frac{\mu_v/N}{1-\mu_v/N} \right) \left( \frac{a}{T_o} \right)^2 \right]$$

From Fig.2 :  $\tan \beta_I = \frac{a - \Delta T}{v + R} = \frac{(1-\mu_s/N)(1-2\mu_v/N)}{(1-\mu_v/N)} \left( \frac{a}{R} \right)$

$$\frac{(\overline{BM})_R}{T_o} = \frac{J_{x'}}{\nabla_o \cdot T_o} = \frac{1}{2} \left( \frac{1-3\mu_s/N}{1-\mu_s/N} \right) \left( \frac{R}{T_o} \right)^2 + \frac{1}{16} \left( \frac{1-\mu_s/N}{1-\mu_s/N} \right) \left( \frac{D}{T_o} \right)^2$$

$$\frac{(\overline{GM})_R}{T_0} = \frac{1}{2} \left( \frac{1-3\mu_s/N}{1-\mu_s/N} \right) \left( \frac{R}{T_0} \right)^2 + \frac{1}{16} \left( \frac{1-\mu_s/N}{1-\mu_v/N} \right) \left( \frac{D}{T_0} \right)^2 +$$

$$+ \frac{1}{2} \left[ 1 + \left( \frac{\mu_v/N}{1-\mu_v/N} \right) \left( \frac{\alpha}{T_0} \right)^2 \right] - \frac{\overline{KG}}{T_0}$$

The angle of inclination can be calculated as shown in Appendix 1. The result is given in Section 2.2, equations (2) and (3).

Region II :  $\varphi_0 > \beta_{II}$

In this case, characterized by a constant amount of flooding water, preference is given to the "added-weight" method of calculating damage stability. From Fig.2 follows:

$$\text{tg } \beta_{II} = \frac{a - \Delta T}{R} = (1 - 2 \mu_v/N) \cdot \frac{a}{R}$$

With  $\Delta T = 2 (\mu_v/N) \cdot a$

the volume of displacement, including the weight of the flooding water, becomes

$$\nabla' = \frac{\pi}{4} N \left[ 1 + 2 (\mu_v/N) \frac{a}{T_0} \right] D^2 T_0$$

The height of the center of gravity after flooding is

$$\frac{(\overline{KG})'}{T_0} = \frac{2(\mu_v/N) \frac{a}{T_0} + \frac{\overline{KG}}{T_0}}{1 + 2(\mu_v/N) \frac{a}{T_0}} \quad \text{Further, one gets}$$

$$\frac{(\overline{BM})'}{T_0} = \frac{\frac{1}{2} \left( \frac{R}{T_0} \right)^2 + \frac{1}{16} \left( \frac{D}{T_0} \right)^2}{1 + 2(\mu_v/N) \frac{a}{T_0}} \quad \text{and}$$

$$\frac{(\overline{KB})'}{T_0} = \frac{1}{2} \left[ 1 + 2(\mu_v/N) \frac{a}{T_0} \right]$$

The results of the "added-weight" method of calculation are:

$$\frac{(\overline{GM})'}{T_0} = \left[ \frac{1}{1 + 2(\mu_v/N) \frac{a}{T_0}} \right] \times$$

$$\times \left[ \frac{1}{2} \left( \frac{R}{T_0} \right)^2 + \frac{1}{16} \left( \frac{D}{T_0} \right)^2 + \frac{1}{2} + 2 \left( \frac{\mu_v/N}{1-\mu_v/N} \right) \left( \frac{\alpha}{T_0} \right)^2 - \frac{\overline{KG}}{T_0} \right]$$

Righting moment  $M_R' = \rho_W \cdot g \cdot \nabla' \cdot (\overline{GM})' \cdot \sin \varphi_0$

Heeling moment  $M_H' = \frac{\pi}{2} \rho_W \cdot g \cdot \mu_v \cdot a \cdot R \cdot D^2 \cdot \cos \varphi_0$

From  $M_R' = M_H'$  follows equation (4) in Section 2.2 which may be used for determining the angle of inclination  $\varphi_0$ .

#### APPENDIX 3 :

Calculation of damage stability of a multi-legged radial symmetric rig with non-subdivided columns and a ringlike hull

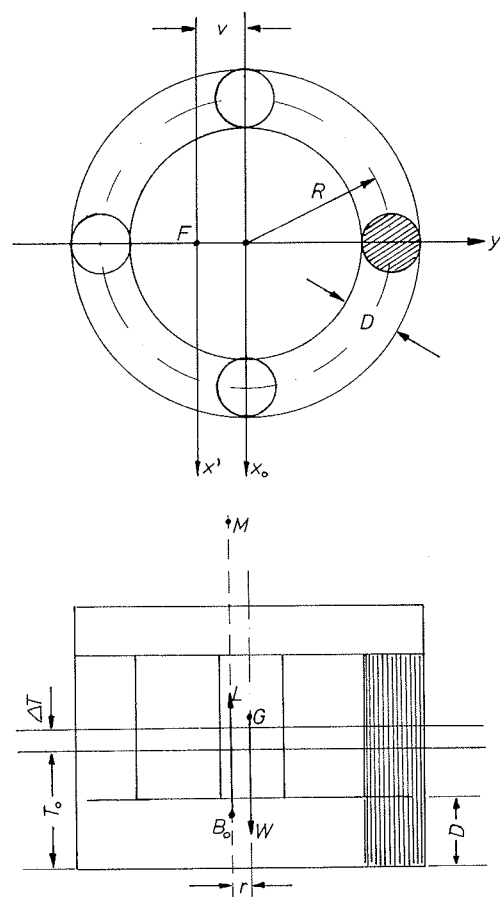


Fig.3 Radial symmetric rig with a ringlike hull in damaged condition

#### a) Intact Stability

$$\nabla_0 = \frac{\pi}{4} N D^2 T_0 \left( 1 - \frac{D}{T_0} + \frac{8R}{N T_0} \right)$$

$$\frac{(\overline{KB})_0}{T_0} = \frac{1 - \left( \frac{D}{T_0} \right)^2 + \frac{8R D}{N T_0^2}}{2 \left( 1 - \frac{D}{T_0} + \frac{8R}{N T_0} \right)}$$

$$\frac{(\overline{BM})_0}{T_0} = \frac{J_{0x}}{\nabla_0 \cdot T_0} = \frac{\left( \frac{R}{T_0} \right)^2 + \frac{1}{8} \left( \frac{D}{T_0} \right)^2}{2 \left( 1 - \frac{D}{T_0} + \frac{8R}{N T_0} \right)}$$

$$\frac{(\overline{GM})_0}{T_0} = \frac{\left( \frac{R}{T_0} \right)^2 + \frac{8R D}{N T_0^2} + 1 - \frac{7}{8} \left( \frac{D}{T_0} \right)^2}{2 \left( 1 - \frac{D}{T_0} + \frac{8R}{N T_0} \right)} - \frac{\overline{KG}}{T_0}$$

#### b) Damage Stability

$$\frac{v}{R} = \frac{\mu_s/N}{1-\mu_s/N} \quad \frac{r}{R} = \frac{\mu_v/N}{\left( 1 - \frac{\mu_v}{N} \right) \left( 1 - \frac{D}{T_0} + \frac{8R}{N T_0} \right)}$$

$$\frac{\Delta T}{T_0} = \frac{\mu_v/N}{1-\mu_v/N}$$

$$\frac{(\overline{KB})_R}{T_0} = \frac{\frac{8RD}{N \cdot T_0^2} + \frac{1}{1-\mu_v/N} - \left(\frac{D}{T_0}\right)^2}{2\left(1 - \frac{D}{T_0} + \frac{8R}{N \cdot T_0}\right)}$$

$$\frac{(\overline{BM})_R}{T_0} = \frac{J_{x'}}{V_0 \cdot T_0} = \frac{\left(\frac{1-3\mu_s/N}{1-\mu_s/N}\right)\left(\frac{R}{T_0}\right)^2 + \frac{1}{8}\left(1-\mu_s/N\right)\left(\frac{D}{T_0}\right)^2}{2\left(1 - \frac{D}{T_0} + \frac{8R}{N \cdot T_0}\right)}$$

$$\begin{aligned} \frac{(\overline{GM})_R}{T_0} &= \frac{\left(\frac{1-3\mu_s/N}{1-\mu_s/N}\right)\left(\frac{R}{T_0}\right)^2 + \frac{8RD}{N \cdot T_0^2} + \frac{1}{1-\mu_v/N}}{2\left(1 - \frac{D}{T_0} + \frac{8R}{N \cdot T_0}\right)} - \\ &\quad - \frac{(7+\mu_s/N) \cdot (D/T_0)^2}{16\left(1 - \frac{D}{T_0} + \frac{8R}{N \cdot T_0}\right)} - \frac{\overline{KG}}{T_0} \end{aligned}$$

$\varphi_0$  follows from  $\sin \varphi_0 = \frac{r/R \cdot R/T_0}{(\overline{GM})_R/T_0}$  (see equation (6) in Section 2.3).

#### APPENDIX 4 :

Calculation of damage stability of a four-legged rectangular rig with non-subdivided columns and a catamaran hull

##### a) Intact Stability

$$V_0 = D^2 T_0 \left(4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}\right)$$

$$\frac{(\overline{KB})_0}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{\pi}{2} - \frac{\pi}{4} \left(\frac{D}{T_0}\right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}}$$

$$\frac{(\overline{BM})_0}{T_0} = \frac{\pi \eta^2 \left(\frac{\ell}{T_0}\right)^2 + \frac{\pi}{16} \left(\frac{D}{T_0}\right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}}$$

$$\frac{(\overline{BM}_L)_0}{T_0} = \frac{\pi \left(\frac{\ell}{T_0}\right)^2 + \frac{\pi}{16} \left(\frac{D}{T_0}\right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}}$$

$$\frac{(\overline{GM})_0}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{\pi}{2} + \pi \eta^2 \left(\frac{\ell}{T_0}\right)^2 - \frac{3\pi}{16} \left(\frac{D}{T_0}\right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}} - \frac{\overline{KG}}{T_0}$$

$$\frac{(\overline{GM}_L)_0}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{\pi}{2} + \pi \left(\frac{\ell}{T_0}\right)^2 - \frac{3\pi}{16} \left(\frac{D}{T_0}\right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}} - \frac{\overline{KG}}{T_0}$$

##### b) Damage Stability

$$\frac{u}{\ell} = \frac{v}{b} = \frac{\mu_s}{4-\mu_s} \quad \frac{r}{\ell} = \frac{p}{\ell} \sqrt{1+\eta^2}$$

$$\frac{p}{\ell} = \frac{q}{b} = \frac{2\mu_v}{(4-\mu_v) \cdot \left(\frac{8\ell}{\pi T_0} + 2 - \frac{D}{T_0}\right)}$$

$$\frac{\Delta T}{T_0} = \frac{\mu_v}{4-\mu_v}$$

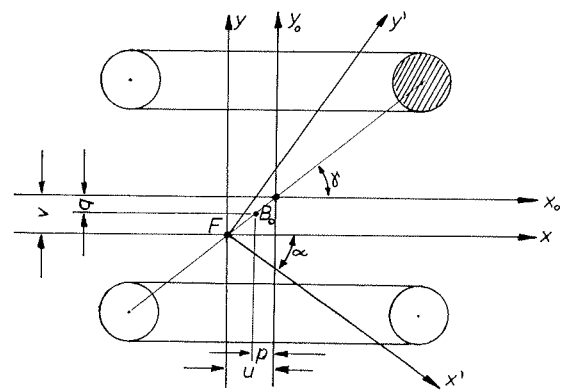
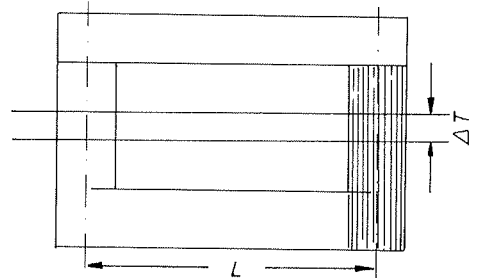
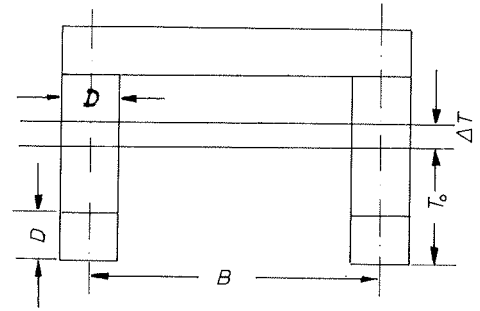


Fig.4 Four-legged rectangular rig with catamaran hull in damaged condition

$$\frac{n}{\ell} = \frac{r}{\ell} \cdot \sin(\alpha + \gamma) \quad \frac{m}{\ell} = \frac{r}{\ell} \cdot \cos(\alpha + \gamma)$$

$$\text{where } \tan 2\alpha = \left| \frac{2J_{xy}}{J_y - J_x} \right| = \frac{\mu_s \cdot \eta}{(2-\mu_s) \cdot (1-\eta^2)}$$

$$\text{and } \tan \gamma = \eta$$

$$\frac{(\overline{KB})_R}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{2\pi}{4-\mu_v} - \frac{\pi}{4} \left(\frac{D}{T_0}\right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}}$$

$$\text{With } \xi \equiv \sqrt{(1-\eta^2)^2 + \left(\frac{\mu_s \cdot \eta}{2-\mu_s}\right)^2}$$

following formulas are obtained :

$$\frac{(\overline{BM})_R}{T_0} = \frac{\pi \left( \frac{2-\mu_s}{4-\mu_s} \right) \left( \frac{\ell}{T_0} \right)^2 (1+\eta^2-\xi) + \frac{\pi}{64} (4-\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}}$$

$$\frac{(\overline{BM}_L)_R}{T_0} = \frac{\pi \left( \frac{2-\mu_s}{4-\mu_s} \right) \left( \frac{\ell}{T_0} \right)^2 (1+\eta^2+\xi) + \frac{\pi}{64} (4-\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}}$$

$$\frac{(\overline{GM})_R}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{2\pi}{4-\mu_v} + \pi \left( \frac{2-\mu_s}{4-\mu_s} \right) \left( \frac{\ell}{T_0} \right)^2 (1+\eta^2-\xi)}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}} -$$

$$- \frac{\frac{\pi}{64} (12+\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}} - \frac{\overline{KG}}{T_0}$$

$$\frac{(\overline{GM}_L)_R}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{2\pi}{4-\mu_v} + \pi \left( \frac{2-\mu_s}{4-\mu_s} \right) \left( \frac{\ell}{T_0} \right)^2 (1+\eta^2+\xi)}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}} -$$

$$- \frac{\frac{\pi}{64} (12+\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \pi - \frac{\pi D}{2 T_0}} - \frac{\overline{KG}}{T_0}$$

Inclination about axis  $x'$  caused by the corresponding component of the revolving couple of buoyancy- and weight-forces:

$$\sin \varphi_0 = \frac{n}{(\overline{GM})_R} = \frac{n \ell \cdot \ell / T_0}{(\overline{GM})_R / T_0}$$

Inclination about axis  $y'$  caused by the corresponding component of the revolving couple of buoyancy- and weight-forces:

$$\sin \psi_0 = \frac{m}{(\overline{GM}_L)_R} = \frac{m \ell \cdot \ell / T_0}{(\overline{GM}_L)_R / T_0}$$

Substituting the expressions given above into the equations for  $\varphi_0$  and  $\psi_0$ , one obtains the full formulas (equations (7) and (8) in Section 2.4). If  $\varphi_0$  and  $\psi_0$  are known,  $\vartheta_0$  can be determined from equation (10).

#### APPENDIX 5 :

Calculation of damage stability of a six-legged rectangular rig with non-subdivided columns and a catamaran hull

##### a) Intact Stability

$$\nabla_0 = D^2 T_0 \left( 4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0} \right)$$

$$\frac{(\overline{KB})_0}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{3}{4} \pi - \frac{\pi}{2} \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}}$$

$$\frac{(\overline{BM})_0}{T_0} = \frac{\frac{3}{2} \pi \eta^2 \left( \frac{\ell}{T_0} \right)^2 + \frac{3\pi}{32} \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}}$$

$$\frac{(\overline{BM}_L)_0}{T_0} = \frac{\pi \left( \frac{\ell}{T_0} \right)^2 + \frac{3\pi}{32} \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}}$$

$$\frac{(\overline{GM})_0}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{3}{4} \pi + \frac{3}{2} \pi \eta^2 \left( \frac{\ell}{T_0} \right)^2 - \frac{13}{32} \pi \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} - \frac{\overline{KG}}{T_0}$$

$$\frac{(\overline{GM}_L)_0}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{3}{4} \pi + \pi \left( \frac{\ell}{T_0} \right)^2 - \frac{13}{32} \pi \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} - \frac{\overline{KG}}{T_0}$$

##### b) Damage Stability

$$\frac{u}{\ell} = \frac{v}{b} = \frac{\mu_s}{6-\mu_s} \quad \frac{r}{\ell} = \frac{p}{\ell} \sqrt{1+\eta^2}$$

$$\frac{p}{\ell} = \frac{q}{b} = \frac{3\mu_v}{(6-\mu_v) \cdot \left( \frac{8\ell}{T_0} + 3 - 2 \frac{D}{T_0} \right)}$$

$$\frac{\Delta T}{T_0} = \frac{\mu_v}{6-\mu_v}$$

$$\frac{(\overline{KB})_R}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{9\pi}{2(6-\mu_v)} - \frac{\pi}{2} \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}}$$

$$\text{With } \xi \equiv \sqrt{1 + \left[ \frac{6\mu_s \eta}{12-5\mu_s-6\eta^2(3-\mu_s)} \right]^2}$$

following formulas are obtained:

$$\frac{(\overline{BM})_R}{T_0} = \frac{\pi \left( \frac{\ell}{T_0} \right)^2 (12-5\mu_s+6\eta^2[3-\mu_s] - |12-5\mu_s-6\eta^2[3-\mu_s]| \xi)}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} +$$

$$+ \frac{\frac{\pi}{64} (6-\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}}$$

$$\frac{(\overline{BM}_L)_R}{T_0} = \frac{\pi \left( \frac{\ell}{T_0} \right)^2 (12-5\mu_s+6\eta^2[3-\mu_s] + |12-5\mu_s-6\eta^2[3-\mu_s]| \xi)}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} +$$

$$+ \frac{\frac{\pi}{64} (6-\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}}$$

$$\frac{(\overline{GM})_R}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{\pi \left( \frac{\ell}{T_0} \right)^2 (12-5\mu_s+6\eta^2[3-\mu_s] - |12-5\mu_s-6\eta^2[3-\mu_s]| \xi)}{4(6-\mu_s)}}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} +$$

$$+ \frac{\frac{9\pi}{2(6-\mu_v)} - \frac{\pi}{64} (26+\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} - \frac{\overline{KG}}{T_0}$$

$$\frac{(\overline{GM}_L)_R}{T_0} = \frac{2 \frac{\ell D}{T_0^2} + \frac{\pi \left( \frac{\ell}{T_0} \right)^2 (12-5\mu_s+6\eta^2[3-\mu_s] + |12-5\mu_s-6\eta^2[3-\mu_s]| \xi)}{4(6-\mu_s)}}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} +$$

$$+ \frac{\frac{9\pi}{2(6-\mu_v)} - \frac{\pi}{64} (26+\mu_s) \left( \frac{D}{T_0} \right)^2}{4 \frac{\ell}{T_0} + \frac{3}{2} \pi - \frac{\pi D}{T_0}} - \frac{\overline{KG}}{T_0}$$

From the different definition of  $\alpha$  (see Fig.6) follow different formulas for the component leverarms  $n$  and  $m$ :

$$\text{if } \eta \leq \sqrt{\frac{12-5\mu_s}{18-6\mu_s}} : \quad \frac{n}{l} = \frac{r}{l} \cdot \sin(\alpha + \gamma) \\ \frac{m}{l} = \frac{r}{l} \cdot \cos(\alpha + \gamma)$$

$$\text{if } \eta > \sqrt{\frac{12-5\mu_s}{18-6\mu_s}} : \quad \frac{n}{l} = \frac{r}{l} \cdot \cos(\alpha - \gamma) \\ \frac{m}{l} = \frac{r}{l} \cdot |\sin(\alpha - \gamma)|$$

$$\text{where } \tan 2\alpha = \left| \frac{6\mu_s \cdot \eta}{12 - 5\mu_s - 6\eta^2(3 - \mu_s)} \right|$$

$$\text{and } \tan \gamma = \eta$$

Angles of inclination about axes  $x'$  and  $y'$ :

$$\sin \varphi_0 = \frac{n}{(\overline{GM})_R} = \frac{n/l \cdot l/T_0}{(\overline{GM})_R/T_0}$$

$$\sin \varphi_0 = \frac{m}{(\overline{GM}_L)_R} = \frac{m/l \cdot l/T_0}{(\overline{GM}_L)_R/T_0}$$

$\varphi_0$  and  $\psi_0$  are functions of  $\mu_s, \mu_v, \eta, l/T_0, D/T_0$  and  $\overline{KG}/T_0$ . These functions are obtained after some transformations by using the above-written formulas (equations (12), (13), (14), (15) in Section 2.5). The final angle of inclination,  $\vartheta_0$ , can be calculated as shown in Section 2.4 (equation (10), Fig.5).

## Discussion

S. Rusaas (Det norske Veritas, Norway)

Is the survivability after damage assessed with buoyancy from the columns alone? In that case, I think that the probability to survive a major damage to one of the columns will be rather hypothetical, since operational requirements always imply limits to the actual design. During our investigation after the Kielland accident, we investigated a number of rig designs with respect to survivability when the buoyancy from one of the columns is lost. We found that no one (even with high no. of columns) were able to survive such a damage without buoyancy in the deck structure. In your concluding remarks in item 3 you list a number of points which should be observed in order to attain a high degree of survivability. We feel that the most important point is missing: Buoyancy in or near the deck structure. For most semi-submersibles, it does not imply severe modifications in order to obtain at least some buoyancy in way of the deck structure, since one always has some sort of superstructure here.

M. Abe (Mitsui Eng. & Shipbuilding Co., Japan)

The author discussed about the static features of damaged stability of semi-submersibles, and also stated on minimization of wave-induced heel moment related to the change of the volume of displacement. If an inclination of heel is caused by flooding with a semi-submersible, particularly of column stabilized type, a larger wave-excited higher-order wave force shall be induced comparing with the state of leveled draft.

It will be explained in principle that vertical force acting on inclined-side is

smaller than the opposite-side. This induces a steady tilt to give a larger inclination. In my opinion the effect of steady tilt mentioned shall not be disregarded when discussing the damage stability.

O. Krappinger (Hamburg Ship Model Basin, FRG)

I congratulate Prof. Abicht for a very nice, and I think very useful paper. I would like to comment on the assumption made for the maximum damage length (see the two lines following equation (17)). The damage length is estimated to be proportional to the height of the column. This is alone in analogy to ships where the damage length is assumed to be proportional to the ship length. There is a difference between ships and semi-submersible with regard to the damage length which is to be expected: As the IMO-collection of damage data has shown, big ships meet in the average bigger damage than small ships do. This results in average damage lengths which are proportional to the ship length. The average size of ships which could collide with a semi-submersible seems to be independent from the size of the semi-submersible. Therefore, I would expect that the average (or what is the same, the maximum) damage length is constant in this case. It might even decrease with increasing dimensions of the submersible.

### Author's Reply

To the questions of Mr. Rusaas I will give the following answers:

1. All damage stability calculations were made for the still water condition and only for rigs which have a final angle of inclination being so small that neither the deck structure immerses nor any column or footing emerges.



2. The characteristics and KG-values which are necessary to fulfil the above mentioned conditions may be read from Table I to V of my paper.

3. The results show that it will be possible to design rigs which are able to survive the flooding of a column without the need of the reserve buoyancy of the deck structure.

4. Of course, the designer of a rig must not only pay attention to damage stability but also to the behaviour in heavy seas and above all, he must try to find an economic construction. Therefore, actual designs will differ from designs which are optimal from the damage stability point of view.

Mr. Abe points out that a steady tilt will be caused by a vertical higher-order-force induced by waves. This force was not taken into account in my damage stability calcu-

lations because they were made for the still water condition. The goal in my paper was to give a first idea how column stabilized rigs should be designed in order to attain rigs which are "unsinkable" in the case of the flooding of one column. Of course, in a second step in the design stage also these forces should be observed.

The author agrees with Professor Krappinger who pointed out that for rigs it seems to be more realistic to assume a maximum vertical extent of damage  $e_0$  which is constant, than a maximum vertical extent which is proportional to the height of the column. Perhaps a final answer can be given some years later because then a sufficient number of damage data may exist. The value  $e_0=H/6$  in my paper is only a value which was taken to demonstrate the way, how the survival probability can be calculated.

# MODEL EXPERIMENTS ON CAPSIZING OF A JACK-UP DRILLING PLATFORM

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## SUMMARY

In order to investigate the mechanism of capsizing of a certain jack-up platform, which happened to be in a low freeboard condition in waves under tow, a model platform was constructed and tested, with different amount of flooding of the mud pump tank. Experiments were conducted in regular waves and irregular waves of a certain realistic spectrum with and without wind loads.

Characteristic phenomena observed lead to the conclusion that non-linear or second order drifting moments came into play and that depending on the amount of loss of righting lever due to flooding, the model may gradually experience a drift in heel and trim which would either lead to motions about an equilibrium inclined condition or capsizing.

Mechanism of the non-linear drifting moment is tentatively explained and quantitatively measured. It was shown that for a given regular wave condition in beam or quartering sea, the non-linear drifting moment increases with the angle of heel, which is of importance in coping with capsizing events of this nature.

## SYMBOLS

|           |  |
|-----------|--|
| L         | length of platform model               |
| B         | breadth of platform model              |
| F.B.      | free-board height of platform model    |
| $\phi$    | rolling angle                          |
| $\phi$    | drift in heel (the mean angle of roll) |
| $\phi$    | roll amplitude                         |
| $\psi$    | pitch amplitude                        |
| z         | heave amplitude                        |
| $\lambda$ | wave length                            |
| h         | wave height                            |
| $\alpha$  | wave slope                             |

|            |   |
|------------|---|
| k          | wave number   |
| $\bar{T}$  | characteristic period of wave spectra                                     |
| $H_{1/2}$  | significant wave height   |
| $\omega_0$ | natural frequency of roll   |
| $\omega_e$ | encounter frequency of wave   |
| $M_f$      | heeling moment by wind  |
| $M_r$      | righting moment of platform model   |
| $M_{cap}$  | drifting moment or capsizing moment                                       |
| m          | symmetrical excitation moment amplitude caused by wave                    |
| $m_1$      | asymmetrical excitation moment amplitude caused by shipping water on deck |
| $H_z$      | relative wave height above deck edge on the side incident to waves        |
| $\beta$    | non-linear coefficient of restoring moment                                |
| 2 $\mu$    | damping coefficient   |
| X          | wave direction  |
| Q          | amount of water admitted into the MPT                                     |

## 1. INTRODUCTION

With increasing amount of offshore engineering carried out around the world, the importance of stability of drilling platforms in station and under tow could not be underestimated. To wit, the capsizing of the Alexander L.Kielland, the Ranger and the Pohai No.2 were three outstanding cases occurring in different parts of the world with heavy losses of lives. The present report is not meant to be a simulation of the actual conditions of capsizing of Pohai No. 2 — which has been investigated, but rather as a series of extended experiments on a hypothetical low-freeboard jack-up drilling platform with an aim to investigate the mechanism of such characteristic capsizing under a realistic background.

The model is mat type jack-up drilling platform, the outline of which is shown in Fig.1. A mud pump tank is installed on the aft deck of the platform, to which different

amount of water is added to investigate the effect of quasi-static stages of flooding of this tank. The principal dimensions of the model are given in Table 1,

Table 1

|                         |          |
|-------------------------|----------|
| Length                  | 1.203 m  |
| Breadth                 | 1.0 m    |
| Displacement            | 320 kg   |
| Height of C.G. above BL | 0.25 m   |
| Mean Freeboard          | 0.038 m  |
| Towing Speed            | 0.25 m/s |

Four kinds of tests and simulations are carried out:

- a) Towing in regular waves representing beam and quartering seas,
- b) Towing in irregular beam and quartering seas,
- c) Bench tests on a plexiglass model of the mud pump tank including forced oscillations of the latter to find out the dynamic moment of moving water acting on the platform under different frequencies,
- d) Simulation on an analogue computer to find out the influence of shipped water on non-linear rolling of the platform in a beam sea.

From analysis of these results, an important view is advanced in the present report which may be quite general to low-freeboard floating platforms or damaged platforms floating under a relatively large angle of heel, i.e. under very unsymmetrical conditions.

## 2. MODEL EXPERIMENTS

Model experiments were carried out in CSSRC 69m x 46m x 4m seakeeping basin. The model was towed by a carriage at a speed of 0.25 m/sec and the tow line was about 12 meters in length. Gyroscopes and accelerometers were installed in a watertight compartment of the model to measure its roll, pitch and heave motions. Two capacity wave probes are fixed on port and starboard respectively of the platform deck to measure the relative wave elevations. There are two holes on the deck, through which water is admitted into the otherwise intact mud pump tank in precalibrated amounts. Watertight covers were provided to seal off these holes after each filling of water. Model experiments in the basin were conducted in two groups:

### 2.1 Motions and Behaviour of the Model in Regular Waves

The amount of water admitted into the aft mud pump tank (MPT) were 0, 3.87kg and 10.10kg respectively. Righting moment curves corrected for free surface effects corresponding to different amount of water in MPT were calculated and shown in Fig.2. Towing

experiments were carried out under these conditions in both beam and quartering seas. The wave length ranges from 2.5 meters to 15 meters and the wave height is fixed at approximately 125mm. The measured roll, pitch and heave response and the mean angle of heel, termed here drift in heel, (and taken as the mean of the asymmetrical roll angles) are shown in Fig. 3 to Fig.6. Fig.7 shows the mean height of shipped water at the deck edge of the wind ward side of the platform in beam seas. The results obtained from regular wave experiments are as follows:

- a) Shipping of water on deck is serious because of the low freeboard of the platform. See Fig.8. For shorter waves, this shipping of water is even more serious. See Fig.7.
- b) Roll amplitude is small, in general, the double amplitude of roll ranges from 5 to 10 in above mentioned waves. But the rolling is not symmetrical, in other words the platform rolls symmetrically about a mean or "drift" angle of heel which is the asymmetric part of the motion. The magnitude of this mean angle varies with wave frequency. The largest drift in heel occurs near heave synchronism. The occurrence of this drift in heel is most obviously traced to the effect of shipping of green water on deck. However, there may be other subtle reasons due to unsymmetrical pressures acting on the underwater hull.
- c) The drift in heel increases with the amount of water added in MPT. See Fig.9.
- d) The peak value of roll response curve decreases with increasing amount of water added to MPT. But for excitation frequencies larger than the natural frequency of roll, there is little difference in roll between the three cases of water admission. This shows that water in the MPT acts as a roll-stabilizer in near resonance frequency, but this effect is not significant when excitation frequencies are higher than the synchronism range. This result agreed with bench test of the MPT. (See section 4).

### 2.2 Capsizing Simulation Experiment in Irregular Seas

From the above experiments in regular waves, it is demonstrated that with greater and greater amount of water in MPT roll amplitude of the symmetrical part decreases at roll synchronism but the asymmetric part of roll increases. That is, with increased amount of water in the MPT the drift in heel is increased. This drift angle is dangerous for low-freeboard platforms. We therefore tested with different amount of

water in the MPT and set to tow the model in irregular waves. The spectrum of irregular wave is shown in Fig.10. with a characteristic period of  $T=1.45\text{sec}$  and significant wave height  $H_w = 120\text{mm}$ . When the amount of water in MPT is increased to 11kg, capsizing of the platform model always occurs even if the wind moment were zero, and the direction of capsizing appears to be diagonalwise i.e. towards the wind and trimming aft, see Fig.11. The time history of roll in the capsizing process is shown in Fig.12. This gradual capsizing process is similar to that of low freeboard ships with shipping of green water on deck (ref. 1). Next, a weight is attached on the leeward side of the deck to simulate a 1.56kg-m wind heeling moment, then an amount of 4.5kg of water added to MPT is sufficient to produce capsizing. The direction of capsizing in this case is also diagonalwise but is away from the wind i.e. towards leeward and trimming aft, see Fig.13. The directions of capsizing are also similar to small low freeboard vessels which capsize in windward direction when it was under a gentle breeze and in leeward direction when it was under a strong wind (ref. 2). The cause of capsizing diagonalwise may be traced to the fact that there is a superstructure at the bow which is not flooded while the flooded mud pump tank is situated aft, and that as the heeling angle increases the trim by the stern also increases. It is observed from the model capsizing experiment that even if the platform is not subjected to any wind heeling moment and that if initially the angle of heel were zero, there will still develop a drift in heeling angle in the windward direction presumably caused by asymmetric moment due to shipping of water on deck as a result of the low freeboard characteristic of the model. Fig.14(a) and (b) show the model without wind moment but with an initial drift angle in heel towards the wind. The initial drift angle is derived by the relative case of shipping water on the windward side. In this condition, when a wave crest arrives at the position shown by Fig.14(a), a drift moment to starboard would be developed due to water shipped on deck. Again, as the wave crest moves over to position shown in Fig.14(b), a drift moment to starboard would still be developed due to the added buoyancy of the port platform which has a higher freeboard. Thus, with every cycle of wave passage, the drift in heel increases, which further aggravates the situation causing greater drifting moment. The vicious cycle continues until either an equilibrium angle of heel is reached, where the righting moment of the model is sufficient to balance the drifting moment at the same angle but with the former having a stiffer slope. In which case the model will roll about the equilibrium drift angle  $\phi$ . However, when the righting moment curve of the model is below the drift moment curve, the model capsizes.

When there is strong wind blowing in the direction of wave propagation there will be wind moment (represented by an offset

weight in the experiment), and the model takes on an initial inclination to leeward as shown in Fig.15(a) and (b). The same reasoning applies as in the case of Fig.14, except that the direction of drifting moment is reversed, and that in Fig.15(a), when the wave crest is over the windward side there is an additional impact force and heeling moment due to the dynamic pressure of wave crest slamming onto the high freeboard side. Besides, when the wave crest moves over to the leeward side as shown in Fig.15(b), although the wave crest is higher than the deck, there is no shipping of green water owing to the fact that the wave is propagating away from the deck side (instead of incident to the inclined deck as in Fig.14 (b)). Consequently, in this half cycle there is little apparent drifting moment acting to the port. Again with every cycle of wave passage, the drift in heel increases, either to an equilibrium value, or until the model capsizes.

With a view to measure this drifting moment quantitatively and validate the above hypothesis, the following treatment and analysis of test results were applied.

### 3. ANALYSIS OF TEST RESULTS

As in the case of measuring slow drifting force of a ship in waves, a soft spring is often applied both to restrain the model and also as a sensor to measure the drifting force. In the present experiment a soft spring which both restrains and measures the drifting moment in heel is required. Now, the "model-ambient water" is by itself a natural soft spring system with a known restoring moment curve. Therefore, the righting moment curves (GZ curves) could be first calculated very accurately, by means of a computer, for each case of water admission in the MPT. The GZ curves corrected for free surface could be calculated for any diagonalwise inclination, i.e. for any combined heel and trim, but in the present paper only GZ curve in the transverse plane is considered and is given in Fig.2. Since the amount of water added to the MPT (ranges from 0-11kg) is only a fraction of the model displacement of 320kg, one could argue that the addition of water in the MPT besides altering the spring characteristic of the "model-ambient water" system, i.e. GZ curve, has little influence on changing the attitude of the model. For instance a change of 9mm in average draft would be obtained corresponding to 11kg of water admission. It may thus be assumed that as far as wave excitation force is concerned, the 3 cases of water admission correspond approximately to one and the same displacement and attitude of the model with respect to the action of wind and waves. One may then think of the righting moments corresponding to mean angles of roll response (the drift in heel) of the model with 3 different amounts of water in MPT in a certain regular wave as a measure of the wave drifting moment developed for different angles of drift in heel. For example, from

beam sea regular wave test Fig.6.(a), if a vertical ordinate is erected at  $\omega_1 T/2\pi = 1.2$ , which corresponds to a wave length of 5.24m and a wave height (double amplitude) of 125mm. One would get intersections of  $\theta = 1.93^\circ, 2.58^\circ, 8.20^\circ (\alpha = 4.29)$  respectively, for amount of water addition  $Q = 0, 3.87$  and  $10.10\text{kg}$  respectively. Looking at Fig.2, One finds  $M_r$  (which is a measure of the drifting moment) equals  $2.37\text{kg-m}, 3.25\text{kg-m}$  and  $6.29\text{-m}$  respectively from the three different "spring" characteristics. Constructing the drifting moment  $M_{cap}$  V.S. drifting angle  $\phi'$  curve (heavy solid line) in Fig.16, one gets a curve representing the capsizing moment as a function of drifting angle  $\phi'$ . It is worth noting that this curve is a monotonic rising curve, which validates the hypothesis that once an initial heel is started either by wave or by wind, the nonlinear or second order wave exciting moment builds up as a monotonic rising function of drift angle in heel. The generation of this second order drifting moment is roughly described as the action of shipping water on deck and nonlinear buoyancy effect in the previous section. Work is continuing at present to give a 3-D numerical pressure analysis of the model under test, so as to illustrate further on the nature of 2nd order drifting moment. However, the 2nd order drifting moment curve (Fig.16) obtained experimentally reveals an important aspect and peculiar nature of the wave drifting moment in action in regular waves, a revelation as important as the added resistance experienced by ships moving in regular waves. Two conclusions may be drawn following this analysis.

- (1) As far as capsizing in above mentioned regular wave is concerned, the model platform would not capsize for the two cases  $Q = 0$  and  $3.87\text{kg}$  respectively. It would only roll (with an amplitude of  $5^\circ - 10^\circ$ ) about an angle of heel  $1.93^\circ$  and  $2.58^\circ$  respectively. The drift angle increases rapidly with more addition of water, and with  $Q = 10.10\text{kg}$ , the  $M_{cap}$  curve almost coincides with the righting moment curve  $M_r$  of the model and the model would only balance precariously at an angle of heel of  $8.25^\circ$ , considering the max.  $M_r$  in this case is at  $\phi = 10^\circ$ , the model would eventually capsize due to insufficient dynamic stability introduced by rolling. Thus with low freeboard ship or platform, the direct and important factor of stability is still the maintenance of sufficient righting moment, which may be provided by adjustment of many factors including the lowering of center of gravity. For a platform damaged and inclined by any reason, it is always important to maintain the maximum of residual stability, which means all access holes, ventilation ports should be seaworthy and should be easily closed off against flooding of sea water.
- (2) In irregular waves extremely slow oscillations may be set up by difference

frequencies and wave grouping phenomena. However, if the stability of the model is low, the slow drifting in heel may just gradually drift the model over and capsizing takes place as a slowly developing process. See Fig.12. This diagram is typical of all the capsizing experiments done for the present model - a total of 20 cases.

#### 4. TESTING OF MUD PUMP TANK MODEL WITH VARIOUS AMOUNT OF WATER ADMISSION ON A ROLL TABLE

The purpose of this test is to find out the dynamical effect of water in the MPT. It is known that water in MPT may have a threefold effect.

4.1 Static effect of a deformable added weight with a free surface. This is taken care of in the numerical calculation of  $GZ$  or  $M_r$  curves as shown in Fig.2.

4.2 Symmetrical part of dynamic effect of this moving water behaving like water in a passive anti-rolling tank. This is to be evaluated by the bench test on a roll table constructed for testing of antirolling tanks.

4.3 Asymmetrical part of this water moving in the MPT, especially when the neutral point of motion is at an inclined position. This is to be evaluated by a separate bench test on the same roll table.

Bench tests were conducted in the hydrodynamic laboratory of Shanghai Jiao Tung University where a small roll table is employed. The MPT model is made of plexiglass and to the same scale as the model in tank tests. The amounts of water admitted into the MPT model were  $3.87\text{kg}, 10.10\text{kg}$  and  $16.35\text{kg}$  respectively. Force gages were placed under the supports of MPT, (Fig.17) so that dynamical moments generated by moving water in MPT were measured. See Fig.19. The excitation moments provided by driving motor of roll table were measured and further converted accordingly to the wave slopes. So that the effect of the MPT on the platform model expressed as a roll amplitude response is obtained (Fig.18). It is to be noted that the amplitude response is high at the natural frequency of roll of the platform model and that the more the amount of water admitted to the MPT the more the damping effect of the latter acting as a passive anti-rolling tank. However, at frequencies higher and lower than the synchronism range, the total effect and the difference between any of the 3 cases of water admission is small (Fig.18). Turning to the amplitude of dynamical moment measured by the force gauges under the supports (Fig.19), it is seen that for cases at and above the natural frequency of roll, the dynamical moment of tank water is at least  $90^\circ$  out of phase and lagging behind the motion of the model. The tendency of

moment amplitude response is approximately the same as that of the roll amplitude response. However, the peak moment for those cases with lesser amounts of water admission is hard to measure exactly. For frequencies lower than  $\omega/\omega_0 = 0.6$ , the moment amplitude generated by MPT rises abruptly. This is explained by the fact that the moment amplitude measured at very low frequencies approaches that of the static moment due to tank water behaving as a shifting weight moving in phase with the angle of roll. This effect has already been taken into account in the static effect of tank water outlined in 4.1, and hence should not be included in the dynamic effect. In any case, since the effect of wave frequencies investigated lies on the higher frequency side of synchronism, it may be concluded that the action of water in MPT when the platform is rolling about its up right position is similar to that of an anti-rolling tank, that the total effect is the reduction of roll in synchronous waves, and that this effect is small in higher frequency waves for all 3 quantities of tank water considered. Tests were also conducted for rolling of MPT about an inclined position. (Table 2)

Table 2

|   |                                       |
|---|---------------------------------------|
| Amount of water in MPT                      | 4.5kg, 11kg                           |
| Preincluded angle about which MPT is rolled | 2°, 4°, 8°, 10°, 12°                  |
| Rolling frequency                           | 2.5rad/sec., 3.0rad/sec., 3.5rad/sec. |
| Double amplitude of forced rolling          | 8°                                    |

A representative time history of force gauge measurement is presented in Fig.20. Enough is to say that no asymmetric part of dynamical moment is observed for all cases considered. The static moment of the tank water as a shifted weight at the preinclined angle of course has been deducted by zero setting of the force gauges before the rolling experiment.

## 5. SIMULATION OF AN ANALOGUE COMPUTER

On the basis of model experiments in regular waves, a simple equation of rolling motion in which the initial heeling angle is zero, i.e. without consideration of wind moment is set up as follows:

$$\ddot{\phi} + 2\mu\dot{\phi} + \omega_0^2\phi - \beta\phi^3 = m\sin\omega_0 t + \begin{cases} m_1\sin(\omega_0 t + k\frac{\phi}{2}) & 0 < \omega_0 t + k\frac{\phi}{2} < \pi \\ 0 & \pi \leq \omega_0 t + k\frac{\phi}{2} < 2\pi \end{cases} \quad (5.1)$$

The first term on the right hand side of eq. (5.1) represents the wave excitation moment, while the second term considers only the moment produced by shipping of water on deck. As the incident wave crest hit the windward side of platform, water is shipped on deck, while for the other half period no water is shipped on deck. It is therefore assumed that moment due to shipping of water varies sinusoidally for half a period,

while it is zero for the other half period. The amplitude of excitation moment  $m_1$ , produced by shipping of water can be calculated by the following formula, see Fig.21.

$$m_1 = \frac{1}{2} H_z \cdot \frac{1}{2} B \cdot l \cdot \gamma \left[ \frac{B}{3} \cos\phi + \left( \frac{H_z}{3} + F.B. \right) \sin\phi \right] \quad (5.2)$$

where

|          |   |
|----------|---|
| $H_z$    | relative wave height above deck edge of the wind-ward side, obtained from experiment.             |
| $B$      | breadth of platform   |
| $l$      | longitudinal extent of water shipped on deck measured along the fore and aft axis of the platform |
| $\gamma$ | specific gravity of water   |
| $F.B.$   | free-board of wind-ward deck edge   |
| $\phi$   | angle of roll   |

It has been seen in regular wave experiments that shipping of water on deck in succession would produce a drift in heel. The aim of the analogue simulation is to check on a rough but simple basis, the action of shipping water on roll and on drift in heel. Deleting the first term on the right hand side of equation (5.1), (Fig.22b), the analogue computation gives an asymmetric roll motion (Fig.22e). Deleting the second term on the RHS of eq.(5.1) (Fig.22a), the analogue computation would give a symmetrical roll motion (Fig. 22d) excited purely by wave excitation moment. Fig. 22c shows the results produced by two excitation moments together and the total motion produced by such an excitation is shown in Fig.22f, it may be seen that shipping of water not only produces a drift in heeling angle but also plays an important role on the amplitude of roll.

Table 3 gives some of the input and output results of analogue computation following the full eq. (5.1)

Table 3

|                                     |       |       |
|-------------------------------------|-------|-------|
| $\omega$                            | 2.82  | 3.42  |
| $2\mu$                              | 1.5   | 1.72  |
| $\omega_0^2$                        | 4.69  | 4.69  |
| $\beta$                             | 27.1  | 27.1  |
| $H_z$                               | 0.057 | 0.057 |
| $m$                                 | 0.204 | 0.356 |
| Computed double amplitude of roll   | 7.73° | 6.3°  |
| Experiment double amplitude of roll | 7.5°  | 6.1°  |
| Computed drift in heel              | 1.7°  | 2.3°  |
| Experiment drift in heel            | 1.4°  | 1.96° |

## 6. CONCLUSION

The motions of drilling platform in beam and quartering seas in low free-board condition are investigated by systematic experiments in regular and irregular waves, supplemented by bench tests of the Mud Pump Tank with varying amount of water and

analogue simulation on a computer. The following conclusions may be drawn.

6.1 A low free-board drilling platform in tow and exposed to beam and quartering seas would sooner or later manifest a list toward the wind when the wind moment is negligibly small, or a list to leeward if the wind moment is sufficiently strong. In the former case the initial list is caused by periodic shipping of water on deck on the side incident to waves.

6.2 With the appearance of an initial list, a vicious cycle is started following each roll. This process is caused by the difference in free-board on the wind-ward and leeward side of the deck. Nonlinear drifting moments is generated, of which the most apparent reason is that due to successively intensified unsymmetrical shipping of water, however there might be more subtle second order moments coming into play, for instance by difference of pressure acting on the underwater hull. The nature of the non-linear drifting moment  $M_{cap}$  in specific regular sea as measured by experiment is a monotonic increasing curve with the angle of list  $\phi'$  (drift in heel).

6.3 The intersection of this  $M_{cap}(\phi')$  curve with that of the righting moment curve  $M_r(\phi)$  determines the final angle of repose of the platform about which the platform will roll.

6.4 If the  $M_r(\phi)$  curve is lower than  $M_{cap}(\phi')$  by reason of decrease in stability by varying degrees of flooding of MPT, the platform will capsize. However, since the development of drift in heel takes time, the capsizing is a slow process. A typical time history is a slow drifting process in heel, on to which is superposed the normal rolling motion.

6.5 A platform with low free-board or a platform inclined to one side (unsymmetrical cross section), would not be lost if its inherent righting moment is sufficiently high. Consequently for future seaworthiness consideration, utmost attention should be paid to safety measures safeguarding against inflow of water through any of the hatches or ports to the internal spaces of the platform. This attention should be paid both in general layout, structural strength design of vent pipes, windows, port covers, water tight doors etc. as well as in the incorporation of automatic closure systems that will close all these ports once a certain critical low free-board or list is exceeded.

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Vehicles, 1975

2. Boroday, I.K. and Rakhmanin, N.N. "State of the Art of Studies on Capsizing of an Intact Ship in Stormy Weather Condition" 14th ITTC Proceedings, Vol.4

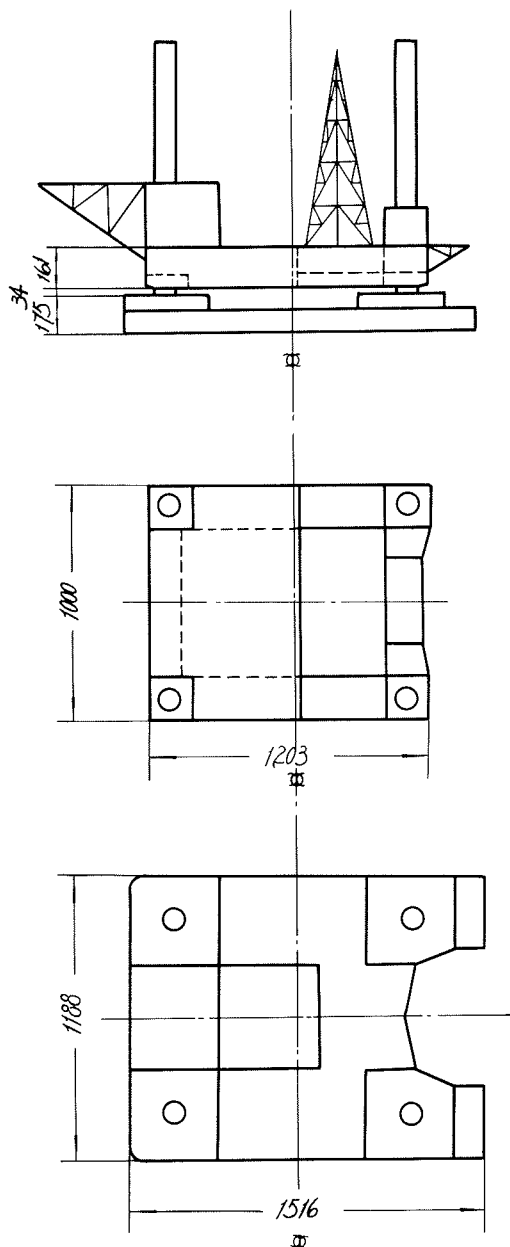


Fig.1. Principal dimensions of the platform model

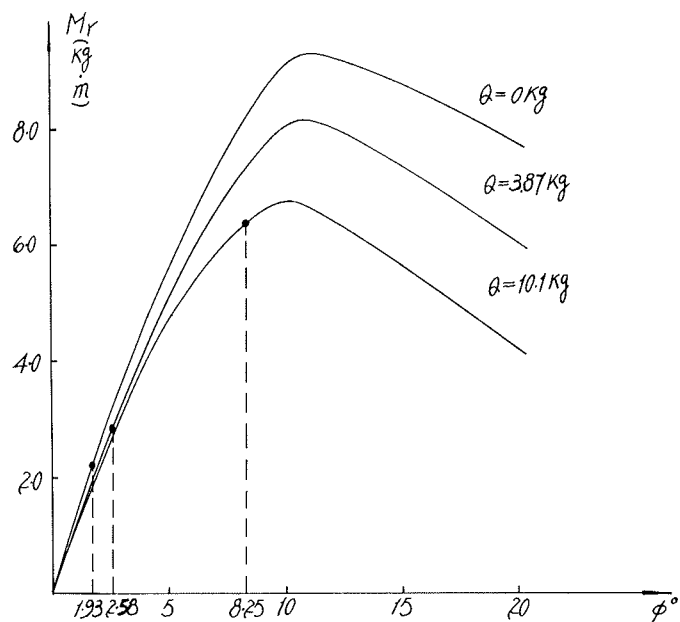


Fig.2. The curves of transverse righting moment of the platform model

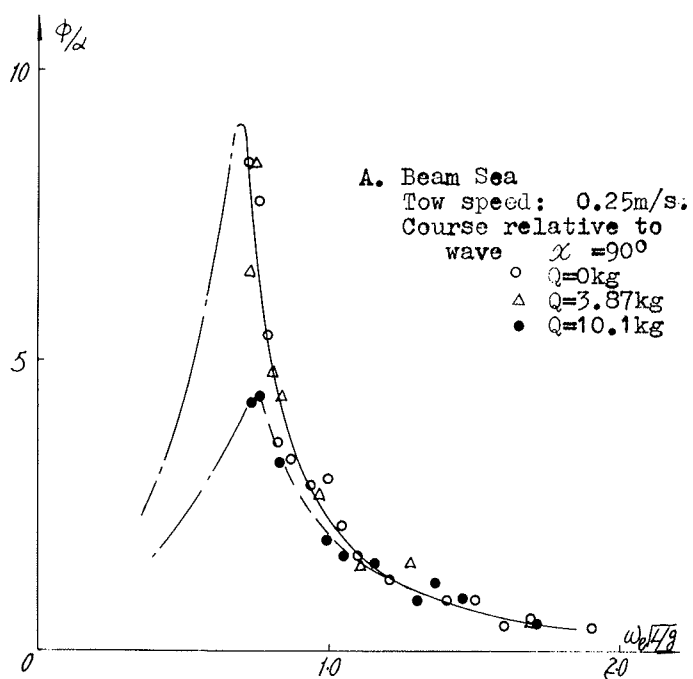


Fig.3.(a) Roll response of the platform model in regular waves



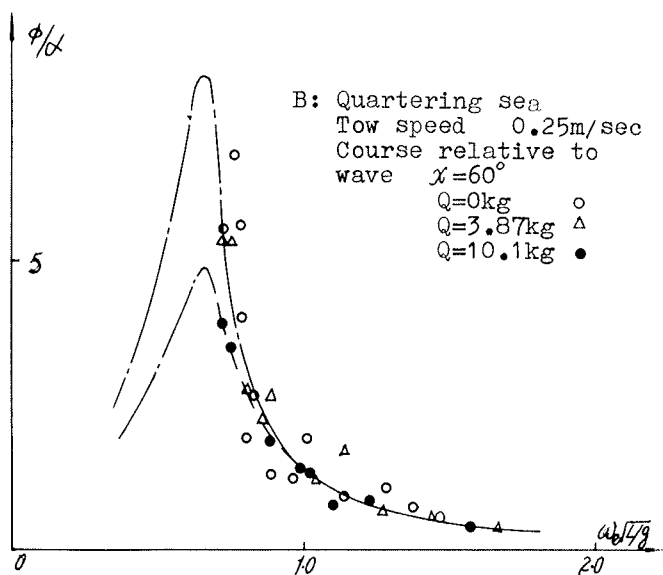


Fig. 3.(b) Roll response of the platform model in regular waves

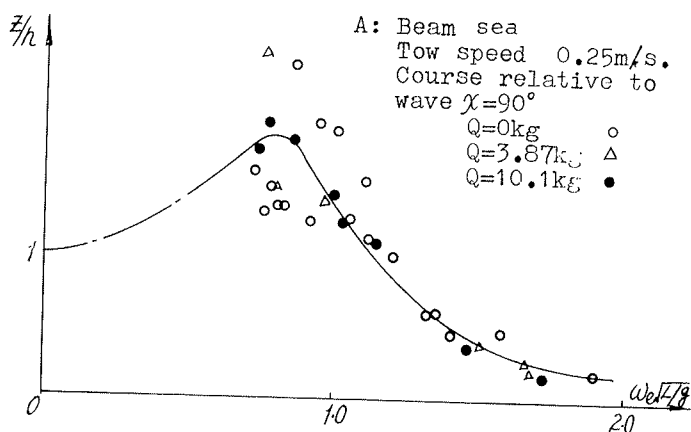


Fig. 5.(a) Heave response of the platform model in regular waves

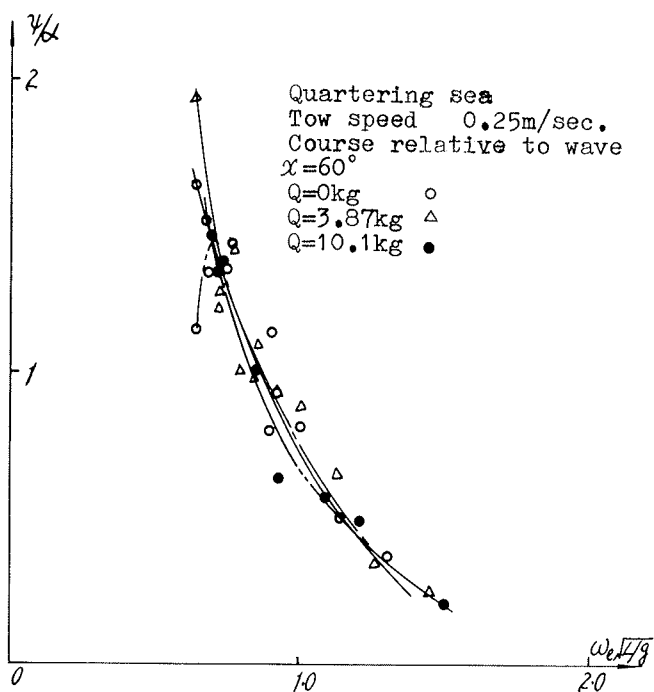


Fig. 4. Pitch Amplitude Response of the Platform model in regular waves

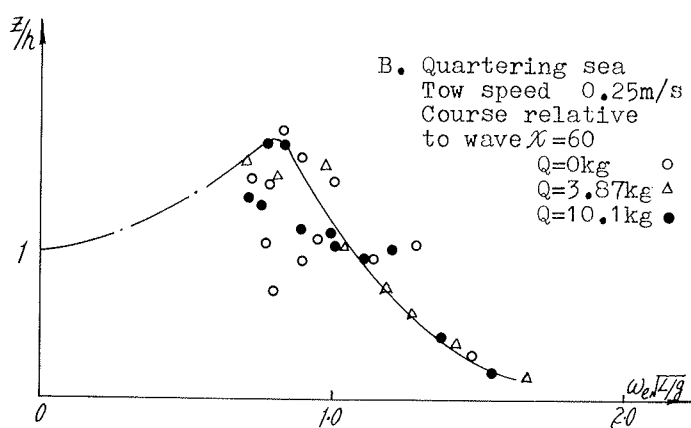


Fig. 5.(b) Heave response of the platform model in regular waves

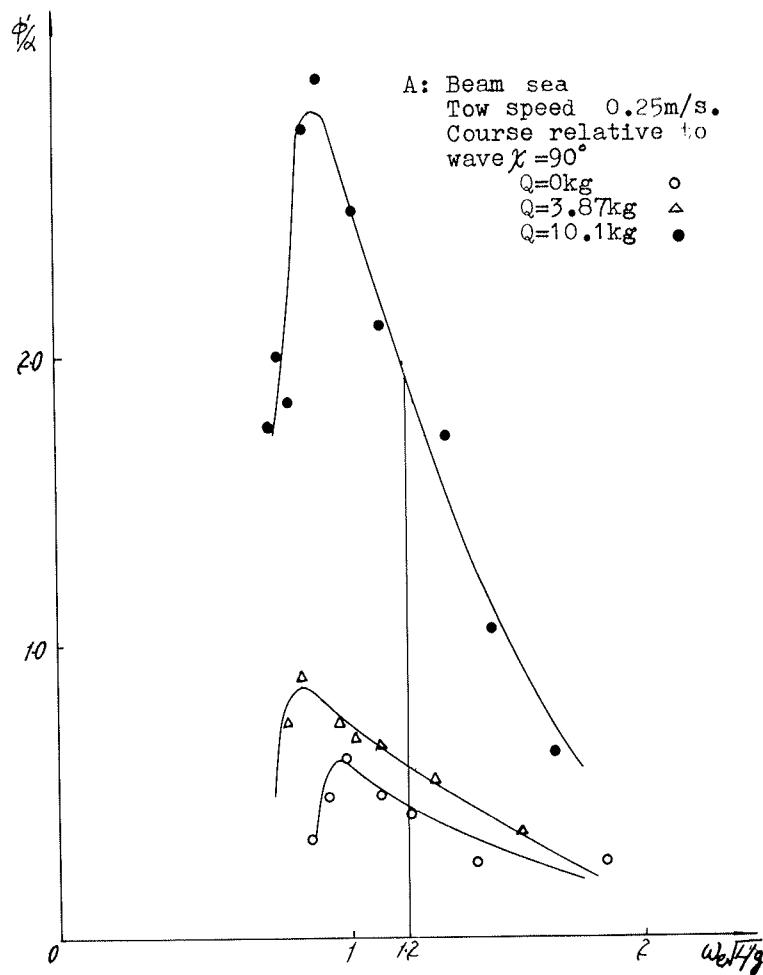


Fig.6.(a) Drift heeling angles of the platform in regular waves

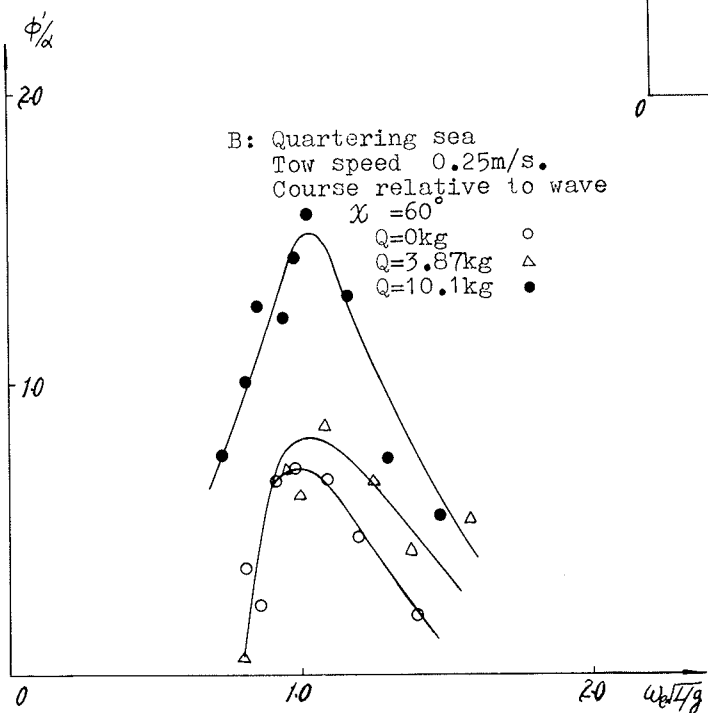


Fig.6.(b) Drift heeling angles of the platform model in regular waves

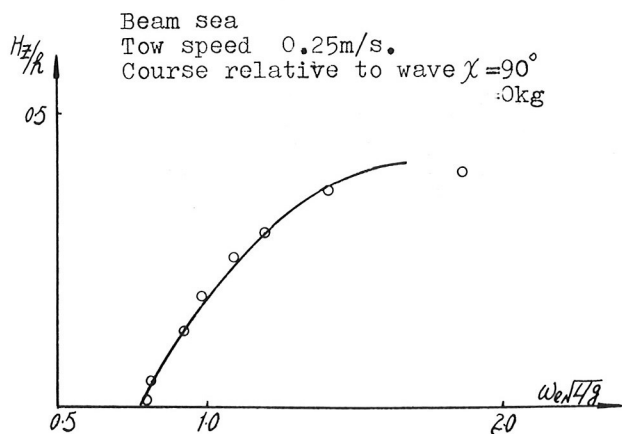
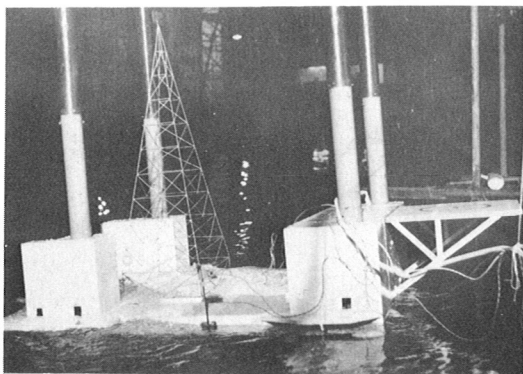
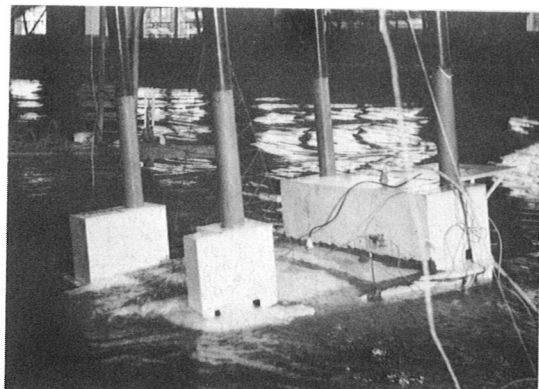


Fig.7. Relative wave height above deck edge on the side incident to wave in regular sea.



A: Water in MPT  $Q=0\text{kg}$  Tow speed 0m/s.  
Course relative to wave  $\chi=90^\circ$   
Wave length  $\lambda=5\text{m}$  Wave height  $h=110\text{mm}$



B: Water in MPT  $Q=3.87\text{kg}$  Tow speed 0m/s.  
Course relative to wave  $\chi=90^\circ$   
Wave length  $\lambda=5\text{m}$  Wave height  $h=110\text{mm}$

Fig.8. Photos showing shipping trapping of water on deck of the platform model in regular waves

|                         |             |             |        |
|-------------------------|-------------|-------------|--------|
| Tow speed               | 0.25m/s.    |             |        |
| Course relative to wave | 90          |             |        |
| Symbols                 | Wave length | Wave height | Q      |
| ○                       | 3.73m       | 128mm       | 0kg    |
| △                       | 4.67m       | 109mm       | 3.87kg |
| ●                       | 3.62m       | 105mm       | 10.1kg |
| ◐                       | 3.97m       | 119mm       | 13.1kg |

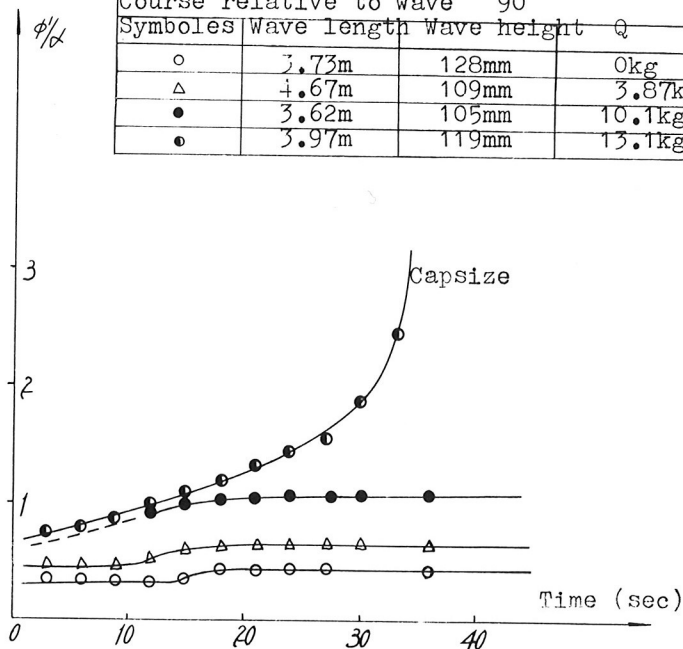


Fig.9. Development of Relative Drift Angles in Heel as Function of Time

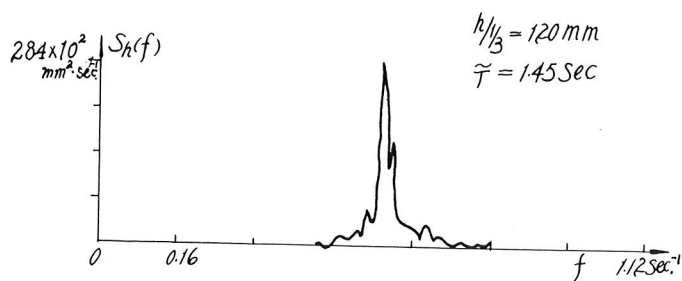


Fig.10. The measured irregular wave spectra in seakeeping basin during capsizing experiments of the platform model

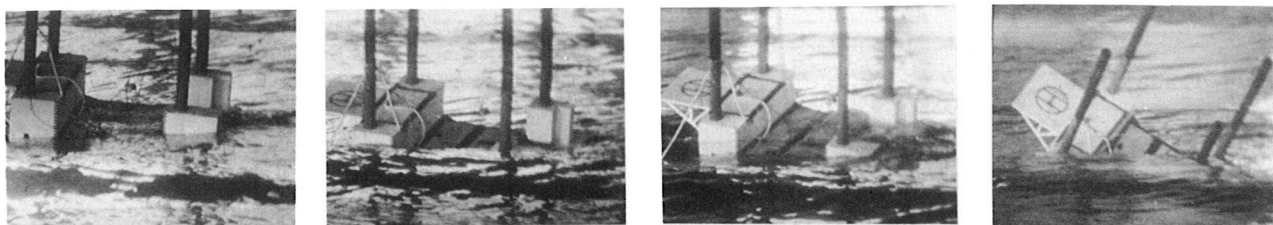


Fig.11. Process of capsizing (wind moment  $M_f=0$ , water in MPT,  $Q=11\text{kg}$ ) in irregular Beam Sea ( $H_1/3=120\text{mm}$ ,  $T=1.40\text{sec}$ ).

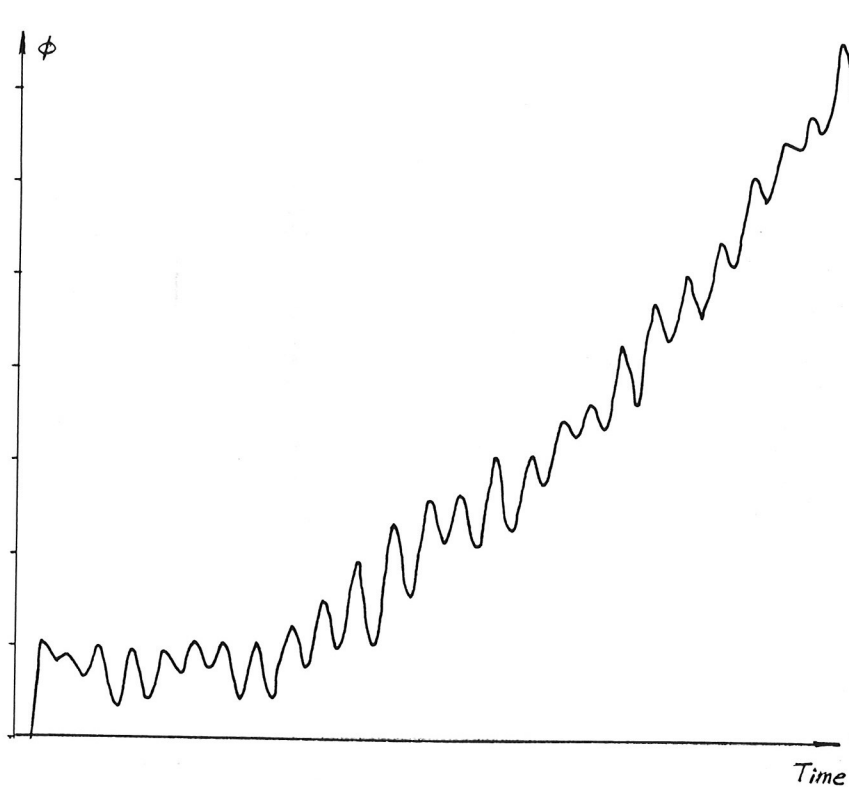


Fig.12. The time of history of capsizing in irregular waves

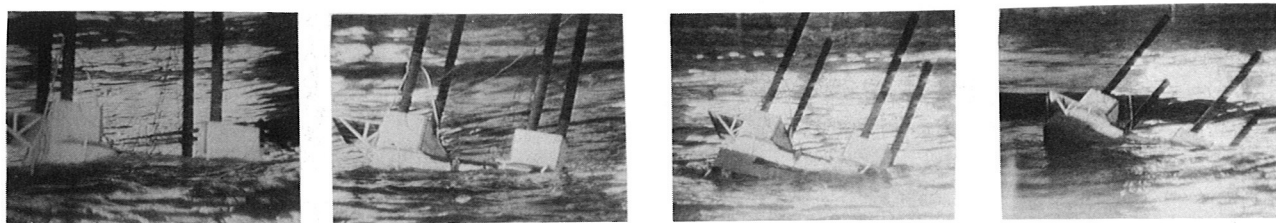


Fig.13. Process of Capsizing (wind moment  $M_f=1.56\text{kg-m}$ , water in MPT,  $Q=4.5\text{kg}$ ) in irregular beam sea ( $H_1/3=120\text{mm}$ ,  $T=1.4\text{sec}$ ).

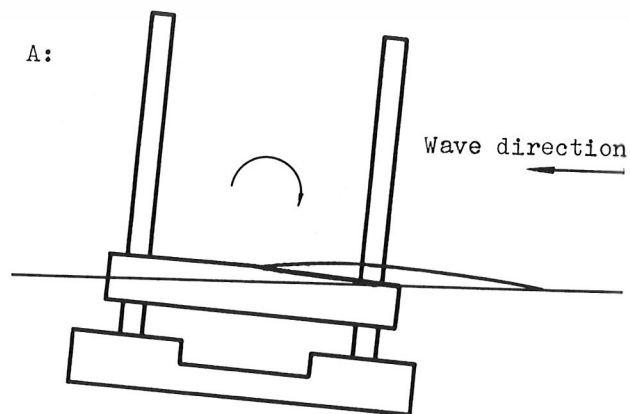
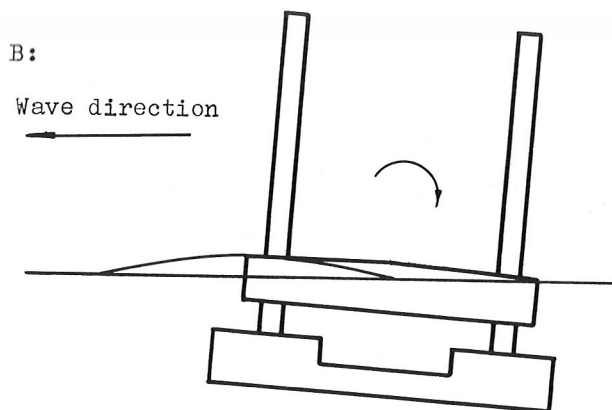


Fig.14. To illustrate the mechanism of non-linear drifting moment when the platform model has initial list forward the waves (wind moment  $M_f$  assumed zero).

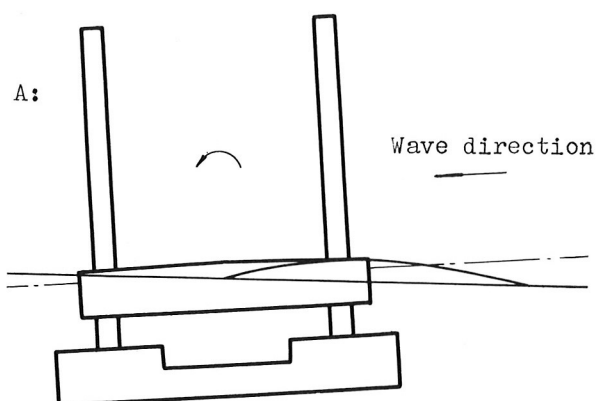
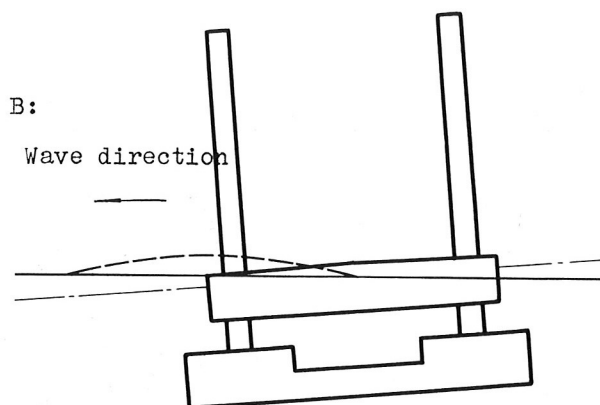


Fig.15. To illustrate the mechanism of non-linear drifting moment, when the platform has an initial list leeward of the wind and waves (wind moment  $M_f$  assumed acting counter clockwise)

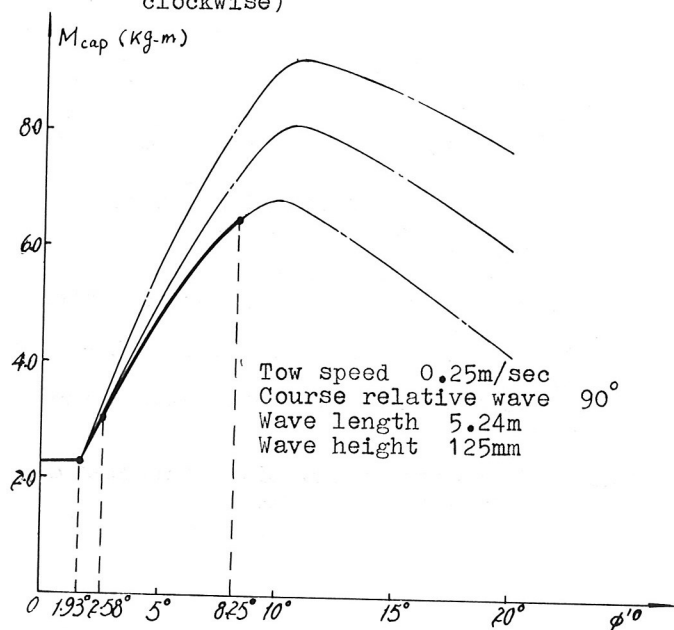


Fig.16. Diagram illustrating the construction of the Drifting moment curve  $M_{cap}(\phi')$  and its relation with  $M_{cap}(\phi)$

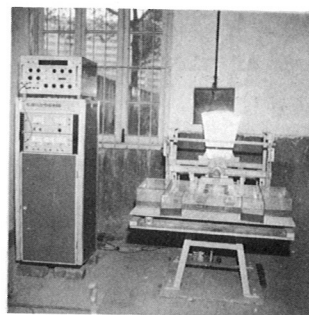


Fig.17.

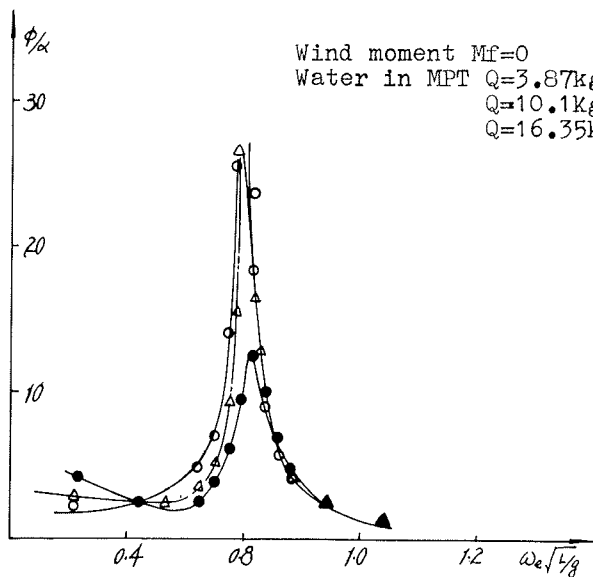


Fig.18. Roll amplitude response of MPT.  
On Bench test.

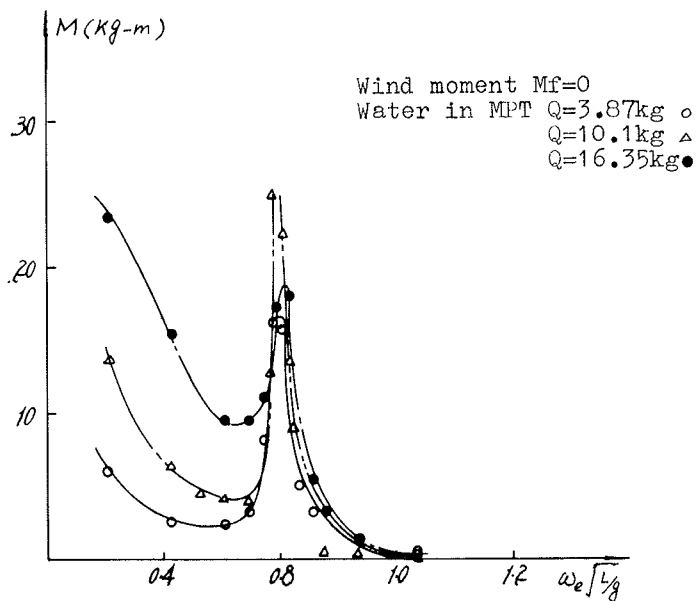


Fig.19. Dynamical moment of MPT during  
Bench tests.

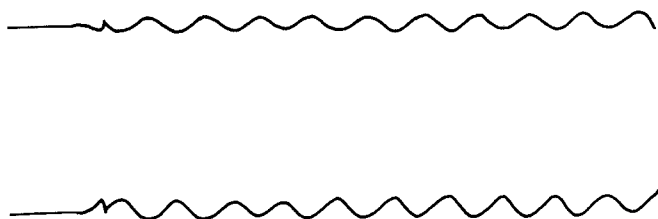


Fig.20. A representative time history of  
force gauge measurement

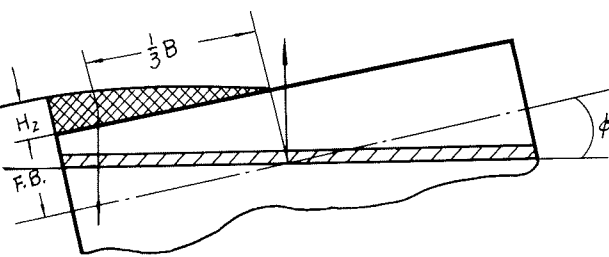
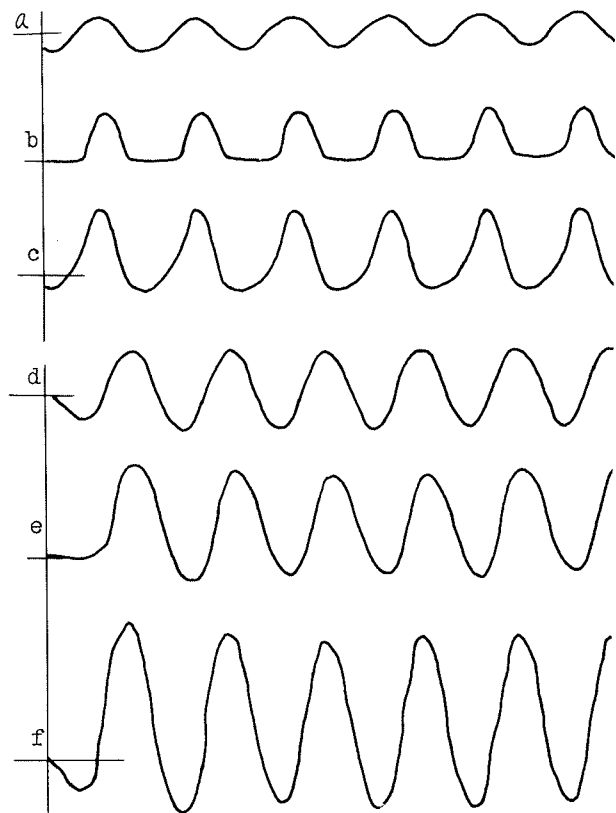


Fig.21. Diagram to illustrate the calculation  
of Drifting moment amplitude  $M1$



- a: Symmetrical wave excitation moment
- b: Asymmetrical excitation moment due to shipping water on deck
- c: Total excitation moment
- d: Roll motion caused by asymmetrical excitation moment
- e: Roll motion caused by asymmetrical excitation moment
- f: Roll motion caused by total excitation moment

Fig.22. Input and output time histories of  
analogue computation

## Discussion

I.A. Manum (Norwegian Maritime Directorate,  
Norway)

Referring to all the losses of units and likes in the offshore oil business I appreciate very much the papers submitted to this session of the conference. It gives hopes for a possible amendment of the IMO MODU-code in a rather near future. As to the safety against capsizing of jackups in our understanding of that expression, I believe that could best be improved by using jack-up carriers for the long transfer mode and only transfer locally when the weather forecast is good. This is not to discourage further studies in order to improve the seaworthiness of jack-up rigs but in order to reduce the probability for capsizing by operational means.

T. Kuragaki (Hitachi Zosen Corp., Japan)

The authors presented an interesting report regarding the capsizing of Jack-up.

My question is that the author's study has any relation with the accident of "Bohai No.2" or not.

Author's Reply

First, I'd like to say that Mr. Manum has put forward a good proposal by using jack-up carrier to reduce the probability of capsizing of jack-up rigs during transfer operation. But I suppose that this transfer mode may be costly and of a lot of technical problems to be solved. In my opinion that to improve the safety against capsizing of jack-up rigs and the other type of platforms by increasing the ability of counter-capsizing of themselves is still an important study aspect.

Second, Mr. Takao Kuragaki asked a question about the relations between my paper and the accident of "Bohai No.2". My reply is that the accident of "Bohai No.2" had been already investigated. My paper is not meant to be a simulation of the actual capsizing condition of "Bohai No.2", but is an extensional research on capsizing mechanism of low freeboard drilling platforms.

*Session X*

## Broaching-to Phenomena

*Chairmen*

Prof. Richard E.D. Bishop  
Brunel University  
U.K.

Prof. Kensaku Nomoto  
Osaka University  
Japan



## ON THE MECHANISM OF BROACHING-TO PHENOMENA

SEIZO MOTORA\*, MASATAKA FUJINO\*\* AND TAKESHI FUWA\*\*\*

\*Nagasaki Institute of Applied Science, \*\*University of Tokyo,

\*\*\*Ship Research Institute

Japan

### ABSTRACT

In order to identify the causal factors resulting in the occurrence of broaching-to phenomena, free running model tests, captive model tests, and full-scale measurements were conducted. Furthermore, numerical simulations of surge, sway, yaw, and roll motions of a ship traveling in following seas were performed by making use of the results of captive model tests. From the free running model tests, it was clarified that broaching-to is likely to occur under the conditions of  $\lambda/L \div 2$ ,  $\chi = 20^\circ \sim 30^\circ$  and  $U \cos \chi \div U_w$ . The propriety of the prevailing opinions with regard to the primary cause of broaching-to was examined on the basis of hydrodynamic data obtained through captive model tests. Based on the investigation results as well as numerical simulations, it was induced that the wave exciting yaw moment which exceeds the course-keeping-ability of the rudder results in broaching-to phenomena. This proposition was verified not only by investigating the dynamic balance of force and moment during broaching-to observed with the free running model, but also by comparing the wave exciting forces acting on a full-scale ship, which were estimated from the ship's measured motions, with the results of the captive model tests. Furthermore, the effects of the heel on the development of sway and yaw motions, and the growth of the heel during broaching-to were discussed.

### 1. INTRODUCTION

A small high-speed craft traveling in the following seas occasionally loses control over the rudder and is forced to turn abruptly against the helm. Rapid develop-

ment of forced turning, which is known as broaching-to, is frequently accompanied by an extraordinary heel of the hull. Hence, broaching-to phenomena have been feared by the seamen. Based on experience, it is known that broaching-to is apt to occur when a craft travels in a following wave, of which the wave length is about twice the length of the craft, at a speed less than or equal to the wave celerity[1,2]. However, the causes of the occurrence of broaching-to are not definitely known. Several different causes are now advocated with regard to the occurrence of broaching-to.

- (a) Loss of directional stability [3,4].
- (b) Decrease in rudder force[4].
- (c) Wave exciting yaw moment; the wave exciting yaw moment induced by the cross flow component of the orbital velocity of wave particles is especially emphasized.

Until the present, however, none of these causes have been verified sufficiently and the mechanism of occurrence of broaching-to also has not been clarified in a definite manner.

First of all, the authors conducted various kinds of captive model tests in order to examine the propriety of the notions (a)~(c). Based on the results, it is concluded that any particular one is not a primary cause of broaching-to. Then, the authors carried out numerical simulations on the lateral motions of a ship traveling in a following sea by making use of hydrodynamic data obtained through captive model tests. The primary aim of the simulation is to reveal the characteristics of sway and yaw motions in a following wave, particularly the effects of ship speed, ship position relative to the wave, encounter

angle, etc. on the development of lateral motions. Based on the results of numerical simulation, the mechanism of broaching-to is discussed.

In addition to captive model tests, free-running model tests were carried out with a large radio-controlled model having a similar hull form to investigate how the occurrence of broaching-to is affected by wave conditions such as wave height, wave length, etc., and the running conditions such as ship speed, ship's initial course, etc.

Based on the results of the experiments, it is found that broaching-to phenomena are likely to occur during a particular combination of wave and running conditions.

The occurrence of broaching-to during the free running model tests was judged by observing the behavior of the model and by inspecting the measurements on ship motion.

Together with the model tests, full-scale measurements were performed with a leisure craft of 7.14m in length, having a typical hull form of that of a Japanese fishing boat. During the full-scale measurements, broaching-to phenomena were observed several times when the craft traveled in a following wave, of which the height and the length were 1.0~1.5m and 10~15m respectively, at a speed of approximately 8.5 knots. A comparison of the full-scale measurements with the model tests shows that both records of ship motion during broaching-to correspond well to each other in terms of pattern and agree well in terms of quantity.

Based on the results of the numerical simulations mentioned previously, it is assumed that broaching-to is the result of the wave exciting yaw moment when it is so enormous that it cannot be overcome even by the hard-over helm. The validity of the proposition was examined not only by estimating the value of wave exciting yaw moment acting on the full-scale ship during broaching-to, but also by examining the balance of force and moment acting on the free running model.

In addition, the effects of the heel on the development of sway and yaw motions during broaching-to were investigated by numerical simulations based on hydrodynamic data obtained from additional captive model tests. Based on the results, it was discovered that for the particular ship used in this experiment, the heel of a moderate angle did not completely affect the development of sway and yaw motions during broaching-to. However, it was shown that the heel of a ship during broaching-to was stimulated by hydrodynamic heel moment induced by hydrodynamic lateral force acting on the hull rather than by wave exciting roll moment.

## 2. NOMENCLATURE

d: draught of ship  
 $F_n$ : Froude number

$\overline{GZ}(\phi)$ : righting arm  
 $H_w$ : wave height  
 $I_{xx}, I_{zz}$ : mass moment of inertia of ship about x and z axes, respectively  
 $J_{xx}, J_{zz}$ : added mass moment of inertia about x and z axes, respectively  
 $L$ : ship length  
 $L_E$ : wave exciting roll moment  
 $\ell$ : horizontal distance between the centre of gravity of a ship and the crest line of wave directly in front of the ship  
 $(\ell/\lambda)_0$ : initial value of  $\ell/\lambda$   
 $m$ : mass of ship  
 $m_x, m_y$ : added mass in x and y directions, respectively  
 $N$ : hydrodynamic yaw moment  
 $N_{\dot{\beta}}$ : hydrodynamic derivative with respect to the rate of drift angle  
 $N_E$ : wave exciting yaw moment  
 $N_{\dot{\phi}}(\phi)$ : roll dumping moment  
 $r$ : yaw angular velocity  
 $u$ : surge velocity  
 $U$ : resultant speed of ship  
 $U_0$ : initial value of  $U$   
 $U_w$ : wave celerity  
 $v$ : sway velocity  
 $W$ : displacement of a ship  
 $X$ : hydrodynamic force in x direction  
 $X_E$ : wave exciting surge force  
 $X_u$ : hydrodynamic derivative with respect to surge velocity  
 $Y$ : hydrodynamic sway force  
 $Y_E$ : wave exciting sway force  
 $Y_H$ : hydrodynamic sway force acting on the hull due to sway and yaw  
 $Y_v, N_v$ : hydrodynamic derivatives with respect to sway velocity  
 $Y_{\dot{\beta}}, N_{\dot{\beta}}$ : hydrodynamic derivatives with respect to drift angle  
 $Y_{\dot{\chi}}, N_{\dot{\chi}}$ : derivatives of wave exciting sway force and yaw moment with respect to encounter angle  
 $Y_r, N_r$ : hydrodynamic derivatives with respect to yaw angular velocity  
 $Y_{\ddot{r}}$ : hydrodynamic derivative with respect to yaw angular acceleration  
 $X_R(\delta), Y_R(\delta), N_R(\delta)$ : hydrodynamic forces in x and y directions, and moment about z axis respectively, caused by steering the rudder  
 $z_H$ : vertical position of application of  $Y_H$   
 $\beta$ : drift angle of ship  
 $\delta$ : rudder angle  
 $\lambda$ : wave length  
 $\xi$ :  $2\pi\ell/\lambda$   
 $\phi$ : heel angle of ship  
 $\chi$ : encounter angle to waves  
 $\chi_0$ : initial value of  $\chi$

non-dimensional values are denoted by the superscript (') as follows.

$m', m_x', m_y'$ :  $m, m_x, m_y / (0.5\rho L^2 d)$

$I_{zz}', J_{zz}': I_{zz}, J_{zz} / (0.5\rho L^4 d)$   
 $Y': Y / (0.5\rho L d U^2)$   
 $N': N / (0.5\rho L^2 d U^2)$   
 $r', \dot{r}': rL/U$  and  $r(L/U)^2$ , respectively  
 $\beta', \dot{\beta}': \beta$  and  $\dot{\beta}L/U$ , respectively  
 $\chi', \dot{\chi}': \chi$  and  $\dot{\chi}L/U$ , respectively  
 $\rho$ : density of fluid

. over a symbol denotes differentiation  
 regarding time.

### 3. CONDITIONS OF THE OCCURRENCE OF BROACHING-TO

Prior to discussing the mechanism of  
 occurrence of broaching-to phenomena, it  
 is necessary to clarify the conditions  
 under which broaching-to is likely to occur.  
 For this purpose, free running model tests  
 were carried out with a radio controlled  
 model of 2.75m in length[6,7].

#### 3.1 Ship Model

The principal particulars of the model  
 and its body plan are shown in Table 1 and  
 Fig.1, respectively. At first, it was  
 difficult to realize broaching-to even  
 though such parameters as wave height, wave  
 length, model speed, and encounter angle  
 were adjusted in varying manners. In fact,  
 broaching-to almost never occurred during  
 the full-scale measurements which were  
 made together with the free running model  
 tests. Therefore, the size of rudder was  
 shortened into 3/4 of its original size  
 both in terms of height and width. Namely,  
 the rudder area decreased to approximately  
 56% of its original area as shown in Fig.2.  
 For the propulsion of the model, a gasoline  
 driven engine (37cc, 1.75PS) was internally  
 installed.

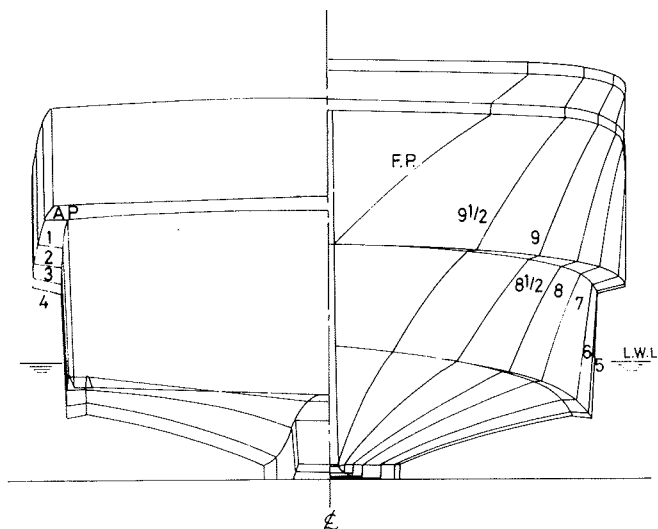


Fig. 1 Body Plan of the Ship Used for Experiments

Table 1 Principal Particulars of Full-scale Ship and Models

| SHIP ITEM             | FULL SCALE SHIP | FREE RUNNING MODEL | CAPTIVE MODEL |
|-----------------------|-----------------|--------------------|---------------|
| Length L              | 7.26 m          | 2.750 m            | 1.017 m       |
| Breadth B             | 1.87 m          | 0.708 m            | 0.262 m       |
| Draught $d_f$         | 0.39 m          | 0.121 m            | 0.0364 m      |
| $d_a$                 | 0.52 m          | 0.197 m            | 0.0710 m      |
| Displacement          | 2.786 ton       | 147.3 kg           | 6.629 kg      |
| C.G. from keel KG     | 0.847 m         | 0.341 m            | 0.0830 m      |
| Metacentric height KM | 1.382 m         | 0.523 m            | 0.204 m       |
| Metacentric radius GM | 0.535 m         | 0.192 m            | 0.121 m       |
| C.G. from midship G   | 0.453 m aft     | 0.172 m aft        | 0.0842 m aft  |
| Longi. gyradius $k_y$ |                 | 0.805 m            | 0.252 m       |
| Trans. gyradius $k_x$ |                 | 0.285 m            |               |
| Rolling period $T_r$  |                 | 1.50 sec           |               |
| PROPELLER             |                 |                    |               |
| No. of blades         | 3               | 4                  | 3             |
| Diameter              | 540.0 mm        | 166.7 mm           | 75.6 mm       |
| Pitch                 | 580.0 mm        | 173.0 mm           | 81.2 mm       |
| E.A.R.                | 0.45            | 0.565              | 0.45          |

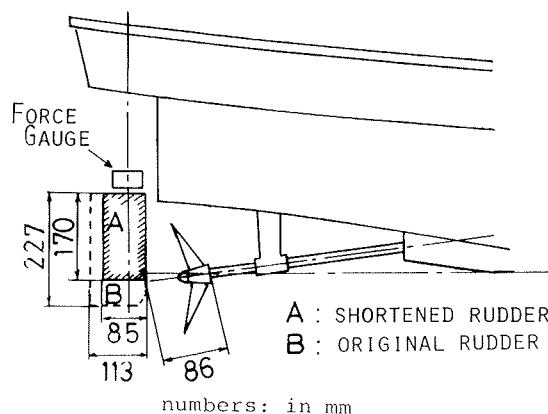


Fig. 2 Stern Profile of Free Running Model

#### 3.2 Experiments

Course-keeping-tests in two dimensional  
 regular waves were performed to investigate  
 into the conditions necessary for broaching-  
 to to occur.

In order to maintain the desired head-  
 ing of the model, every possible effort for  
 steering was made by radio controlled  
 manual steering. When broaching-to occurs,  
 the heading of model is forced to turn in a  
 direction parallel to the crest line of the  
 waves.

During the experiments, various changes  
 were made in wave conditions such as wave  
 height and wave length, and in running con-  
 ditions such as encounter angle to waves  
 and ship speed.

The travelling speed of the model was  
 adjusted by the engine throttle so as to  
 maintain the required nominal speed in calm  
 water. During one run, the engine throttle  
 position was fixed so that the speed of the  
 model in the waves was below nominal speed  
 in calm water.

Experimental conditions were as follows.

- (1) wave conditions
  - wave length to ship length ratio  $\lambda/L = 1.25 \sim 3.0$
  - wave height to wave length ratio  $H_w/\lambda = 1/50 \sim 1/15$
- (2) running conditions
  - encounter angle to waves  $\chi = 0^\circ \sim 45^\circ$  (see Fig.6)
  - nominal ship speed in calm water  $F_n = 0.4 \sim 0.7$

Fig. 3 shows a typical example of broaching-to phenomena observed during free running model tests.

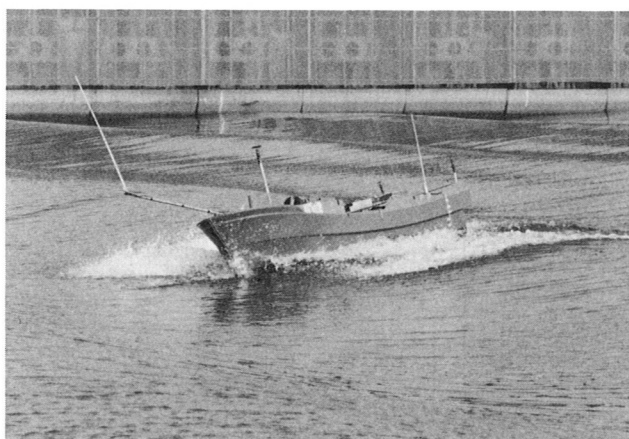


Fig. 3 Broaching-to Observed at Free Running Model Tests

### 3.3 Results of Experiments

It is known that various conditions are clearly related to the occurrence of broaching-to. They are (a) hull form and loading condition, (b) wave condition, and (c) running condition. During the present free running model tests, the effects of the latter two conditions were extensively investigated.

The results of the course-keeping-tests with the free running model are summarized in Table 2. As shown in this table, parameters which varied during the experiments are wave length to ship length ratio  $\lambda/L$ , wave steepness  $H_w/\lambda$ , encounter angle  $\chi$ , and ship speed in terms of the Froude number in calm water. The following facts are seen in the table.

- (1) Broaching-to is likely to occur when there is a particular combination of parameters. The condition of  $\lambda/L = 2.0$  and  $\chi = 22.5^\circ$  represent the center of the broaching-to region, in other words, the condition under which broaching-to occurs most frequently, as shown in Table 2.
- (2) When broaching-to has occurred, the ship model runs on the down slope of the wave, and it is under surf-riding condition.
- (3) In the case where  $\chi$  equals zero, broaching was not observed at all.

(4) As far as the present results are concerned, the lower limit of wave steepness necessary for broaching-to to occur is estimated as  $H_w/\lambda = 1/25$ .

In addition to these conclusions, there is one thing which should be pointed out. Even in similar test conditions, different results were occasionally observed. In other words, certain other factors in addition to the parameter mentioned previously influence the occurrence of broaching-to. They might be the acceleration timing of the model to a required speed, the manner of steering the model to make it enter the prescribed course and to hold the course during the initial stage of the run, and so on. On the whole, however, similar results were obtained under similar conditions.

Incidentally, based on past experience of numerous full scale measurements, it is known that minor changes in hull form drastically affect the probability of broaching-to. Therefore, the conditions obtained by the present investigation are not sufficient conditions but only the necessary conditions for the occurrence of broaching-to.

Table 2 List of the Occurrence of Broaching-to

| $\lambda/L$ | $H_w/\lambda$ | $F_n$ | $\chi = 45^\circ$ |     |     |     | $\chi = 30^\circ$ |     |     |     | $\chi = 22.5^\circ$ |     |     |     | $\chi = 11.25^\circ$ |     |     |     | $\chi = 0^\circ$ |     |     |     |
|-------------|---------------|-------|-------------------|-----|-----|-----|-------------------|-----|-----|-----|---------------------|-----|-----|-----|----------------------|-----|-----|-----|------------------|-----|-----|-----|
|             |               |       | 0.4               | 0.5 | 0.6 | 0.7 | 0.4               | 0.5 | 0.6 | 0.7 | 0.4                 | 0.5 | 0.6 | 0.7 | 0.4                  | 0.5 | 0.6 | 0.7 | 0.4              | 0.5 | 0.6 | 0.7 |
| 3.0         | 1/25          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |
| 2.5         | 1/20          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |
| 2.0         | 1/25          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |
|             | 1/20          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |
|             | 1/15          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |
| 1.5         | 1/20          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |
| 1.25        | 1/20          |       |                   |     |     |     |                   |     |     |     |                     |     |     |     |                      |     |     |     |                  |     |     |     |

Marks: ● Broaching ○ Nearly-Broaching ○ Non-Broaching

### 3.4 Comparison with the Results of Full-scale Measurements

Using a ship of 7.14m in length (see Table 1), full-scale experiments were performed off Imagiri which is located on an estuary of Lake Hamana, and some broaching-to phenomena were observed. During free running model tests, the occurrence of broaching-to was determined by visually observing the behavior of the model and by inspecting the recorded time histories of ship motion. During the full-scale tests, on the other hand, broaching-to was deter-

mined by a pilot driver on board, who could feel the behavior of the ship by himself and judge the ship's response to the helm. When the heading angle of the ship cannot be checked by the hard-over helm and when drastic development of turn and heel is found, the driver determines that broaching-to has occurred. Therefore, it is necessary to examine whether the occurrence of broaching-to is similarly determined or not for the free running model tests and for full-scale measurement.

In Fig. 4, the time histories of rudder angle and ship motion recorded during broaching-to are shown. According to observations made during full-scale tests, it is confirmed that when broaching-to has been determined to have occurred, the ship runs under a surf-riding condition on the down slope of a wave and shows extraordinary turn and heel. Furthermore, it is discovered that the time histories of ship motion recorded during both model and full-scale tests are quite similar to each other as seen in Fig.4(a) and (b). Furthermore, the peak values of motion of the full-scale ship during broaching-to correspond well to those of the free running model in quantity.

Therefore, it is concluded that judgment of broaching-to during the free running model tests is reasonable and that the conditions required for the occurrence of broaching-to which are induced from the results of model tests, are of practical value.

#### 4. MECHANISM OF OCCURRENCE OF BROACHING-TO

The necessary conditions for broaching-to phenomena to occur have been revealed to a great extent by free running model tests and full-scale measurements. In the following, the mechanism of occurrence of broaching-to will be considered.

##### 4.1 Review of Prevailing Opinions

As mentioned at the beginning of this paper, there exist various opinions with regard to the cause of broaching-to phenomena. Captive model tests were conducted to obtain hydrodynamic data which would be used to examine the propriety of those opinions. The model used during the captive tests is smaller than that for the free running tests, and its principal particulars are shown in Table 1. It would be desirable for the captive tests to be carried out under conditions in which broaching-to frequently occurs, for example, under the conditions of  $\lambda/L = 2.0$  and  $U/\cos\chi \div U_w$ . Nevertheless, most of the captive tests were performed under the conditions of  $\lambda/L = 1.6$  and  $U/U_w = 0.9$

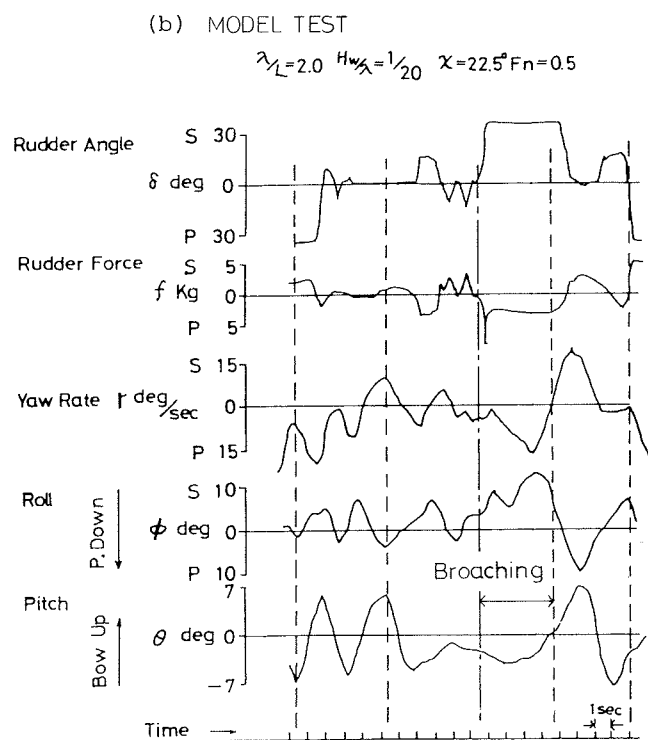
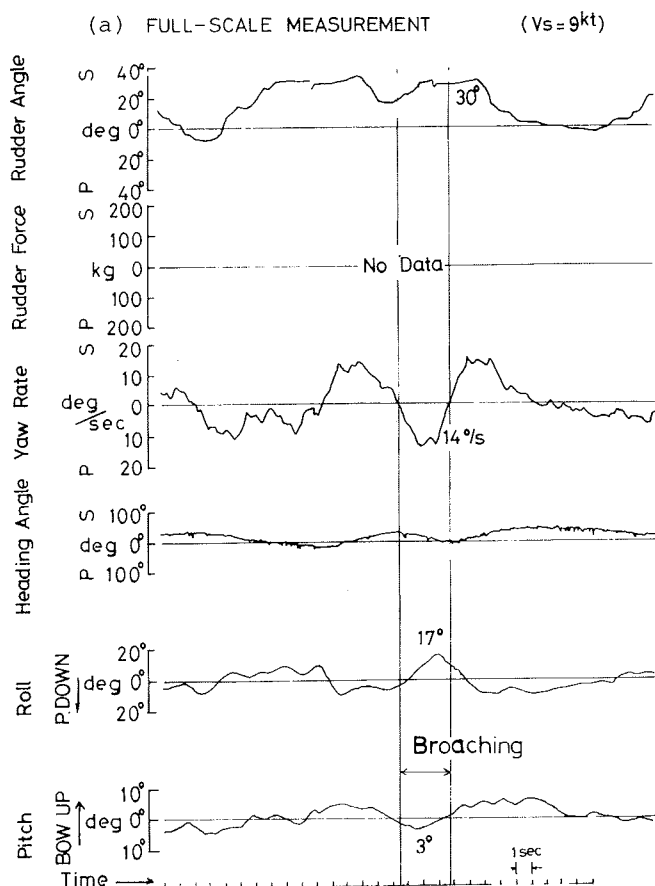


Fig. 4 Time Histories of Ship Motions During Both Model and Full-scale Tests

primarily due to the restrictions on the attainable speed of the towing carriage.

#### (1) Decrease in Rudder Forces

As one of the plausible causes of broaching-to, the drastic decrease in ruddering force which a ship suffers when running on the down slope of waves is advocated. In order to ascertain this fact for the case of the model used in the present investigation, the hydrodynamic moment about the C.G. of the model which is induced by steering the rudder was measured at various encounter angles. Here, it should be noted that the rudder used during the experiments was of original size, and was not shortened. The test results are shown in Fig. 5, where the abscissa denotes the non-dimensionalized ship position relative to the crest line of the wave (see Fig. 6), say  $l/\lambda$ . The  $l/\lambda$  value between  $4/8$  and  $8/8$  indicates that the C.G. of a ship is situated on the down slope of the wave. The ordinate denotes the hydrodynamic yaw moment generated by 30-degree helm. From this figure, it is revealed that in the case of starboard rudder there exists a tendency for the hydrodynamic yaw moment to change according to the ship's relative position to the wave, but for the port rudder such a tendency does not clearly exist. On the whole, however, the hydrodynamic yaw moment due to the helm does not change significantly according to the ship's position on the wave. On the contrary, Renilson, et al. reported that when a ship is located on the down slope of a wave, the hydrodynamic derivatives of rudder become less than half of those measured when the ship is on the up slope of a wave[4]. This difference in rudder effectiveness between Renilson's data and the present experimental results is assumed to be primarily the result of the variation in the hull forms of the ships including rudder configuration.

The ship model which was used by Renilson, et al. has a hanging rudder, and its immersion depth in water is shallow in comparison to that of the model used during the present investigation. In fact, Renilson, et al. reported that in their experiments, the rudder sometimes drew the air particularly when the model was situated on the down slope of a wave. Incidentally, the rudder's normal force was measured not only during the free running model tests but also during the full-scale tests. As a result, it is confirmed that rudder effectiveness of the present ship hardly changes according to the ship's position on the wave. Nevertheless, typical broaching-to phenomena were observed many times during the free running model tests and full-scale measurements. Hence, it is concluded that a decrease in ruddering force is not the primary cause of broaching-to.

#### (2) Loss of Directional Stability

There exists an opinion that the loss of directional stability on the down slope of a wave results in broaching-to.

In this section, the directional stability of the ship traveling on a straight course at encounter angle  $\chi$  is examined. Assuming that the ship travels on the wave surface without any sway or yaw, small perturbations of sway and yaw are described by the following linear equations:

$$\begin{aligned} -(m' + m_y') \dot{\beta}' - Y_{\dot{\beta}}' \dot{\beta}' - Y_{\beta}' \beta' - \\ (Y_{\dot{r}}' - m') \dot{r}' - Y_{\chi}' \chi' &= 0 \\ (I_{zz}' + J_{zz}') \dot{\chi}' - N_{\dot{\beta}}' \dot{\beta}' - N_{\beta}' \beta' - \\ N_{\dot{r}}' \dot{r}' - N_{\chi}' \chi' &= 0 \end{aligned} \quad (1)$$

where the symbol of dash, say, (') denotes non-dimensional quantities. In the following analysis, the coupling terms  $Y_{\dot{\beta}}' \dot{\beta}'$  and  $N_{\dot{\beta}}' \dot{\beta}'$  will be neglected. Taking account of the relationship

$$\dot{\chi}' = r' \quad (2)$$

the characteristic equation corresponding to eq.(1) is as follows:

$$As^3 + Bs^2 + Cs + D = 0 \quad (3)$$

where the coefficients A,B,C and D are

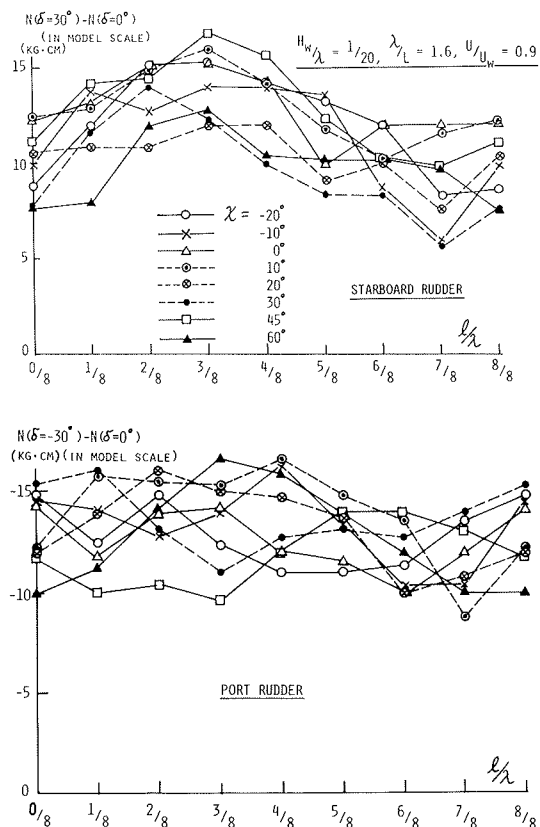


Fig. 5 Rudder Effectiveness in Following Seas ( $H_w/\lambda=1/20$ ,  $\lambda/L=1.6$ ,  $U/U_w=0.9$ )

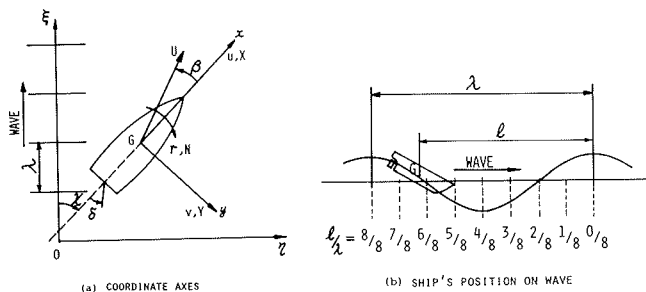


Fig. 6 Coordinate System

$$\begin{aligned} A &= (m' + m_y')(I_{zz}' + J_{zz}'), \\ B &= -(m' + m_y')N_r' + (I_{zz}' + J_{zz}')Y_\beta', \\ C &= -(m' + m_y')N_\chi' - N_r'Y_\beta' + \\ &\quad (Y_r' - m')N_\beta', \\ D &= -Y_\beta'N_\chi' + Y_\chi'N_\beta'. \end{aligned} \quad (4)$$

As is well known, necessary and sufficient conditions for directional stability are obtained by the Routh-Hurwitz criteria as follows.

$$\begin{aligned} B/A > 0, C/A > 0, D/A > 0, \\ BC-AD > 0 \end{aligned} \quad (5)$$

For the purpose of examining directional stability, the hydrodynamic derivatives should be known as functions of encounter angle  $\chi$  and ship position on the wave  $\ell/\lambda$ . For this purpose, the static derivatives  $Y_\beta'$ ,  $N_\beta'$  and the derivatives  $Y_\chi'$ ,  $N_\chi'$  were measured through captive model tests. The results of the oblique towing tests in the waves are shown in Fig.7. However, the acceleration derivatives  $m_y'$  and  $J_{zz}'$  were assumed to be constant and estimated from Motora's chart[5]. Furthermore, the rotary derivatives  $Y_r'$ ,  $N_r'$  were assumed to be constant and were replaced by their values in calm water. They are as follows:

$$\begin{aligned} m' + m_y' &= 0.491, I_{zz}' + J_{zz}' = 0.0260 \\ Y_r' - m' &= -0.312, N_r' = -0.0763 \end{aligned}$$

Instead of examining the Routh-Hurwitz criteria, the characteristic equation (3) was solved to obtain the eigen values of the equation. The maximum value of the real part of the solution,  $\sigma_{\max}$ , is shown in Fig. 8. When it is positive, the ship's course is unstable and vice versa. According to the figure, it is evident that the ship will always be unstable when traveling on the down slope of waves, but is stable on the up slope of waves. The contour map in Fig.8 is not strictly correct, because the acceleration and rotary derivatives are assumed to be independent of  $\chi$  and  $\ell/\lambda$ . Nevertheless, it is not expected from Fig.8 that the degree of instability on the down slope of a wave significantly depends on the encounter angle  $\chi$ , since one of the main criteria to determine course stability is  $D/A > 0$ .

On the contrary, broaching-to phenomena were likely to occur at an encounter angle of approximately 20 to 30 degree during both free running model tests and full-scale measurements. Therefore, it is not concluded that the inherent instability of a ship traveling on the down slope of wave is the primary cause of broaching-to.

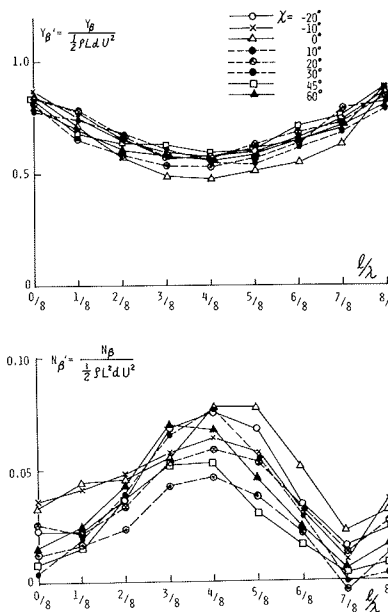


Fig. 7 Results of Oblique Towing Tests ( $H_w/\lambda=1/20$ ,  $\lambda/L=1.6$ ,  $U/U_w=0.9$ )

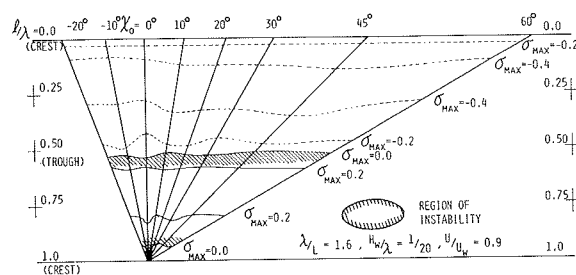


Fig. 8 Contour Map of  $\sigma_{\max}$  Values

### (3) Yaw Moment due to the Cross Flow Component of the Orbital Motion of Wave Particles

When a ship travels on the down slope of a wave at a non-zero encounter angle, the cross flow drag which acts on the bow and the stern of the ship in the opposite direction makes a couple to turn the ship toward the crest line of the wave. This is another prevailing explanation of the occurrence of broaching-to phenomena. In order to examine the effects of the turning moment caused by cross flow drag, several two-dimensional cylinders of which the transverse sections are similar to those of the prototype ship, were made and towed in



calm water in a direction normal to the axis of the cylinder in order to measure the drag force. From the experimental results, yaw moment resulting from the cross flow drag was estimated for the three-dimensional captive model. When comparing them with the estimated values of yaw moment based on Froude-Krylov force, it was found that sway force and yaw moment resulting from cross flow drag are negligibly small.

When summarizing the results of the examination described above, it can be said that none of three plausible causes (a)~(c) is a primary factor in the occurrence of broaching-to.

#### 4.2 Numerical Simulation of Lateral Motions in Waves

Taking account of the results obtained in the previous section, it was decided that the numerical simulation of lateral motions of a ship traveling in the following seas would be performed in order to understand the characteristics of ship behavior under such a condition and then to uncover factors in considering the mechanism of broaching-to phenomena[8].

Most numerical simulations are carried out without taking the heel of a ship into consideration. However, the heel of a ship should be considered in the following two aspects. One is that the heel of the ship's hull changes the numerical values of the hydrodynamic derivatives, particularly the static derivative  $N_{\beta}'$  and the rotary derivative  $N_r'$ [10] as well as the wave exciting force. The other is the development of heel angle during broaching-to. The effects of heel will be briefly examined at the end of this section. Hence, for the time being, surge, sway, and yaw motions are taken into consideration, and are described as follows:

$$\begin{aligned} m(\dot{u} - rv) &= X + X_E \\ m(\dot{v} + Ur) &= Y + Y_E \\ I_{zz}\dot{r} &= N + N_E \end{aligned} \quad (6)$$

The first terms on the right-hand side of the equations represent the hydrodynamic forces resulting from ship motion and steering of rudder, and the second terms the wave exciting forces which are functions of  $\chi$  and  $\ell/\lambda$ . By expanding  $X, Y$ , and  $N$  in terms of the perturbations  $u, v, r, \dot{u}, \dot{v}, \dot{r}$ , and  $\delta$ , and then by only retaining linear terms, the following equations of motion are obtained.

$$\begin{aligned} (m + m_X)\dot{u} &= X_{u0} + X_E(\chi, \ell/\lambda) \\ (m + m_Y)\dot{v} &= Y_{v0} + (Y_r - mU)r + Y_{\delta}\delta + Y_E(\chi, \ell/\lambda) \\ (I_{zz} + J_{zz})\dot{r} &= N_{v0} + N_r r + N_{\delta}\delta + N_E(\chi, \ell/\lambda) \end{aligned} \quad (7)$$

where the derivatives  $-X_{\dot{u}}$ ,  $-Y_{\dot{v}}$  and  $-N_{\dot{r}}$  are replaced by  $m_X$ ,  $m_Y$  and  $J_{zz}$  respectively, and the coupling terms  $Y_{\dot{r}}$  and  $N_{\dot{v}}$  are neglected. To perform the numerical simulations, the hydrodynamic derivatives should be known as functions of  $\chi$  and  $\ell/\lambda$ . Such data are not available for the hydrodynamic derivatives except  $Y_v$  and  $N_v$ . Therefore, the values of the acceleration derivatives  $m_X$ ,  $m_Y$  and  $J_{zz}$ , and the rotary derivatives  $Y_r$  and  $N_r$  were substituted by their values in calm water as was the case during the analysis of directional stability mentioned previously. On the other hand, the values of the wave exciting forces  $X_E$ ,  $Y_E$  and  $N_E$  were calculated as functions of  $\chi$  and  $\ell/\lambda$  based on the results of the captive model tests. The derivative  $X_u$  denotes the increase in resistance and was determined from the captive tests in calm water. The equations(7) were solved in a step by step manner as described below.

- The initial values of encounter angle  $\chi_0$ , ship position on the wave  $(\ell/\lambda)_0$  and resultant speed  $U_0$  are prescribed. The ship has neither sway speed nor yaw angular velocity at the beginning.
- Corresponding to the initial values  $\chi_0$ ,  $(\ell/\lambda)_0$  and  $U_0$ , the wave exciting forces and the static derivatives  $Y_v$ ,  $N_v$  are estimated, and they are fixed at an interval  $\Delta t$  from  $t=0$  to  $t=\Delta t$ . Solving the equation(7), the ship's position on the wave  $(\ell/\lambda)_1$ , encounter angle  $\chi_1$ , resultant speed  $U_1$ , etc. at the time of  $t=\Delta t$  are determined.
- Corresponding to the new parameters  $\chi_1$ ,  $(\ell/\lambda)_1$ , and  $U_1$ , the static derivatives and the wave exciting force are calculated again, and then the equations(7) are solved for the interval  $\Delta t$  from  $t=\Delta t$  to  $t=2\Delta t$ . This procedure is repeated in a step by step manner. When the encounter angle  $\chi$  deviates from range  $-20^\circ \leq \chi \leq 60^\circ$ , the calculation is interrupted because the data on wave exciting forces are not available.

In the numerical calculation, the time step  $\Delta t$  was 0.05 seconds with the scale of the small captive model.

The values of parameters  $\chi_0$ ,  $(\ell/\lambda)_0$  and  $U_0$  are as follows:

$$\begin{aligned} \chi_0 &: 0^\circ, 10^\circ, 20^\circ, 30^\circ \\ (\ell/\lambda)_0 &: 0, 0.25, 0.50, 0.75 \\ U_0 \cos \chi_0 / U_w &: 0.8 \sim 1.2 \end{aligned}$$

Some of numerical simulations are shown in Fig.9 where only the trajectory of the center of gravity and the heading of the ship are shown. The rudder was kept amid-ship during all simulated runs. Moreover, it should be noted that the linear scale in the direction of the crest line of the wave is concentrated to 1/6.25 times the scale in the direction of wave propagation, and the interval between any adjacent circles drawn on the trajectories indicates



one second in the scale of time of the small captive model. Fig.9 shows the effects of initial position  $(\ell/\lambda)_0$  on the lateral motion of a ship traveling in following seas. It is evident in Fig.9(d) that a ship traveling on the down slope of a wave is forced to turn rapidly toward the crest line behind the ship if the initial encounter angle  $\chi_0$  ranges from  $20^\circ$  to  $30^\circ$ . On the other hand, Fig.10 shows the effects of ship speed on the development of the lateral motions of a ship traveling in following seas. In any case, as shown in Fig.10, the ship was initially situated in the middle of the down slope of the wave. When  $U_0 \cos \chi_0 / U_W$  ranges from 0.8 to 0.9, that is to say,  $U_0 \cos \chi_0$  is slightly less than the wave celerity, the ship enters a surf-riding condition and is forced to turn its bow toward the following crest line of the wave. This fact is in good agreement with the observations of the free running model tests.

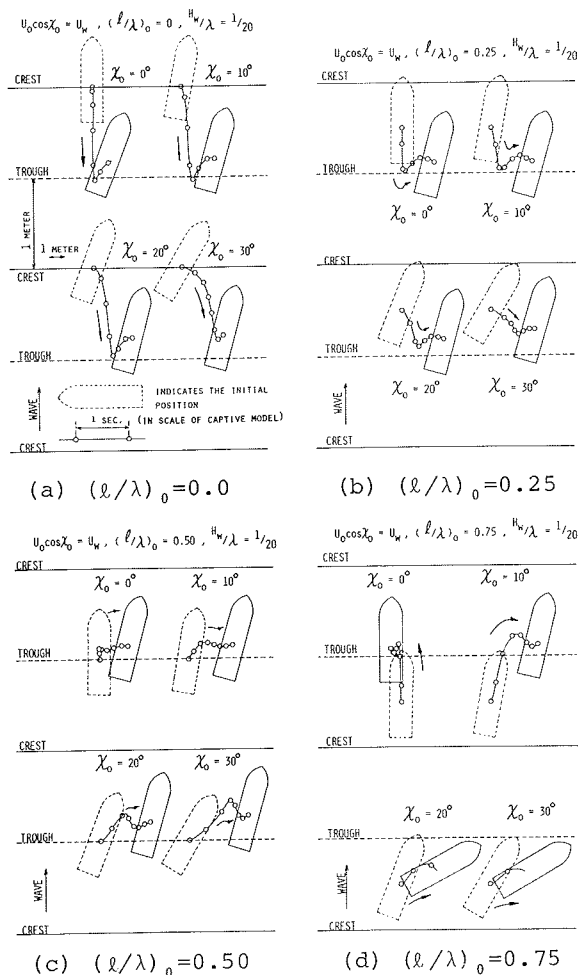


Fig.9 Ship's Trajectories in Following Seas ( $U_0 \cos \chi_0 / U_W = 1$ ,  $H_W/\lambda = 1/20$ )

In Fig.10 as well, it is evident that a drastic yawing motion develops when the initial encounter angle  $\chi_0$  is  $20^\circ$  to  $30^\circ$ .

When  $\chi_0$  is less than  $20^\circ$ , sway and yaw motions do not develop significantly since the wave exciting sway force and yaw moment are not sufficiently large. When a ship is situated on the up slope of a wave, on the contrary, it does not develop yaw motion at all because the wave exciting yaw moment forces the ship to turn so that the longitudinal center line of the ship may be in a position normal to the crest line of the wave. Here, it should be pointed out that together with longitudinal acceleration resulting from wave slope thrust, fast development of sway motion toward the trough of the wave makes the ship situated on the down slope of wave remain there for a while in spite of the development of strong yaw motion.

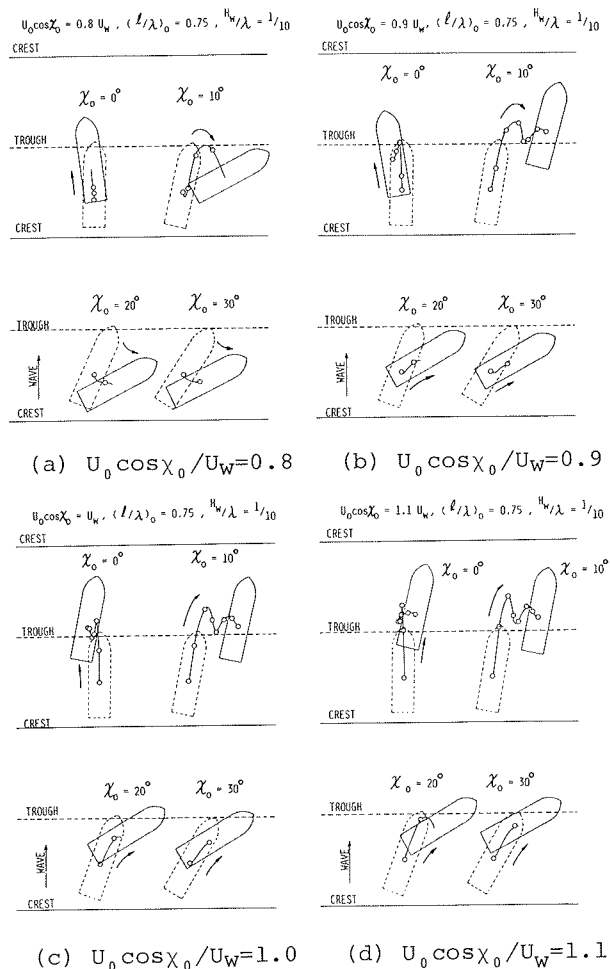


Fig.10 Effects of Initial Speed on Ship's Trajectories

When summarizing the results of the numerical simulations, the following conclusions are obtained.

- (a) When a ship travels on the down slope of a wave at a speed satisfying the condition where  $U_0 \cos \chi_0$  is slightly less than or equal to wave celerity, and furthermore, when the initial

encounter angle  $\chi_0$  is approximately 20° or 30°, the state of surf-riding is more likely to occur than otherwise.

- (b) The extent of the development of yaw motion is directly related to the slope of the wave, that is to say, the larger the wave slope, the faster the development of yaw motion.

In the preceding simulations, the rudder was not steered at all. If the steering of rudder could generate turning moment sufficient to overcome the wave exciting yaw moment acting on the hull, broaching-to would be prevented. In fact, during the free running model tests and during full-scale measurement, broaching-to phenomena had hardly occurred prior to the decrease in rudder size. Hence, in order to examine whether or not a drastic yaw motion is stimulated even when the hard-over helm is applied, additional numerical simulations were performed in the case corresponding to Fig.10 (c). During those simulations, the rudder area was reduced to approximately 56% of its original area, following the free running model tests as well as full-scale measurement. In the case where  $\chi_0$  equals 30°, the initial wave exciting yaw moment is approximately 1.6 times as much as the moment caused by the hard-over helm. Therefore, in spite of a hard-over helm, a drastic increase in yaw motion could not be prevented, although it took more time for yaw motion to fully develop in comparison to the case shown in Fig.10 (c). Then, it is concluded that significant development of yaw motion which cannot be prevented by a hard-over helm is caused by the generation of wave exciting yaw moment which exceeds the course-keeping-ability of the rudder.

#### 4.3 Examination of Broaching-to Phenomena Observed during Free Running Model Tests and Full-Scale Measurement

Based on the results of numerical simulations described in the preceding section, a hypothesis is proposed that the occurrence of broaching-to is caused by wave exciting yaw moment which exceeds by far the course-keeping-ability of the rudder. In this section, the propriety of the hypothesis will be examined by investigating the balance of sway force and yaw moment acting on the free running model, and by estimating the wave exciting forces acting on the full-scale ship from an analysis of ship motions during broaching-to.

- (1) Evaluation of the Balance of Force and Moment

A method of system identification is applied to analyze the data on ship motions recorded during broaching-to observed during the free running model tests in order to evaluate the optimal equations of manoeuvring equations and to

determine the unknown coefficients involved in the equations[6,7]. Only surge, sway, and yaw motions are taken into consideration, and their small perturbations from a straight advance under the constant speed  $U$  are described as follows:

$$\begin{aligned}(m + m_X)\dot{u} &= X_{\dot{u}}u + X_R(\delta) + X_E(\chi, \xi) \\ (m + m_Y)\dot{v} &= Y_{\dot{v}}v + (Y_r - mU)r + \\ &\quad Y_R(\delta) + Y_E(\chi, \xi) \\ (I_{ZZ} + J_{ZZ})\dot{r} &= N_{\dot{v}}v + N_r r + N_R(\delta) + \\ &\quad N_E(\chi, \xi)\end{aligned}\quad (8)$$

$$\text{where } \xi = 2\pi l/\lambda$$

As previously stated, the hydrodynamic derivatives of equations(8) vary according to the ship's position on wave  $\xi$  and encounter angle  $\chi$ . Their dependence on the relative position on the wave, that is to say,  $\xi$ , is assumed to be described approximately by a simple trigonometric polynomial as follows.

$$F(\chi, \xi) = F_0(\chi) + F_C(\chi)\cos \xi + F_S(\chi)\sin \xi \quad (9)$$

where  $F(\chi, \xi)$  represents, in general, a function of the variables  $\xi$  and  $\chi$ . And then,  $F_0(\chi)$ ,  $F_C(\chi)$  and  $F_S(\chi)$  are expanded by the Taylor series in  $\chi$ .

$$\begin{aligned}\begin{Bmatrix} F_0(\chi_0 + \Delta\chi) \\ F_C(\chi_0 + \Delta\chi) \\ F_S(\chi_0 + \Delta\chi) \end{Bmatrix} &= \begin{Bmatrix} F_0(\chi_0) \\ F_C(\chi_0) \\ F_S(\chi_0) \end{Bmatrix} + \\ &\quad \begin{Bmatrix} F_{0\chi}(\chi_0) \\ F_{C\chi}(\chi_0) \\ F_{S\chi}(\chi_0) \end{Bmatrix} \Delta\chi + \\ &\quad \begin{Bmatrix} F_{0\chi\chi}(\chi_0) \\ F_{C\chi\chi}(\chi_0) \\ F_{S\chi\chi}(\chi_0) \end{Bmatrix} \Delta\chi^2 + \dots \quad (10)\end{aligned}$$

where the subscript  $\chi$  on the right-hand side of the equation of eq.(10) stands for the differentiation with respect to  $\chi$ . That is to say,  $F_{0\chi}(\chi_0)$ , for example, indicates the partial derivative  $\partial F_0/\partial\chi$  at  $\chi=\chi_0$ . When combining the equations(8), (9) and (10), a general form of manoeuvring equations of motion in waves is obtained. To determine the optimal and significant forms of equations describing ship motion in waves, the AIC (Akaike's Information Criteria) technique[9] was used and the identification of unknown coefficients was performed by the least mean square evaluation in the balance of force and moment. This method was applied to the recorded data of broaching-to, of which the time histories of ship motion, rudder force, etc. are shown in Fig.11. For the analysis, three different parts shown in Fig.11 were used. Case I includes the entire region of observed broaching-to, and Case III the region just after the beginning of measure-

ment, during which broaching-to was not observed. Since the available data length is short and the amount of information is very limited, it is important to restrict the number of coefficients to be identified. Therefore, added mass and added mass moment of inertia  $m_x$ ,  $m_y$ , and  $J_{zz}$  are estimated by Motora's chart [5], and their values were assumed to be unchanged in waves. The results of analysis for sway and yaw motions are shown in Table 3. The amount of each component of sway force and yaw moment during a few instants are shown in the column of the table in MKS unit. Concerning the balance of sway force and yaw moment during sampled instants, the results of identification were successful but not necessarily satisfactory as shown in the table. Quantitative investigation into the balance of sway force and yaw moment leads to some interesting results. The most dominant term in yaw moment during the broaching-to is wave exciting yaw moment  $N_E$ , and the second one is the moment of rudder force  $N_R(\delta)$  which is approximately half of  $N_E$ . Furthermore, the identified amount of wave exciting yaw moment  $N_E$  and sway force  $Y_E$  correspond well in quantity to those estimated from the results of the captive model tests. Although the results of identification shown in Table 3 represent only one example and are not yet confirmed, it is induced that any particular component in the balance of force and moment does not change significantly during the broaching-to in comparison with non-broaching conditions. Based on the results mentioned above, it is concluded that any extraordinary change in sway force and yaw moment due to the following sea condition is not found even during the occurrence of broaching-to, and that the primary cause of broaching-to is the generation of wave exciting yaw moment exceeding rudder force that is required to keep the ship on a prescribed course in the waves.

## (2) Estimation of Wave Exciting Forces During Full-Scale Measurement

As previously stated, broaching-to was observed several times during full-scale measurements. However, it is impossible to apply the identification technique mentioned above to the data recorded during the full-scale measurement due to the lack of sufficient data on ship motion. Hence, in what follows, the amount of wave exciting forces which act on the full-scale ship during broaching-to will be roughly estimated by analyzing the recorded ship motions, and then they will be compared with the wave exciting forces obtained during captive model tests [8]. The wave exciting sway force  $Y_E$  and yaw moment  $N_E$  are evaluated in the following manner. From eq. (8),

$$\begin{aligned} Y_E &= (m + m_y) \dot{v} - Y_{\dot{v}} - (Y_r - mU) r - Y_R(\delta) \\ N_E &= (I_{zz} + J_{zz}) \dot{r} - N_{\dot{v}} - N_r r - N_R(\delta) \end{aligned} \quad (11)$$

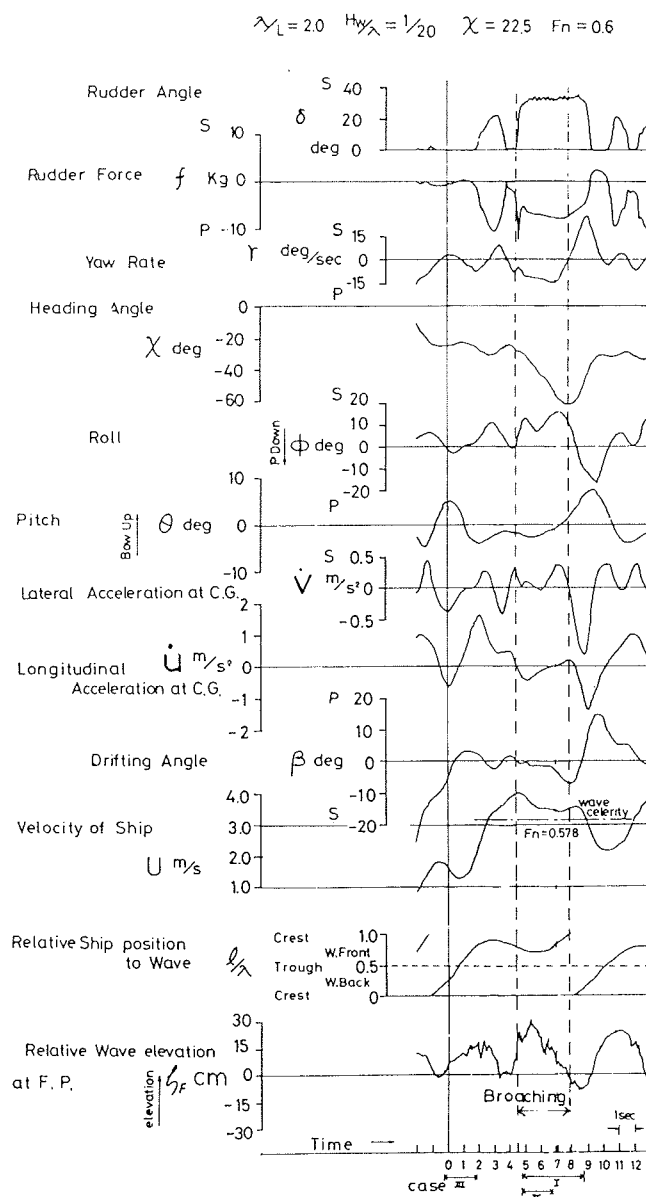


Fig.11 Time Histories of Ship Motion Used for Identification

- Yaw rate  $r$  was determined by a rate gyroscope on board. Yaw acceleration  $\dot{r}$  is calculated by the numerical differentiation of yaw rate.
- Sway acceleration  $\dot{v}$  at the centre of gravity of the ship is estimated from the lateral accelerations measured at the bow and stern by subtracting both components of the centrifugal and gravity acceleration from the measured acceleration. Sway velocity  $v$  is obtained by numerical integration of  $\dot{v}$ .
- The lateral component of rudder force  $Y_R(\delta)$  was determined by the force gauge installed at the rudder post. Yaw moment due to rudder force,  $N_R(\delta)$ , is calculated by multiplying

Table 3 Composition and Value of Each Component Included in Equations of Motion

$\lambda/L=2.0$ ,  $H_w/\lambda=1/20$ ,  $\chi=22.5^\circ$ ,  $F_v=0.6$ ,  $\xi=2\pi/\lambda$

| Mode       | Component                | Case          | Case I (Whole Region of Broaching)                                       |        |         | Case II (Broaching)               |         | Case III (Ordinary Response) |        |
|------------|--------------------------|---------------|--|--------|---------|-----------------------------------|---------|------------------------------|--------|
|            |                          |               | Data Length  |        |         | 4.8~8.8                           |         | 4.8~6.8                      |        |
|            |                          |               | Time   |        |         | $t_1$                             | $t_2$   | $t_1$                        | $t_2$  |
| Yaw (kg-m) | ① $(I+J)\ddot{\theta}$   | Coef. Assumed | 6.44   | 7.44   | 8.64    | 5.24                              | 6.44    | 0.44                         | 1.44   |
|            | ② $N_r \cdot r$          | Identified    | 0.313  | -2.262 | 4.796   | 6.865                             | 8.897   | 1.097                        | -2.167 |
|            | ③ $N_{\dot{v}} \cdot v$  |               | $(N_{r0} + N_{r1} \cos \xi + N_{r2} \sin \xi) \cdot r$                   |        |         | $(N_{r0} \cdot r)$                |         | $(N_{r0} \cdot r)$           |        |
|            | ④ $N_E$                  |               | 3.374  | 9.056  | 0.059   | -0.803                            | -0.917  | 0.190                        | -1.144 |
|            |                          |               | $(N_{\dot{v}0} + N_{\dot{v}1} \cos \xi + N_{\dot{v}2} \sin \xi) \cdot v$ |        |         | $(N_{\dot{v}0} \cdot v)$          |         | $(N_{\dot{v}0} \cdot v)$     |        |
|            |                          |               | $-14.289$<br>$+2.766^*$<br>$+3.344^*$<br>$-1.834^*$                      |        |         | $-12.425$<br>$-2.322$<br>$-1.834$ |         | $-13.481$<br>$-15.217$       |        |
|            | ⑤ $N_R(\delta)$          | Measured      | 6.534  | 6.689  | 5.374   | 6.181                             | 6.518   | 0.563                        | -0.162 |
|            | ⑥ $(m+m_R)\ddot{\theta}$ | Calculated    | -7.835   | -2.076 | 0.582   | -5.609                            | -7.850  | -1.571                       | -0.753 |
|            | ⑦ $mUr$                  |               | -7.846   | -2.287 | 0.700   | -7.419                            | -7.237  | -1.375                       | -1.566 |
|            | ⑧ $(m+m_R)\ddot{\theta}$ | Balance       | -7.846   | -2.287 | 0.700   | -7.419                            | -7.237  | -1.375                       | -1.566 |
| Sway (kg)  | ① $(m+m_R)\ddot{y}$      | Coef. Assumed | 3.858  | 6.892  | -32.208 | 2.256                             | 3.798   | -7.143                       | -0.400 |
|            | ② $mUr$                  | Identified    | -12.321  | -6.741 | 19.478  | -10.325                           | -12.325 | 0.744                        | -1.616 |
|            | ③ $Y_r \cdot r$          |               | -4.063   | 3.256  | -18.532 | -3.757                            | -4.868  | -1.235                       | 2.234  |
|            | ④ $Y_{\dot{v}} \cdot v$  |               | $(Y_{r0} + Y_{r1} \cos \xi + Y_{r2} \sin \xi) \cdot r$                   |        |         | $(Y_{r0} \cdot r)$                |         | $(Y_{r0} \cdot r)$           |        |
|            | ⑤ $Y_E$                  |               | -0.728   | -2.634 | 0.301   | -4.394                            | -5.042  | -0.138                       | 0.795  |
|            |                          |               | $(Y_{\dot{v}0} + Y_{\dot{v}1} \cos \xi + Y_{\dot{v}2} \sin \xi) \cdot v$ |        |         | $(Y_{\dot{v}0} \cdot v)$          |         | $(Y_{\dot{v}0} \cdot v)$     |        |
|            |                          |               | 1.699  | 5.765  | 9.766   | -4.377                            | -4.997  | -4.302                       | -4.129 |
|            | ⑥ $Y_R(\delta)$          | Measured      | -5.295   | -5.421 | -4.355  | -5.009                            | -5.282  | -0.456                       | 0.132  |
|            | ⑦ $(m+m_R)\ddot{y}$      | Calculated    | -3.169   | 5.571  | -8.375  | -3.060                            | -3.246  | -5.943                       | -2.148 |
|            | ⑧ $(m+m_R)\ddot{y}$      |               | -3.092   | 6.387  | -8.464  | -3.773                            | -4.914  | -5.675                       | -1.100 |

\* influence of heel

the horizontal distance between the centre of gravity of the ship and the rudder post by the rudder force  $Y_R(\delta)$ .

- (d) Hydrodynamic derivatives  $m_Y$ ,  $J_{zz}$ ,  $Y_r$  and  $N_r$  were assumed to be unchanged in waves, and their values were estimated in the same manner as in the numerical simulations of ship motion in waves. Derivatives  $Y_{\dot{v}}$  and  $N_{\dot{v}}$  were estimated from the results of captive model tests.

In Fig.12(a), the time series of yaw rate, rudder angle and sway velocity estimated in the manner stated previously are shown. In this case, where wave height was 1.0~1.5m and wave length 10~15m and ship speed was approximately 8.5 knots, the ship turned violently in the starboard direction in spite of the hard-over helm to check the starboard turn. The time series of the wave exciting sway force estimated by eq.(11) is shown together with each sway force component in Fig.12 (b) and (c). At the same time, the wave exciting sway force was again estimated from the results of captive model tests and is entered on the ordinate of Fig.12(c). The time series of yaw moment analyzed in the same manner and the amount of yaw moment estimated from the results of captive model tests, are also shown in Fig.12

(d) and (e). The peak value of yaw moment estimated from equation(11) agrees well with that estimated directly from the results of the captive model tests. On the contrary, agreement of the peak values of sway force estimated separately is not satisfactory as shown in Fig.12(c). This is because although sway force  $Y_{\dot{v}}$  due to sway velocity is dominant as seen in Fig.12(c), the accuracy of sway velocity estimated in the manner described previously is not sufficiently satisfactory. In Fig.13, similar results by estimating the wave exciting forces are shown in another case of broaching-to. In this case as well, agreement of yaw moment estimated by different methods is fairly good, but agreement of sway force is not especially good.

When comparing Fig.12(d) and Fig.12(e), it is evident that the moment of rudder force during broaching-to is approximately 600 kg-m and approximately 1/1.6 times smaller than the amount of wave exciting yaw moment for the encounter angle  $\chi=30^\circ$ , and approximately 1/2.5 times the peak value of wave exciting yaw moment estimated from eq.(11). Based on the results of analysis performed in this section, it is confirmed that broaching-to observed during the full-scale measurements was the result of wave exciting forces.

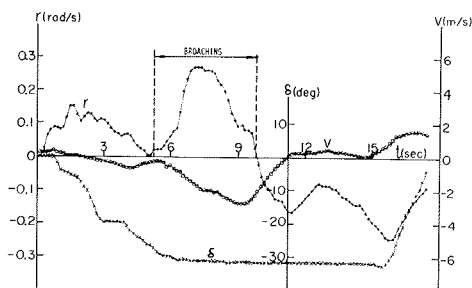
#### 4.4 Development of the Heel

To evaluate exactly the effects of heel on sway and yaw motions and development of the heel during broaching-to, the sway-roll-yaw coupled equations of motion should be considered. In the following, however, an alternative approach which has been advocated successfully by Hirano, et al.[10] to investigate the development of the heel during a steady turning motion, will be used.

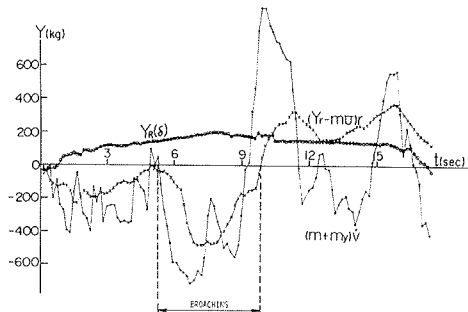
Additional captive model tests in calm water were performed to identify the effects of the heel on the hydrodynamic derivatives, particularly on the static derivatives  $N_Y$  and the rotary derivative  $N_r$ , because Hirano, et al. had reported that these derivatives were significantly affected so that the ship became more turnable than when the heel of the hull was not taken into consideration.

As a result of these experiments, it was found that the hydrodynamic derivatives  $Y_v$ ,  $N_v$ ,  $Y_r$  and  $N_r$  of the present model are almost unchanged even for the heel until reaching approximately  $20^\circ$ , and that either wave exciting sway force or yaw moment were of almost the same value as in the upright condition. Therefore, as far as the ship used during the present investigation is concerned, it is supposed that the development of sway and yaw motion during broaching-to are not fostered by the occurrence of heel of the hull.

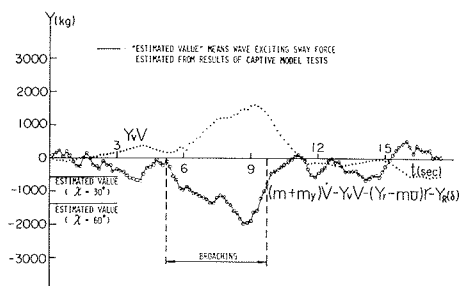
On the contrary, the wave exciting roll moment was affected to a certain extent by the presence of heel. Following



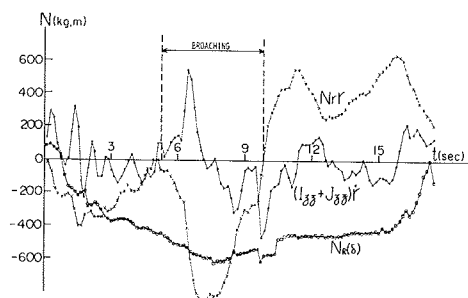
(a) Time Histories of Ship Motion



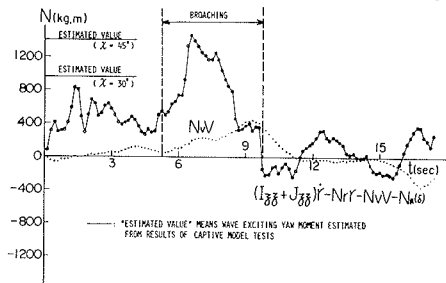
(b) Time Histories of Sway Force



(c) Time Histories of Sway Force



(d) Time Histories of Yaw Moment



(e) Time Histories of Yaw Moment

Fig.12 Ship Motion and Estimated Hydrodynamic Forces During Broaching-to

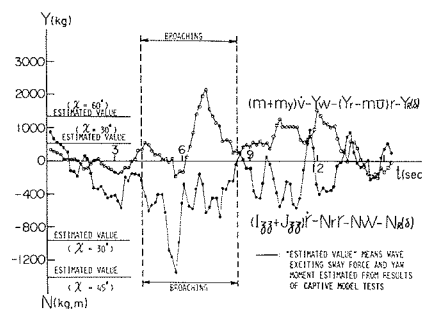


Fig.13 Sway Force and Yaw Moment Estimated by Analysis of Ship Motion Recorded During Broaching-to.

Hirano, et al., numerical simulations of ship motion in waves were performed with the heel of the hull being taken into consideration.

$$(I_{xx} + J_{xx}) \ddot{\phi} = -N \dot{\phi}(\dot{\phi}) - W \cdot \overline{GZ}(\phi) - Y_H \cdot z_H + L_E(\chi, \xi) \quad (12)$$

where  $Y_H$  stands for the hydrodynamic sway force acting on the hull below the water line and  $z_H$  the vertical position of application of  $Y_H$ . The roll damping moment was described as follows:

$$N \dot{\phi}(\dot{\phi}) = A \dot{\phi} + B \dot{\phi} |\dot{\phi}| \quad (13)$$

where  $A$  and  $B$  are constant coefficients, and determined by free rolling tests of the model. Righting moment  $W \cdot \overline{GZ}(\phi)$  was also determined by measuring the heel angle  $\phi$  of the model traveling in calm water caused by shifting a balast of known weight in a transverse direction. Besides, the vertical position of the point of application of lateral force  $Y_H$ , that is to say,  $z_H$ , was determined from the heel angle measured during the steady turning tests. The value of  $z_H/d$  thus determined is equal to 3.3.

By making use of these hydrodynamic data, several numerical simulations of lateral motions were performed. As previously stated, the hydrodynamic forces related to sway and yaw motions are not clearly affected by the heel. Therefore, the rate of development of sway and yaw motions in the following seas are nearly unchanged notwithstanding the ship heels.

However, it is important to note that the heel develops rapidly for a brief time when broaching-to has occurred. Furthermore, it should be pointed out that the heeling moment due to the hydrodynamic lateral force acting on the hull, that is to say,  $Y_H z_H$ , has a dominant effect rather than the wave exciting heel moment during broaching-to. An example of numerical simulations performed in the case where the parameters describing the wave conditions, say,  $\lambda/L$  and  $H_w/\lambda$  are equal to 2 and 0.05 respectively, with the rudder kept amidship is shown in Fig.14. Fig.14(a) shows the results of simulation obtained by neglecting

the heel, and Fig.14(b) those obtained by taking the heel into account. As shown in the figures, the rate of development of sway and yaw motions does not differ significantly in both cases, though the heel develops remarkably for a short time as shown in Fig.14(b).

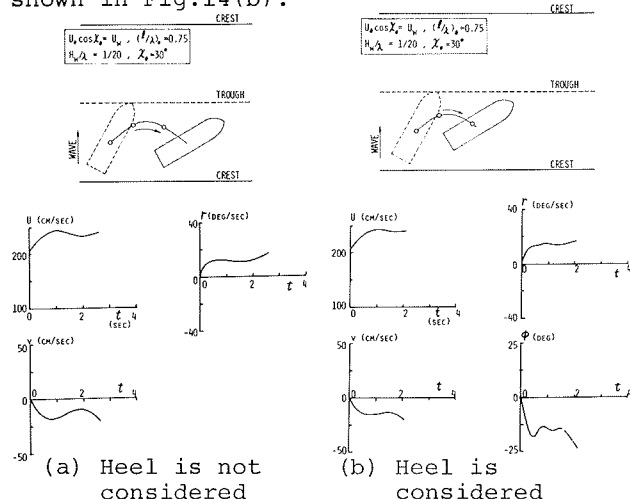


Fig.14 Effects of Heel on Development of Sway and Yaw Motions

## 5. CONCLUSIONS

The mechanism of the occurrence of broaching-to phenomena has been investigated by examining the results of free running model tests, captive model tests, and full-scale measurements. As a consequence, it is concluded that the primary cause of broaching-to is the generation of wave exciting yaw moment which is so enormous that even the hard-over helm cannot check the drastic development of yaw motion. In this sense, the decrease in rudder force on the down slope of the following wave is, to a certain extent, related to the occurrence of broaching-to, but is not the primary cause of broaching-to.

With regard to the loss of directional stability on the down slope of the wave, the authors also believe that it is not the primary cause of broaching-to, although it is true that the ship loses its directional stability when it is situated on the down slope of a wave.

From the free running model tests, it has been observed that broaching-to phenomena are likely to occur when wave conditions such as wave length, wave steepness, etc., and the running conditions such as ship speed, encounter angle, relative ship position on the wave, etc. are satisfied at the same time. The conditions under which broaching-to is likely to occur are not described here again since they were previously mentioned in detail. Nevertheless, it should be noted that these are not sufficient conditions but only necessary conditions for the occurrence of broaching-to.

One of the reasons why broaching-to has been long feared by seamen is generation of extraordinary heel during broaching-to. In the worst case, broaching-to is accompanied by the capsizing of the ship. As a result of numerical simulation, it is found that the heel moment due to hydrodynamic lateral force may exceed the wave exciting heel moment while broaching-to occurs. On the other hand, effects of the heel on the development of sway and yaw motions are extremely small as far as the ship used in the present investigation is concerned. As understood immediately from the conditions necessary for broaching-to, it is possible to understand a practical method to avoid broaching-to. When a ship is required to travel in the following seas, the heading angle of the ship should be kept parallel to the direction of wave propagation. Furthermore, it is advisable to make the speed of the ship vary to some degree from the wave celerity. If the engine has sufficient power, it is recommended to increase the engine power so that the ship may stay just in the trough of the wave or stay on the up slope of the wave. By the present investigation, the mechanism of broaching-to phenomena has been clarified to a great extent, but there still remain many problems to be solved. Particularly, the relationship between the probability of broaching-to and the hull form of a ship including the rudder configuration should be investigated first. Then, the development of extraordinary heel during broaching-to should be studied in more detail.

## ACKNOWLEDGEMENT

This investigation was performed as one part of the research project supported by Japan Craft Inspection Organization. Free running model tests, captive model tests, and full-scale measurement would not have been successful without the cooperation of many members participating in the research committee organized by JCIO. The authors wish to express their sincere gratitude to the members for their cooperation.

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## Discussion

M.R. Renilson (University of Glasgow, UK)

The systematic experiments carried out by the authors have shed considerable light on the cause of a broach. I agree with their main conclusion that a broach is caused by the large wave induced yawing moment being greater than the available corrective moment from the rudder. This yawing moment is a function of the ships position in the wave and a broach can occur if the ship spends a long time in this position, i.e. it is travelling at approximately wave speed.

I would be interested in how the authors obtained their  $Y_\beta$  and  $N_\beta$  experimentally as shown in Figure 7. I note they state  $e$  that they measured  $Y_x$  and  $N_x$  too but they don't seem to have quoted these results. I'd very much appreciate if they could include them in their written reply.

I wonder if the authors could say why they did not use on autopilot in their free running model experiments. In my opinion it is better to use an autopilot with known characteristics which can be modelled in any simulation attempt.

In the analysis of equations (1) to (5) the authors do not include rudder angle. Since  $A$  will always be positive, the criteria that  $D/A$  is greater than zero will reduce to  $D$  being positive - or essentially that of  $N_x$  being negative as  $Y_\beta$  is always positive and  $Y_x$  and  $N_\beta$  are small. This is simply stating that if there is a positive yawing moment from the wave the ship will yaw as there is no restoring moment. (Incidentally if  $N_x$  is negative then the ship will settle on a stable course at zero heading angle.) This is not meant to be a criticism of what the authors have done. I am nearly saying that what they have proved

mathematically seems reasonable physically.

The rudder derivatives are included later on in the paper and I would like to ask the authors what autopilot equation they used to obtain the rudder angle?

I was very interested by the time history of a broach presented in Figure 11. In my opinion the important thing to note is the large yaw acceleration which occurs just before the region denoted broach. I think it is this large yaw acceleration which causes a considerable yaw rate and leads to the broach. It is interesting to note that the heel angle is very low over this region and that the ship was surfing just slightly faster than the wave over this critical period.

Finally, I would like to congratulate the authors for accomplishing an extensive series of model experiments and for analysing then in such a way as to give a good insight into the cause of a broach.

### Author's Reply

I greatly appreciate Dr. Renilson's comment. I would like to reply to Dr. Renilson's discussions. First of all, I should say that the static derivatives  $Y_\beta$  and  $N_\beta$  were obtained through oblique towing tests performed in the waves. Namely,  $Y_\beta$  and  $N_\beta$  were determined as the factors which indicated the rate of increase of hydrodynamic sway force and yaw moment when the drift angle  $\beta$  was varied. Besides, we measured the wave exciting sway force and yaw moment for the various values of encounter angle, of which the results are shown in Fig. 15. The various lines drawn in Fig. 15 indicate the mean lines of the experimental results. The values of  $Y_x$  and



$N_x$  were determined by representing the experimental results of sway force and yaw moment with Spline functions.

The reason why the autopilot was not used during the free-running model tests is that the model did not have enough displacement to install the autopilot apparatus. As for Dr. Renilson's comment with regard to our examination of directional stability in the waves, I would like to emphasize that the instability in the waves is almost independent of the encounter angle, but we had many experiences telling us that the broaching-to is most likely to occur at a particular range of encounter angle.

We did not use the autopilot equation when examining the effects of steering on the drastic sway and yaw motions induced by the wave exciting forces, because the rudder was deflected to the maximum angle during the the broaching-to even when the rudder angle was governed by the autopilot equation. Therefore, we used the hard-over helm to check the rapid turning motion immediately after the simulation started.

force and moment in following sea. However, the final stage of the phenomena with eventual capsizing has not been clarified fully in here.

In this symposium, two papers are presented about roll-induced instability of high speed craft. And, it is shown that variation of GM is the most dominant factor on the phenomenon. And in several other papers, transverse stability of ships in a following sea is dealt with experimental and theoretical approaches, and the variation of the righting moment is investigated in reference to the relative position of the ship to the wave.

So, I think, the dominant factor on capsizing in broaching-to phenomena is the variation of the apparent righting moment by the effect of hydrodynamic moments acting on the hull. And investigation of transverse stability along this view point is needed for future.

This is my understanding about broaching-to phenomena. Any comment by the authors would be highly appreciated.

#### Author's Reply

I greatly appreciate Mr. Kobayashi's comment. I would like to reply to his comment. Our attention was particularly concentrated to the occurrence of the drastic development of sway and yaw motions of a craft traveling in the following seas, because even drastic development of sway and yaw without extraordinary heel are called broaching-to among the seamen. Discussor stated that the broaching-to phenomena are initiated by the instability of sway-yaw coupled motion. However, the authors do not conceive that the instability of sway-yaw coupled motions in the wave is not the primary factor to induce the broaching-to, because the ship is always unstable when traveling on the down slope of the following seas. Nevertheless, the broaching-to does not always occur as described in our paper.

At the initial stage of our investigation, however, we performed additional free-running model tests to investigate whether the transverse metacentric height GM has significant influence on the occurrence of broaching-to phenomena. As far as the experimental results are concerned, it is found that the effects of changing GM are negligibly small compared with such other conditions as wave length, wave height, ship speed, encounter angle and so on.

With regard to development of the heel during broaching-to, we mentioned in the last section of our paper as well as in the end of this presentation that the hydrodynamic moment due to the hydrodynamic lateral force acting on the hull has a dominant effect rather than the wave exciting heel moment.

Needless to say, I agree with Mr. Kobayashi that the further investigation should be performed as for the mechanism of development of heel during broaching-to.

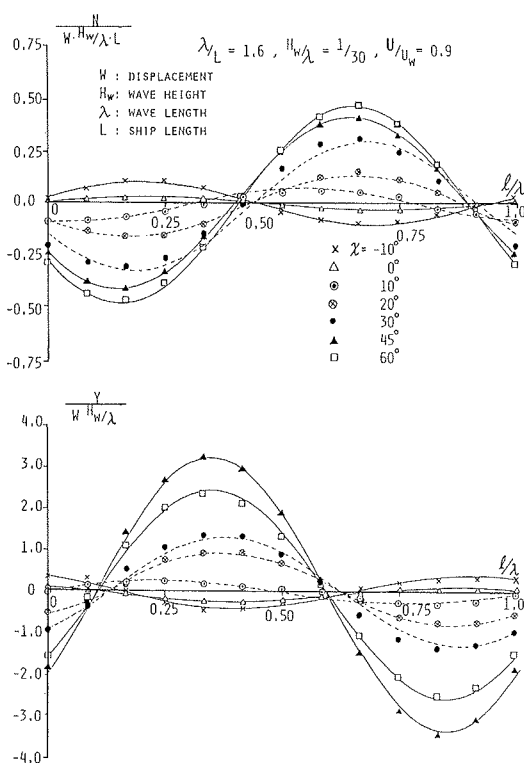


Fig. 15 Wave exciting sway force and yaw moment measured during captive model tests

E. Kobayashi (Nagasaki Experimental Tank, MHI, Japan)

The authors should be congratulated for the presentation of this very interesting study. In my understanding of this paper, broaching-to phenomena are initiated by the instability of sway-yaw coupled motion caused by the effect of wave exciting



S X - 2

# AN INVESTIGATION INTO THE FACTORS AFFECTING THE LIKELIHOOD OF BROACHING-TO IN FOLLOWING SEAS

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## ABSTRACT

One of the most dynamic modes of capsize can occur as a result of broaching-to in severe following seas. This paper covers the final part of an investigation into broaching, carried out jointly between the Admiralty Marine Technology Establishment (Haslar) and the University of Glasgow. A mathematical model for use on an analogue/digital hybrid simulation is developed in order to predict the conditions which could lead to broaching. A theoretical method is also developed for calculating the coefficients for use in the model and the results from this are compared with those obtained experimentally using a planar motion mechanism in a circulating water channel. The experimental technique is described fully in a previous paper (14), so only a brief outline is given here. The results of the simulation are compared with the results of free running model experiments and the effect of varying rudder size is shown.

## NOMENCLATURE

|           |   |
|-----------|---|
| A         | Wave amplitude  |
| $\bar{A}$ | Effective aspect ratio  |
| $B_1$     | Local half breadth  |
| $b_1$     | Half breadth  |
| C         | Calm water stability criteria   |
| C         | Wave speed  |
| $C_{al}$  | Added mass coefficient of half a rhombus calculated using the Schwarz-Christofel transformation |
| $C_H$     | Local transverse added mass coefficient   |
| $C_L$     | Lift coefficient  |
| $C^3$     | Maximum rudder rate   |
| $D^3_T$   | $z^*$ displacement of bottom of strip   |
| $d^T$     | Depth of rudder   |
| h         | Wave height   |

|            |  |
|------------|--|
| $I_z$      | Moment of inertia about the z-axis                     |
| K          | Wave number ( $= 2\pi/\lambda$ )                       |
| L          | Ship length, lift on rudder                            |
| $\ell_R$   | x co-ordinate of rudder                                |
| m          | Ship mass  |
| $N_r$      | Yaw moment derivative with respect to angular velocity |
| $N_v$      | Yaw moment derivative with respect to sway velocity    |
| $N_\alpha$ | Yaw moment derivative with respect to heading angle    |
| $N_\delta$ | Yaw moment derivative with respect to rudder angle     |
| P          | Pressure   |
| $P_1$      | Autopilot proportional control constant                |
| $P^2_r$    | Autopilot rate control constant                        |
| r          | Angular velocity                                       |
| $S_R$      | Rudder area  |
| T          | Draft  |
| $U_0$      | Self-propulsion speed                                  |
| u          | Surge velocity   |
| $V_R$      | Relative velocity of water past the rudder             |
| $V_S$      | Ship speed   |
| v          | Sway velocity  |
| $V_O$      | Orbital velocity at centre of area of rudder           |
| X          | Component of force along x-axis                        |
| $X_u$      | Surge force derivative with respect to surge velocity  |
| $X_\delta$ | Surge force derivative with respect to rudder angle    |
| $X_\xi$    | Wave induced surge force                               |
| $X^{PROP}$ | Thrust from the propeller                              |
| $X^a$      | Total force in x direction due to acceleration         |
| $X^P$      | Total force in x direction due to pressure             |
| $X_G$      | x co-ordinate of the centre of gravity                 |

|            |   |
|------------|---|
| $Y$        | Component of force along y-axis   |
| $Y_r$      | Sway force derivative with respect to angular velocity  |
| $Y_v$      | Sway force derivative with respect to sway velocity   |
| $Y_\alpha$ | Sway force derivative with respect to heading angle   |
| $Y_\delta$ | Sway force derivative with respect to rudder angle  |
| $z_G$      | z displacement of centre of gravity   |
| $\alpha$   | Heading angle   |
| $\alpha_0$ | Desired heading angle   |
| $\alpha_0$ | Yaw amplitude of PMM oscillation  |
| $\beta$    | Drift angle   |
| $\gamma$   | Rudder chord  |
| $\delta$   | Rudder angle  |
| $\zeta$    | z co-ordinate of centre of area of rudder   |
| $\eta$     | Ratio of relative velocity of water past the rudder to the ship velocity  |
| $\theta$   | Angle between tangent to the waterline and the centreline measured in such a way that it is negative towards the bow and positive towards the stern |
| $\mu_G$    | Aspect ratio factor for rudder caused by its proximity to the hull  |
| $\xi$      | Non-dimensional $x^*$ co-ordinate of the stern  |
| $\sigma$   | Clearance between the baseline and the bottom of the rudder   |
| $\tau$     | Pitch angle   |
| $\psi$     | Heading error ( $= \alpha - \alpha_d$ )   |
| $\omega$   | Frequency   |
| $\omega_e$ | Encounter frequency   |

Superscript "." indicates that the quantity has been differentiated with respect to time, and superscript "'" indicates that it has been non-dimensionalised as follows:

Non-dimensional mass  $= m' = m/\frac{1}{2}\rho L^3$

Non-dimensional force  $= X' = X/\frac{1}{2}L^2U^3$

Non-dimensional velocity component  $= v' = v/U$

Non-dimensional angular velocity component  $= r' = rL/U$

Subscripts c and w indicate the calm water and wave conditions respectively, and subscripts p and s stand for port and starboard respectively.

Two right handed co-ordinate systems are used.  $(x^*, y^*, z^*)$  is a wave fixed system with  $x^*$  in the direction of wave travel,  $z^*$  vertically downwards and origin on the calm water line in the wave crest.  $(x, y, z)$  is a body fixed system with the positive x-axis forward and parallel to the load waterline, the y-axis to starboard and the origin on the centreline amidships.

## 1. INTRODUCTION

Considerable difficulties have often been experienced with steering when travelling in severe following seas (1). The danger is that the ship will suddenly yaw from its desired course, ending up almost beam on to the wave direction despite application of maximum opposite rudder.

This is known as broaching-to and the associated large heel angles can cause considerable damage and possibly even a capsize. The danger has been appreciated for many years by mariners, who can give graphic, but unscientific descriptions of the behaviour of their ships under extreme conditions. The fact that the frequency of encounter is low and that surging velocities can be high makes the problem extremely non-linear and hence very difficult to investigate scientifically (2).

There have been a few attempts over the years to study the phenomenon of broaching, notably (3)-(12). This present study is based upon work done by Nicholson (13) on a free running radio controlled model in regular waves in a large manoeuvring tank. He identified the region on a plot of  $\lambda/L$  against nominal  $F_n$  where a broach occurred. Figure 1 is an example of such a plot given by Lloyd in an unpublished report. The

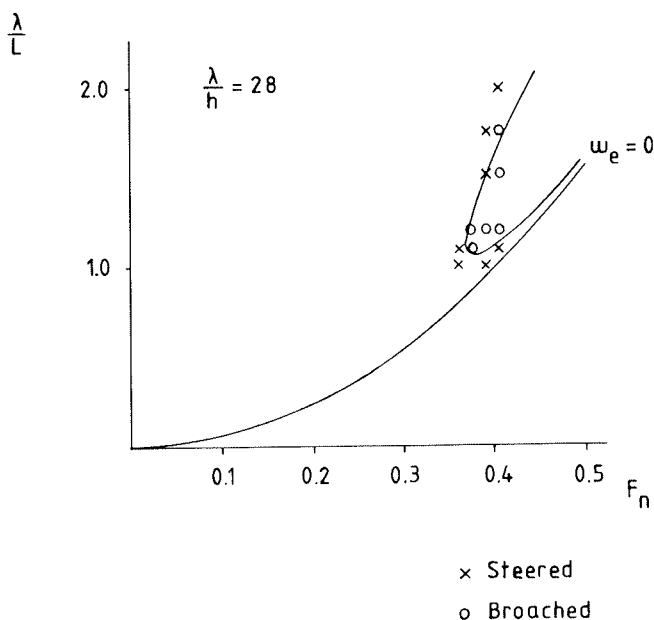


Figure 1. Broaching zone for standard rudder.  
(From an unpublished report by Lloyd).

suggestion made was that free running models should be run for each proposed design at various  $\lambda/L$ 's and their performance compared to that of an acceptable existing ship. It was found that the size of the rudders affected the broaching zone considerably and so it was suggested they could be made larger in a proposed design until an acceptable criteria was met.

This procedure involved considerable sophisticated model experiments and did not shed light on the quantitative nature of the various factors involved. The next step was to attempt to predict the broaching zone shown in figure 1 theoretically. This involved setting up a mathematical model,

obtaining the values of the coefficients and simulating the resulting motion in the time domain.

## 2. MATHEMATICAL MODEL

The mathematical model used is based on that used for manoeuvring in calm water as described in (14). The coefficients are all functions of the longitudinal position of the ship in the wave,  $\xi$ , but assumed to be independent of encounter frequency. This is because, for a true broach to occur, as discussed in (15), the encounter frequency will be almost zero since the wave induced longitudinal force will accelerate the ship to close to wave speed just prior to the broach.

The model is based on the linear equations of motion to reduce complexity and it is assumed that they will be able to indicate whether a broach will occur or not under any given wave and ship conditions.

Neglecting the roll equation the equations are therefore:

$$Y_V'v' + (Y_V' - m')\dot{v}' + (Y_R' - m')r' + (Y_R' - m'x_G')\dot{r}' + Y_\delta'\delta' + Y_\alpha'\alpha' = 0 \quad (1)$$

$$N_V'v' + (N_V' - m'x_G')\dot{v}' + (N_R' - m'x_G')r' + (N_R' - I_z')\dot{r}' + N_\delta'\delta' + N_\alpha'\alpha' = 0 \quad (2)$$

$$X_u'u' + X_V'v' + X_R'r' + (X_u' - m')\dot{u}' + X_\delta'\delta' + X_\xi'\xi' = 0 \quad (3)$$

In order to reduce complexity eqs. (1)-(3) are not written in functional form. However, it is important to remember that all the coefficients ( $Y_V'$ , ( $Y_V' - m'$ ), ....etc.) are functions of  $\xi$  as described above.

Equations (1)-(3) contain terms dependent on the rudder angle,  $\delta$ . This rudder angle is often prescribed by a helmsman but, for the model being developed, some means of determining it is required. The most convenient way to do this is to adopt the standard autopilot equation discussed in Ref. 16:

$$\delta_d = P_1\psi + P_2\dot{\psi} \quad (4)$$

where  $P_1$  and  $P_2$  are known as the autopilot constants, and  $\psi$  is the heading error ( $\psi = \alpha - \alpha_d$ ).

The problem with simply substituting (4) into (1)-(3) is that of time lags since neither the control system nor the rudder respond instantly.

## 3. CALCULATION OF THE COEFFICIENTS

In order to determine the values of the coefficients experimentally, a unique facility was set up at the National Maritime Institute (14). This involved oscillating a model using a PMM whilst balanced in a fixed longitudinal position

on a wave created by a wanedozer in a circulating water channel. The technique was similar to that used by Hamamoto (9,10) and is described fully in (14).

The theoretical calculation of the coefficients is based on a strip theory. Figure 2 shows the ship in the wave and it can be seen that the wave height will not be small compared to the draft. In

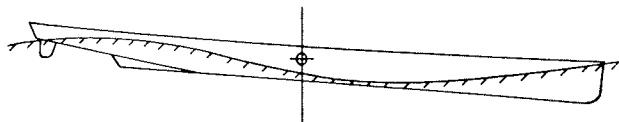


Figure 2. Profile Of The Hull In A Wave.

addition, the wave length will be of the order of the ship length, or greater, and so it is not possible to assume infinitesimal wave height or that the ship will remain in its undisturbed position in the vertical plane. Since the coefficients are assumed to be independent of encounter frequency they can be calculated for the zero frequency of encounter condition. For this case the ship will be in its equilibrium position in the vertical plane so the pitch angle ( $\tau$ ) and vertical displacement ( $z^*$ ) will be functions of the wave position only. They are calculated by a trial and error method which adjusts the position of the ship in the vertical plane until the displacement in the wave equals that in calm water, and the longitudinal centres of buoyancy (LCB) and gravity (LCG) are in the same longitudinal position. This has the effect of ignoring the vertical component of the Froude-Kryloff and inertia forces and assumes that the pressure varies linearly with depth from the surface of the wave. Since the vertical position of the ship in the wave is only used for calculating the lateral and longitudinal coefficients and will be altered by the fact that the ship adopts an additional trim angle due to the high speed, it is thought that the above approximation is sufficiently accurate for the present purpose.

### 3.1 Sway Force and Yawing Moment Due to Heading Angle

In order to calculate  $Y_\alpha(\xi)$  and  $N_\alpha(\xi)$  it is convenient to consider the ship with a small heading angle  $\alpha_0$ . The ship is then divided into transverse strips  $\delta x$  wide, distance  $x$  from the origin (Figure 3).

Using the slender body assumption, the side force can be obtained by integrating the horizontal component of the force on each strip along the length of the hull. (Since the  $X$  component of the force is assumed to be negligible compared to the  $Y$  component).

The velocity forces are ignored throughout the wave force calculations as they were found to be negligibly small compared to the pressure and acceleration terms.

The lateral pressure force on each strip

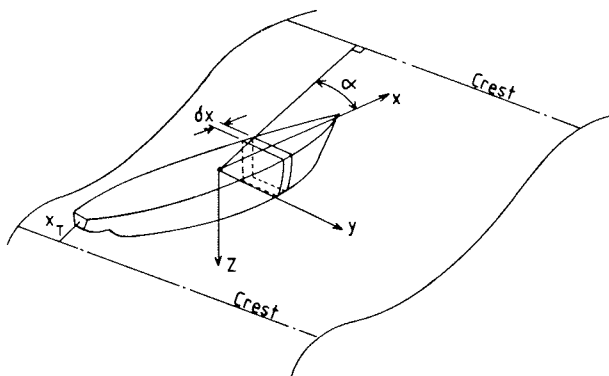


Figure 3. Schematic View Of The Hull In A Wave.

will be given by:

$$Y_{STRIP}^P = - \int_{D_T} \frac{-A \cos Kx_C^*}{\rho g (A e^{-Kz^*} \cos Kx_P^* + z^*)} \delta x dz + \int_{D_T} \frac{-A \cos Kx_C^*}{\rho g (A e^{-Kz^*} \cos Kx_S^* + z^*)} \delta x dz \quad (5)$$

$$x_S^* = x_T^* + x + \frac{L}{2} - b(z) \alpha \quad (6)$$

$$x_P^* = x_T^* + x + \frac{L}{2} + b(z) \alpha$$

for small  $\alpha$ .  $x_S^*$  and  $x_P^*$  are the values of  $x^*$  and  $x^*$  respectively on the free surface.

Hence, the total lateral pressure force and yawing moment can be found by integrating equation (5) along the length of the ship in the usual way. Thus:

$$y_{STRIP}^a = \frac{AVM_{STRIP} \alpha g A}{D_{WC}} \sin Kx_C^* (e^{-KD_T} - e^{KA \cos Kx_C^*}) \delta x \quad (7)$$

where  $AVM_{STRIP}$  is the two-dimensional added mass value for the strip and can be calculated using the Schwarz-Christoffel transformation (17) for straight-sided sections, or by the Frank-Close-Fit method (18) for ship shapes.

### 3.2 Longitudinal Force

The slender body assumption does not hold for longitudinal motions since the gradient  $dy/dx$  cannot be assumed to be zero. A further complication is the trim angle which introduces a component of the vertical gradient,  $dy/dz$ , into the required gradient on the  $x^*-y^*$  plane,  $dy^*/dx^*$ . It can easily be shown that:

$$\frac{dy^*}{dx^*} = \frac{dy}{dx} \cos \tau + \frac{dy}{dz} \sin \tau \quad (8)$$

In principle it should be possible to make use of eq. (8) when computing the X force on the hull. It was found, however, that the numerical errors introduced by this

method were considerable and an alternative approach was used.

Instead of attempting to calculate the force in the  $x^*$  direction directly, the forces in the  $x$  and  $z$  directions were found and resolved to obtain the  $x^*$  and  $z^*$  forces. This procedure involved less error, as the  $z^*$  force is known to equal  $-mg$  for equilibrium and the  $x$  force is small.

Figure 4 shows the forces acting on a ship with a trim angle of  $\tau$ .  $Z$  and  $X$  are the hydrodynamic and hydrostatic forces acting;  $mg$  is the gravitational force on the body and  $X_\xi$  is the resultant wave force.

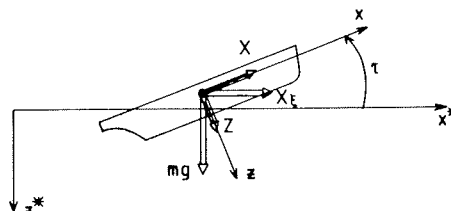


Figure 4. Longitudinal Forces Acting On The Hull.

Thus:

$$X_\xi = mg \tau - X \quad (9)$$

for small  $\tau$ .

As  $mg$  and  $\tau$  are known, only  $X$  is required to be calculated and this can be done as follows:

#### Pressure Force:

As for the sway force and yawing moment, the longitudinal pressure force is calculated by integrating the pressure over the entire wetted surface of the hull. The essential difference between the calculation for the longitudinal force and the lateral force is that, in the case of the lateral force, the longitudinal component of the normal to the ship's hull is neglected, whereas for the longitudinal calculation it must be taken into account.

Thus, the longitudinal component of the pressure force on a transverse strip will be:

$$X_{STRIP}^P = -2\rho g \int_{D_T} \frac{-A \cos Kx_C^*}{(A e^{-Kz^*} \cos Kx_C^* + z^*)} \theta \delta x dz \quad (10)$$

for small  $\theta$ , where  $\theta$  is a function of  $z$  as well as of  $x$ .

In addition to the pressure force contribution to the  $X$  force from the longitudinal component of the pressure at the sides of the ship, there will be a contribution due to the transom stern. Here  $\theta = \pi/2$  and the contribution to the  $X$  pressure force will be:

$$X_{TRANSOM}^P = -2\rho g \int_{D_T} \frac{-A \cos Kx_T^*}{(A e^{-Kz^*} \cos Kx_T^* + z^*)} b dz \quad (11)$$

Thus, the total longitudinal pressure force is given by:

$$X^P = \int_{-\frac{L}{2}}^{\frac{L}{2}} X_{STRIP}^P dx + X_{TRANSOM}^P \quad (12)$$

#### Acceleration Force:

The problem with calculating the longitudinal acceleration force is in obtaining the longitudinal added mass and, in particular, since the acceleration will be varying over the ship's length, in obtaining the longitudinal spread of the added mass.

The method used is to divide the ship into transverse strips as before. These transverse strips are further divided into horizontal strips resulting in regular trapezohedrons, as shown in Figure 5.

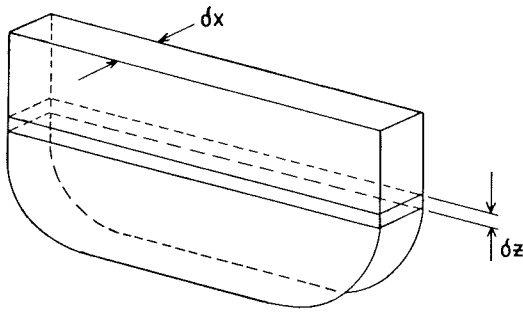


Figure 5. Schematic View Of A Transverse Strip Showing A Trapezohedron  $\delta z$  Thick.

Each of these trapezohedrons can be considered in two dimensions as a trapezium which forms part of a rhombus, as shown in Figure 6. Now, using the Schwarz-Christoffel transformation, the AVM of this rhombus can be obtained and, assuming that this is spread evenly over the entire shape, the AVM of the trapezium can be found. Thus, what in fact is being obtained is the AVM due to an element of the ship's surface. It is then assumed that the acceleration is constant over this element in order to calculate the acceleration force.

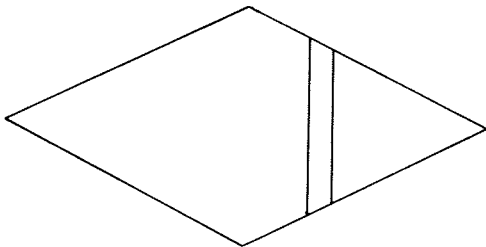


Figure 6. Rhombus Formed From Trapezium.

Using this method, which is explained more fully in Ref. (19), the longitudinal acceleration force at a point on the ship's hull can be expressed as:

$$X_{POINT}^a = \frac{\rho \pi^2 B_1 C_{al} A e^{-Kz^*} \tan(1-\gamma)\pi \sin Kx_C^*}{\lambda} \quad (13)$$

Thus, the longitudinal acceleration force on a strip will be:

$$X_{STRIP}^a = \int_{D_T} \frac{-A \cos Kx_T^*}{D_T} X_{POINT}^a \delta x dz \quad (14)$$

In a similar way to the pressure force there will be a contribution due to the force on the transom. This will depend on the AVM of the transom which can be calculated using half of the value of an equivalent flat plate. Thus, the total longitudinal acceleration force is given by:

$$X^a = \int_{-\frac{L}{2}}^{\frac{L}{2}} X_{STRIP}^a dx + X_{TRANSOM}^a \quad (15)$$

### 3.3 Rudder Derivatives

The rudder derivatives,  $N_\delta$  and  $Y_\delta$ , are both dependent on the side force generated by the rudders operating at an angle to the centreline of the ship. The rudders operate as low aspect ratio hydrofoils with a limited groundboard effect, caused by their proximity to the hull, in a complex flow which is affected by the hull and screws upstream. For this reason an absolute theoretical calculation of the rudder derivatives will be complex and inaccurate, so the method used here is to calculate the ratio between calm water and wave derivatives. This is then used, together with the experimental calm water value, to obtain  $Y_\delta$  and  $N_\delta$  in the wave. In order to simplify the procedure the effect the vertical component of the orbital velocity will have on the rudder derivatives is ignored.

For simplicity, the velocity at the rudders can be calculated from:

$$V_R = \eta V_S \quad (16)$$

where  $\eta$  is assumed constant for small changes in speed.

Considering only rectangular geometry rudders with no sweepback, the lift slope can be expressed as (20).

$$\frac{dC_L}{d\delta} = \frac{0.9 (2\pi)}{\sqrt{AR^2 + 4} + 1.8} \quad (17)$$

Now, the lift on the rudder is given by:

$$\frac{dL}{d\delta} = \frac{1}{2} \rho S_R V_R^2 \frac{dC_L}{d\delta} \quad (18)$$

giving:

$$Y'_{\delta} = \frac{1}{L^2 V_S^2} S_R V_R^2 \frac{dC_L}{d\delta} \quad (19)$$

and

$$N'_{\delta} = \frac{l_R}{L^3 V_S^2} S_R V_R^2 \frac{dC_L}{d\delta} \quad (20)$$

Hence,

$$\frac{Y'_{S_W}}{Y'_{S_C}} = \frac{N'_{S_W}}{N'_{S_C}} = \frac{d^2 (\eta V_S - V_O)^2 \Lambda_C \mu_W}{d_C^2 (\eta V_S)^2 \Lambda_W \mu_C} \quad (21)$$

where

$$\Lambda_C = \sqrt{(\mu_C^2 \frac{d^2}{\gamma^2} + 4) + 1.8}$$

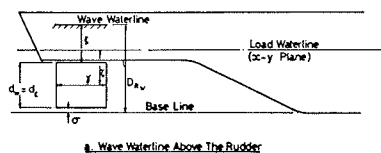
$$\Lambda_W = \sqrt{(\mu_C^2 \frac{d^2}{\gamma^2} + 4) + 1.8}$$

$$V_O = KAC e^{-Kz} \cos Kx^*$$

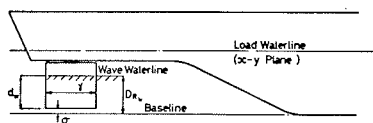
and  $\mu_C, \mu_W$  are the aspect ratio factors caused by the proximity of the rudders to the hull in the calm water and the wave conditions respectively.  $V_O$  is the horizontal component of the orbital velocity which is assumed to be constant over the rudder with the value taken at its centre of area.

There are two separate possibilities for the wave condition dependent on the position of the wave waterline, as shown below in (a) and (b). In the first case the aspect ratio and rudder area are unchanged from the calm water condition to the wave condition and hence eq. (21) simplifies to:

$$\frac{Y'_{\delta W}}{Y'_{\delta C}} = \frac{N'_{\delta W}}{N'_{\delta C}} = \frac{(\eta V_S - V_O)^2}{(\eta V_S)^2} \quad (22)$$



a. Wave Waterline Above The Rudder



b. Wave Waterline Below The Rudder

In the second case both the aspect ratio and the rudder area will be reduced from the calm water condition to the wave condition.  $\mu_W$  will equal one, since the hull will now no longer have a groundboard effect on the rudder in the wave condition, and  $d_W = D_{R_W} - \sigma$ . Thus eq. (21) becomes:

$$\frac{Y'_{\delta W}}{Y'_{\delta C}} = \frac{N'_{\delta W}}{N'_{\delta C}} = \frac{(D_{R_W} - \sigma)^2 (\eta V_S - V_O)^2 \Lambda_C}{d_C^2 (\eta V_S)^2 \Lambda_W \mu_C} \quad (23)$$

The rudder derivatives are then obtained from:

$$Y'_{\delta} = Y'_{\delta E} \frac{Y'_{\delta W}}{Y'_{\delta C}} \quad (24)$$

and

$$N'_{\delta} = N'_{\delta E} \frac{N'_{\delta W}}{N'_{\delta C}} \quad (25)$$

where  $Y'_{\delta E}$  and  $N'_{\delta E}$  are the calm water values obtained experimentally.

### 3.4 Manoeuvring Derivatives

It is not possible, using the present state-of-the-art, accurately to predict theoretically the sway velocity or rotational velocity derivatives even at low speeds in calm water. For this reason no attempt has been made here to try to do so at high speeds in waves. Instead, a calculation of the change in the derivatives caused by the wave is made and applied to the derivative obtained experimentally in calm water. The method used here is based on the assumption that the derivatives are made up of potential and viscous flow components which are independent. The potential flow component is that which exists in an ideal fluid and hence can be calculated using strip theory, whereas the viscous flow component is related to the lift and cross-flow drag effects and cannot be readily calculated theoretically. The assumption made here is that the viscous component will remain unchanged in the wave condition and hence it is only required to calculate the change in the potential flow component.

The acceleration derivatives can be calculated directly using strip theory, as they comprise entirely of potential flow contributions. The cross coupling acceleration derivatives are calculated here although it is recognised that they are very small and often assumed to be zero.

The strip theory used is due to Clarke (17) with the added mass values obtained from the Frank-Close-Fit method, as for the transverse force calculation. The rudders are assumed to be at the stern so that an addition due to their added mass

is made to the added mass coefficient of the stern.

Thus:

$$Y'_{\dot{V}} = -\pi \left(\frac{1}{L}\right)^2 \int_{\text{STERN}}^{\text{BOW}} T^2 C_H dx' \quad (26)$$

$$N'_{\dot{V}} = -\pi \left(\frac{1}{L}\right)^2 \int_{\text{STERN}}^{\text{BOW}} T^2 C_H x' dx' \quad (27)$$

$$Y'_{\dot{F}} = -\pi \left(\frac{1}{L}\right)^2 \int_{\text{STERN}}^{\text{BOW}} T^2 C_H x' dx' \quad (28)$$

$$N'_{\dot{F}} = -\pi \left(\frac{1}{L}\right)^2 \int_{\text{STERN}}^{\text{BOW}} T^2 C_H x'^2 dx' \quad (29)$$

$$Y'_{\text{V POTENTIAL}} = -\pi \left(\frac{1}{L}\right)^2 T_{\text{STERN}}^2 C_{H \text{STERN}} \quad (30)$$

$$N'_{\text{V POTENTIAL}} = Y'_{\text{V POTENTIAL}} x'_{\text{STERN}} + Y'_{\dot{V}} \quad (31)$$

$$Y'_{\text{r POTENTIAL}} = Y'_{\text{V POTENTIAL}} x'_{\text{STERN}} \quad (32)$$

$$N'_{\text{r POTENTIAL}} = Y'_{\text{V POTENTIAL}} x'_{\text{STERN}} + Y'_{\dot{F}} \quad (33)$$

using the generally accepted assumption that  $C_{H \text{BOW}} = 0$ , since a finite value of kinetic

energy cannot be instantaneously imparted to the fluid at the bow.

Comparison between the theoretical results and those obtained from the experiment in Ref. 14 is given in Figures 7-19. Although there is considerable disagreement with some of the coefficients, the more important ones ( $X'$ ,  $N'_{\alpha}$ ,  $N'_{\delta}$ ) are predicted reasonably well. It is therefore assumed to be possible to apply the method developed above to predict the values of the coefficients for a range of wave conditions for use in the time domain simulation.

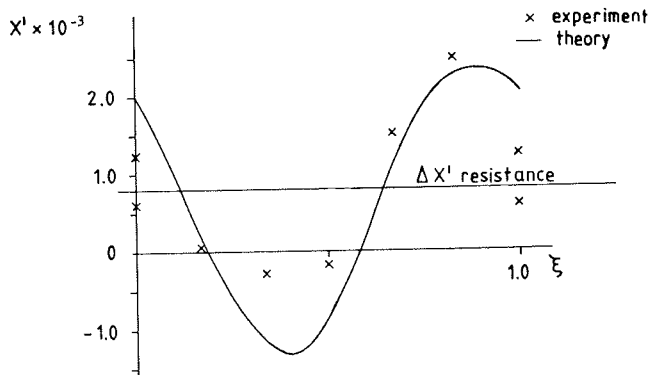


Figure 7. Non-dimensional X-force as a function of  $\xi$ .

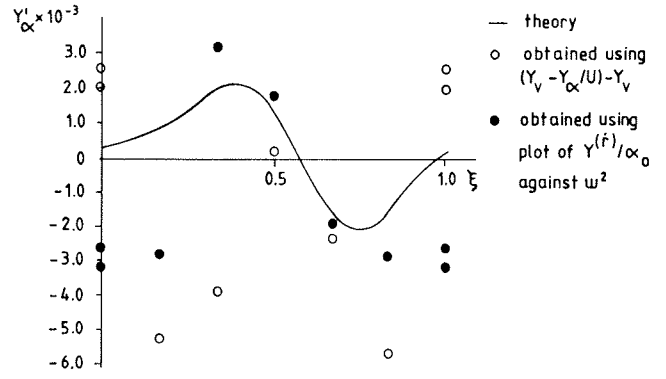


Figure 8.  $Y'_{\alpha}$  as a function of  $\xi$ .

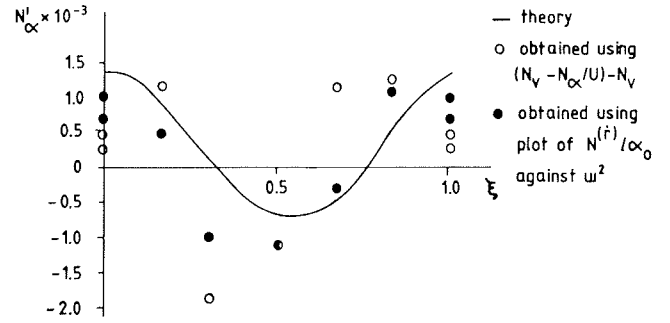


Figure 9.  $N'_{\alpha}$  as a function of  $\xi$ .

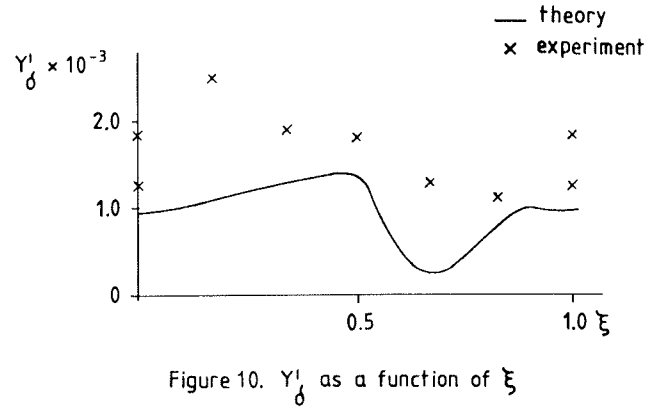


Figure 10.  $Y'_{\delta}$  as a function of  $\xi$

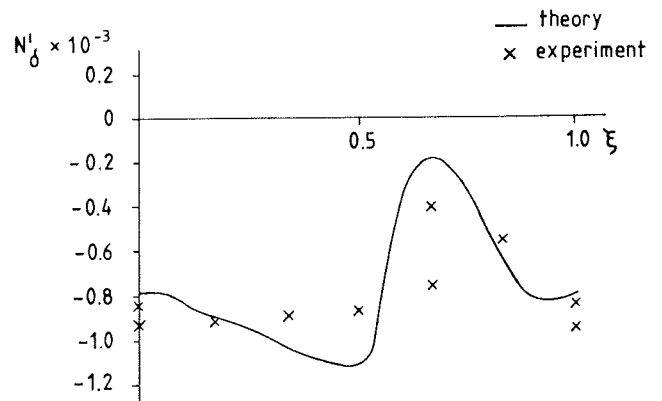


Figure 11.  $N'_{\delta}$  as a function of  $\xi$

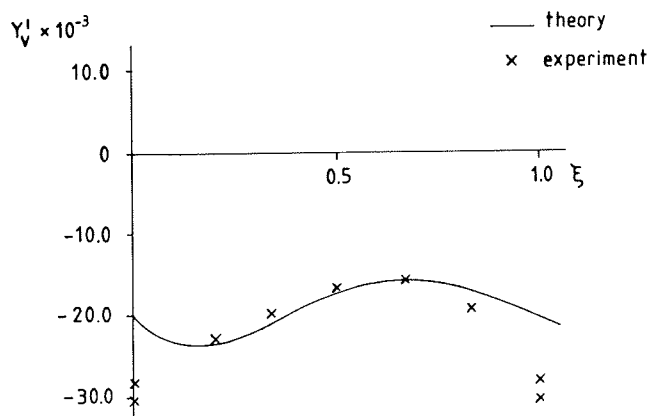


Figure 12.  $Y_V^I$  as a function of  $\xi$

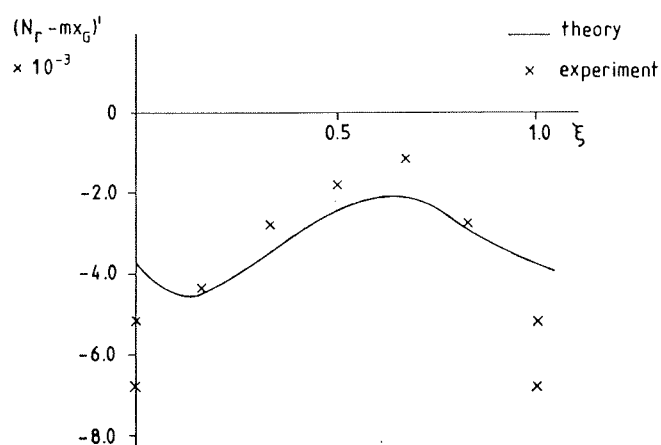


Figure 15.  $(N_r - mx_G)^I$  as a function of  $\xi$

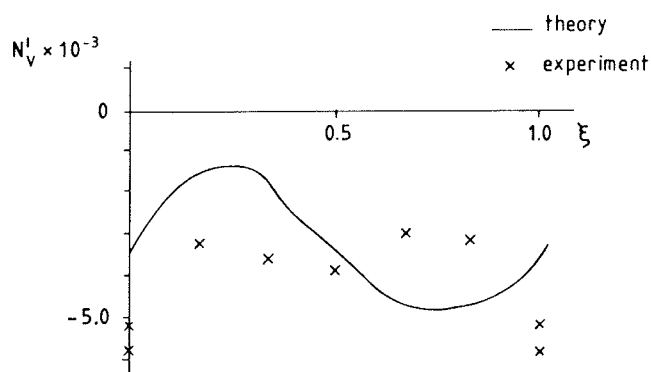


Figure 13.  $N_V^I$  as a function of  $\xi$

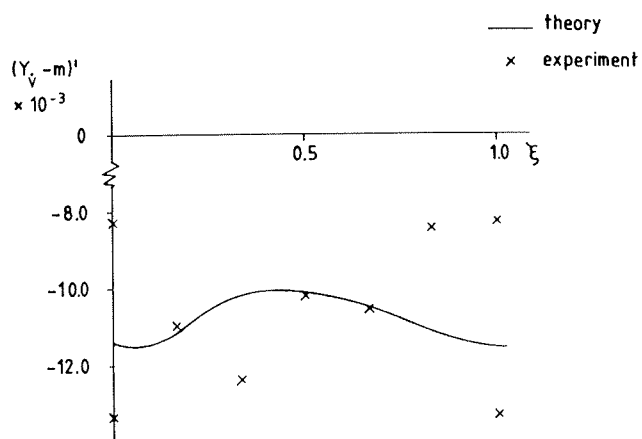


Figure 16.  $(Y_V - m)^I$  as a function of  $\xi$

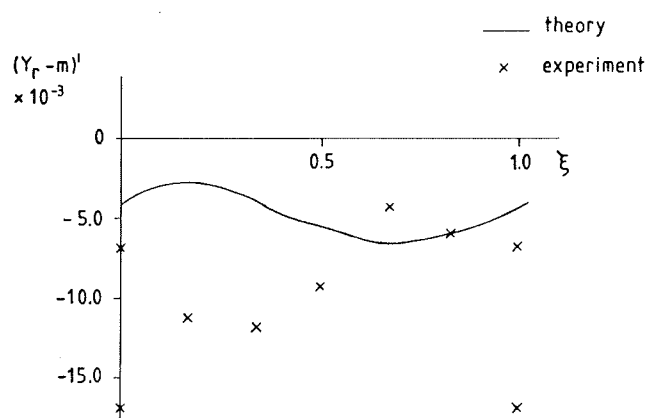


Figure 14.  $(Y_r - m)^I$  as a function of  $\xi$

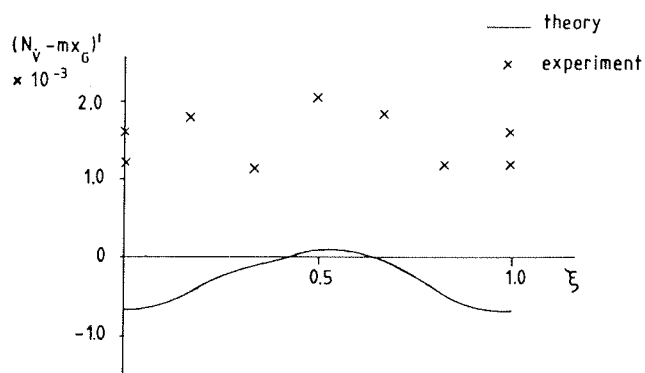


Figure 17.  $(N_V - mx_G)^I$  as a function of  $\xi$



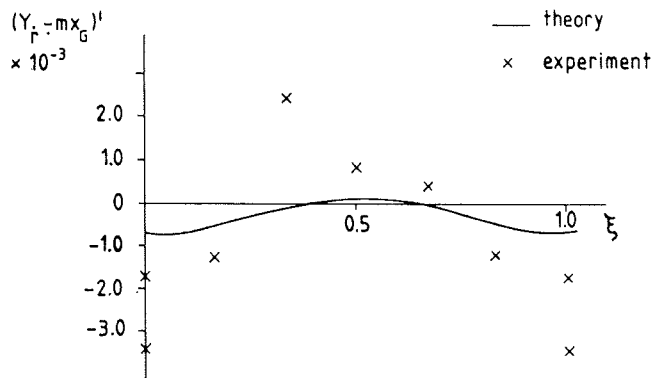


Figure 18.  $(Y_r - m x_G)'$  as a function of  $\xi$

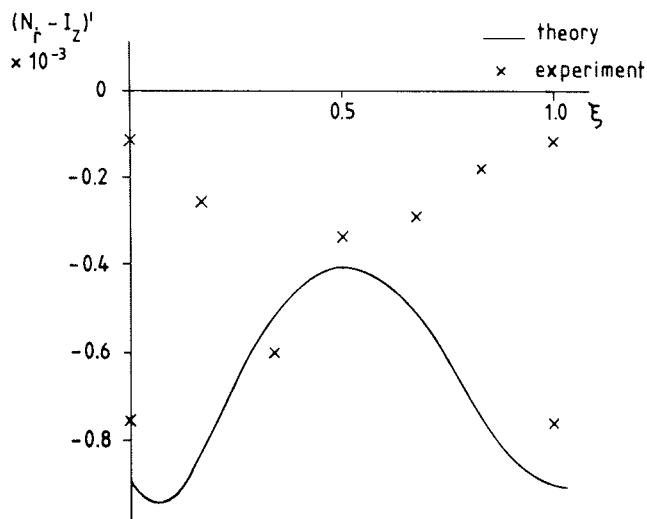


Figure 19.  $(N_r - I_z)'$  as a function of  $\xi$

#### 4. TIME DOMAIN SIMULATION

In order to predict whether the ship will broach in certain conditions it is necessary to couple the lateral equations to the surge equation. It is also desirable to model the maximum rudder velocity, maximum rudder angle, and any time delays which may occur in the autopilot system. This was done using a hybrid simulation which involved an EAI 2000 analogue computer for the real time integration, together with a PDP 11/45 digital computer which provided the coefficients.

##### 4.1 The Simulation Technique

When dimensionalised and simplified the equations become:

$$0 = Y_v v + (Y_v - m) \dot{v} + Y_\alpha \alpha + (Y_r - mU) \dot{\alpha} + Y_\delta \delta \quad (34)$$

$$0 = N_v v + N_\alpha \alpha + (N_r - m x_G U) \dot{\alpha} + (N_r - I_z) \ddot{\alpha} + N_\delta \delta \quad (35)$$

$$0 = x_{u2} U^2 + (x_{u1} - m) \dot{u} + x_\xi + x_{\text{prop}} + x_\delta |\delta| \quad (36)$$

$$\xi = \frac{1}{\lambda} \int_0^t [U - C - v\alpha] dt \quad (37)$$

$$\delta_d = P_1 \psi + P_2 \dot{\psi} \quad (38)$$

$$\delta_a = \int_0^t \dot{\delta}_a dt$$

$$-c_3 < \dot{\delta}_a < c_3$$

$$-\delta_{a_m} < \delta_a < \delta_{a_m}$$

where  $\cos \alpha$  is assumed to be equal to 1 and  $\sin \alpha$  to be equal to  $\alpha$ . The cross coupling acceleration terms and the coupling between  $v$  and  $r$  and  $X$  have been ignored since they are small and there was a shortage of coefficient units. In addition to these equations, a first order Padé circuit has been incorporated between the demand for the rudder and the start of the rudder movement in order to provide an approximation to a time delay (21). The main problem with the simulation on the analogue computer was that many of the coefficients of the equations were dependent on  $\xi$ . A digitally set function generator (DSFG) was used to provide the dependence of  $X_\xi$  on  $\xi$ , but as there was only one of those available, an alternative method had to be used for the remaining coefficients. The technique involved using the PDP 11/45 digital computer which is connected to the EAI 2000. The procedure was as follows. The digital computer set up the initial conditions as normal using the serial port, and the counter on the logic part of the analogue computer was set to a pre-determined value. The '2000 was then put into the operate mode by the '11 and the counter started counting down to zero. The simulation continued with  $X_\xi$  being varied by the DSFG, but with all other coefficients remaining constant. When the counter reached zero (this took 0.05 of a second) the '2000 was put into the hold mode using the patch panel control. The digital computer then sampled the output corresponding to  $\xi$  through an analogue to digital converter and calculated the values of the variable coefficients using interpolation from previously fed data. The appropriate coefficient values were then set at the 6 digital to analogue converter's and the '2000 put back into the operate mode. This continued until (a) the ship was overtaken by three waves, (b) it was broached, or, (c) the time limit elapsed.

Unfortunately there were only 6 digital to analogue converter's available and 8 required. This was solved by assuming  $[Y_v/(Y_v - m)]$  and  $[(Y_r - mU)/(Y_v - m)]$  remained constant at their calm water value, and the effect of this on the yaw motion is thought to be negligible.

##### 4.2 Comparison of Simulation with Free Running Model Experiments

Since the relationship between motion and force was assumed to be linear, prediction of the model's path during a broach

was not attempted. Instead, the simulation was used to predict the broaching zones discussed in the introduction. Values of  $P_1$  and  $P_2$  were chosen to be 3 and 1 respectively and the rudder rate was scaled from the ship which had a value of  $3^\circ/\text{sec}$ . It was assumed that once the rudder started to move it would reach maximum speed almost immediately. However, the time delay from the moment it was required to when it started moving was estimated to be 3 seconds model scale. The standard rudder and half-depth rudder conditions were both investigated in waves of  $\lambda/h = 28$ , which corresponded to the free running experiments discussed in the introduction. Comparison between the broaching zones obtained using the simulation and the experimental results are given in Figures 20 and 21. A broach was considered to have occurred if either there was an overload\* or the heading angle exceeded  $40^\circ$ . If the model was overtaken

by three waves without broaching then the run was assumed to be steered. The other way in which a steered run could be obtained was if the model was carried along by the wave at a constant heading angle. This generally happened if the self-propulsion speed was nearer to wave speed as the model settled into its longitudinal equilibrium position almost immediately.

Looking at Figures 20 and 21, the comparison between the predicted zones and the experimental results is quite good. This is especially so considering that there are a large number of imponderables, such as, calculation of coefficients, neglect of heel angle, simplification of equations and, not least, the modelling of the rudder.

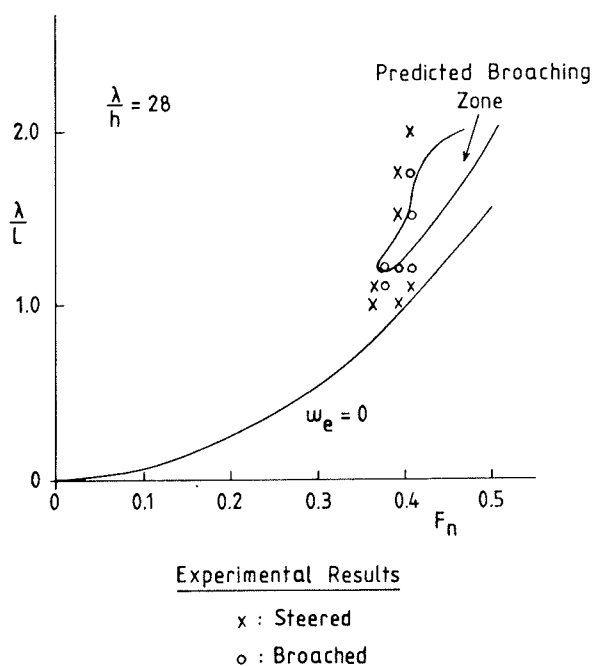


Figure 20. Comparison between predicted and experimental results for standard rudder.

\*On an analogue computer all variables must be within the range -1 to +1 and the equations are scaled using expected maximum values before being patched up. If the value of a variable goes outside these limits during a simulation an overload is said to occur, implying that the value has exceeded the expected maximum.

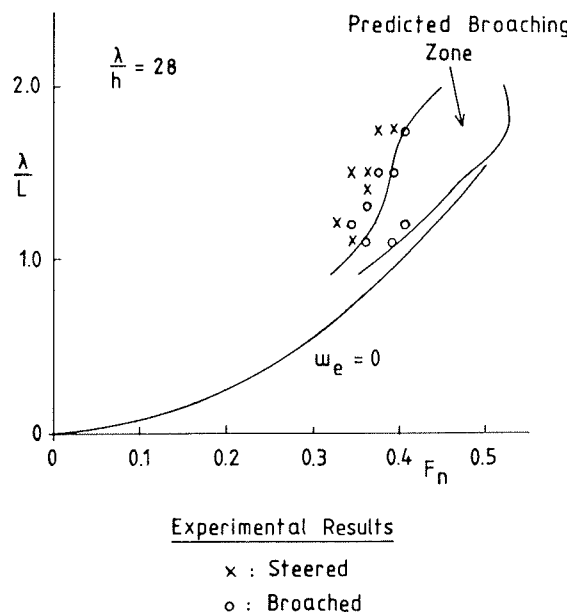


Figure 21. Comparison between predicted and experimental results for 1/2 depth rudder.

#### 4.3 Discussion

From studying the records of the simulation it is possible to conclude that the principle cause of a broach is the large wave induced yaw moment which occurs around the same longitudinal position as a reduction in rudder effectiveness. The length of time a ship spends in this longitudinal position is crucial and in regular waves this will depend on its self-propulsion speed. A high speed (near wave speed) will enable the ship to settle in its stable longitudinal position immediately, and a low speed will result in the waves overtaking the ship without accelerating it significantly. It is the speeds which just allow the ship to be accelerated to wave speed which cause it to remain in the critical region for long enough to result in a broach.

In an irregular sea the length of time spent in the critical region will also depend on the previous wave.

## 5. GUIDELINES FOR REDUCING THE LIABILITY TO BROACH

From studying the simulation records and the plots of the broaching zones for various conditions, it is possible to give some tentative guidelines for reducing the liability to broach. It must be borne in mind that the guidelines are based on observations of the results of the simulation which has certain limitations. The most important of these are that roll is neglected and only regular waves are considered. The guidelines can be split into those which apply at the operating stage and those which apply at the design stage.

### 5.1 Guidelines Applicable at the Operating Stage

The first step is to recognise when the sea is severe enough to cause a broach. The predominant wave length must be of the order of the ship length or greater, with the wave amplitude the order of the ship draft or greater. In these conditions broaching may occur if running in following seas and the safest advice is to avoid that heading.

The most obvious factor, next to heading angle, which is under the control of the captain, is the ship speed. The situation to try and avoid is being accelerated to wave speed by a steep wave as long as, or longer than, ship length. Thus, if the ship takes on a low frequency surging motion with the maximum speed near to wave speed, then it is time to slow down - before encountering a slightly steeper wave which might just carry the ship along for long enough to be broached. The speed ought to be reduced till the surging motion becomes less noticeable and the waves are overtaking the ship with a reasonably high frequency.

As can be seen from Figures 20 and 21 no broaching occurs when ship self-propulsion speed is close to wave speed (i.e. faster than the broaching zone) and it may be presumed from this that an alternative to slowing down might be to speed up. Unfortunately, it is unlikely that this would work in a real irregular sea since there is always the possibility of encountering a longer wave resulting in acceleration to this wave's speed and hence broaching.

Loss in rudder effectiveness, due to emersion, is an important factor and this can be reduced by submerging the rudders as much as possible, either by increasing draft or increasing trim by the stern, or both. This is particularly important if the rudders are near the surface in the calm water condition - such as with short spade rudders.

### 5.2 Guidelines Applicable at the Design Stage

When a ship is being designed which may have to travel at high Froude numbers (greater than about 0.25) in rough seas, it is necessary to give considerable attention to its susceptibility to broaching. Since the critical wave lengths are dependent on the ship length, shorter ships will encounter more severe conditions and, hence, are more likely to broach.

By far the most important variable at the design stage is the rudder, and it has been shown how the loss in rudder effectiveness caused by emersion has an important influence on broaching. The spade rudders common to twin screw ships are prone to emersion in waves, since they are near the calm waterline and tend to be short to reduce the bending moment on the stock. Increasing their depth can result in a marked reduction in the liability to broach. However, this may make the ship difficult to dry dock. The method developed in this paper can be used to determine how big the rudders need to be in order to meet a desired criteria. The single deep skeg-mounted rudder is less likely to emerge and may prove to be better at preventing broaching, although more work will have to be done in this direction.

A reduction in beam over the stern region may decrease the amount by which the stern is lifted by the waves permitting the rudders to emerge. The reduction would have to be continued forward for a short distance and would also have to include a reduction in flare over the stern region. This could tend to imply that the wider transom-sterned ships may be more vulnerable particularly if associated with considerable flare to increase deck area aft.

## 6. CLOSURE

It is thought that this work represents a step forward in isolating the predominant factors contributing to broaching, and in quantifying the forces and moments involved. Although considerable work still has to be done, the simulation method developed here can be used to determine whether a new design will meet a given criteria and to carry out parametric studies in order to reduce its liability to broach.

## ACKNOWLEDGEMENTS

The work described in this paper was carried out at the University of Glasgow and was supported by a Science and Engineering Research Council Studentship. The author would like to thank his colleagues in the Department of Naval Architecture and Ocean Engineering for their useful discussions. The work carried out on the hybrid simulation could not have been completed without the considerable help received from the staff of the Electrical Engineering Department.

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## Discussion

M. Fujino (University of Tokyo, Japan)

I greatly appreciate Dr. Renilson's work which provides information useful for considering what the primary factors affecting the occurrence of broaching-to phenomena are. Dr. Renilson concluded that the principal cause of broaching-to is the large wave yaw moment. In this respect, I completely agree with Dr. Renilson. I would like to state two comments.

(1) As I described in our paper, I conceive that the encounter angle or the heading angle is one of the most important factors affecting the occurrence of broaching-to. Even if the conditions of wave length, wave height and ship speed are satisfied, broaching-to does not always occur as shown in Table 2 of our paper. For instance, in the

case of initial encounter angle  $X_0$  equal to zero, broaching-to was not observed at all during the free-running model tests. Therefore, I conceive that Figs. 20 and 21 of Dr. Renilson's paper do not provide information sufficient for predicting the occurrence of broaching-to because the difference in the probability of broaching-to due to the difference in the encounter angle is not shown in a definite manner.

(2) Dr. Renilson calculated the values of hydrodynamic derivatives by taking account of the change of the profile of the hull in a wave. However, the effects of the change of the inflow velocity due to the orbital motion of water particles are not negligible. Furthermore, it should be pointed out that the strip method cannot predict negative values of added mass moment of inertia

although the experimental results show negative added mass moment of inertia as shown in Fig. 19.

#### Author's Reply

I would like to thank Professor Fujino for his useful discussion. In answer to his first question, encounter angle was not varied in our work. The free running model experiments were carried out at 20° and so was the simulation. I agree that encounter angle is an important factor in broaching, however, the aim of the work described in the paper did not require that it be investigated. Now that a method for comparing with existing ships has been developed, it is possible to extend the use of the simulation technique to study the effect of heading angle.

As far as the calculation of the hydrodynamic coefficients is concerned I agree that some of the coefficients are predicted very badly by the theory and look forward to further discussion with Professor Fujino in order to improve this. I should say that the poor prediction tends to be for the less important coefficients.

T. Fuwa (Ship Research Institute, Japan)

The methodical way of investigation into broaching-to phenomena and its fruitful results shown in this paper are to be highly appreciated. The explanation how the broaching-to occurs and what are the predominant factors for it is quite reasonable. Especially guidelines to avoid broaching-to both for the operating stage and the design stage are very important in practice. It is nothing but a surprise that these results are quite similar to those obtained by the broaching-to study for a small fishing boat carried out in Japan, which have already been shown by Prof. Fujino in the previous presentation. This coincidence of the results between the different researches performed independently shows that the mechanism of the broaching-to phenomena is correctly understood and well described in the scientific manners, though it had been thought to be quite difficult.

I would like to discuss on the following two points.

(1) As stated in the paper, there are a few factors having predominant effects on the occurrence of broaching-to. Among them the factors related to the loading condition (GM, trim, etc.) and running condition (ship speed and encounter angle to waves) are the first to be investigated thoroughly. Because they are the conditions in the operating stage and ready to be improved. My first question is about Figs. 20 & 21. We know the initial encounter angle and relative ship position to waves are quite sensitive parameters for the results of the simulation as well as the model test.

I wonder whether the results in Figs. 20 & 21 cover the whole range of these parameters or they are examples for a certain condition.

(2) Second of my question is whether the same or similar criterion is applied for the judgement of the model test results and the simulation results. Criterion of broaching-to is very important to evaluate the ship characteristics for broaching-to. In my opinion, the same and simple criteria should be applied for practical purposes, and considerable efforts should be made to establish the criteria for the next stage of the research. According to my experience, it is not easy to find out a proper definition and criterion of broaching-to. However, the combination of the duration of the yaw angular velocity opposite to the ordinary ship response to the rudder action and the relative ship position to waves could be the criterion of broaching-to. They are indices of "rudder ineffectiveness" and surfriding condition respectively, both of which are essential to broaching-to phenomena as the author has correctly pointed. Moreover, combining with the change of the heading angle, they can be the measure of the broaching-to severity. Therefore, time histories of ship behavior should be examined carefully even they are the results of the linear estimation, and shall be compared with the experimental results. I would like to ask for the author's comments on the definition and the criterion of broaching-to.

#### Author's Reply

Dr. Fuwa's comments are very welcome. He points out the fact that he has come to very similar conclusions in a completely independent study and I agree wholeheartedly with him that this implies that the mechanism of broaching has been correctly understood in both studies. His first question is similar to Professor Fujino's point so I would like to refer him to my reply to Professor Fujino on this. His second question is very relevant and difficult to answer. I did not carry out the radio controlled model experiments so I can't say exactly what criteria was used for them. I spent a lot of time speaking to the people who did carry them out and was given the impression they used a very subjective method. It is easy to say when a ship has been violently broached and when it has been well steered, but it is difficult to judge the in-between cases. Although I may have given the impression in the paper that I used an objective criteria for judging the results of the simulation there was in fact a certain amount of subjectivity in the discussion as to whether a run was broached or not. I do not consider it possible to give a universal criteria at a broach based on heading error, yaw rate etc., it is a very subjective thing.

I used the time histories of the simu-

lation to gain insight into what causes a broach. Unfortunately, I was not able to compare them with the time histories of the radio controlled models as they did not exist. I look forward to comparing them with Dr. Fuwa's model experimental results.

N. Matsumoto (Tsu Research Laboratories, NKK,  
Japan)

Thank you for your excellent paper.  
I have one question.

There is considerable disagreement in Fig. 7 - 19. I think you applied these calculated derivatives to time domain simulation.

If derivatives obtained from the experiment would be adopted for the simulation, does the predicted broaching zone change in Fig. 20 and 21?

#### Author's Reply

Mr. Matsumoto is correct in saying that there is considerable disagreement between some of the experimental and theoretical results quoted in figures 7 - 19. Fortunately, more important ones are predicted quite well.

The captive model experiment was only carried out at one wave condition and so it would not be possible to obtain the plots of figures 20 & 21 from these results. On the other hand, the simulation was carried out at that wave condition and the time histories obtained compared with that obtained by using the theoretical results. The agreement was quite good.

*Session XI*

## Stability of Offshore Structures (Part II)

*Chairmen*

Prof. Sigismund Kastner  
College of Engineering Bremen  
F.R.G.

Prof. Makoto Ohkusu  
Kyushu University  
Japan

# THE EFFECT OF LOW FREQUENCY ROLL MOTION ON UNDER-DECK CLEARANCE OF A SEMI-SUBMERSIBLE PLATFORM

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## ABSTRACT

In this paper a method is presented in which the allowable under-deck clearance in irregular waves is predicted for a column-stabilized type semi-submersible platform (Photo. 1) which is subjected to the non-linear effect induced by the second-order wave force in roll mode.

First, extensive model experiments were carried out in regular waves with a column-stabilized type semi-submersible platform model to certify the existence of the steady tilt and to obtain the corresponding steady tilt moment by changing the metacentric height of the model, where the effect of wave height was considered as well. In addition, the model experiments in irregular waves were carried out for the purpose of providing a method for predicting the under-deck clearance, where both the irregular waves having the ordinary spectrum of the Pierson-Moskowitz type and the regular wave groups are used.

Second, computations were made iteratively to obtain the motion spectrum in roll mode by taking into account the steady tilt moment established by the model experiments in regular waves. Further, the expected values of roll motion and relative displacement in irregular waves were predicted by introducing the statistic process, and those were compared with the results of model experiments in irregular waves with good agreement.

It seems in conclusion that the present method will reasonably explain the non-linear effect of low frequency roll motion on the estimation of under-deck clearance in spite of the fact that it is still unsatisfactory in qualitative discussion. Here the statistical analysis was also tried for a spectrum having double peaks.

## NOMENCLATURE

|    |   |  |
|----|---|--|
| b  | : | Horizontal distance from the center plane to the center of column                              |
| B  | : | Horizontal distance between the center of each lower hull                                      |
| d  | : | Height from the center of lower hull to center of gravity                                      |
| r  | : | Equivalent radius of lower hull ( $=\sqrt{\text{midship sectional area of lower hull}/\pi}$ )  |
| g  | : | Gravitational acceleration   |
| G  | : | Transfer function of roll amplitude by the unit regular wave                                   |
| G' | : | Transfer function of roll amplitude by the unit regular wave not being coupled with sway force |



Photo. 1 A column-stabilized type semi-submersible platform



$G_1$  : Transfer function of roll moment by the unit regular wave  
 $GRD$  : Transfer function of relative displacement by the unit regular wave  
 $GM$  : Metacentric height  
 $H_{1/3}$  : Significant wave height  
 $H_w$  : Regular wave height  
 $h_0$  : Depth from the water level to the half height of a lower hull in upright condition  
 $h_1, h_2$  : Depth from the water level to the half height of both port and starboard lower hulls in tilt condition, respectively  
 $I_1$  : Bessel function of the first kind  
 $\lambda$  : Wave length  
 $K$  : Wave number ( $=2\pi/\lambda$ )  
 $L$  : Length over all of a semi-submersible platform  
 $l$  : Horizontal distance from the center of gravity to measured point of relative motion  
 $M_T$  : Steady tilt moment  
 $N$  : Data sampling number  
 $m_{OL}, m_{OH}$  : Variance  
 $R$  : Steady tilt moment coefficient  
 $SEW$  : Equivalent wave spectrum  
 $SR$  : Roll motion spectrum  
 $SRL$  : Low frequency component of the roll motion spectrum  
 $SRH$  : High frequency component of the roll motion spectrum  
 $STM$  : Steady tilt moment spectrum  
 $SRD$  : Relative displacement spectrum  
 $SW$  : Real wave spectrum  
 $T_\phi$  : Resonant period of roll  
 $T$  : Mean wave period  
 $T_w$  : Regular wave period  
 $YA$  : Sway amplitude  
 $ZA$  : Heave amplitude  
 $ZRA$  : Relative displacement amplitude  
 $\rho$  : Density of water  
 $\zeta_A$  : Wave amplitude of regular or regular wave group  
 $\zeta_0$  : Elementary wave amplitude  
 $\phi_s$  : Steady tilt angle  
 $\phi_A$  : Roll amplitude  
 $\omega$  : Circular frequency of wave, motion etc.  
 $\omega_1, \omega_2$  : Circular frequency of elementary wave  
 $\Delta\omega$  : Difference of the circular frequency of elementary wave ( $=\omega_1-\omega_2$ )  
 $\zeta_{RD}^{(+)}, \zeta_{RD}^{(-)}$  : Positive and negative relative displacement, respectively  
 $\nabla$  : Displacement of a lower hull

## 1. INTRODUCTION

Shipbuilding industries have been actively developing offshore floating structures, especially semi-submersible platforms since their advent in the early 1960's (Ref.1). At present, a large number of semi-submersible platforms have been put in operation under the rapid development of sub-sea oil production in the North Sea and other sea areas, where shore distances are

great and weather conditions hostile.

In these circumstances, column-stabilized type semi-submersible platforms have proven advantageous for maintaining stable performance and bearing a heavier deck load. In addition to oil drilling rigs, crane barges and hotel barges with a deck load of more than 4,000 tons have been put into operation (Ref.2). Such a type of semi-submersible platform can be designed to optimize its motion characteristics, especially heave, in the design wave by selecting the preferable dimensions of columns and lower hulls in order to maintain a highly efficient operation capability.

When designing a semi-submersible platform, safe operation in a hostile environment as well as maintaining consistent operation are extremely important. A semi-submersible platform is subjected to extreme disturbances of wind, wave and current in its operation areas. Ideally, their effects should be discussed in such combined environments, and the interaction between the hull and mooring lines should also be considered. However, in realistic design, the governmental authorities and classification societies (Ref.3) permit us to assume the effect of wind and current as steady disturbances put on a hull by taking into account the fruits of recent researches. The design criteria submitted by the authorities may become a little conservative due to the paucity of information on the dynamic behaviors of semi-submersible platforms which should be carefully investigated in detailed considerations.

It is indicated, in fact, that the semi-submersibles with large lower hulls come to be steadily tilted by the tilt moment induced by the second-order wave force, and to be oscillated around the steady tilt angle of roll. This steady tilt is deeply related to the choice of the design criteria so that it brings about a decrease of under-deck clearance. Further, the low frequency motion induced by the non-linear force, namely the second-order wave force, effects a vital relation to the design of the mooring lines, too.

Numata et al.(Ref.4) indicated the existence of steady tilt in regular wave which amounted to 10 through 15 degrees in its amplitude for a semi-submersible platform with a smaller GM, and it was certified by the theoretical calculation. Further, they carried out model experiments in irregular waves to make clear the effect of steady tilt, and indicated that the experimental results showed lower under-deck clearance when compared with the prediction derived from the motion response in regular wave. Martin and Kuo (Ref.5) calculated the steady tilt moment precisely by considering the effect of the motions of a hull, and discussed the stable tilting of semi-submersible platforms.

However, regarding the behavior of a semi-submersible platform in irregular waves which bring about the non-linear response

characteristics to be connected with the steady tilt, any realistic solution has not yet been established to be applied for a practical design. As mentioned previously, the phenomenon of steady tilt can not be ignored in order to keep the under-deck clearance and to judge the submergence of deck under water on the survival environment. Further, when evaluating the external force acting on the mooring lines, consideration of the steady tilt may be necessary.

## 2. SCOPE OF WORK

Floating structures at seas are oscillated due to the wave exciting forces of various orders. In general, significant motion characteristics of a floating structure are subjected to first-order wave forces. When the floating structure is highly tuned in motion characteristics, however, a significant motion is possibly induced by the higher-order wave forces at a resonant frequency of the motion, even if the first-order wave forces are small at the resonant frequency.

The low frequency drifting motion of a moored structure, for instance, has been observed in irregular waves (Refs.6 and 7). The structure softly moored by chains or cables usually has a longer natural frequency in lateral motion due to the smaller restoring force in such a direction of motion. In this case, the lateral motion becomes highly tuned, and the second-order wave force induces a large lateral motion at the resonant frequency of motion even when random waves encountered do not include the component wave of low frequency to be resonant.

Such a behavior could be found in the other modes of motion when a floating structure is subjected to a highly tuned motion. Generally, semi-submersible platforms are designed with a large natural frequency of motion so as to avoid the resonance of motion with waves and to realize good performance at the significant range of wave frequency observed at seas. This indicates the possibility of the existence of highly tuned motions and low frequency motions. The low frequency motions are related to the natural frequency of motion and to the magnitude of the second-order wave forces.

Referring to the above discussion, the authors investigated the allowable under-deck clearance of a column-stabilized type semi-submersible platform in irregular beam seas. They considered that the low frequency motion in roll mode which has already been examined by Numata et al. (Ref.4) subjects to the second-order wave exciting roll moment. In such a case, the method is applicable with which the low frequency drifting motion of moored floating structures in irregular waves was discussed by Pinkstar (Ref.8).

The authors introduced a spectrum of low frequency roll moment using the steady

tilt moment which was obtained from the motion behavior in regular waves as subjected to the second-order wave force. The steady tilt moment induced by the second-order wave force in regular waves has been measured experimentally and theoretically by Numata and others (Refs.4 and 5). They have also suggested that at the present stage the theoretical calculation of the steady tilt moment could not be quantitatively agreed with experimental results.

Then the authors carried out the model experiments to investigate the steady tilt angles in regular waves, and obtained the steady tilt moment with respect to the wave frequency, for which an assumption was made that the steady tilt moment is equivalent to the righting moment at the corresponding steady tilt angle.

In the next stage, the spectrum of the low frequency roll motion was approximately obtained multiplying the roll response amplitude operator by the equivalent wave spectrum introduced by the low frequency roll moment spectrum. It was compared with the results of the model experiments in irregular waves. Prior to the above discussion, a simplified process applied to the regular wave groups was tried to confirm the validity of the method developed here, and was compared with the experimental results in the mentioned wave, too.

In the final stage, the spectrum of relative displacement was obtained superposing low frequency roll motion spectrum over ordinary relative displacement spectrum. This spectrum has double peaks at low and high frequencies corresponding to the natural frequency of roll motion and the significant wave frequency, respectively. Then the spectrum was statistically processed to obtain the expected values at arbitrary oscillation numbers of random waves. The predicted results in relative displacement were compared with the results obtained from the model experiments in irregular waves.

## 3. MODEL EXPERIMENTS

The semi-submersible platform tested here is of a column-stabilized type, having two lower hulls and eight columns and being symmetrical lengthwise with respect to the midship. The dimensions and rough sketches are shown in Fig. 3.1. A model of it was made to a scale of 1/90. All the figures shown in this paper, including tables and figures, are of full-scale values or non-dimensional to facilitate understanding.

A vertical gyroscope and acceleration meters were set on the top of the model to measure the angular and linear displacements, respectively. To measure the relative displacement from wave elevation to the under-deck, three sets of wave height probes were set at the lee side (44.80m from the center plane), mid part (11.70m lee side from the center plane), weather side (44.80m from the center plane).

Model experiments were carried out in the small towing tank (100m long, 5m wide and 2.15m deep) of the Akishima Laboratory, Mitsui Engineering & Shipbuilding Co.. As shown in Table 3.1, the model was adjusted at a draft with two different metacentric heights. Prior to the experiments in waves, righting moment was measured as shown in Fig. 3.2, and free roll and heave motions were checked, too.

At experiments in waves, long term measurements were necessary to measure the low frequency roll motion which is expected to happen in the experiments. Therefore the model was drifted leeward and the towing carriage followed the model at the same speed in order to measure the incident wave

at the position of the model.

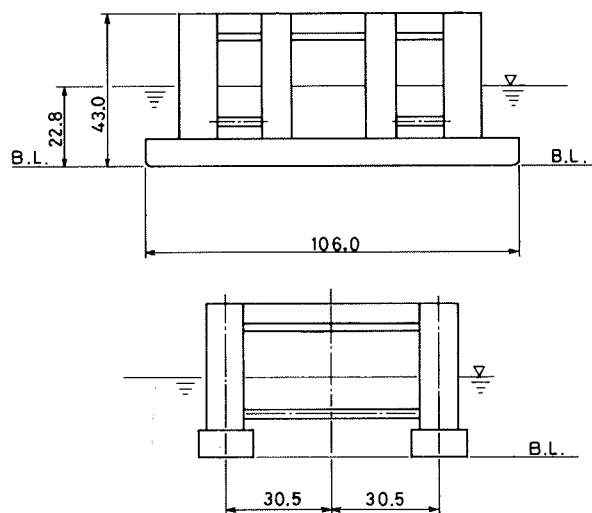
Experiments were carried out both in regular and irregular beam seas. The experiments in regular waves were aimed to obtain the steady tilt induced by waves and the response amplitude of various components of motion. The waves generated in the model tank have a period of 6 to 27 seconds and a steepness of 1/15 to 1/50. Included in the experiments in regular waves, a number of regular wave groups were generated to verify the occurrence of low frequency roll motion. Each elementary wave has a period of around

Table. 3.1 Testing conditions

|                                      |       |             |
|--------------------------------------|-------|-------------|
| Draft                                | (m)   | 22.8        |
| Displacement                         | (ton) | 36500       |
| CG above Base Line                   | (m)   | 22.6 / 21.0 |
| Radius of Gyration in Roll Direction | (m)   | 30.5 / 31.1 |
| Transverse Metacentric Height        | (m)   | 0.4 / 2.1   |
| Natural Period                       | (sec) |             |
| : heave                              |       | 23          |
| : roll                               |       | 152 / 66    |

Table. 3.2 Characteristics of irregular waves

| Series No. | Significant Wave Height (m) | Mean Wave Period(sec) |
|------------|-----------------------------|-----------------------|
| 1          | 4.0                         | 9.8                   |
| 2          | 6.3                         | 11.9                  |
| 3          | 8.8                         | 14.0                  |
| 4          | 11.6                        | 16.1                  |



DIMENSIONS in meters

Fig. 3.1 A semi-submersible platform

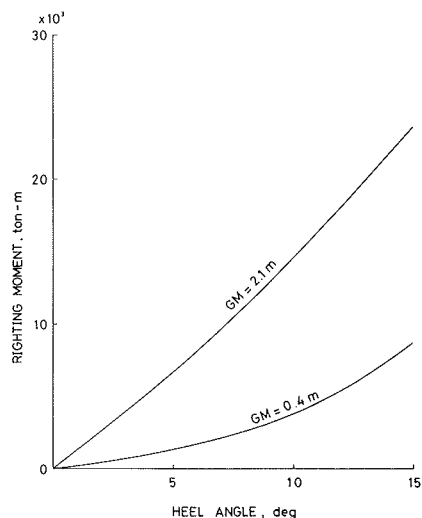


Fig. 3.2 Righting moment curves

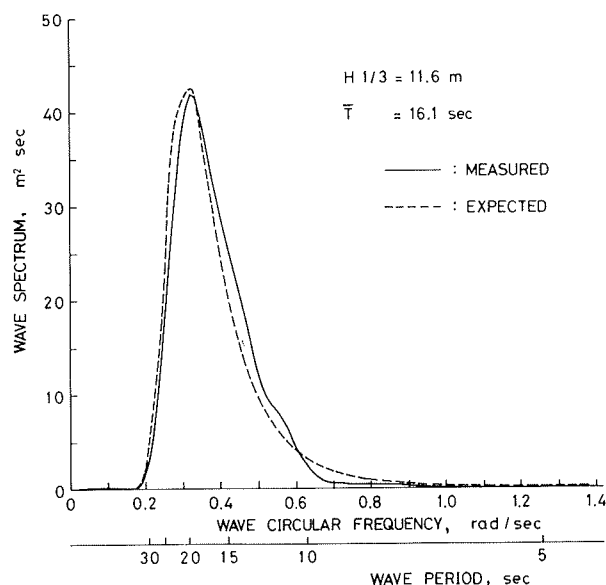


Fig. 3.3 Wave spectrum

8 seconds and the frequency of beat is approximately equal to the resonant frequency of roll motion. The experiments in irregular waves were carried out using four kinds of wave spectra of Pierson-Moskowitz type, significant wave heights and mean wave periods of which are presented in Table 3.2, and the spectra expected and measured are shown in Fig. 3.3.

All the measured data were recorded on magnetic tapes and were processed by the electronic computer.

#### 4. STEADY TILT IN REGULAR WAVE

A roll motion record in regular beam seas is shown in Fig. 4.1. This figure indicates that the model is forced to incline to leeward with some steady tilt angles so as to be oscillated around the steady tilt position, while steady displacement could not be seen in the heave motion. This suggests the existence of steady tilt moment due to the second-order wave force, too. It is confirmed from the above discussion that the column stabilized type semi-submersible platform is subjected to a steady tilt phenomenon in regular waves. Fig. 4.2 shows the experimental results in which the steady tilt angle depends on wave height, wave period and metacentric height.

As illustrated in Fig. 4.3, the steady tilt moment should be equal to the righting

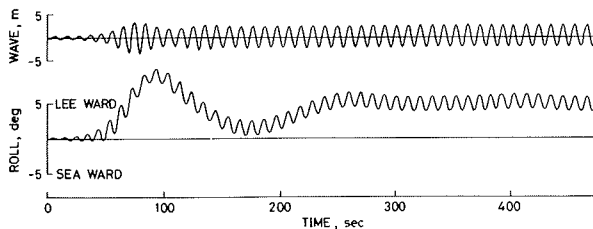
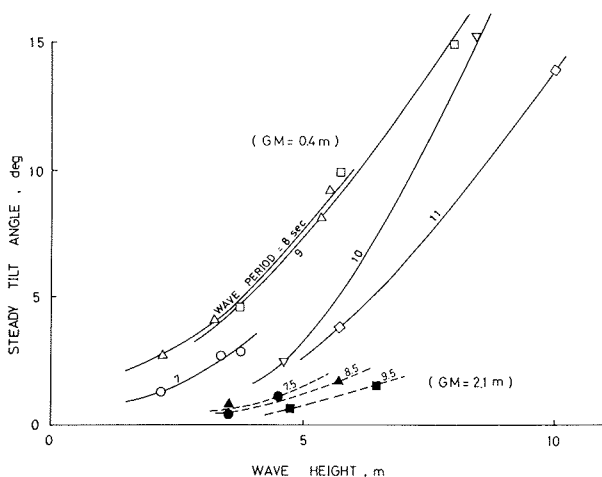


Fig. 4.1 Roll motion recording in regular wave



Unfortunately the results of calculation shown in Fig. 4.4 do not quantitatively agree with experimental ones in this case. However, it was encouraging that the calculation showed a similar tendency with experiments so that the steady tilt moment coefficient,  $R$  changes with respect to wave periods having a peak of around 8 seconds. Regarding the experimental results, the second peak at 22 seconds may arise due to the coupling effect of heave motion.

For reference, the motion responses of heave, roll, sway and relative displacement are presented in Figs. 4.5 through 4.10. Compared with the experimental results, the calculated ones are in good agreement.

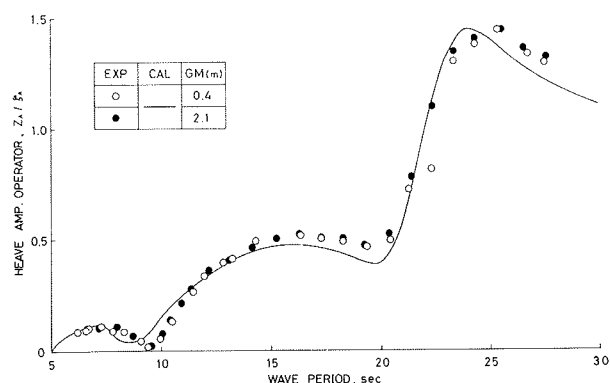


Fig. 4.5 Heave amplitude in regular wave

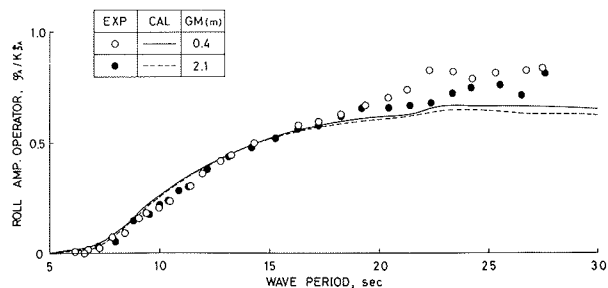


Fig. 4.6 Roll amplitude in regular wave

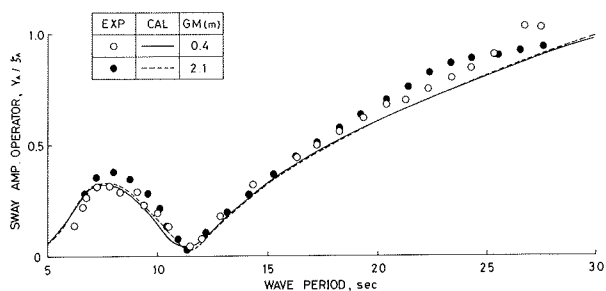


Fig. 4.7 Sway amplitude in regular wave

## 5. LOW FREQUENCY ROLL MOTION IN IRREGULAR WAVES

### 5.1 Tilt moment in regular wave groups

In order to verify the aforementioned analogy, the tilt moment in regular wave groups (Ref.10) were calculated for comparison with the experimental result.

A regular wave group consisting of two regular waves with slightly different frequencies are written as

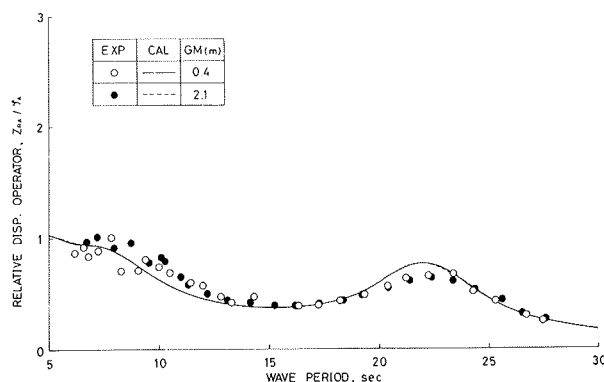


Fig. 4.8 Relative displacement amplitude in regular wave (lee side)

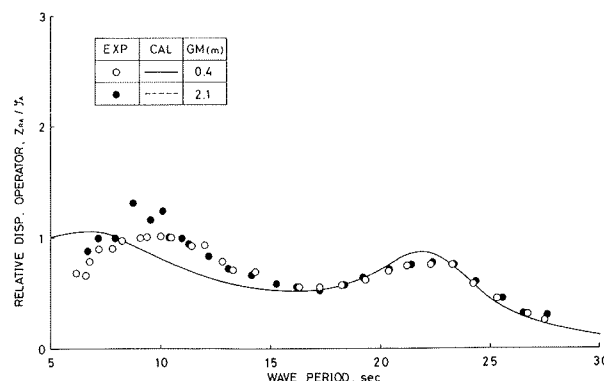


Fig. 4.9 Relative displacement amplitude in regular wave (center)

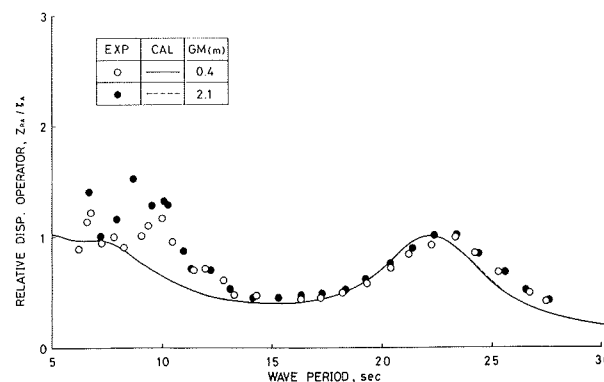


Fig. 4.10 Relative displacement amplitude in regular wave (weather side)

$$\begin{aligned}\zeta &= \zeta_0 \cos \omega_1 t + \zeta_0 \cos \omega_2 t \\ &= \zeta_A \cos \frac{\Delta \omega}{2} t \cos \omega t\end{aligned}\quad (3)$$

where

$$\begin{aligned}\zeta_A &\equiv 2\zeta_0 \\ \Delta \omega &\equiv \omega_1 - \omega_2 \\ \omega &\equiv (\omega_1 + \omega_2)/2\end{aligned}$$

The steady tilt moment in such a wave train can be calculated using the steady tilt moment coefficient obtained in regular wave.

$$M_T = \frac{1}{2} R^2(\omega) \rho g L B \zeta_A^2 (1 + \cos \Delta \omega t) \quad (4)$$

The AC component in equation (4) is approximately equal to the roll moment induced by a regular wave which has an amplitude of

$$\frac{1}{2} R^2(\omega) \rho g L B \zeta_A^2 / G_1(\Delta \omega)$$

and a circular frequency of  $\Delta \omega$ .  $G_1$  is a roll moment operator in a regular wave and is obtained from the theoretical calculation (Ref.11).

Then the roll motion in a regular wave group can be calculated by the simultaneous equations of motion in sway and roll modes. Here the wave exciting sway force is put at

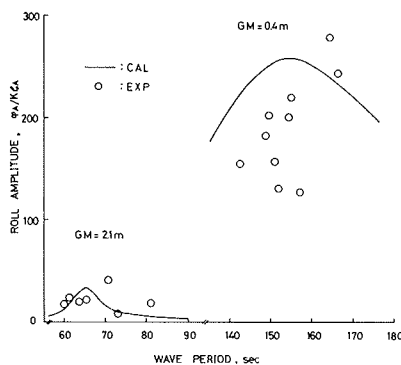


Fig. 5.1 Roll amplitude in regular wave group

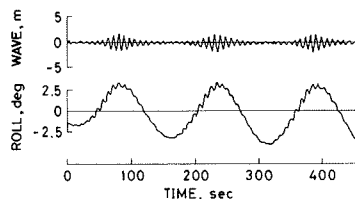


Fig. 5.2 Roll motion recording in regular wave group

zero because the AC component in sway mode could be disregarded. In this calculation, the natural frequency and damping coefficients obtained from the free roll motion experiment were used. The results of calculation are shown in Fig. 5.1 as compared with the experimental ones. In this figure, the experimental values are scattered around the calculated ones depending on the resonant frequency of roll. This indicates that equation (4) will be established in case of the low frequency roll motion.

A time history of a regular wave group and the corresponding roll motion obtained from the model experiments are also shown in Fig. 5.2.

## 5.2 Low frequency roll motion in irregular waves

Applying the method which has been developed by Pinkster (Ref.8), the low frequency tilt moment spectrum in irregular waves can be written as

$$S_{TM}(\omega) = 8(\rho g L B)^2 \int_0^\infty S_W(\mu) S_W(\mu + \omega) R^4(\mu + \frac{\omega}{2}) d\mu \quad (5)$$

In this formulation,  $R$  is the steady tilt moment coefficient shown in Fig. 4.4. Calculations were actually carried out in accordance with equation (5) as shown in Fig. 5.3.

The equivalent wave spectrum,  $S_{EW}(\omega)$  (Ref.12) which induces the roll moment corresponding to the low frequency tilt moment, is expressed as

$$S_{EW}(\omega) = S_{TM}(\omega) / [G_1(\omega)]^2 \quad (6)$$

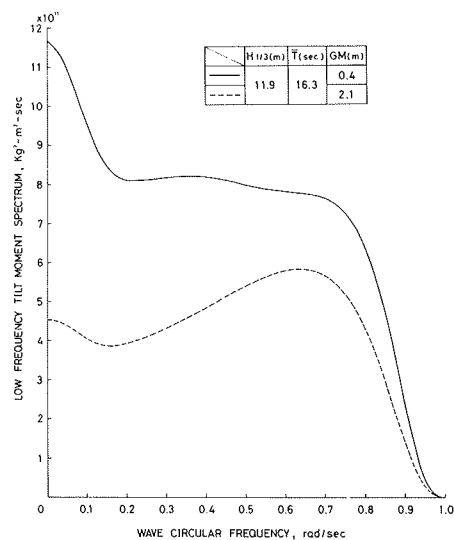


Fig. 5.3 Low frequency tilt moment spectrum

Fig. 5.4 shows the result of calculation, in which the equivalent wave spectrum has larger power than that of the wave obtained from experiments at the region of low frequency. Here the roll motion spectrum can be easily calculated by applying the above equivalent wave spectrum and the roll amplitude operator  $G'(\omega)$ .

$$S_{RL}(\omega) = S_{EW}(\omega) [G'(\omega)]^2 \quad (7)$$

In this equation,  $G'(\omega)$  is calculated by simultaneous equations of sway-roll motion in the same manner with which the roll moment was previously calculated (Ref.11), and where the wave exciting sway force is disregarded. Fig. 5.5 shows the results of calculation on  $G'(\omega)$  as compared with the roll amplitude operator  $G(\omega)$ .  $G'(\omega)$  is highly tuned around the natural frequency of roll, while  $G(\omega)$  is not significant as compared with  $G'(\omega)$ . This indicates that the wave exciting sway force was assumed to be null in  $G'(\omega)$ , and the

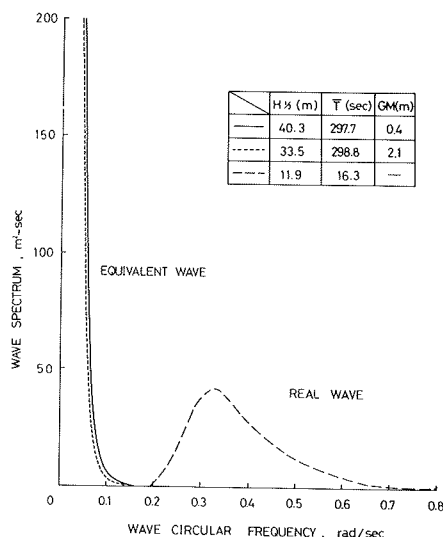


Fig. 5.4 Equivalent wave spectrum

coupled effect of sway and roll motions is less in  $G'(\omega)$  than in  $G(\omega)$ .

On the other hand, the roll motion spectrum when only considering the first-order wave exciting roll moment is

$$S_{RH}(\omega) = S_W(\omega) [G(\omega)]^2 \quad (8)$$

Accordingly the roll motion spectrum in irregular waves can be written using equations (7) and (8).

$$S_R(\omega) = S_{RL}(\omega) + S_{RH}(\omega) \quad (9)$$

Of course since the roll damping force is non-linear, the roll amplitude operator  $G'(\omega)$  depends on wave heights. When calculating  $G'(\omega)$ , therefore a wave steepness was reasonably defined as (significant wave height / 1.56 (mean wave period)<sup>2</sup>), while a wave steepness of 1/50 was used for obtaining  $G(\omega)$ .

In Fig. 5.6 the result of calculation on roll motion spectrum is shown as compared with the experimental one. As seen in the figure, the calculated spectrum is in good agreement with the experimental results.

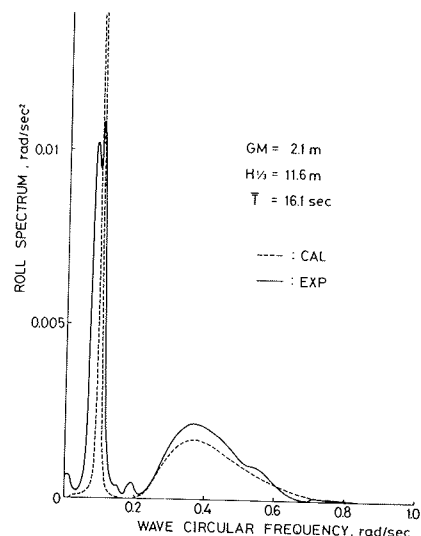


Fig. 5.6 Roll motion spectrum

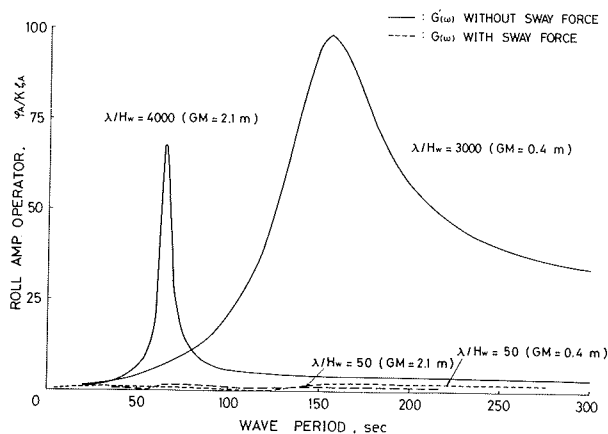


Fig. 5.5 Roll amplitude

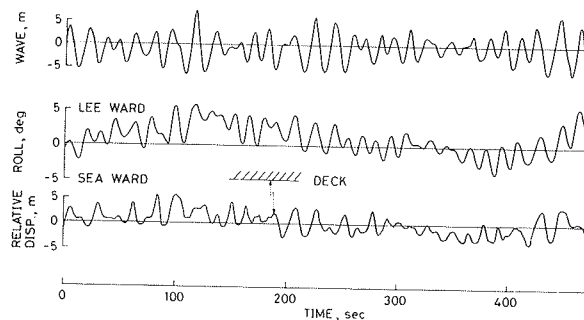


Fig. 5.7 Roll motion recording in irregular waves

A time history of irregular waves, the corresponding roll motion and relative displacement obtained from the model experiments are also shown in Fig. 5.7.

## 6. CALCULATION OF UNDER-DECK CLEARANCE

### 6.1 Relative displacement

In general the relative displacement at an arbitrary position of the platform is calculated by heave, roll motion and wave elevation taking into account the phase differences among these components. However, the low frequency motion is discussed here. In the region of low frequency, roll motion is only predominant while heave motion is almost null, and the low frequency wave components are non-existent. As discussed in Section 5, the

spectrum of the low frequency relative displacement can be processed only by the low frequency roll motion spectrum. Then the overall spectrum of relative displacement is expressed as follows,

$$S_{RD}(\omega) = 1^2 S_{RL}(\omega) + S_W(\omega) [G_{RD}(\omega)]^2 \quad (10)$$

where  $G_{RD}$  is the amplitude operator of the relative displacement for real irregular waves.

The calculated results of the relative displacement spectrum are shown in Figs. 6.1 and 6.2 as compared with the experimental ones. The calculated spectra agree approximately with the experimental results, especially at the region of low frequency.

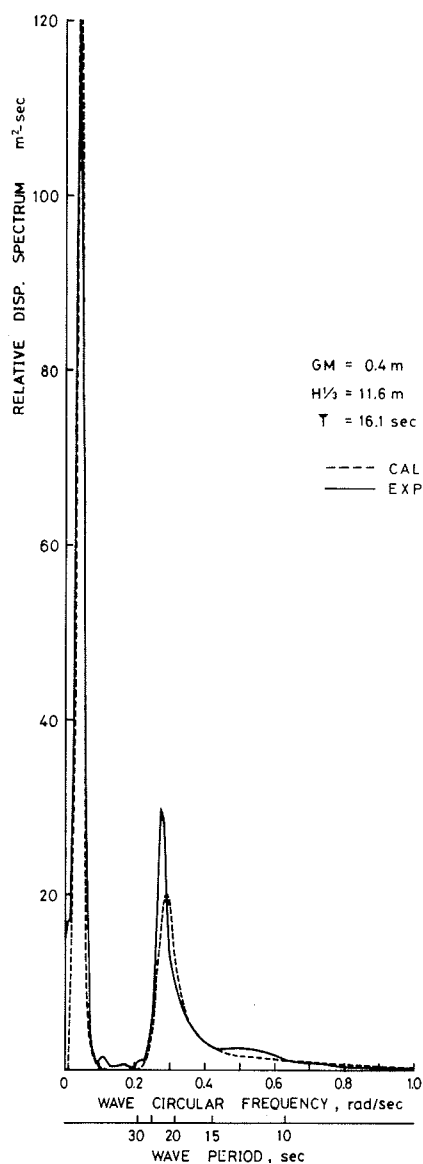


Fig. 6.1 Relative displacement spectrum (GM=0.4m)

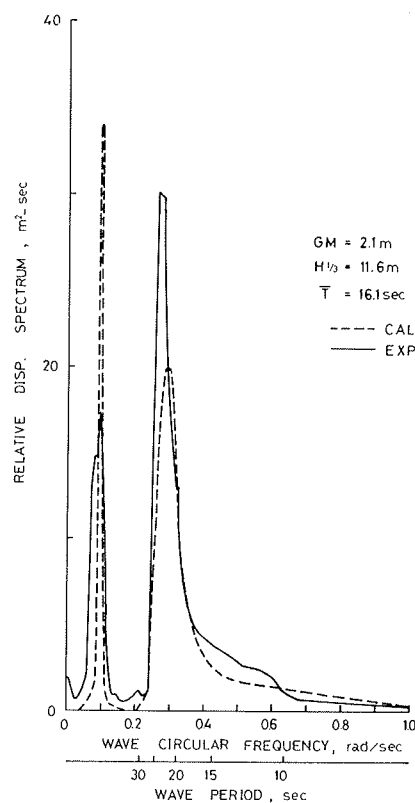


Fig. 6.2 Relative displacement spectrum (GM=2.1m)

### 6.2 Estimation of the expected values

The relative displacement spectra obtained in Figs. 6.1 and 6.2 have double peaks at low and high frequencies, respectively. In such a case, a conventional statistic processing is not useful. Then the authors developed a method in the assumptions that :

- low frequency roll motion is independent of high frequency one without the relative interference in phase, and



b. the spectrum at the roll resonant frequency predominates at the region of low frequencies.

The detailed discussion of introducing the method developed here is made in the Appendix.

The results of computation in the expected values of roll motion are shown in Fig. 6.3 in good agreement with the experimental ones.

In the same manner, the expected values of the relative displacement were calculated as shown by the solid line in Fig. 6.4. However, there exists a discrepancy with the experimental results. The reason for this will be found in the record of the relative displacement shown in Fig. 5.7.

Carefully examining the record, the relative displacement is asymmetric with respect to the still water level. It seems to correspond to the asymmetric wave elevation. Thus the following consideration could be made: When the positive displacement (to decrease the under-deck clearance) is noted by  $\zeta_{RD}^{(+)}$  and the negative one by  $\zeta_{RD}^{(-)}$ , the following relation is found from the record of relative displacement.

$$\frac{1}{N} \sum_{i=1}^N [\zeta_{RD}^{(+)}]^2 > \frac{1}{N} \sum_{i=1}^N [\zeta_{RD}^{(-)}]^2 \quad (11)$$

while the following is approximately kept.

$$\frac{1}{N} \sum_{i=1}^N \zeta_{RD}^{(+)} \approx \frac{1}{N} \sum_{i=1}^N |\zeta_{RD}^{(-)}| \quad (12)$$

The relation (11) indicates that the standard deviation of  $\zeta_{RD}^{(+)}$  should be defined ignoring  $\zeta_{RD}^{(-)}$  to obtain the realistic values of the under-deck clearance. In Fig. 6.4 the result of the revised computation based on the experimental data was also shown by the chain line in good agreement with the experimental plots.

Thus the under-deck clearance could be theoretically predicted by the method developed here with the effect of low frequency roll motion. However, non-linear oscillations of the incident waves which are probably related to the non-linear relative displacement, could not be treated theoretically. Coming back to Fig. 6.4, the present method is not necessarily satisfactory, while the linear theory prediction shown by the dotted line is never a safe estimation of the under-deck clearance. Consequently it is needed at the present stage to keep a safety margin of several ten percent for predicting the under-deck clearance, even if the present method is used.

## 7. CONCLUDING REMARKS

To estimate accurately the under-deck clearance of a semi-submersible platform in waves is directly related to safety of operation at sea.

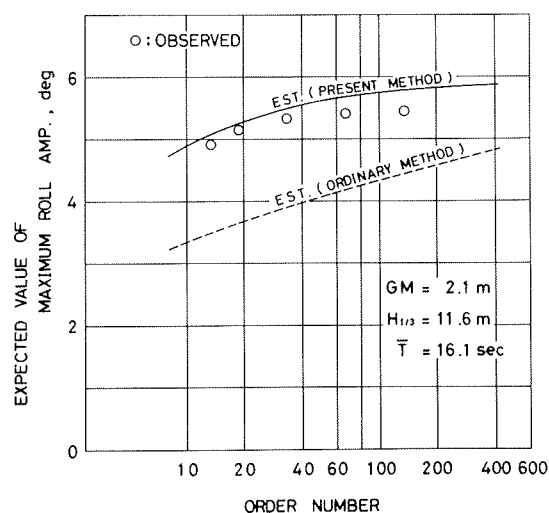


Fig. 6.3 1/n-th highest expected values of roll motion

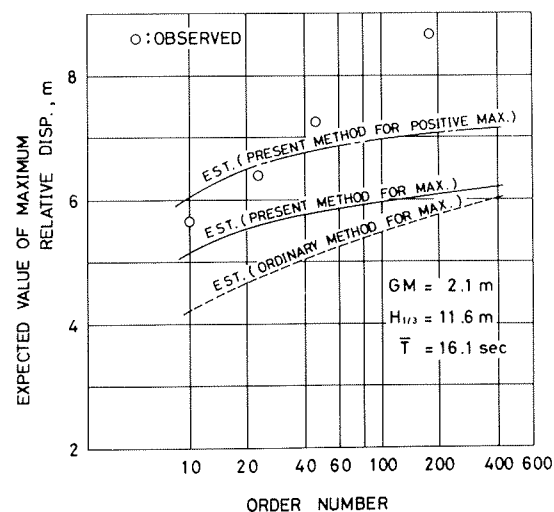


Fig. 6.4 1/n-th highest expected values of relative displacement

The objective of this research was to identify the method which predicts the allowable under-deck clearance of a semi-submersible platform in irregular waves, especially in cases that are subjected to the non-linear effect due to the second-order wave force in roll motion.

The main conclusion is that the non-linear effect of roll motion processed in the present method gives a good estimation of the roll motion as compared with the linear response assumption. However, satisfactory results could not be necessarily obtained in the estimation of the under-deck clearance due to the lack of introducing the non-linear effect of irregular waves while

the non-linear effect of roll motion was successfully introduced in the present method. Here an interesting treatment in statistical analysis was tried to obtain the expected value of a spectrum having double peaks.

Further investigations are necessary in theoretical aspects. However, the second-order wave force is relatively small compared with the first-order hydrodynamic forces to be quantitatively processed, while it affects the motion behavior when a semi-submersible platform has highly tuned motion characteristics. In fact, the theoretical calculation expressed by Numata et al. (Ref.4) could not necessarily explain the result of the present experiments at small wave excitation.

This indicates that detailed theoretical modelling is necessary including the viscous effect and the interaction among under-water structures of a semi-submersible platform.

#### ACKNOWLEDGEMENT

The authors are indebted to The Offshore Co. (presently SONAT) for having provided the opportunity to investigate the bi-stable roll phenomenon of a semi-submersible platform, JOHN SHAW which has been constructed by Mitsui Engineering & Shipbuilding Co..

Dr. Y. Yamanouchi, the General Director of Akishima Laboratory encouraged us to present the paper to this conference. He and Dr. M. Kobayashi of Akishima Laboratory discussed with us the theoretical profiles. The authors wish to pay sincere thanks to them.

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#### APPENDIX: EXPECTED VALUES OF THE SPECTRUM WITH DOUBLE PEAKS AT LOW AND HIGH FREQUENCIES

In order to obtain the expected value of motion whose spectrum has double peaks at low and high frequencies, respectively, the following assumptions were made.

- (1) Low frequency motion is independent of high frequency motion with respect to the phase lag between the two.
- (2) Spectrum of the roll resonant frequency predominates at the region of low frequency.

Thus motions in irregular waves can be divided into two components: a low frequency motion at the roll-resonant frequency and a high frequency motion at the wave frequency.

With these assumptions the low frequency motion displacements are subjected to the Gaussian distribution. Its probability density is expressed as

$$P(x;1)dx = \frac{1}{\sqrt{2\pi}} \exp(-\frac{1}{2}x^2)dx \quad (A.1)$$

where  $x = \xi_L / \sqrt{m_{0L}}$ ,  $\xi_L$  is a low frequency motion displacement and  $m_{0L}$  is the 0-order moment of the spectrum. From the equation (A.1), the  $1/n$ -th highest expected value,  $\bar{x}_{1/n}$  can be derived.

$$\bar{x}_{1/n} = n \int_{x_n}^{\infty} x P(x) dx \quad (A.2)$$

where

$$\int_{x_n}^{\infty} P(x) dx = \frac{1}{n}$$

Here the probability of exceeding an amplitude  $\xi_{Li}$  is  $p_{Li}$  and the probability that the value lies in the range ( $\xi_{Li}$  and  $\xi_{Li-1}$ ) can be denoted by  $q_{Li}$  ( $=p_{Li}-p_{Li-1}$ ).

On the other hand, the high frequency motion amplitude is considered being subjected to the following distribution with a band width parameter  $\epsilon$ . The probability density of the high frequency motion amplitude is expressed as

$$P(x;\epsilon) dx = \frac{1}{\sqrt{2\pi}} \left\{ \epsilon \exp\left(-\frac{x^2}{2\epsilon^2}\right) + \sqrt{1-\epsilon^2} x \exp\left(-\frac{x^2}{2}\right) \frac{x\sqrt{1-\epsilon^2}}{\epsilon} \int_{-\infty}^{\infty} \exp(-\alpha^2/2) d\alpha \right\} dx \quad (A.3)$$

where  $x = \xi_H / \sqrt{m_{0H}}$ ,  $\xi_H$  is a high frequency motion amplitude and  $m_{0H}$  is the 0-order moment of the spectrum.

In the same process as in case of the low frequency motion, the probability of exceeding  $\xi_{Hj}$  is  $p_{Hj}$  and that the value lies at the range ( $\xi_{Hj}$  and  $\xi_{Hj-1}$ ) is  $q_{Hj} (= p_{Hj} - p_{Hj-1})$ .

From the above, the probability of exceeding an amplitude, which is simultaneously arised by low and high frequency motions, could be found by searching the combination with the following relation.

$$\xi_K < \xi_{Li} + \xi_{Hj} \quad (A.4)$$

Then the probability of exceeding  $\xi_K$  is denoted as

$$p_K = \sum_{i=1}^m \sum_{j=1}^n q_{Li} q_{Hj} \quad (A.5)$$

where the indices of m and n indicate the range of combination satisfying the relation shown in (A.4). Here the probability that the value lies in the range ( $\xi_K$  and  $\xi_{K-1}$ ) is  $q_K (= p_K - p_{K-1})$ , and the 1/n-th highest expected amplitude can be expressed as follows,

$$\bar{\xi}_{1/n} = \frac{1}{\sum_K} \xi_K q_K / \frac{1}{\sum_K} q_K \quad (A.6)$$

where  $\frac{1}{\sum_K} q_K = 1/n$

## Discussion

N. Matsumoto (Tsu Research Laboratories, NKK, Japan)

Thank you for your excellent presentation. I have two questions.

(1) Long term measurements are necessary to measure the low frequency motion at experiments in waves. In that case, transverse waves usually become higher with time in a tank, and are recorded together with incident waves height. Did you adopt any analysis to separate this undesirable one from recorded data?

(2) You wrote that the calculated spectrum is in good agreement with the experimental results in Fig. 5.6.

However, at low frequency ( $\omega < 0.15$ ), experimental spectrum has wider range peak than calculated one and it has two peaks. Is there any cause explaining the difference?

### Author's Reply

The first your question is about the transverse wave observed through model experiment.

In our case, little transverse wave was observed. If any, it has a different frequency from the frequencies of motion. The transverse wave is probably separated in course of frequency analysis.

The second one is concerned with our Fig. 5.6.

Small double peaks appeared at low frequency is not clear to be explained. As concerns with wider band of spectrum measured comparing with calculation, there are two reasons.

The first one is due to the method of frequency analysis. The other is contained in the process of calculation. The spectrum involves an integration of the fourth power of R. This means there is a

little inaccuracy in numerical calculation.

G. Nakai (Kawasaki Heavy Industry, Ltd., Japan)

We have got deep interest in 4. STEADY TILT IN REGULAR WAVE because United Kingdom DOE (Department of Energy) by their guidance on stability requires that units should have GM large enough to withstand the wave induced tilting moment. And they propose the Dr. Numata's paper in which the calculation method of required minimum GM is shown.

According to your Fig. 4.4, the measured steady tilting moments are generally larger than the calculated moments which were obtained by following the procedure of Dr. Numata's paper.

Do you think the procedure of deciding the minimum GM would give the smaller values than required?

### Author's Reply

As you pointed out there is a discrepancy between the calculated and measured tilt moment in Fig. 4.4. Following this figure, measurement gives larger value than calculation. However, it will be risky to understand the required minimum GM changeable, because it was just a particular case.

Hopefully, realistic prediction will be established in theoretical aspect.

C. Kuo (University of Strathclyde, UK)

The problem of deck clearance is very important to the designer and the contributions of the authors are very important. I would like ask the opinion on two points:

Firstly, should there not be more parameters if we are to arrive at the option deck clearance? For example, we need to trade

air gap with the need to meet the strength of the deck structure.

Secondly, how do you think the results of your research can be used for practical design purpose? For example, steady tilt can be removed by increase the value of GM.

#### Author's Reply

In this research we have a simplified method of analysis to obtain a basic solution only on the effect of low frequency roll motion with a un-moored semi-sub.

Further investigations are necessary including the dynamics of wind and current and the effect of mooring system.

Cao Zhen Hai (China Ship Scientific Research Center, China)

The authors are to be congratulated for this comprehensive paper on the low frequency effect of roll on a semi-submersible platform. I am specially interested in the good agreement found between experimental results and the approximate calculated results on second order roll motions. In my paper on the capsizing of a certain jack-up rig, we found the same steady tilt in regular waves which I called drift in heel. Although in my case the phenomenon was complicated by the shipping of water on deck, we are also calculating the second order roll moment on the rig using a 3-D diffraction theory at the present, the results of which unfortunately are not available for this conference. I am encouraged by the results obtained by you and also by the same kind of experimental results obtained by Dr. Takarada et al. However, my opinion would be that a 3-D diffraction theory approach would be more appropriate in doing the calculations, though the computer time is high.

Finally I would suggest that a standard term should be agreed internationally. Steady tilt, drift in heel (my paper) and pseudo-static heel angle, in fact all means the same thing.

#### Author's Reply

We agree with your opinion. Three dimensional diffraction theory could be applied at primary stage. But the final goal will be hopefully to obtain the non-linear equations of motion of six freedoms taking into account of the viscous damping, in which higher-order wave forces could be reasonably processed.

E. Numata (Davidson Laboratory, USA)

Reference 4 showed that linear superposition procedures underpredict the relative displacement between the deck edge of a semisubmersible platform and the surface of long-crested irregular waves. The authors of the present paper have significantly improved the prediction of relative displacement by using a suitable combination of theoretical and experimental procedures to treat non-linear aspects of platform motion responses.

A key step in the procedure is the use of an experimentally determined coefficient  $R$  to calculate a "tilt moment" spectrum, Equation (5). Experimental determination of tilt moment for the authors' semisubmersible configuration required extensive model testing with GM and wave steepness as parameters, Figure 4.4. Since the moment spectrum involves an integration of the fourth power of the experimentally-determined  $R$ , the success of the authors' roll prediction method must be due to their careful measurements of tilt moment. Do the authors believe that a semisubmersible design must be model tested to obtain  $R$ , or is there hope of developing a generalized formulation for  $R$ ?

We look forward to further developments in treating the effects of non-linear irregular waves in calculating the relative displacement of the deck of a semisubmersible. Safe operation of these large, costly vessels in rough seas will be enhanced by results of research such as reported by the authors.

#### Author's Reply

Mr. Numata is one of the pioneers who indicated the phenomena of steady tilt on semi-subs.

To obtain theoretically the tilt moment is really complicated. The center of force is not determinant even if the amount of horizontal and vertical forces could be obtained.

In addition, as Ogilvie has pointed out (Reference 9), non-linear effect should be considered when lower hulls are moving near the surface of water.

At present stage, however, many have obtained the drifting force successfully by potential theory. Maybe the exact formulation including viscous effect will lead to the final goal.

## A COMPARISON OF STABILITY CHARACTERISTICS OF SHIPS AND OFFSHORE STRUCTURES

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### ABSTRACT

Most of the established stability rules and criteria deal with both ships and floating offshore platforms by using similar physical considerations. As a result of the wide diversity of geometric forms, however, both the static and the dynamic response characteristics of offshore platforms differ in several essential respects from those of conventional ships. The objective of this paper is to point out some of these differences, especially as they may affect the ship or platforms' survivability in both a static and dynamic environment. From these results may be deduced some suggestions for modifying the focus of stability criteria and needed areas for research if more rationally-based criteria are to be developed.

### 1. INTRODUCTION

Many of the established stability rules and criteria which are in common use for the regulation or assessment of the stability of either ships or offshore structures are strikingly similar, and important features of each can often be traced to the classic work of Rahola, Reference [1]. The similarities in the rules become somewhat puzzling when one considers the great differences between both the principal geometric features and the related static and dynamic response characteristics of the two types of floating structures.

These stability rules can usually be characterized as falling into one of two categories: those based upon the Rahola criterion which specifies the GM, angle of occurrence and value of the maximum righting arm, and dynamic stability, but

which make no explicit reference to any disturbing influence; and wind heel criteria, in which the static or dynamic heeling moment caused by wind is related to the righting moment. The U.S. Navy dynamic wind heel criterion, Reference [2], is an example of the latter type of rule. It should be noted that the concept of a wind heeling moment which is applied as a stepwise function of time was included in Rahola's original analysis, but the stability criteria which he proposed did not contain the wind heel in an explicit form.

Intact stability criteria for floating offshore platforms, as required by many of the major classification societies, for example, References [3], [4], and [5], are generally similar to each other and to the U.S. Navy wind heel criterion. Essentially, these rules state that the dynamic stability must exceed the heeling work done by a stepwise wind heeling moment up to a limiting angle determined by downflooding or other considerations. The factor of exceedance must be either 30 percent in the case of semisubmersibles or 40 percent in the case of most other types of platforms. It should be noted that the factor of 40 percent is used in the U.S. Navy criterion where the rationale for the value is that it allows a margin for computational inaccuracies and the effect of wind gusts. In addition, the Navy criterion requires that the wind heeling work and dynamic stability be based upon an initial heel of 25 degrees into the wind as an approximate means of including the effect of wave-induced rolling. The rules for offshore platforms are usually based upon an initial condition of zero heel angle.

In view of the pronounced differences in geometrical configuration, and the

related differences in dynamic response to disturbing influences such as wind and waves of ships and platforms, we might expect somewhat greater differences in stability criteria for the two types of floating structures. Some of these differences between ships and platforms are as follows.

Ships are long in relation to their transverse and depth dimensions. Platforms may be equal in width and length but of much smaller depth (jackup rigs), or they may consist of a space frame assembly of cylindrical members (semisubmersibles), in which case, the length, width and depth dimensions may be of similar magnitude.

Ships have a relatively large range of stability, but low righting arms over the entire range. Platforms, especially the barge and jackup types, often have high righting arms over a relatively short range of stability.

The natural period of roll of a ship is typically in the range of 10 to 20 seconds, that of a semisubmersible is typically 30 to 40 seconds. As a result, the ship natural period of roll may fall within the range of appreciable energy content of a realistic seaway and resonant amplification of roll response may occur. For the semisubmersible this seldom occurs, and in fact, such detuning of roll response is a design criterion for semisubmersibles.

The speed and heading of the ship relative to the waves strongly influence the frequency of wave encounter. Thus, for a ship which is free of resonant roll amplification in one condition of operation, a change to another heading or speed may induce the possibility of roll resonance. The platform, being immobile, encounters waves only at the wave frequency and this is usually well above the resonant frequency.

The form of the above and below water parts of the ship hull are such that the transverse stability characteristics may be strongly modified by the presence of waves having crests normal to the ship centerline. This effect, which is almost purely geometric in origin, results in a reduction of stability when the crest of a wave is at amidships, while a trough at amidships causes the stability to be increased as compared to still water. Under certain combinations of ship speed and heading, this wave-induced stability variation may lead to an unstable rolling motion and even to capsizing as described in Reference [6].

In the following sections, we will briefly investigate the manner in which some of these differences between ships and semisubmersible platforms may influence the intact stability. Our ultimate objective is to point out some areas in which it may be fruitful to conduct research intended to produce more rational and consistent platform stability criteria.

## 2. STATIC STABILITY IN STILL WATER AND IN FOLLOWING WAVES

As noted in the INTRODUCTION, the profound differences in geometry between ships and semisubmersible platforms result in markedly different stability characteristics of the two types of structures. Figure 1 illustrates a composite semisubmersible having a simplified form but principal dimensions somewhat similar to one described in Reference [7]. Static stability curves for this platform, computed under several different conditions are shown in Figure 2. The two curves labelled "still water" were obtained assuming the tops of the eight vertical column members to be 112.5 feet and 130 feet above the baseline respectively, and are intended to illustrate the effect of this dimension. It should be noted that, in order to satisfy certain recently stiffened platform damaged stability requirements for North Sea and other severe areas, the trend has been to design the upper deck volume of newer platforms to be a watertight and buoyant portion of the structure. This has not been followed in the platform of Figures 1 and 2, and the results may be taken as representative of current design practice for less severe operating areas.

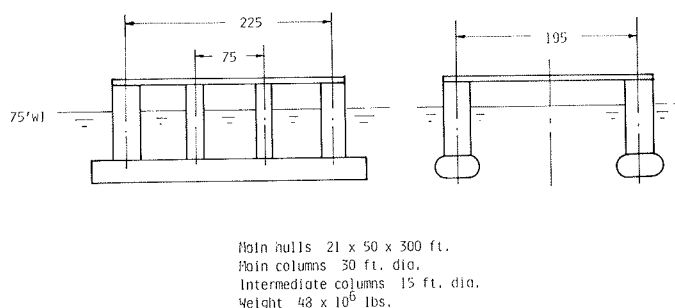


Fig. 1 Twin hull semisubmersible.

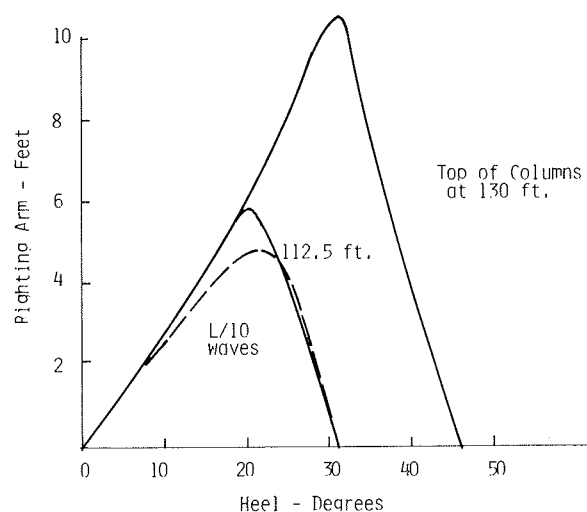


Fig. 2 Static stability in calm water and waves for example semisubmersible GM = 14 ft.

Next consider Figure 3 which illustrates the righting arm curves for a typical large container ship. In this figure, the solid curve corresponds to the righting arm in calm water. Above and below this are two dashed curves representing the righting arms computed respectively for the ship in an assumed position of static equilibrium with a wave trough at amidships and a wave crest at amidships. Referring again to Figure 2 for the platform, we see in the vicinity of the still water righting arm for the 112.5 foot deck height, that there are two nearly coincident curves giving the righting arms for this platform corresponding to the wave crest and wave trough at the amidships position. The wave effect on stability is clearly much less pronounced than it is for the ship.

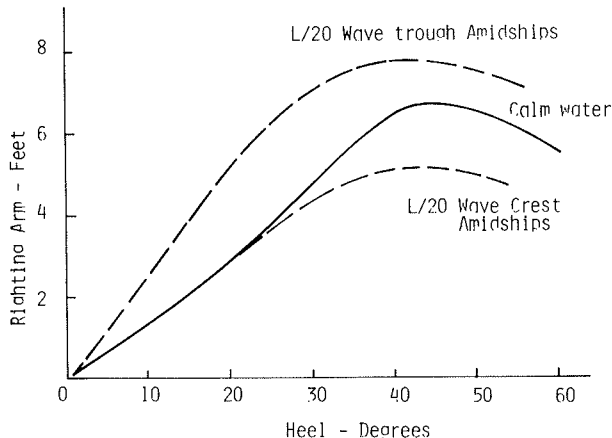


Fig. 3 Static stability in calm water and waves for 900 ft. LBP container ship.

A rather unpleasant consequence of the wave-induced variation of righting arm in the case of the ship may occur if the ship is operating at a particular speed in following or quartering seas, and this is described in Reference [6]. It is found that if the frequency of wave encounter is in the vicinity of twice the natural roll frequency, an unstable rolling motion may occur even though the ship operates in following seas so that there is no wave induced rolling moment. A severe case of this motion instability is illustrated in Figure 5 for a ship whose stability in still water and waves is shown in Figure 4. The solid curve in the upper part of Figure 5 is a graph of the roll motion versus time measured during a towing tank test in which a model was towed in following waves. The broken curve gives the results of a numerical simulation of the motion in the same conditions of speed and waves as those of the model test. In this simulation, the nonlinear equations of large-amplitude motion, including the wave-induced stability variations, were numerically integrated. The curves in the lower part

of the figure represent the trace of the encountered waves.

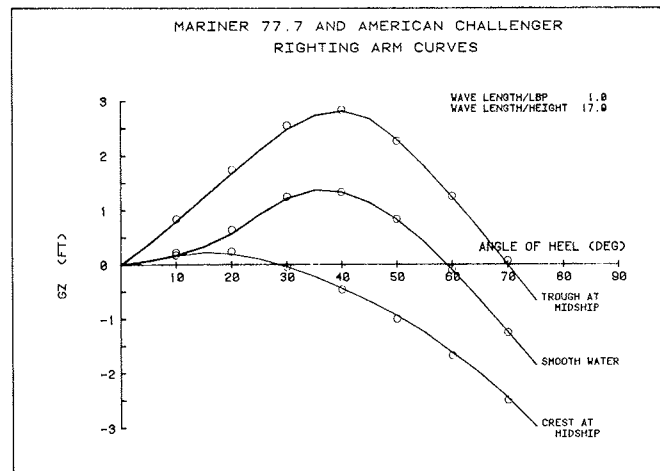


Fig. 4 Righting arm curves for ship of Figure 5.

The roll motion takes place at one-half the encounter frequency, and is seen to build up until the model capsizes at about 80 seconds into the run. As shown in Figure 4, the effects of waves on the stability of this model were quite pronounced and the initial stability was somewhat less than would be considered the usual operating practice for such a ship. Nevertheless, such unstable motion has been shown to occur for realistic conditions, usually with less extreme consequence than is illustrated in Figure 5.

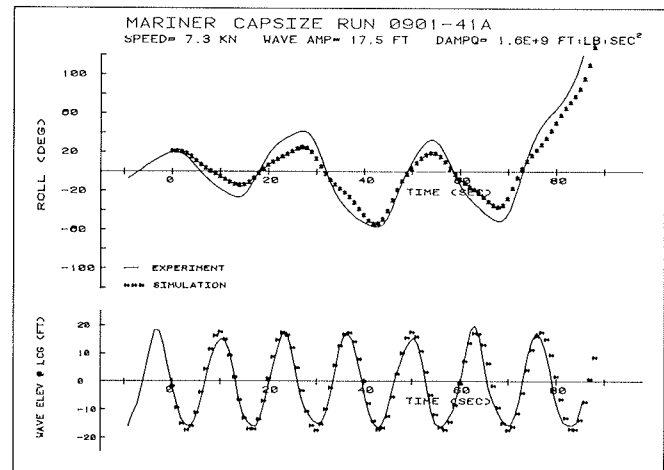


Fig. 5 Experimental and numerical simulation of ship roll in following waves.

In order for this motion instability to occur, two ingredients are necessary. First, the encountered waves must cause a pronounced variation in the stability between the crest amidships and trough amidships condition. Second, the ship must be moving at such a speed and on such a heading as to cause the frequency of encounter to equal twice the natural frequency of roll.

Almost invariably, the ship must be operating in following seas in order for the frequency ratio to take on the critical value.

Referring again to Figure 2, we see that the wave-induced stability variations in the case of the platform are quite small in comparison to those of the ship. This is not surprising when we observe that the variations are closely related to the variation in topside flare along the length of the ship. A ship having vertical sides at all sections would have little or no wave-induced stability variation, and the semisubmersible platform is also a vertical-sided vessel. In addition, we observe that the semisubmersible is usually expected to have a lower frequency of roll than the ship. Since the platform is normally stationary except for periods of low speed transit between operating sites, the possibility of encountering waves at the critical frequency which is needed to induce the roll instability illustrated in Figure 5 is seen to be rather remote.

### 3. RESPONSE OF A SEMISUBMERSIBLE TO A STEPWISE HEELING MOMENT

The wind heel criteria for both ships and platforms are based on the assumption that the heeling moment is applied in a stepwise manner such that the magnitude increases instantaneously from zero to its final value, and thereafter remains unchanging with time. In the U.S. Navy criterion, the effect of beam seas is taken into account by assuming an initial value of the roll angle to windward. The stability criteria, in the case of a semisubmersible, however, assume zero initial roll, and it is of some interest to investigate possible effects of waves and the resulting initial roll angle on the response for such a platform.

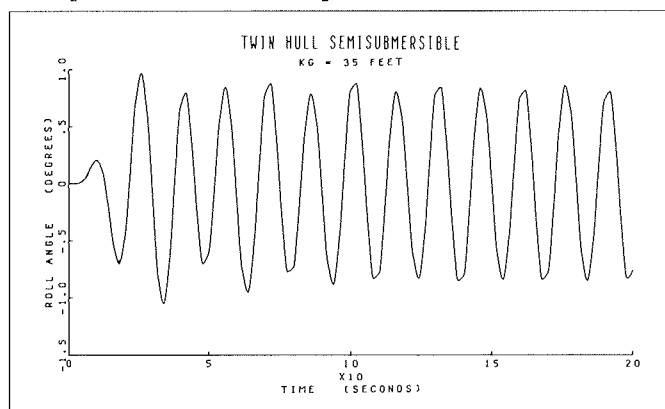


Fig. 6 Roll motion of semisubmersible, Wave Height = 20 ft., Period = 15 sec.

Figures 6, 7, 8 and 9 contain the results of simulations in which the nonlinear equations of large-amplitude platform motion were integrated utilizing a procedure similar to that which was used in obtaining

the results of Figure 5. In Figure 6 is shown the rolling motion of the platform exposed to regular beam seas alone. After a period in which an initial transient motion decays, the predicted roll motion is apparently periodic except for slight irregularities attributable to the finite time step used in the integration.

Figure 7 illustrates the response of the platform, floating in calm water, to a stepwise heeling moment, and Figures 8 and 9 illustrate the computed response to the simultaneous effects of waves and the stepwise moment. The moment, in these latter cases, was applied at the instant of maximum roll to windward, thus simulating the condition of the Navy criterion with the additional dynamic effects of the waves and the platform motion. The magnitude of the heeling moment was estimated using results given in Reference [8] for an assumed wind speed of 100 knots. The results of Figures 7 and 8 were computed for the same vertical center of gravity position. The results of Figure 9 correspond to a somewhat higher center of gravity, thus a condition of lower initial stability.

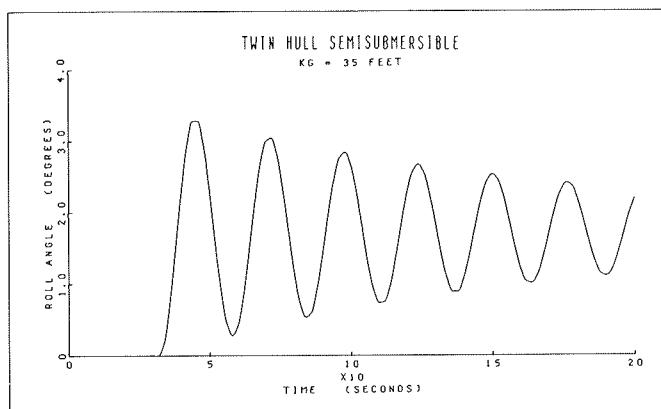


Fig. 7 Roll motion of semisubmersible under stepwise heeling moment, Heeling Arm = 1 ft.

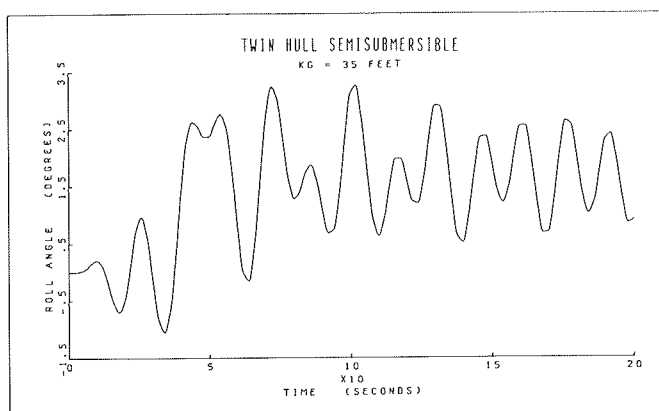


Fig. 8 Roll response of semisubmersible under combined action of waves and stepwise wind moment.



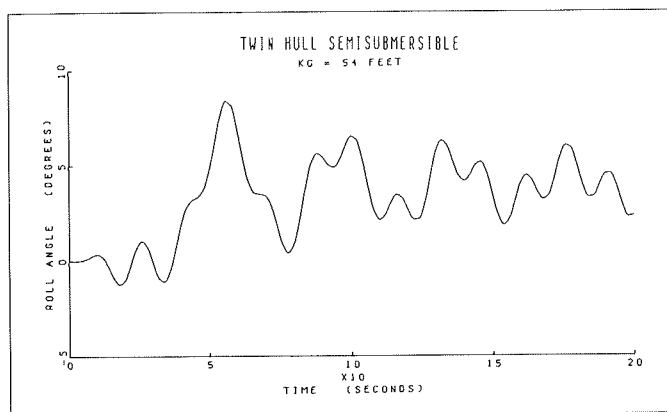


Fig. 9 Roll response of semisubmersible under combined action of waves and stepwise wind moment.

The combined response of Figure 8 has the appearance of a simple superposition of the wave- and stepwise moment responses, and this is not unexpected in view of the moderate roll angles involved in this particular example. For more severe waves and larger heeling moments, some irregularities would be expected as a result of nonlinear aspects of the motion. For extreme cases, the numerical integration scheme which provided the data for these figures would be capable of simulating motions up to and including capsizing.

In comparing the combined response for the two different values of initial stability, Figures 8 and 9, we see that the maximum value of the first roll following the application of the wind moment is strongly affected by the relationship between the natural period of roll, which is the period of the transient response to wind heel, and the period of the waves which is the period of the wave-induced motion. In Figure 8, the natural period is approximately twice the wave period, with the consequence that the wave-induced roll is of nearly opposite phase to the first peak of the transient wind-induced roll, and the two effects tend to cancel each other during the first transient swing. For the less stable case of Figure 9, the longer natural period of roll results in the two components being nearly in phase at the first transient peak, thus two effects tend to reinforce each other, causing a large initial roll.

In a real random seaway, of course, the phase of the wave-induced roll as compared to the wind heel effect would be a random quantity, and the occurrence of such reinforcement of the two effects would have to be expressed in probabilistic terms.

#### 4. THE EFFECT OF TIME VARIATIONS IN WIND STRENGTH

In computing the wind-induced heeling moment, it is usually assumed that this

moment remains constant in time following its initial stepwise application. The effect of gustiness and other uncertainties or irregularities is then taken into account through the factor of 1.4 or 1.3 by which the dynamic stability is required to exceed the heeling work.

The time-varying part of the wind velocity is typically expressed as a spectral density function in a form proposed by A. G. Davenport, References [9] and [10]. The nondimensional form of the Davenport spectrum is given in equation (1).

$$\bar{S}(f) = \frac{x^2}{(1+x^2)^{5/3}} \quad (1)$$

where

$$S(f) = \text{Velocity spectrum} = \frac{kV_{10}^2}{f} \bar{S}(f)$$

$k$  = Roughness factor characteristic of the terrain

$f$  = Frequency in Hz

$V_{10}$  = Wind velocity in m/s at height of 10 m

$x = \frac{1200f}{V_{10}}$

If we differentiate this expression and set the result equal to zero, the peak frequency of the spectrum,  $f_0$ , is found to be given by,

$$f_0 = \sqrt{\frac{3}{5}} \frac{V_{10}}{1200}$$

For a mean wind speed of 100 knots, this gives a spectral peak (maximum gust energy) at a frequency of 0.0332 Hz or a period of 30 seconds. For 50 knots mean speed, the spectral peak occurs at a period of 60 seconds. This range of periods includes the range of typical semisubmersible roll natural periods, but is somewhat longer than typical roll periods of ships. The possibility of wind heeling moments exciting resonant rolling motion of a semisubmersible is suggested by this overlap of frequencies.

The effect of a time-varying wind heeling moment on the example semisubmersible is explored in Figure 10. For this simulation, the moment was assumed to consist of a constant mean value plus a sinusoidally varying part equal to 0.25 times the mean value. Since the wind moment varies as the square of the velocity, this would correspond to a variable component of about twelve percent of the mean velocity. The period of the variation was taken to be equal to 35 seconds, corresponding to a wind speed of slightly less than 100 knots. For the assumed platform KG, the natural period of roll of the platform is 33 seconds, which is typical of the operating condition for a platform of this type and size.

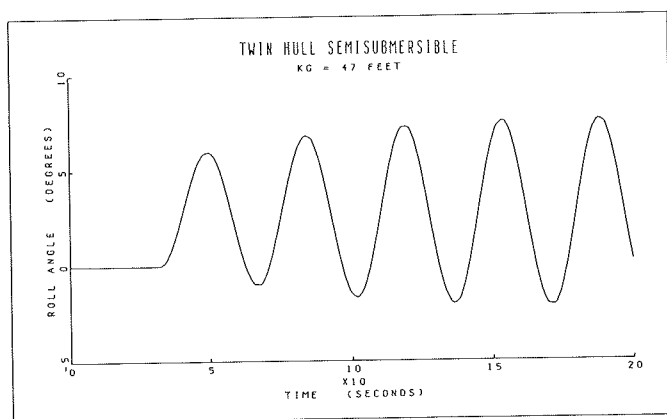


Fig. 10 Roll Motion of semisubmersible under periodic wind heeling moment.

The peak values of roll shown in Figure 10 are similar to the peaks of the roll obtained under the combined action of the impulsively applied wind plus waves as is illustrated in Figure 9. The results of Figure 9, however, were obtained for a considerably less stable platform than that of Figure 10. Assuming that the 25 percent variation in the wind force is reasonable, this suggests that resonant amplification of roll in an unsteady wind field may be a factor to consider in assessing the stability of a semisubmersible if the natural period of roll may be expected to fall within the range of significant energy of the gust spectrum.

In order to put this effect in another perspective, we note that the wind heeling arm corresponding to a steady wind of 100 knots is about one foot for this platform, thus the time varying arm is 0.25 foot. The periodic heeling moment caused by long waves corresponds to a heeling arm approximately equal to the metacentric height multiplied by the wave slope. For the case illustrated in Figure 10, the metacentric height is 21 feet, consequently the wave heeling arm is given by,

$$HA = 21 \times 2\pi a/\lambda$$

where  $\lambda$  is the wave length, and  $a$  is the amplitude.

If the wave steepness, i.e., the ratio of wave length to wave height is  $1/20$ , the wave heeling arm is approximately 3.3 feet. For wave components having periods in the vicinity of the 33 second natural period, the steepness will be much lower than  $1/20$ , and it may reasonably be expected that the wind-induced component in this range will be of at least equal importance in comparison to the wave component.

In concluding this discussion of unsteady wind heel effects, we must observe, as in the case of wave-induced roll, that the random character of the gust effects will require that the behavior of the moments and the platform motion must be assessed through a probabilistic approach.

## 5. CONCLUSIONS

In this paper, we have attempted to compare several aspects of the intact stability of ships and semisubmersible platforms in which differing behavior might be expected as a result of differences in the geometry and dynamic behavior characteristics of the two classes of floating vessels. One of our objectives is to suggest areas in which research may be needed in order to develop a more rational basis for stability criteria for platforms. Some significant differences in the stability characteristics of ships and platforms have been pointed out, and among these are the following.

(1) The effect of following seas on the righting arm curve is much more pronounced for a ship than it is for a semisubmersible platform. As a result of this, the semisubmersible is much less likely than a ship to experience the autoparametrically-induced roll motion instability in a following sea which sometimes results in heavy rolling and, in extreme cases, capsizing of a ship. Contributing further to the semisubmersible's immunity to this motion instability, is the combination of a long natural period of roll and a non-mobile mode of operation which, together, result in the platform's reduced probability of being exposed to the critical wave period of encounter needed for the instability to occur.

(2) The roll response of a semisubmersible to the stepwise wind heel moment assumed in most intact stability rules may be increased somewhat if the effect of waves acting simultaneously is included. That is, the dynamic roll caused by the waves may augment the wind heel, resulting in somewhat larger dynamic roll than predicted using the calculation procedure typical of published rules.

(3) Unsteadiness of the wind field is usually characterized by the Davenport spectrum. If this spectrum is suitable for describing the wind over water, the resulting unsteady part of the wind heeling moment may contain an appreciable energy component in the frequency range which is typical of the natural roll frequency of a semisubmersible, resulting in resonant amplification of the wind-induced dynamic rolling motion. This frequency range is substantially lower than the frequency of roll of a ship and, therefore, resonant amplification is probably of no importance in that case. Further study of the wind spectrum over water is as well as improved accuracy in estimating the unsteady wind forces on the structure is suggested in order to fully evaluate the importance of this effect.

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## Discussion

T. Oda (Mitsui Engineering & Shipbuilding Co., Ltd., Japan)

In Fig. 2 the effect of wave on righting moment is almost null.

However, I think wave effect is related to the breadth of semisub say distance between two lower hulls on beam condition.

Is this trend not changed by wave condition and geometry of semisub?

### Author's Reply

Mr. Oda is quite correct in stating that the effect of the wave on stability is related to the beam of the ship or platform. The most important parameter, however, seems to be the relationship between the shape of the above water part of the hull in the ends of the vessel as compared to amidships. In the case of a ship, the factors which cause a pronounced loss of stability in the wave crest are low freeboard amidships and pronounced flare with high freeboard in the ends of the ship. For the example semisubmersible in the paper, there is only a slight lack of similarity between the amidships and the ends. This is the difference between the diameters of the main corner columns and the smaller columns in the amidships position.

I. Manum and E. Dahle (The Norwegian Maritime Directorate, Norway)

In the paper, the author discusses some characteristic differences between ships and semi-submersibles.

In our opinion, the most important difference between these types of vehicles is the much greater vulnerability to unsymmetric flooding for the semisubs. The reason is that present recommendations, as in the MODU-code, only give protection against low-energy impact damage from moderate size supply ships operating at rather low speeds. For higher energy damages, as well as for other reasons (overloading, faulty ballasting, combined smaller damages), the MODU-code is quite

inadequate, and the semi-sub will capsize. For this reason, the Norwegian Maritime Directorate in February 1982 imposed new rules for Norwegian semisubs operating world-wide, regardless of operating area and their weather condition. The important requirement for reserve buoyancy to protect against unsymmetric flooding for semisubs could well have been included among the list of research projects.

### Author's Reply

Professor Dahle quite correctly points out another very important characteristic of semisubmersibles. This effect is perhaps recognized indirectly by the requirement that the stability be computed about an axis of heel which is oriented in the horizontal plane so as to result in the lowest stability. In contrast to ships which almost always rotate about a longitudinal axis, the computation of the stability of a semisubmersible must be carried out for roll axes at several different orientations in order to find the one corresponding to least stability.

The great breadth in relation to length of the semisubmersible definitely makes a asymmetric flooding more severe than in the case of a ship. Similarly the great height of the structure makes it more prone to capsize than a ship, in the case of damage.

S. Kastner (College of Engineering Bremen, FRG)

Professor Paulling has been applying his simulation program in order to study the effects of different external parameters such as the combination of wind gust with waves on semisubmersibles as compared with ships. I wonder, if any of the applied wind spectra apply to the whole length of the structure in question, either semisub or ship? While applying Davenport's wind spectrum we assume an even distribution of wind forces on the whole lateral area of the structure. I think experimental data are needed, from measurements of the wind-

field at different positions within the lateral field at a platform, including the correlation of the data. The same uncertainty applies for the effect of wind gust on ships, and measurements at large scale would give an answer.

#### Author's Reply

I certainly agree that more detailed information, especially of wind-wave correlation is needed in order to fully describe the spatial distribution of the wind field. The Davenport Spectrum applies to wind velocity fluctuations over land. The irregular, moving contour of the sea may be expected to induce turbulence in the wind boundary layer which differs appreciably in content from that induced by irregularities such as hills, valleys, trees and tall buildings which remain stationary.

An attempt to include the spatial variation of the wind is included in the velocity profile or height coefficient of the stability rules published by classification societies. Horizontal variation is not included in these formulations. In the wind field over water, the scale of horizontal variation could be expected to be similar to the wave length or crest length dimensions. For the Davenport spectrum corresponding to a 100 knot wind speed, the typical gust wave length, given by the product of wind speed and peak period of the spectrum, is slightly less than one mile.

#### I. Manum (The Norwegian Maritime Directorate, Norway)

When comparing stability characteristics of ships and offshore structures the practical side should also be taken into consideration.

Re-inclination tests of 20 rigs under Norwegian flag after the capsizing of A.L. Kielland showed rather alarming results, very difficult from what you could expect for ships. See overhead. The overhead shows that prior to the re-inclination test 75% of the platforms had stability documents onboard which were based on too low light weight displacements and vertical centres of gravity. Therefore, the maximum permissible deck-load had up to then been 500 high, because the basic data were uncorrect. If such platforms, when loaded to their maximum permissible deckload limit, had experienced a damage similar to one assumed and approved limiting damage condition, the consequence could have been fatal.

The situation might be the same for rigs under other flags.

#### Author's Reply

Mr. Manum's data emphasize once again that stability is not purely a matter of technical computations, but it must also take into consideration the manner in which the floating vessel will be operated, and

the customs and habits of the operating personnel. The effects of such operational factors need to be quantified and either included in the criterion as an additional safety margin or controlled through regulations and operational directives. The designer and research worker must be aware of their importance if they are to do their jobs properly.

#### D. Vassalos (Univ. of Strathclyde, UK)

I have been trying to find out a justifiable reason why do researchers concern themselves with quasi-static calculations in following waves when quartering waves always give the worst GZ curve.

This is my opportunity to get an answer from an authority on this matters and I would like to take it and ask Prof. Paulling's opinion.

I would also like to take this opportunity to congratulate Prof. Paulling for initiating research continuously in so many and diverse fields.

#### Author's Reply

The most direct answer to Dr. Vassalos' question would be that following waves are used most frequently because it is easier to make the computations for this case. The ship-wave system is symmetrical in the upright condition, and the waterlines at each section of the ship are straight lines even when the ship is heeled. When the computations are performed by machine, as they almost always are today, the complications involved in integrating the wave-induced pressure distribution over the immersed surface of the ship in quartering waves are easily overcome.

Conceptually, the quartering sea situation is more complex in that there is now a significant wave heeling moment in addition to the restoring moment term. At large angles of heel, the two terms interact in such a way that they cannot be thought of as simply two additive components, but must be computed simultaneously. Here, we are considering only the moment due to the undisturbed wave pressure force, commonly known as the Froude-Krylov term, and are not considering other effects related to motion such as diffraction of the wave system, damping and added mass.

When we take dynamic effects into consideration, experience has shown that the response of the ship in pure following seas can be very well characterized by considering only the GZ curve, including, of course the effect of waves on this curve. In quartering seas, however, the dynamic terms assume much greater importance, and a dynamic motion analysis including large amplitude effects must be performed in order to accurately characterize the vessel response. At the present state of the art, the accuracy of such an analysis is limited primarily by our ability to predict the hydrodynamic effects at large motion amplitudes.

THE STABILITY ON SEMI-SUBMERSIBLE PLATFORM IN WAVES  
( ON THE CAPSIZING OF MOORED SEMI-SUBMERSIBLE PLATFORM )

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Sumitomo Heavy Industries, Ltd.

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ABSTRACT

In this paper, the authors describe briefly the problems regarding the stability of semi-submersible platform under wave conditions. It is believed that most studies of the stability concerned have been concentrated in the intact stability of the platforms in free floating condition. The main emphasis of this paper, however, is to take the influence of mooring lines on the stability of the semi-submersible platforms in waves.

Deeper experimental studies were subsequently attempted, with concentration on the steady and oscillatory tilts of the moored semi-submersible platform in waves varying the metacentric height, fairleader height and so forth.

Some interesting results have been obtained from these experiments and are discussed in this paper.

1. INTRODUCTION

In recent years, the semi-submersible type platform is used for many kinds of offshore works including the oil rig. The reason this type of platform is widely used is due to the minimum motion characteristics of such a structure.

Even though a large number of these are under construction, many unknown factors still remain to be solved due to their short historical background comparing with that of the conventional ships. These unknowns which are based on the difference of ship characteristics, involve the exact solution of hydro-dynamic forces and moments on the immersed body, mooring problems, problems such as a stational structure and so on.

It seems, therefore, that investiga-

tion of capsizing and/or the stability of the moored semi-submersible platform in waves has not been done sufficiently comparing with that of the platforms motion in waves.

Authors had carried out series of tank tests on the capsizing of the semi-submersible platform starting from the Summer of 1981. Various problems and facts on the stability of semi-submersible platforms have been discovered through experiments. We hope that the individual detailed study of those experiments will be published later.

This paper presents the summary of some important characteristics to be considered for designing those structures and some interesting phenomena observed in the tank tests on the stability of semi-submersible platforms.

2. SOME ARGUMENTS ON STABILITY CRITERIA

2.1 Current Stability Criteria for Semi-Submersible Platform

The existing stability criterion of semi-submersible platforms, as established by rules of many classification societies, are mainly based upon the idea that the platform should have sufficient righting energy to the overturning energy due to highest wind force. This is achieved by calculating the areas under the righting moment and the wind heeling moment curves so that the ratio of these areas (R) meets the requirement of rule.

Fig. 1 shows a typical intact stability curve for offshore drilling platform, the ratio (R) is given by the ABS rule, for example [1];

$$R \leq \frac{\text{Area (A + B)}}{\text{Area (B + C)}}$$

The stability requirement by the ABS rule gives this ratio as 1.3 currently for the semi-submersible platform.

Other requirement on the stability is that the metacentric height (GM) should be positive.

Many arguments about the stability requirement of semi-submersible platform have been made and reported by some technical papers during the 1970's.

The essential problem mostly discussed was if the ratio (R) 1.3 was sufficiently large or not at the first stage (many designers and operators had questioned that the value is unreasonably high).

Over many years, attempts to study on the stability of semi-submersible platform have been done since the ABS rule on the static stability criterion came into force in 1968.

The experimental study on the stability of semi-submersible platforms by Numata et al. has revealed that no capsizing or near capsizing had occurred even when the stability of the tested model was less than that of the requirement of the rule [2] [3] [4]. The paper concluded, therefore, that it has not been possible to cause a capsizing of semi-submersible platform with any reasonable combination of wind force and wave actions.

This indicates that the capsizing of semi-submersible platform could be caused by other factors if the capsizing exists in reality. Finally, Numata et al. noticed the so called steady tilt behavior of semi-submersible platforms which could be a very serious problem. In such a case, the upper deck would be frequently struck by waves and occasionally washed over. This curious phenomena tends to develop when the metacentric height is relatively small.

On the other hand, since the existing computation of static stability are referred to the mean water level and the presence of wave action is generally omitted, Kuo et al. had suggested that the simple triangular wave profile is superposed on the still water level so that a quasi-static steady treatment is performed [5].

In conclusion, it became understood that the existing stability criterion on semi-submersible platform is somewhat insufficient from the safety point of view and the prediction of safety of intact semi-submersible platforms at sea is still a problem.

## 2.2 Further Considerations on Stability of Semi-submersible Platforms

So far the current requirements on the stability of semi-submersible platform and some arguments on the rules have been discussed. It seems that very few arguments have been made about the influence of the mooring line on the stability of platform simply because they are generally considered

to be small in effect.

The experimental studies are performed with the presence of the mooring lines in order to investigate the effect of the mooring line on the stability of the semi-submersible platforms. The tested model is the twin lower hull type of semi-submersible platform and is shown in Appendix.

## 3. OVERTURNING LOADINGS ON SEMI-SUBMERSIBLE PLATFORMS UNDER MOORED CONDITION

### 3.1 Introduction

As is described in the previous chapter, the current stability criterion require only the overturning moment due to wind force as basic input and disregards any other loadings. One of the reasons is that the existing regulation is originally applied to the offshore drilling units by modifying the U.S. Coast Guard stability criteria for a surface vessel which generally exclude mooring conditions.

Most offshore drilling platforms, however, are usually positioned by several mooring lines during their operating condition. Accordingly, loadings on the platform are considered to be more complicated nature and numerous types of loadings are encountered.

The possible steady loadings acting on the platform under moored condition are listed below;

- 1) Wind Force
- 2) Current Force
- 3) Wave-induced Drift Force
- 4) Wave-induced Impact Force
- 5) Mooring Force
- 6) Others

Clearly, it is important to consider these items if a more realistic representation on the stability of semi-submersible platforms is desired.

The wind loads are believed to be the most important load on the static stability of the floating structure. Since these loads have already been taken into consideration for the regulation, they are not discussed in this paper.

### 3.2 Current Forces

Since a greater portion of the semi-submersible platform is submerged in water, the overturning moment due to current force may result to be significant for the moored platforms. Especially, the author's attention is paid to the condition where both current and wave are included at the same time. In this case, the platform behavior's will become serious condition.

For the sake of overturning moment due to current forces, platform behavior due to current and/or waves had been conducted (See Appendix). The 1/60 scaled model consists of 8 columns with two separated lower hulls at the base of columns and is moored at the center of tank (56 m length x 30 m

width x 2.5 m depth) by 8 chains of 11.5 m in length.

The experimental results of horizontal drift, inclination and the mooring force of weather side chain under beam sea conditions are plotted in Fig. 2. In this figure, one can see the difference in inclination between two different positions of the fairleader; one is the same height of CG (Center of Gravity of the platform) and the other which is 20 cm higher than that of the CG in the model scale.

The theoretical results obtained by the quasi-static solution are also plotted with good agreement.

### 3.3 Wave Induced Drift Forces

Drift forces due to incident waves are caused by the non-linear terms in the hydrodynamic pressure distribution and is generally considered to be small in magnitude to the excitation of the platform motions. But for situations where there is large amplitude wave with a short period, the drift forces become larger steady forces acting on the platform.

A large amount of experimental studies on the platform stability in waves had been carried out. At the tank test of unmoored condition, the model was hinged at the CG to allow roll or pitch motions freely while the wave induced drifts were restrained by the weight for the measurement of their magnitude.

In Fig. 3, non-dimensional drift forces of the tested model are plotted against various wave frequencies. It can be seen that the peak values of these appear to be about 8.5 seconds of a wave period in real scale from these figures.

On the other hand, the second order forces in vertical direction also exists in waves. The earlier studies on this force had been investigated by Goodman (6).

In 1973, Numata et al reported that the semi-submersible platforms having small metacentric height had a tendency to behave with a steady tilt in addition to the oscillatory motions in waves [3] [4].

This curious phenomena generates from the overturning moment due to the horizontal drift forces at the beginning and starts to incline until equilibrium is established.

At the equilibrium point, the heeling moment caused by the difference in vertical second order lift force on each lower hull and the horizontal wave-induced drift force is balanced out by the righting moment of the platform.

This steady lift force which is a function of the added mass of the immersed body increases exponentially as the distance to the water surface is decreased.

### 3.4 Other Considerations on Loads

When the platform heels considerably due to overturning moment, it is possible that water rise on the upper deck sometimes

and may run into openings of upper deck. Therefore, the accumulation of greenwater on deck could be problem. During the tank test, it is observed that the platform heeled drastically due to greenwater on the upper deck in high steep waves.

In addition to the greenwater on upper deck, the accumulation of icing on the upper deck and of snows should be taken into consideration as additional loadings especially when the platform operates in cold weather.

During some severe sea condition, a large volume of water caused by the wind shear force and atmospheric pressure travels around the platform. This phenomena called storm surge is considered for long time as a serious problem especially for the fixed structure in civil engineering. Although those effect on the moored platform should be taken into consideration, very little study on those effects have been done. Further investigations for those are to be done in future.

## 4. STEADY TILTS OF MOORED SEMI-SUBMERSIBLE PLATFORM

### 4.1 Introduction

In the previous chapters, the general problems on the static stability and its criteria of semi-submersible platforms have been briefly discussed. Generally speaking, it is more important to investigate the behavior of the moored semi-submersible platform under severe sea conditions which could result in capsizing of platform.

However, very little work has been previously done on this problem. In this chapter, results obtained from the experiments which mainly aim to simulate the dangerous behavior of the semi-submersible platform in regular waves are discussed.

As the model shifts due to wave-induced drift forces in the wave tank, the mooring lines located on the weather side of the platform become taut until equilibrium of the net horizontal forces is achieved. In this case, it has been observed that the model with low metacentric height has a tendency to develop a steady tilt and the upper deck sometimes becomes immerse in waves. Photo 1 shows the example of this dangerous situations observed in the wave tank test.

Since this dangerous behavior is considered to be strongly influenced by mooring lines and the metacentric height, the following test conditions were imposed for further analysis of the behavior of semi-submersible platform.

- 1) Varying Fairleader Heights
- 2) Varying Metacentric Heights
- 3) Wave Frequency

### 4.2 Influences of Fairleader Height

During the tank test conducted, effects of the fairleader height on the



steady tilt of semi-submersible platform had been investigated under both beam sea and head sea conditions. The details of the test are summarized in the Appendix.

Results obtained from this test session have shown that the angle of the steady tilts was strongly influenced by the location of the fairleader height. Fig. 4 illustrates the comparison of the heeling angles of varying fairleader heights under beam sea condition. When the location of fairleader is higher than that of the CG, the platform has a tendency to tilt to the weather side direction. It is interesting to note that the angle of tilt increases rapidly as the wave period decreases.

The larger angle of steady tilt allow waves to ride on the upper deck sometime, which causes a larger inclination of the model. This indicates that the influence of the fairleader height could no longer be disregarded for the safety of the semi-submersible platform as well as of wave frequency.

To determine the mean tilt angle, the quasi-static analysis of the steady tilt was conducted. Forces and moments acting on the semi-submersible platform are balanced in equilibrium condition and are expressed by the equations in Fig. 5.

#### 4.3 Influences of Metacentric Height

From the stand point of static stability, it is obvious that the steady tilt most likely occurs with the model of low metacentric height simply because it's righting energy is sufficiently small against the overturning energy induced by wave forces.

It has already been discussed in previous chapters that the steady tilt will occur when the metacentric height GM of the platform is small. Numata et al. pointed out this problem by saying that the magnitude of the tilt angle is a function of GM [3] [4].

Fig. 6 shows the comparison of the steady tilt of the semi-submersible platform having different metacentric height. It is obvious that the steady tilt of the platform having a lower GM is much larger than that of the one having a larger GM.

### 5. DYNAMIC BEHAVIOR OF MOORED SEMI-SUBMERSIBLE PLATFORM IN IRREGULAR WAVES

#### 5.1 Introduction

The previous chapter discussed the steady motions of moored semi-submersible platforms due to the effects of various factors such as fairleader height, metacentric height, etc. These motions were considered in order to introduce the serious situation of the semi-submersible platform in waves.

In this chapter, some serious behaviors observed in the tank test including slow oscillatory motions of the moored semi-submersible platform are discussed in

irregular wave conditions.

It is extremely difficult to assess the unsteady behaviors of the moored semi-submersible platform in waves by means of the quasi-static solution. In such a case, non-linear analysis of time domain may be applicable to simulate the dynamic behaviors of the platform.

#### 5.2 Unsteady Tilts in Regular Wave Groups

Regular wave groups which can be regarded as the simplest case of irregular waves, consists of two regular components having a small difference in frequency. It has been already pointed out in various writings that such regular wave groups have two components of drift force; one is a constant part and the other is a slowly varying part [7] [8].

When the regular wave group is propagated toward a moored platform, the slowly varying part of drift force creates a so-called slow drift oscillation in surge and/or sway while the constant part of it causes the steady drift of the platform.

On the other hand, it is anticipated that the overturning moments caused by these drift forces may induce the steady and unsteady tilt of the platform in regular wave groups.

The tank test of both steady and oscillatory tilts were therefore, conducted in regular wave groups. Fig. 7 illustrates the records of platform motions in regular wave groups of three different wave periods obtained by experiments. It is interesting to note that the magnitude of the slowly varying tilt increases drastically as the wave period becomes smaller where the drift force is significant.

In addition to the overturning moment due to slow drift forces, there exists other moments due to the steady vertical lift forces acting on lower hulls.

The slowly varying tilt of the tested model were reconstructed by time domain computer simulation and are shown in Fig. 8. It should be mentioned in this figure that the effects of vertical lift force results as smaller than those of horizontal drift forces.

#### 5.3 Slow Varying Oscillations in Irregular Waves

Wind wave in open sea is irregular in nature rather than regular wave trains. In this case, a varying sequence of drift forces arises in correspondence to change in wave height and period.

In the irregular wave trains, which are obtained by ISSC Spectrum, several tests were conducted with the semi-submersible platform moored with 8 spread mooring lines.

Fig. 9 and Fig. 10 illustrate the typical results of motions in irregular waves from the model test. It is noted that both sway and roll motions involve the slowly varying oscillatory part with the



oscillatory part of the period being the same as the wave period.

The slow varying oscillations in sway considered to be caused by the wave-induced drift force now become common as a curious behavior of the moored object.

However, few studies have been done about the slow varying tilt which is considered to be induced by both horizontal and vertical steady forces.

## 6. CONCLUDING REMARKS

Several tank tests on the stability of moored semi-submersible platforms under both regular and irregular wave conditions were carried out varying some factors such as metacentric heights and fairleader heights.

From these experimental studies, the following conclusions were made.

- (1) It has been made clear that the influences of the mooring lines become significant under severe sea conditions and cannot be disregarded for the stability of the platform.
- (2) In regular waves, the steady tilt caused by the difference in vertical second order steady force on each lower hull becomes significant in the case of the platforms having lower metacentric height. Especially, when the platform has large angle of tilt by mooring lines, the heeling moment due to the vertical lift force increases drastically and causes the serious situation of the platform. In addition, it is found from experiments that the greenwater on the upper deck could be a problem to increase the angle of tilt.
- (3) In irregular waves, however, it is considered that the heeling moment due to horizontal drift force induces slow oscillatory tilt.
- (4) The existing static stability criteria based on comparison of areas below the heeling and righting moment curves be insufficient to avoid the serious situation such as down flooding type of casualty.

## ACKNOWLEDGEMENTS

The authors deeply appreciate the valuable discussions by Professors M. Bessho of Defense Academy and S. Takezawa of Yokohama National University.

Moreover, special thanks are due to Mr. S. Nagamatsu and the staffs of the Hiratsuka Research Laboratory of Sumitomo Heavy Industries, Ltd.

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## APPENDIX

The series of tank test on the stability of the semi-submersible platform in waves and/or currents were carried out in the wave tank of the Hiratsuka Research Laboratory of Sumitomo Heavy Industries, Ltd. during 1981 and 1982.

The tested model of the twin lower hull type was of 1/60 scale and is shown in Fig. A.1 and Photo. A.1 while particulars of the model are tabulated in Table A.1. The model was moored by 8 spread mooring lines and the arrangement of the lines is shown in Fig. A.2.

The test conditions of the model are summarized in Table A.2. The following variables of the tank tests were conducted to investigate the platform motions and the stability of the semi-submersible platform in waves and/or currents.

- 1) Influence of fairleader heights (See Fig. A.3)  
Location of fairleader  
5 cm below the CG (CG-5)  
at the CG (CG)  
10 cm above the CG (CG+10)

- 20 cm above the CG (CG+20)
- 2) Influence of metacentric heights  
 $GM_T = 0.048 \text{ m}, 0.028 \text{ m}, 0.008 \text{ m}$   
 $GM_L = 0.064 \text{ m}, 0.044 \text{ m}, 0.024 \text{ m}$
- 3) Influence of depths of upper deck  
 7 cm deck depth  
 3 cm deck depth  
 No deck
- 4) Influence of broken mooring lines

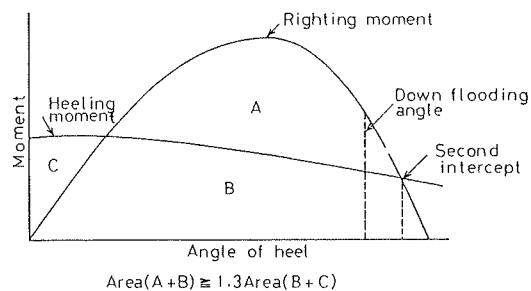
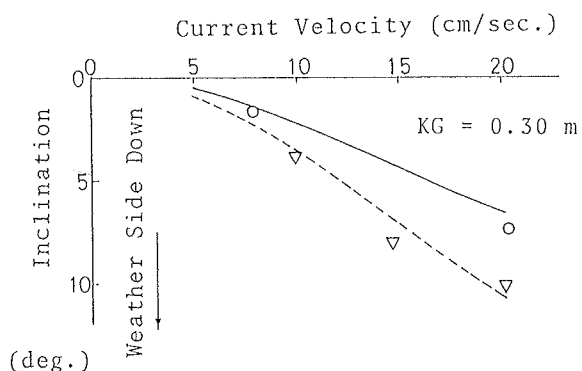


Fig. 1 Intact Stability Criterion for Semi-Submersible Drilling Units



| Fairleader Height | Exp. | Cal.  |
|-------------------|------|-------|
| CG                | ○    | —     |
| CG + 20 cm        | ▽    | - - - |

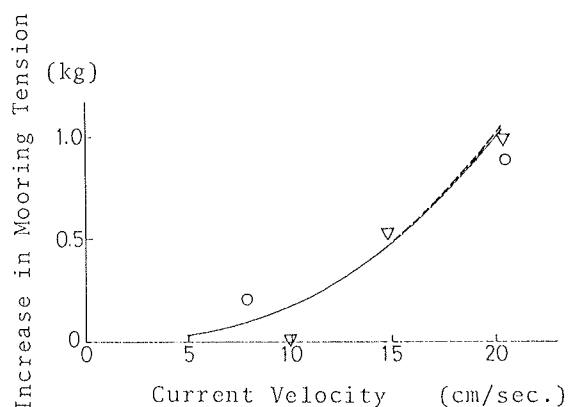
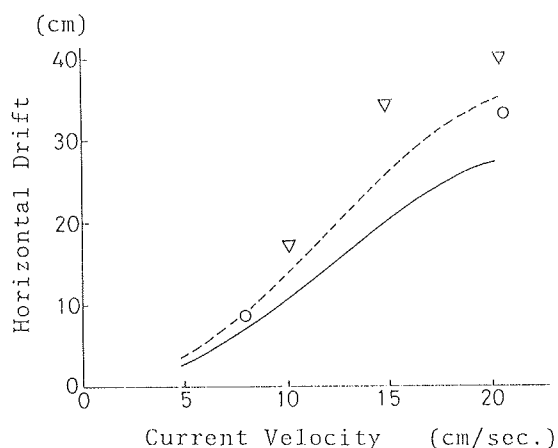


Fig. 2 Horizontal Drift, Inclination and Increase in Mooring Tension due to Current

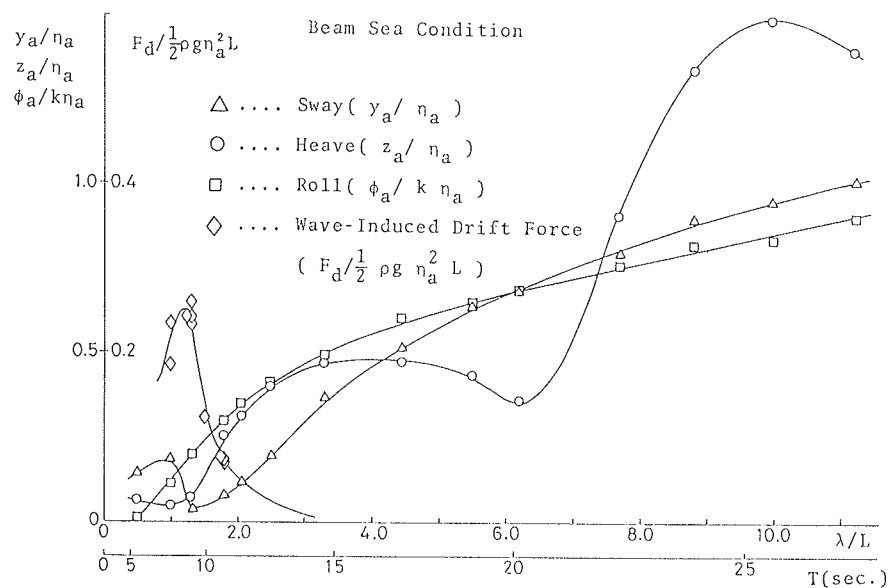


Fig. 3 Response Amplitude of Motions and Wave-Induced Drift Force in Waves

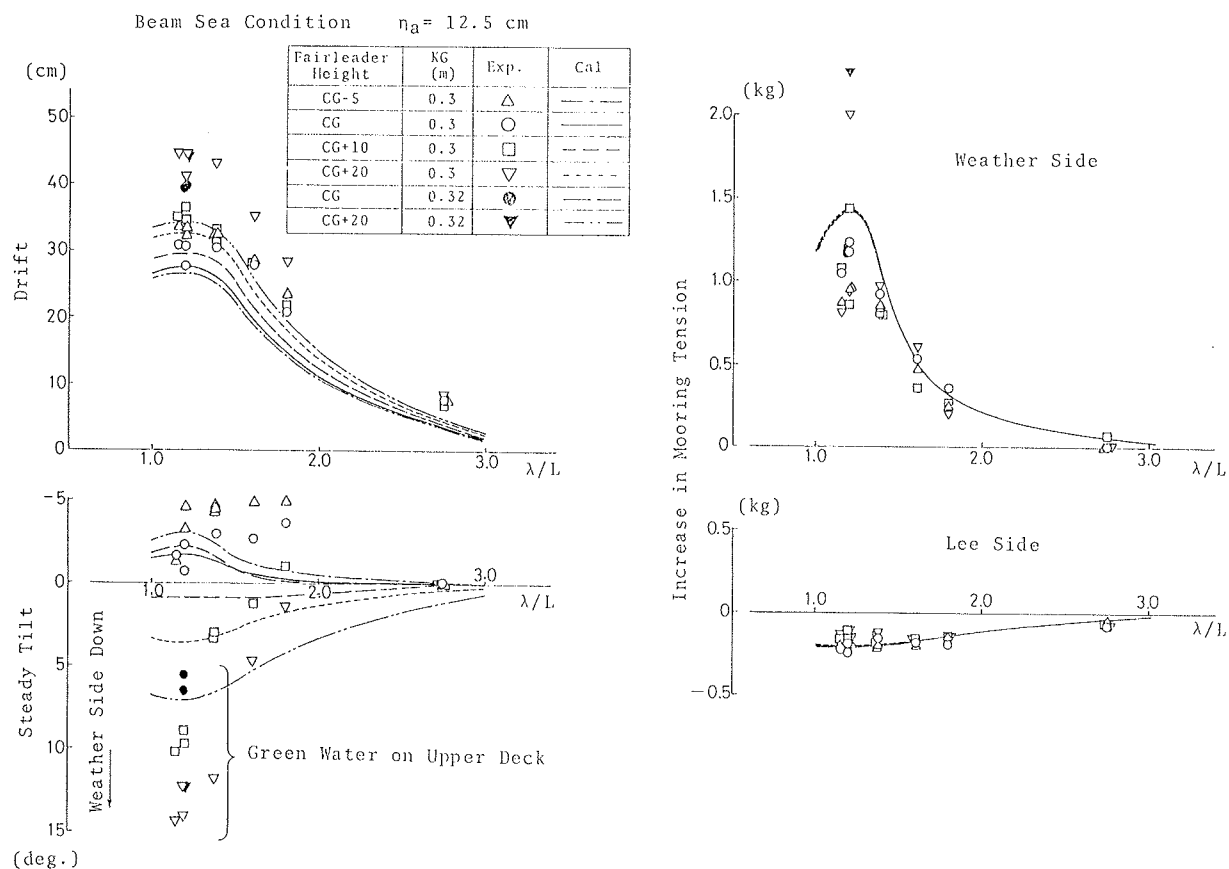
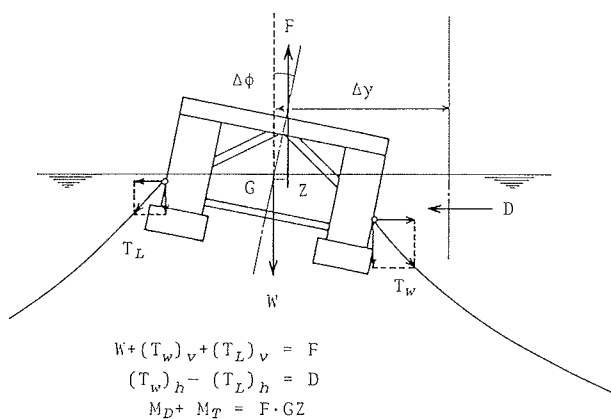


Fig. 4 Comparison of Drifts, Steady Tilts and Increases in Mooring Tensions Varying Fairleader Heights



where

$\Delta y$  : Horizontal Drift

$\Delta \phi$  : Inclination (steady tilt)

$W$  : Displacement

$F$  : Buoyancy

$T$  : Line Tension

suffix  $w$  : weather side

$L$  : lee side

$h$  : horizontal direction

$v$  : vertical direction

$D$  : Steady Force

$M_D$  : Overturning Moment due to the Steady Force

$M_T$  : Overturning Moment due to the Mooring Line Tension

Fig. 5 The Quasi-static Analysis of The Steady and Horizontal Drift

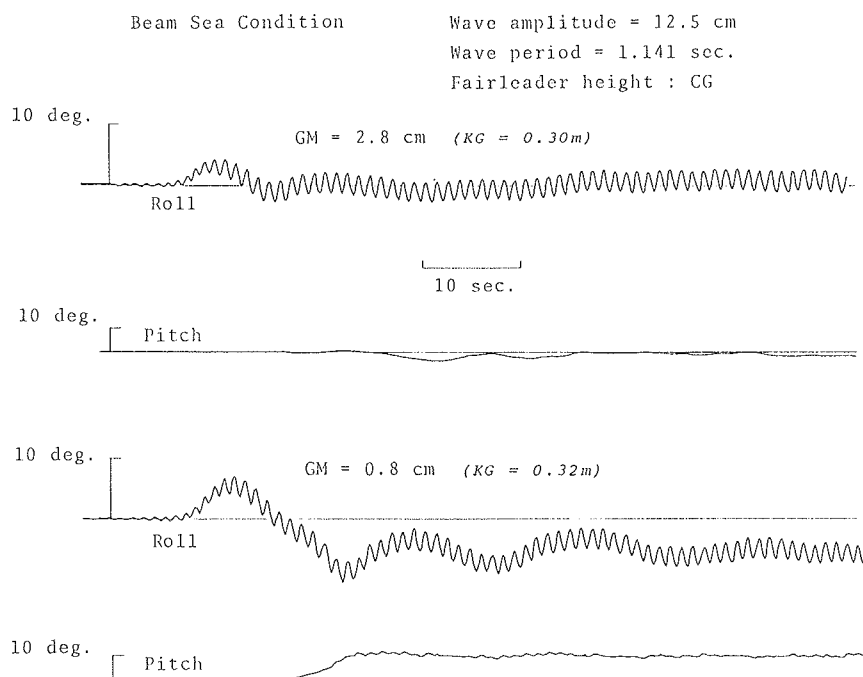


Fig. 6 Comparison of the Steady Tilts of the Semi-submersible Platform

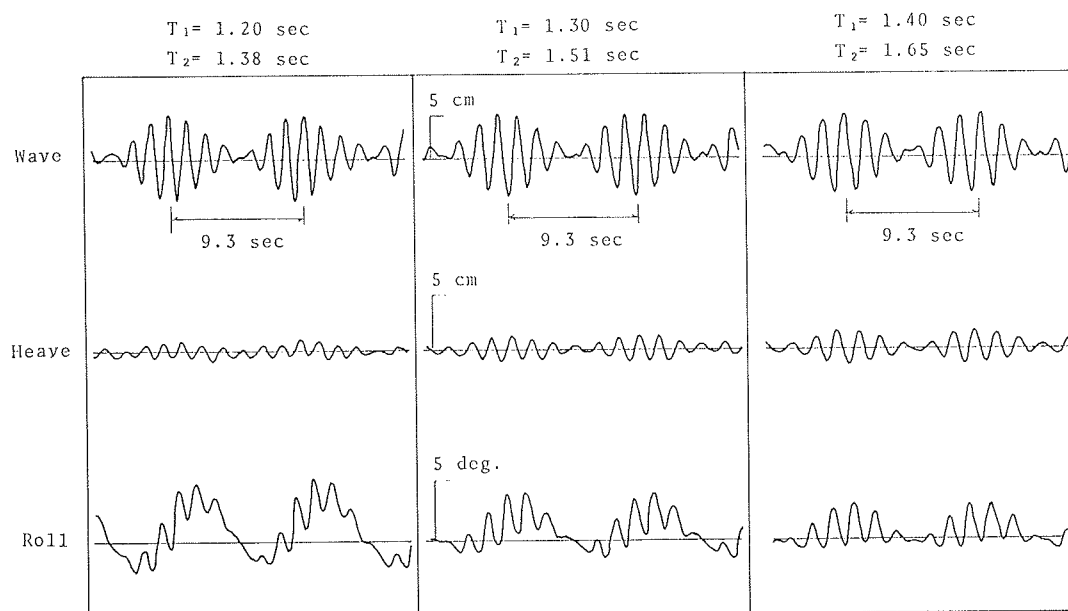


Fig. 7 Experimental Records of Platform Motions  
in Regular Wave Groups (Unmoored cond.)

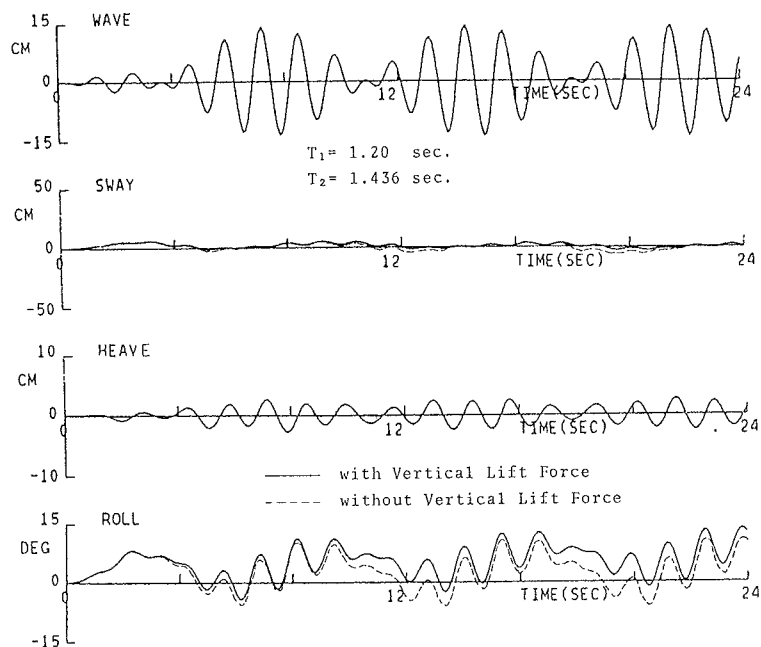
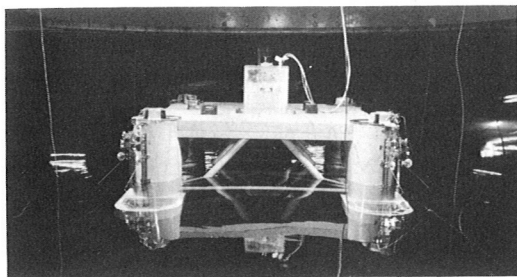
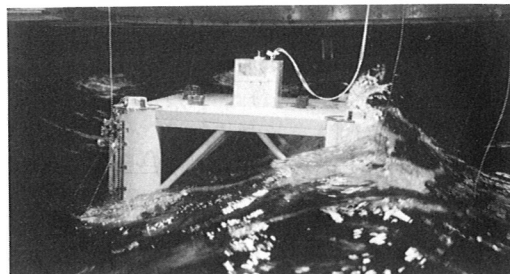


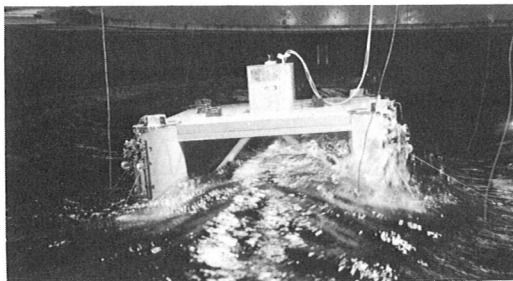
Fig. 8 Computer Simulation of Platform Motions  
in Regular Wave Groups (Unmoored cond.)



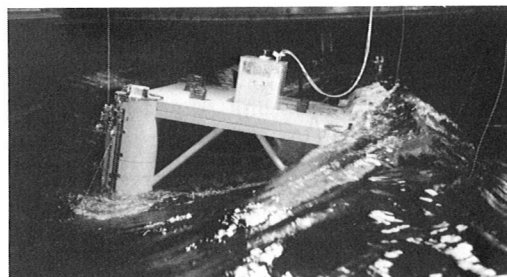
(1) time = 0 sec.



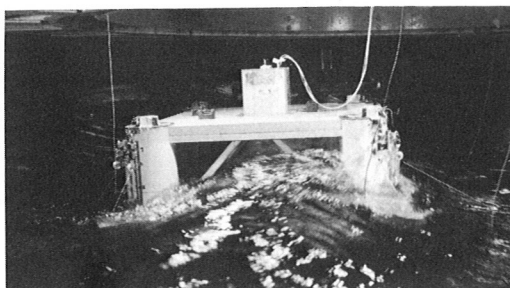
(5) time = 8 sec.



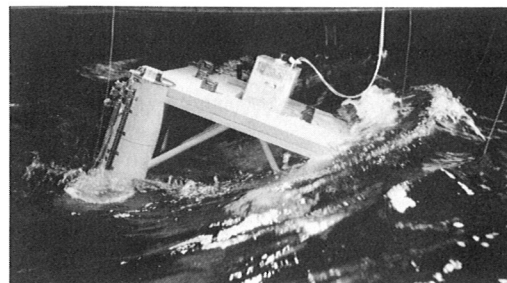
(2) time = 5 sec.



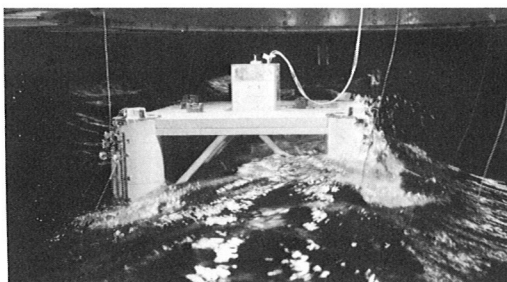
(6) time = 9 sec.



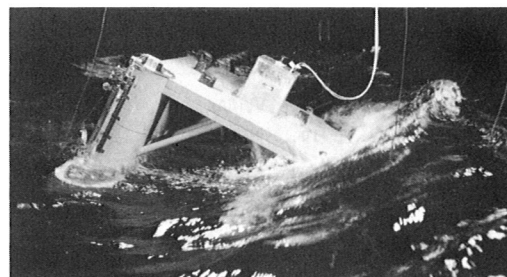
(3) time = 6 sec.



(7) time = 10 sec.



(4) time = 7 sec.



(8) time = 11 sec.

Photo. 1 Serious Behavior of Semi-Submersible Platform  
Observed in the Wave Tank Test  
(  $KG = 0.32$  m, Fairleader Height =  $CG + 20$  cm  
 $\eta_a = 12.5$  cm,  $T = 1.141$  sec. )

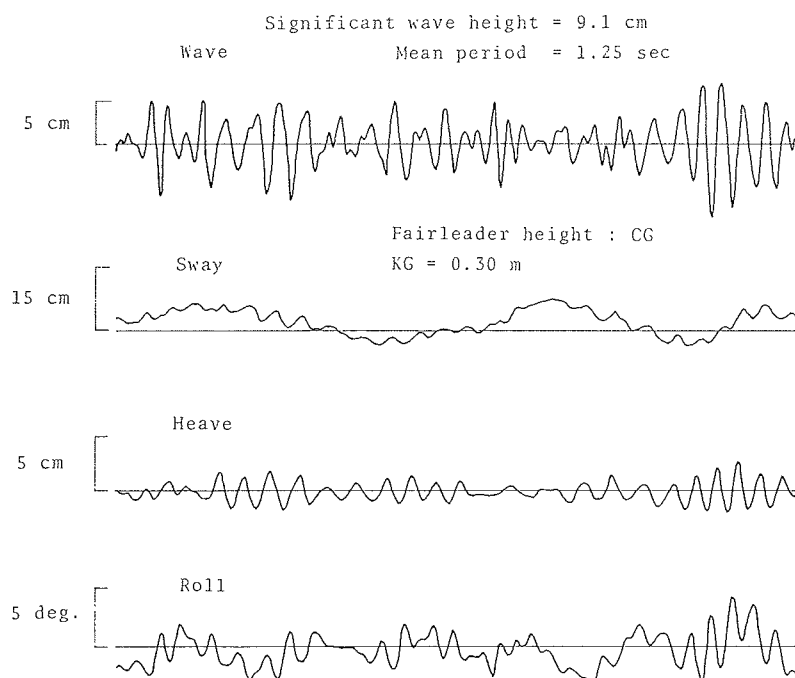


Fig. 9 Experimental Records of Platform Motions  
in Irregular Waves (Moored condition)

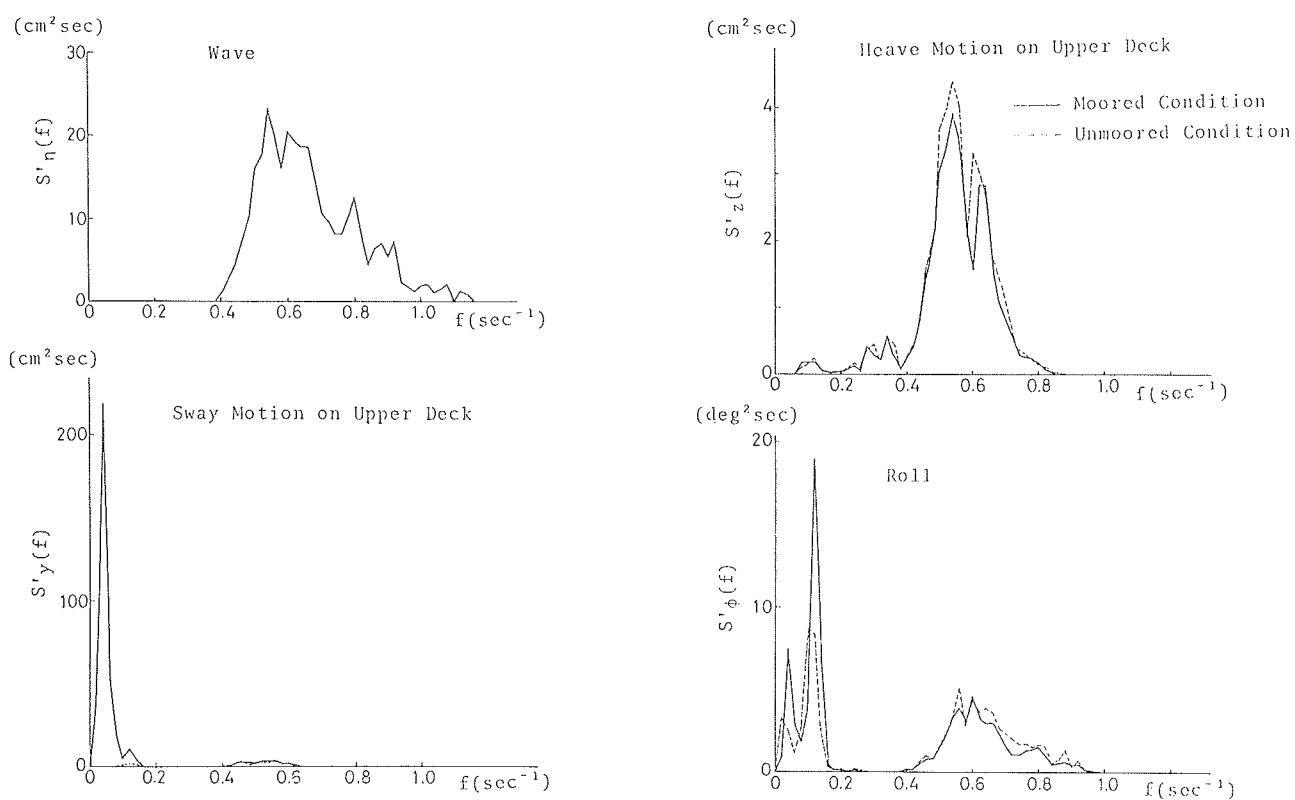
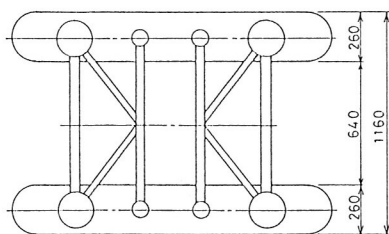


Fig. 10 Spectra of Wave and Motions



Unit : mm

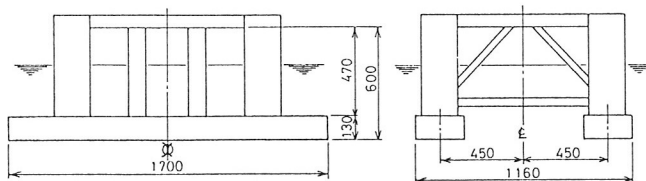


Fig. A.1

Tested Model of Semi-Submersible Platform

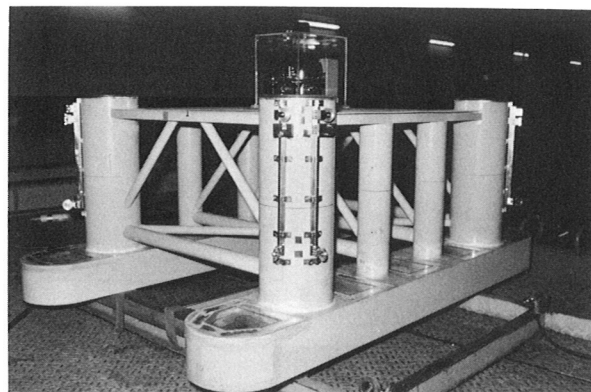


Photo. A1

Tested Model of Semi-Submersible Platform

Table A.1 Particulars of Model

|   | 1/60 Model             | Prototype                  |
|---|------------------------|----------------------------|
| Length of Lower Hull( L )   | 1.7 m                  | 102.0 m                    |
| Distance between Large Columns<br>in Longitudinal Direction( $L_{CC}$ ) | 1.02 m                 | 61.2 m                     |
| Breadth over Lower Hulls( B )   | 1.16 m                 | 69.6 m                     |
| Distance between Lower Hulls  | 0.9 m                  | 54.0 m                     |
| Draft( d )  | 0.4 m                  | 24.0 m                     |
| Displacement( $\Delta$ )  | 161.4 kg<br>(163.5 kg) | 35,730 ton<br>(36,200 ton) |
| Height of the CG( KG )  | 0.3 m                  | 18.0 m                     |
| Metacentric Heights   |                        |                            |
| ( $GM_T$ )  | 0.027 m<br>(0.028 m)   | 1.62 m<br>(1.68 m)         |
| ( $GM_L$ )  | 0.041 m<br>(0.044 m)   | 2.46 m<br>(2.64 m)         |

( ) indicates the moored condition



Table A.2 Test Conditions of Semi-Submersible Platform in Wave Tank

| Condition  |   | Mooring |    | Direction |    | Fairleader<br>Height<br>Series | Metacentric<br>Height<br>Series | Upper Deck<br>Depth<br>Series | Broken<br>Mooring line |
|--|---|---------|----|-----------|----|--------------------------------|---------------------------------|-------------------------------|------------------------|
|  |   | M       | UM | BS        | HS |                                |                                 |                               |                        |
| Regular Waves  | $T=1.12 \quad 1.73 \text{ sec.}$<br>$H_W=10 \quad 25 \text{ cm}$                                  | ○       | ○  | ○         | ○  | ○                              | ○                               | ○                             | ○                      |
| Transient Water Waves<br>( One Shot )                            |   | ○       | ○  | ○         | ○  |                                |                                 |                               |                        |
| Trangent Water Waves<br>(Four Shots of T.W.W.<br>in Succession ) |   | ○       |    | ○         | ○  | ○                              |                                 |                               |                        |
| Regular Wave Groups  | $T=1.1 \quad 1.4 \text{ sec.}$  |         | ○  | ○         | ○  |                                |                                 |                               |                        |
| Irregular Waves  | $H_S=9.1 \text{ cm, } T=1.25 \text{ sec.}$<br>$H_S=12.7 \text{ cm, } T=2.25 \text{ sec.}$<br>etc. | ○       | ○  | ○         | ○  |                                |                                 |                               |                        |
| Currents   | $V_C=7.5 \quad 20.0 \text{ cm/sec.}$  | ○       |    | ○         |    | ○                              |                                 |                               |                        |
| Waves & Currents   |   | ○       |    | ○         |    | ○                              |                                 |                               |                        |

M : Moored Condition  
UM : Unmoored Condition

BS : Beam Sea Condition  
HS : Head Sea Condition

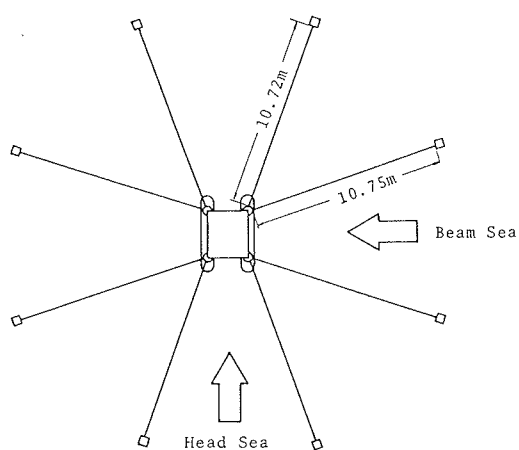


Fig. A.2

Arrangement of Mooring Lines

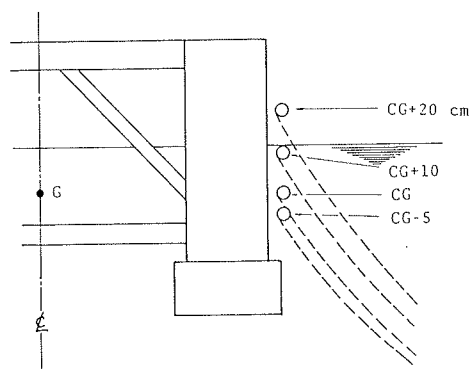


Fig. A.3

Positions of Fairleader

## Discussion

T. Oda (Mitsui Engineering & Shipbuilding, Co., Ltd., Japan)

I am very much interested in the computer time-domain simulation shown in Fig.8.

To show a phenomena physically, time domain analysis is the most sufficient way.

However, the roll damping may be not determined in general.

- (1) How did you determine it?
- (2) Does it agree with experimental result?

It's my brief question.

### Author's Reply

1) In general, there are two kinds of damping, namely potential or wave damping and viscous damping. Since the former is quite small in magnitude for the submerged body, we neglected it while the other damping was taken into account using the relative velocity of the lower hulls and adequate values of  $C_d$ . As an approximation, roll damping were determined by the viscous damping of lower hulls in both vertical and horizontal direction, and the lever between CG and the location of lower hulls.

2) We haven't completed the theoretical problems yet including the comparison between theory and experiment. However, we have tried to compare several experimental results with the theoretical ones even though some reasonable assumptions and approximations are involved. According to those comparisons, the theoretical results are agreeable with experiments as long as the wave steepness becomes higher ( $H/L > 0.06$ ).

Y. Takaishi (Ship Research Institute, Japan)

I deeply appreciate this presentation by Dr. Takarada et al. which concerns with test results on stability problem of offshore semi-submersible platform, because this problem is rather new and needs more effort on the research activity from various points of view, i.e. the environmental condition, wind and wave as well as current loads, independently or combined, the motions of unmoored or moored condition.

We, in Japan, are going to promote a new research project under such circumstances of the arts concerning design loads and stability of offshore structures. I expect very much that the present paper should be very useful guidance for this project.

Finally, I would like to ask the authors the effects of wind loads which they have disregarded in their experiments. Will this load be included in the future study. How can this load be treated in the experiments. Does the wind load act in safe side or unsafe side, or are the effects of wind comparable to those wave and current on the steady and unsteady tilt of the structure?

### Author's Reply

We thank you very much for your kind statement which encouraged us so much. We agree with the importance of wind loads on the floating structure when we discuss about the stability of such structure. Your first question is our researches in future. Yes, we hope we'll try to conduct the experimental study of the stability of semi-submersible platform in wind next. Your second question is the treatment of wind loads in the test. There are many ways to treat wind loads in the experiment. For example, wind load can be directly measured by the wind tunnel test and/or the model test in the combined loading condition may be conducted by using the wind generator in the wave tank. On the other hand, it is also possible to incline the model initially for the simulated heeling condition of the model as a treatment of the model in wind. Your third question is somewhat hard to answer for us. We consider it seems extremely difficult to obtain the general conclusion if the wind loads act in safe side or not by the tank test. Your last question is hard, too. Since our studies concerning with various kinds of loads such as waves, winds and currents have not been completed, it is difficult to answer your last question this time. Thank you.

K. Amako & R. Kojima (Hitachi Zosen Corp., Japan)

The authors should be highly appreciated on this experimental investigation into the influence of mooring lines on the stability of a semisubmersible and also the results obtained by their study are valuable for us to design offshore structure. For this paper, let us ask the following two questions.

1. For the capsizing problem of a semi-submersible, it is very important to predict relative motion of upper deck to wave crest. According to the author it seems that in irregular waves slowly varying tilt induced by second order horizontal drift force should be taken into account for the estimation of such a relative motion. As regards roll motion in low frequency range, it is shown in Fig.10 that the power of spectrum of slowly varying roll motion at unmoored condition has nearly the same magnitude as that at moored condition. The discussers are uncertain on the influence of mooring lines judging from this. Can the authors discuss on the effect of mooring lines quantitatively or qualitatively?

2. Fig.6 shows that the model has a steady trim by about 10 degree in spite of beam sea condition. What causes such a large longitudinal tilt?

#### Author's Reply

1) The slowly oscillatory motions are induced by the 2nd order wave forces when the platform has small restoration in such modes of oscillation. The slow drift oscillation which is now well-known phenomena is caused by wave-induced drift force and the restoration in surge and/or sway due to mooring lines. On the other hand, the slow varying tilts also exist when the platform has small restoration in roll and/or pitch.

When the platform is moored, the influences of the mooring lines, which are generally considered to be small, may exist.

2) When the GM of the model becomes very small, the stability of the model decreases a lot. In this case, the model may incline in the longitudinal direction (even though the test is conducted in beam sea condition) due to the differences of mooring forces in longitudinal direction coming from unsymmetry of the lines and so forth, though the small disturbance of hydrodynamic forces.

M. Ohkusu (Kyushu Univ., Japan)

Rational way of predicting the stability of semi-submersible platforms in waves have been studied by quite a few research workers. I am very pleased new research on this problem is attempted also in Japan and published.

My first comment is concerned with wind force on semi-submersible platforms. The authors stated that they do not discuss it since they are considered in the regulation. Even so, wind force is one of the most serious uncertainty in predicting the stability of the platform. I don't think we are able to come to the exact prediction of the stability of the platform without research works and discussion on the wind force.

Secondly, the authors compared theoretical computation of current induced inclination with experimental results in Fig.2. I hope the authors give us the information of what method they used for this computation.

The third question is about the comparison of experimental records and computer simulation illustrated in Figs. 7 & 8. Although the experimental record corresponding exactly to the simulation is not given, the experimental records seem to generally half as much as the simulation, for instance, of the rolling motion. Do you have any explanation of this discrepancy?

#### Author's Reply

1) We agree with your opinion. We also consider wind force is extremely important for the stability of semi-submersible platform. But, in general, many existing design rules show how to treat wind force acting on the platform even if still many unknowns concerning with wind force exist. Therefore, we decided to concentrate on the study of influence of hydrodynamic forces in this study.

2) We have shown how we calculated the quasi-static results of steady tilt due to current in Fig.5. In the equation in Fig.5, we just input the current force in term D. We used iterative procedure to satisfy these three equations at the same time.

3) Since the wave conditions are different between the experiment and the simulation, the comparisons among those are somewhat uncertain.

However, we have made such comparisons for some cases with good agreement.

N. Matsumoto (Tsu Research Laboratories, NKK, Japan)

Thank you for your valuable presentation. I have two questions.

(1) In Fig.10, the roll spectrum has two peaks at low frequency ( $f < 0.2$ ) in moored condition. Are the lower one induced by swaying motion and the higher one caused by the theory of Mr. Hineno's presentation in this Conference?

(2) In un-moored condition, the roll spectrum also has two peaks at low frequency ( $f < 0.2$ ) in Fig.10.

What phenomenon causes the lower peak?

#### Author's Reply

1) We agree with your comments, generally. The lower peak in Fig.10 may be caused by the coupling term with sway motion while the other peak are induced by 2nd order wave forces.

2) Since the lower peak is located at a frequency of 0.02 Hz where experimental data are somewhat unreliable, we consider it as an error of experimental data itself.

E. Tajima (Kawasaki Heavy Ind., Ltd., Japan)

(1) The paper points out that the mooring system of the unit has very large influence over the stability of the semi-sub. unit. Therefore, I would like to know the detail of the simulated mooring system adopted for the subject model test and the depth of the water during the test. In other words, did you calibrate the effect of mooring line tension over stability of the unit between the tested model and the actual unit?

(2) In table A.1 in the paper, the particulars of the prototype unit is shown. However, please let me know the following, if possible.

1. What are the designed survival wave height and maximum operational (drilling) wave height?
2. What is the designed air gaps for survival condition and operational condition?
3. Is the survival draft the same as the operational (drilling) draft? If different, the model test should be carried out separately for the survival condition and the operational

(drilling) condition.

(3) You will carry out the further study changing fairleader height, metacentric height and wave frequency. Isn't it necessary to carry out the further test changing the wave clearance?

(4) How do you simulate the wind force enacting on the actual unit during the model test?

#### Author's Reply

Thank you very much for your kind discussion.

1) Eight spread mooring lines made of aluminum (0.038 kg/m or 136.8 kg/m in prototype) are used for the model test. The total length of each mooring line is about 11.5 m and the water depth is 2.5 m.

The mooring tension can be separated into the steady component and oscillatory component. The former which is considered to be dominant for the stability of the platform to the extent of our study may be of obtained by the experiment correctly.

2) Since our interest is the basic study on the semi-submersible type platform and therefore, we haven't decided the detailed design criteria of the platform such as design survival wave height, maximum operational wave height, operational draft and so forth. We have just selected the model of the platform which size and proportion are widely designed.

3) We agree your suggestion which may be called as the air gap series test. We consider that lots of other tests should be carried out to obtain more information about the stability of the moored semi-submersible platform.

4) Strictly speaking, it is very difficult or impossible to simulate the actual wind in the wide wave tank. Generally, you may simulate wind moment by putting counter weights on the upper deck of the model.

C. Kuo (Univ. of Strathclyde, UK)

Congratulations on your paper.

I wish to ask the authors to comment on

the photograph I in the paper. Between the 0 to 7 seconds, the tilt is in the leeward direction while between time interval 7 to 11 seconds it tended towards seaward direction. However, our theoretical studies as quoted in Reference 5 of your paper, showed that the model should be able to tilt in either direction. However, our practiced model experiments in regular waves showed tilt in one direction, i.e. leeward and the seaward tilt was unstable and this is true even when we give an initial tilt in the seaward direction. I wonder whether this difference can be accounted by the fact the model is moored?

#### Author's Reply

Thank you very much, Prof. Kuo.

We consider the steady tilt may be induced by the drift force acting on the columns and lower hulls in the horizontal direction at the first stage. Generally, the platform inclines in the leeward direction when the platform is unmoored simply because the wave drift induced moment above CG is larger than that below CG and the resultant overturning moment of the platform is in the direction of leeside down or C.C.W. in Fig. 5.

Once the platform tilts, the additional overturning moment due to the vertical lift force acting on the lower hulls increases the angle of tilt. In this case, this additional tilting moment is in the direction of leeside down because the lower hull in the weather side locates closer to the water surface than the other lower hull.

When the platform is moored, however, the platform may tilt in the leeside or weather side direction depending upon where the fairleaders locate. The additional overturning moment due to the mooring force influences on the initial angle of tilt in this case. When the location of fairleader is somewhat higher than the position of CG, the platform tilts in the weather side direction. Once the platform tilts in the weather side direction, again the additional tilting moment due to the vertical lift force acting on the lower hull (leeside lower hull in this case) increases the angle of tilt.

*Panel Discussion I*

## Philosophy and Research

*Chairmen*

Prof. Chengi Kuo  
University of Strathclyde  
U.K.

Prof. J.R. Paulling  
University of California  
U.S.A.

*Panel Discussion II*

## Criteria and Regulations

*Chairmen*

Mr. Edward H. Middleton  
MIRAID  
U.S.A.

Dr. Nicolay N. Rakhmanin  
Krylov Ship Research Institute  
U.S.S.R.

## PANEL DISCUSSION I: PHILOSOPHY AND RESEARCH

### PANELISTS:

NICOLAY N. RAKHMANIN

Krylov Ship Research Institute  
Union of Soviet Socialist Republics

HARRY BIRD

Department of Trade  
United Kingdom

HARTMUT HORMANN

Germanischer Lloyd  
Federal Republic of Germany

WILLIAM A. CLEARY, JR.

U. S. Coast Guard  
United States of America

ANTHONY MORRALL

National Maritime Institute  
United Kingdom

GÜNTER HELAS

DDR-Schiffs-Revision und  
-Klassifikation  
German Democratic Republic

### CHAIRMAN (Kuo):

Good afternoon, ladies and gentlemen. Welcome to the first panel session dealing with Philosophy and Research. This afternoon, we plan to divide the topics of philosophy and research into two parts as shown on Slide 1. First of all we will spend the first 90 minutes on philosophy with three principal speakers, and this will be followed by research after the tea break, between 3:30 and 5:00, with three other principal speakers. We are going to rearrange the discussion on the panel in a structured way, so that we can encourage as many speakers from the audience as possible. The way we will be dealing with the subject of philosophy will be given shortly.

In the meanwhile I would like to list the three principal speakers for the Philosophy Session. They are Mr. Boroday, Mr. Hormann, and Dr. Morrall. Before begin the discussion, I will pass to my co-chairman Prof. Paulling to introduce the three principal speakers.

### PANEL MEMBERS

Part 1:           Philosophy

#### PRINCIPAL PRESENTERS

N.N. Rakhmanin  
H. Hormann  
A. Morrall

Part 2:           Research

#### PRINCIPAL PRESENTERS

H. Bird  
W.A. Cleary  
G. Helas

Slide 1

CHAIRMAN (Paulling):

Prof. Kuo has indicated to you the topics which we are going to cover this afternoon in these two panel discussion sessions. I take great pleasure in introducing first the three panelists who will make presentations, hopefully, in the intention of stimulating some very lively and interesting discussion on the topic of philosophy of ship stability. You have the names of the panelists before you and I will give a brief introduction of each one.

Dr. Tony Morrall, who is from the U.K., did his studies at King College, University of Durham. Following his undergraduate work, he held the post of Research Assistant and was awarded the PhD degree in 1967, from the same university which is now known as the University of Newcastle. His professional career following his university studies has been spent again with a single institution under two names: the Ship Division of the National Physical Laboratory, which is now known as the National Maritime Institute. Dr. Morrall is the first person at the table on my right.

The next member of the first panel is Mr. Hartmut Hormann from Germanischer Lloyd. Mr. Hormann graduated as a naval architect from the Technical University of Hannover and Hamburg, and joined Germanischer Lloyd in 1965. He has served various functions within that society, mainly in the areas of ship safety and has been continuously engaged in IMCO work. After a short period with a German shipyard, he returned to the Ship Safety Department of Germanischer Lloyd, and since mid seventies has been the head of that Department.

The third member of the panel as originally constituted, Mr. Boroday, is not with us today. But I will read his curriculum vitae. He is a graduate of Leningrad Ship-building Institute in 1957, and received a PhD degree from that Institute in 1962. He became a professor of Krylov Institute in 1974 and his technical studies have been concerned with ship motions in regular waves and ship stability. Dr. Boroday's presentation this afternoon will be given us by Dr. Rakhmanin who was introduced in the session this morning.

CHAIRMAN (Kuo):

Thank you very much. Just to give you some idea how we will conduct this panel session, I will now show you Slide 2.

There will be nine items to be discussed at around 10 minutes approximately on each area. We have received certain written contributions and some of them are very long. We will ask them to be as brief as possible when they are called upon to take part in the discussion.

In the last item my co-chairman Prof. Paulling will try to summarize the highlights of the discussion both from the panel members as well as from the audience.

PANEL DISCUSSION SESSIONS - PHILOSOPHY AND RESEARCH

PART 1 - PHILOSOPHY

Questions/Views - (10 minutes per item)

1. What is stability?
2. The Aims
3. Forms of Criteria?
4. What gap between excitation (forces) - response (restoring)?
5. Judgement of stability
6. How to link with designers?
7. Possible approaches
8. How to link with practice?
9. Summary

Slide 2

Again to give you some idea how we will approach the first item, we would like to show you the sort of questions perhaps we will address ourselves when we are dealing with the question: "What is stability?"

WHAT IS STABILITY?

Possible areas for discussion:-

- a) Safety against capsizing
- b) Stability of equilibrium
- c) Large rolling

Slide 3

For example, in "What is stability?" (Slide 3) we may well ask: "Stability is concerned with:

- (a) Is safety against capsizing, (b) stability of equilibrium, (c) large rolling?

If we could have the slide 2 back again, you will notice that for the items with question marks, we hope to have you, the audience, to give comments. Whereas those without the question marks will be presented by the various panel members with the aim of highlighting the key features of their papers.

So, let's start with "What is stability?" and we have roughly ten minutes to work on this topic item. Who would like to be the first one to give a view or comment on "what is stability?" Any volunteers? Yes, Mr. Cleary.

CLEARY:

We have so little time this week that we should not use any of it for thinking but most of it talking.

Stability already has been defined this morning by the various speakers in several different ways. In our country, we have naval architects trying to decide stability criterion on the basis of economics for the owner. They do not speak of safety. So, I always remind them of safety. Other naval architects design a stability criterion on the basis of operational excellence of the vessel and leave out safety. So, I remind them of safety.

I think there is not one single answer to the question. We have to remain open to the fact that we need to have protection for the operation of the vessels, so that it will not roll so much that passengers will refuse to use the vessel, or cargo will not fall over and be destroyed or lost over the side. Finally we need to examine the same model, the same ship, and the same set of mathematics in a very severe sense so that we will not lose the entire ship.

CHAIRMAN (Kuo):

Thank you very much. Now, would anyone else from the floor like to give a definition? Yes. Could you give your name and organization?

O'DOGHERTY:

I am Admiral O'Dogherty from Pardo.

I agree with the views expressed on defining stability and I feel that stability has two aspects. One is safety and the other one is good movement of the ships. That is, safety is often sensed to be related to the pleasant movement of the vessel. It is necessary then, to find a sort of middle ground since having excessive stability often means having excessive motions. So, one must endeavor to establish a compromise in which ship's safety is fulfilled and at the same time, the rolling movements are not so excessive, that they will cause the possibility of moving or shifting of the cargo. GM must not be too high but nevertheless righting arms must be satisfactory.

Thank you.

CHAIRMAN (Kuo):

Well, thank you. Now, we have two views, safety and good motions. Does anyone else have any views on what is stability?

HELWIG:

My name is Helwig from Rotterdam Marine Academy. Speaking from the view of a captain of a ship, stability might be defined as the power to withstand heeling effects without an extreme behavior of the ship in

the seaway.

Thank you.

CHAIRMAN (Kuo):

Thank you very much for the contribution. Any other views? Yes, Dr. Odabasi. Briefly, please.

ODABASI:

One can define stability in two ways. The first one is the intuitive definition. The intuitive definition is that the ship will remain around its upright equilibrium position and the oscillations around it will not create any danger to the operability of the ship or to the mission of the ship. If you try to translate it to an abstraction, then it will mean that oscillations of the ship will remain within the domain of attraction and within the bounds defined by practice.

Thank you.

CHAIRMAN (Kuo):

Well, thank you for that definition. Someone from the back please.

ST. DENIS:

St. Denis, United States. Things are getting very complicated. One is not only talking about a definition of stability but also about criteria. The two are different. Stability is very simply defined. If a ship is disturbed from her position of equilibrium, it returns or does not return to her point of equilibrium. If she does, she is stable. If she doesn't she is not. Now, there is no discussion about that. That is the only definition of stability. The problem arises in defining the criteria by which to judge stability. And these criteria are related to the techniques of making calculations. That is where all the argument is. There is no use of talking about extremes and this and that. The techniques will probably be statistical, and therefore will involve extreme values, if you will. But this is a separate discussion.

CHAIRMAN (Kuo):

Right. Thank you very much, Dr. St. Denis. Mr. Manum at the back, I notice your hand was up.

MANUM:

Thank you, Mr. Chairman. I agree that you could divide it into the behavior of the ship and the safety of the ship. But I think that we should use our time, the seconds and the minutes as Mr. Clearly said, we should use it as good as possible. And then we should not deal very much with the behavior of the ship. That is self-regulating. If no passenger will use the ship, then, of course, the passenger ship in the



future will be re-designed. The same concerns the cargo-ships. If we cannot carry the cargo, we are sure that the owner will re-design the ship. But as I said, during the previous session as relates to capsizing, the experience shows that is not self-regulating. That has to be regulated. It has to be taken care of by scientists in a proper way. It has to be regulated by the administration. As you can see from the paper by Hormann, there has been no progress at all since Rahola, since 1939 or something like that. So, I think we should use our time on the part of stability that deals with safety, that should prevent against capsizing.

Thank you, Mr. Chairman.

CHAIRMAN (Kuo):

Well, we have already got several interpretations of stability. Would anyone else like to put the last one in? Yes, Mr. Fulford in the front.

FULFORD:

Fulford, British Shipbuilders.

I think, to begin with, we've got to remind ourselves what we are trying to do, which is "to produce better ships" and "the ships that are safe". And there is little point defining stability as such but we should really concentrate on defining the characteristics that are acceptable from carriage of passengers, crew, and cargo. And that should be the only base for any criterion rather than an academic definition of stability, whether from a mathematical or physical point of view.

CHAIRMAN (Kuo):

Well, thank you Mr. Fulford. This leads us directly to the second point about the "Aims". Dr. Morrall, perhaps you can make a presentation for ten minutes or so on what you believe the aim should be.

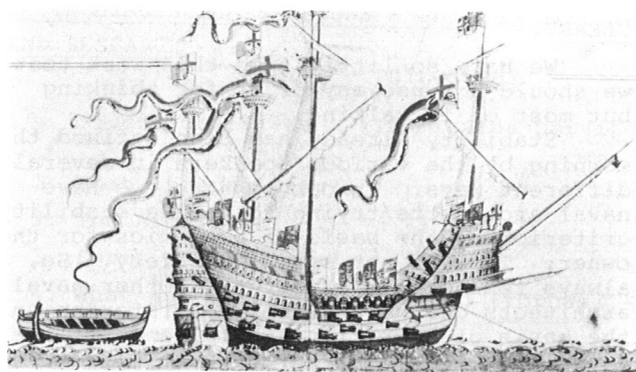
MORRALL:

Thank you, Mr. Chairman.

Ladies and Gentlemen:

What I hope to do in the next few minutes is to go through the main field, if you like, of the present position on stability assessment and end up with aims which may be controversial, but which in my view, provides one avenue of solution to tackling this difficult problem.

I would start, therefore, by considering our data base of casualties. I am sure we know the casualty data can provide some information which is of use in a conventional risk analysis sense. But unfortunately the causes cannot always be ascertained with the great degree of accuracy needed to enable design improvements to be made.



Slide 4      The Mary Rose

Let me just illustrate this by having the first slide, if I may, please.

I am sure those from the U.K. will recognize this. It is the Mary Rose which was recently brought up from the seabed. Let us just consider this example because it pulls out and illustrates two main points.

The Mary Rose which was the Vice Flagship of King Henry VIII capsized in the year 1554. The reasons why she capsized are not known. This is nothing new, of course.

There are two possible explanations. One is the so-called gunports, which you will see with the guns poking out, were too close to the waterline. So, this could lead to flooding of the ship. A so-called design problem, you might say. Or it could have been carrying too many crew on board. A question of overloading or an operational problem, perhaps. We don't know. This is our first dilemma that the casualty data that we look up are not always helpful, because reasons of capsize can't be established within any degree of confidence. It is perhaps interesting to point out that some 400 years later we invented the IMCO criteria. What did we use before we had IMCO criteria? And how long is it going to be before we come up with the new criteria? I will leave that to one side for a moment.

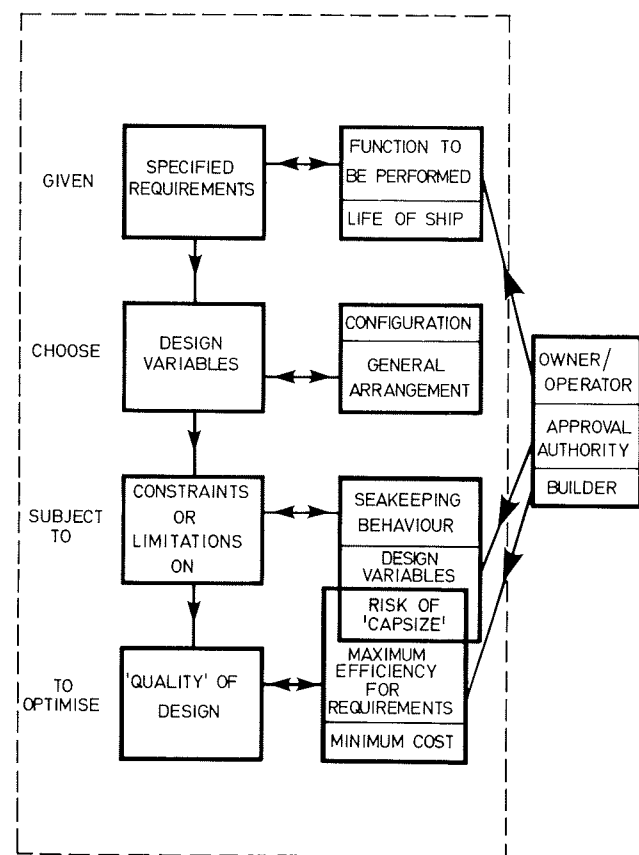
Let us move onto the second dilemma. In recent years, there seems to have been an overemphasis on the modelling of ships using theory. In contrast, very little attention has been paid to developing a framework for future stability criteria. In my view I think this is essential. And I will illustrate this in a minute. There seems to be little point in pursuing any further work on either linear or non-linear single degree of freedom equations because these are not very helpful. The theory in this form is not suited to our present purpose of predicting capsizing.

Two reasons suggest themselves. One is that the theory cannot predict capsize in its present form and there is little hope of ever calculating, or even a sort of estimating the probability of capsizing using these simple ideas. The second is that the problem is too poorly defined. So,

here we have our second dilemma. What do we do? In the first case, we have poor casualty data, and in the second instance we have poor theoretical models.

Let us just examine how the theory in the present form can help us. And I believe there are three ways that it can be of some use. It can help us understand some of the important parameters that are involved. It can also be used to extrapolate the best available data to more extreme conditions, and perhaps over the lifetime of the vessel. And thirdly, it can help us understand the deficiencies of the present techniques, and point us in ways which can help us formulate better theoretical models and better experimental models.

Let's look at the next slide.



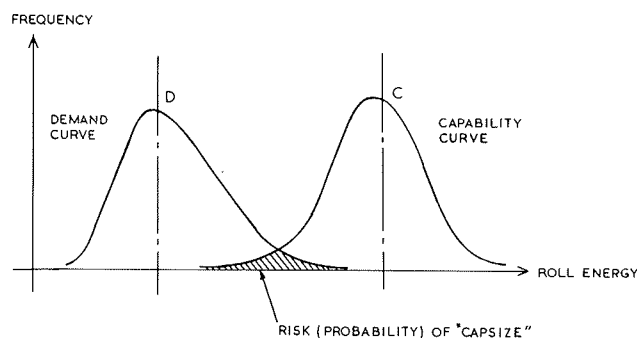
Slide 5

#### A Design Process for Stability Assessment

Coming back at this point of time to something which I was mentioning earlier on the framework, it seems to me that we are faced with the situation where the designs of ships in current use are changing very rapidly, and extrapolation of present ideas on stability are no longer possible. Just to mention one example, the so-called SWATH-ship or the twin-hulled ship. It is clearly not possible to extrapolate the IMCO criteria into the regime of SWATH-ships. So what do we do? What I am suggesting and

what we need to do is to develop a rational framework where we treat this problem of stability in the design stages. The slide behind me illustrates this point. We should also consider the specified requirements. These are the so-called environmental conditions, not just a design wave but a whole set of waves. If you come down the diagram, we can see the seakeeping behavior or features in one of the boxes and below that box--I have put this point of "risk of capsize". My view is that if you use this technique for all types of ships, it will be found to be a much more rational way of approaching the problem of designing adequate levels of safety and stability. This is because it will, in the first place, give a consistent approach and also show us where the uncertainties lie.

I will now go onto my final slide. It is my view that the future infact stability criteria and the related safety of ships should be ideally rationally quantified in terms of the risk of loss or the risk of exceeding certain bounds of motion, or more specifically roll motion. Here I am advocating the use of the environmental risk model as I quote it.



Slide 6 Probability Course

In this, we set up, in the top of the diagram the so-called environment, the ship is likely to be encountering during its lifetime, and end up with the response, the overlap indicates the risk of loss. This should provide the designer with some indication of the vessel's capability.

In conclusion, the aim, in my view, should be to provide the designer with the indication of the vessel's capability. And this I believe can be carried out by comparing the so-called environmental demand over the lifetime of the vessel to the vessel's capability measured by its response.

Thank you very much.

CHAIRMAN (Kuo):

Thank you very much, indeed, Dr. Morrall for both keeping to exactly ten minutes and pointing out the aims.

Can we return to the slide 2 again.

Now the next item to be looked at will be the forms of criteria. We talked about

the aim, we know something about stability. Now, let's go on and discuss in what form should we try to achieve the aims and to obtain the desired stability by looking at the criteria.

Who wants to give some views on the criteria or the form of criteria that will take? Yes, Mr. Rakhmanin, would you like to speak briefly on this?

RAKHMANNIN:

Thank you, Prof. Kuo. Just a couple of words. About the form of stability criteria, I think that one should think not only about the quality of ships, namely stability, but also about the external forces which adversely affect this quality. So, it seems to me that when we speak about the form of stability criteria, this is the stability criteria should be based upon some physical models on the forces acting on or heeling the ship.

Thank you.

CHAIRMAN (Kuo):

Right. Anyone else would like to give a view? Yes, I see a hand up at the back.

HONKANEN:

My name of Dr. Max Honkanen from Finland. There has been quite a lot of talk about the vessel's capability of withstanding the forces of the nature, but as it is pointed out, the stability is mostly a problem of safety. This can be affected by other means as well, including the operational means, for example, how the crew act. There is also another problem. How does the crew know what the actual situation of the vessel is right when it is sailing in these stormy waters? The development today is going very much towards the direction that there will be computer on board, which will perform these calculations. Therefore, it is equally important to establish criteria that are applicable to that particular vessel in question and not only general criteria that can be accepted by all possible vessels.

That was my comment. Thank you.

CHAIRMAN:

Well, thank you very much. Any other comments or putting forward of an idea? Yes, Dr. Odabasi.

ODABASI:

Odabasi from BSRA. It is obviously right that the ultimate form of stability criteria, must be set forth in the form of a risk assessment of some sort. But that assessment will immediately require the definition of "What is demand?" and "What is capability?" The idea put forward by Dr. Morrall, using simulation as a means of assessment, does not quite fit the bill. In

order to determine stability criteria you must first be able to satisfy the conditions in deterministic cases. You have to demonstrate that your model is satisfactory under a controlled condition, and then generalize it to the random environment. If you can't do that in steps, although it may not be satisfactorily achieved in practice right now, that is still the proper way to go about finding the proper cause/effect relationship. We shall try to get the right results for right reasons, rather than right results for wrong reasons.

CHAIRMAN:

Okay, thank you very much. I have another speaker in the front row and at the back. Please.

BARR:

Roderick Barr from the United States. It seems that the goal that we are slowly moving toward, which may be quite difficult to achieve, is the evaluation of capsizing and stability on the basis of risk analysis. However, we must realize that when this goal is achieved or achievable, we must then, in a sense, turn backwards and be able to provide criteria which are understandable to mariners. Mr. Honkanen has mentioned the use of on-board computers. But I think many of us are aware that the ship operators are inclined to rely very much on their own experience and wish to do their own decision-making, and it may take many years before many mariners--particularly those involved in small vessels like fishing vessels--are willing to rely on computers. So, once we have improved criteria which we can use to design ships that we believe to be safe, we must then provide means for the masters of ships to interpret what their operation of the ship will do to the stability. No matter how well the vessel is designed, the operators may find means of exceeding the balance that has been established in the risk analysis, therefore, leading to a situation where a casualty can occur.

Thank you very much.

CHAIRMAN (Kuo):

Thank you very much for the comment. Anyone else? Yes, Dr. St. Denis.

ST. DENIS:

A little while ago I suggested that the definition of stability, to me that is, be simply that a ship is stable if she returns to equilibrium when disturbed by the actions of environment. However, whether she returns or not is depending not only her design but also on the way she is operated. In my experience, we found out that the ships have failed to return to the position of equilibrium because the skipper or the master had not sufficient judgement. Now,

when you are going to talk about criteria, you will have to introduce the judgement of the master somehow. Nature is far more predictable than man. And what is being done now is to simply assume that whoever runs the ship is a person of adequate judgement. This may not be the truth, and in many cases it is not. I challenge the panel on how to introduce this criterion of adequate judgement.

CHAIRMAN (Kuo):

Right. Any other comments? Thank you. Anyone else who wishes to comment on "What form of stability criteria should take?" Yes.

MACNAUGHTON:

MacNaughton, Department of Trade, United Kingdom. We are talking about two different kinds of criteria. One, which is the one we are really concerned with here, is first of all the design criteria. And then we are talking about criteria as used by the operator. We should never lose sight of the fact that the operator, we talking about the small vessels, just does not understand very much about the subject of stability at all, particularly if one is talking about fishing vessels. They have no conception of stability. They pay no attention to it in my experience. As far as the masters of larger ships are concerned, who can predict their vessel's stability state before leaving port and during the voyage, which is normal in a well-run company, it is in my view extremely important that whatever criteria you use have to be simple. The present criteria we have are not simple and not to them. If we talk about the application of IMCO A-167 standards, very, very few ship masters in my experience actually ever compare the stability of the ship against the A-167 criteria. They establish through practice and experience a simple GM value for the ship in various conditions. And once they have become used to that, that is the approach they take. It is fair to say that as far as the operator is concerned, you can only change two things---the center of gravity of the ship and the weight. That is all he can do. And I will suggest to the panel that as far as the criteria for the operator is concerned, that is all that we should give them.

Thank you.

CHAIRMAN (Kuo):

Thank you, Mr. MacNaughton, you have given a view which I think is interesting. Can I have perhaps one more comment?

HELVIG:

To my opinion, criteria should contain minima and maxima, and both should bring into the field the type of ship, the cargo

the ship is carrying and the voyage, that means the external conditions.

CHAIRMAN (Kuo):

Well, thank you very much. I think the points being raised have been very interesting. And let's move on again, because we may have time to come back again. Dr. Morrall raised the point, about the "demand and capability approach". Do we really know enough about what are the forces, excitation and restoring, and on the points mentioned earlier in the discussion by Mr. Rakhmanin? So, perhaps, someone in the audience would like to comment on this item of excitation and response. Well, while you are thinking about it, perhaps Dr. Morrall, you would like to amplify a little on this?

MORRALL:

Yes, Mr. Chairman, you want me to talk about demand and capability. I would like to make the point -- on two points -- on this concept.

Those of you that are familiar with structural engineering will be familiar with this concept of demand and capability. Two diagrams appear on the conference proceedings, these two, one in my own paper and one in Mr. Bird's paper to be given to you later on, (Slide 10).

Now, I would like to tell you that this is just a concept for which we are trying to put out the view that there are two elements. One is the so-called environmental demand, and the other is the ship's capability measured in terms of response.

It is clear that this is too simple an approach to transfer from structural engineering straight into our concepts of stability assessment. Mainly for the limitation that the two values of demand and capability are assumed to be interdependent. But in practice this is not the case. There is a coupling between the two. But I think we can overcome this problem, and work going on in U.K. is looking at this particular problem.

CHAIRMAN (Kuo):

Thank you, Dr. Morrall. Perhaps, we can now find anyone else who would like to discuss the gap between the excitation and restoring. Anyone? Yes, Dr. Odabasi.

ODABASI:

Sorry, Mr. Chairman, I have to talk again I guess.

The gap can be talked of in terms of two items, a) the definition which Dr. Morrall was trying to give, in terms of demand or capability, or as I frequently refer to as normal measure, and b) the second one is how well you represent those things, that is another gap there, which is I think more to the point at present.

In the paper by Dr. Morrall, it was

mentioned that there was, in the first place, an overemphasis on the mathematical modelling of roll motion. It was also mentioned that there was a lack of a reliable mathematical model. I guess both can be true, and I know from practice that the latter is true.

So, I will suggest that the first gap is to represent those quantities in a reasonable way by establishing a proper cause/effect relationship, and then, derive design guidance. That is the only logical way to go about. Because if we can determine what parameters are important and how the hull-form and the environment regulate them, then we can at least find some way out in the design procedure.

CHAIRMAN (Kuo):

Right. Thank you very much. Anyone else? Yes, right at the back, front of the second half?

JOHNSON:

Bruce Johnson, Naval Academy. Are we allowed to show transparencies?

CHAIRMAN (Kuo):

Yes, certainly, if you could perhaps arrange for someone to take it up while you are talking?

JOHNSON:

Could I speak right from here?

What I want to speak to is the problem of characterizing the excitation. And I think this is really the central problem in capsizing. We have been doing quite a number of capsizing studies and this is a very incomplete presentation of a paper I will be presenting next year. But right now we are considering several characteristics. For instance, if we take time domain measurements of the front of a wave as it is about to break and cause a capsize, then I have been looking at a number of different parameters, the depression and elevation parameters, the rise rate of the front of the wave, the rise rate of the back of the wave, things of this nature. The reason I raised this is the very excellent criteria set up by Kjeldsen in Norway only work in the spatial domain. And we don't have spatial domain records. We only have time domain records. So, no one, to date, has characterized the excitation forces in a capsize. I think this is a very central problem. Rakhmanin and I have talked about this somewhat at the last ITTC. In other words we have very inadequate characterization data of the wave, itself.

Now, in response to Prof. Ursell, I will show him just a small number of the parameters that we are currently investigating. This is work in progress.

These are all non-dimensional parameters. We look at the elevation of the

wave, the differences between the velocity, the front and back of the wave. There are all kinds of dimensional parameters. The matrix is incredible. But we are currently working on that. Without boring you any further, let me just show you a couple of the problems.

This slide is a plot versus another parameter that I won't explain. But the important thing is that the abscissa is the velocity ratio--that is the velocity of the crest related to the velocity of the zero crossing. And notice that it is not one. It varies anywhere from less than one up to about 1.26. And since that is true you cannot make spatial transformations. That is a non-linear function. And we do not have any way of doing this. So, with that in mind, I will just show you one last slide.

Here is a breaking/non-breaking criterion which we have just recently measured. This is the average rise rate of the front of the wave divided by the zero crossing speed---I am sorry I didn't write this down---as a function of non-dimensional spacing downstream of your measuring point. And we have found that we can predict a breaking wave out away from the point of measurement. That is to say, the problem with most data currently available is that, if it is a single probe measurement, you have no indication whatsoever whether a capsizing wave is likely. We seem to be finding a minimum value above which the wave will break and below which the wave will not break. That is in the characterization of the excitation forces. Until we can do this, I don't think you are going to make any progress in your criteria.

CHAIRMAN:

Right, thank you very much, Prof. Johnson. I think the point is that this is a complicated problem and the gap is large. Can I perhaps ask Mr. Hormann to comment before we return to the audience again. Mr. Hormann.

HORMANN:

Thank you, Prof. Kuo. I am a little bit puzzled by this obviously demanding question, "What is the gap?" The audience understood it in a different way to that I do. And especially the last discussor and part of the discussion, made it entirely clear to me that we do not just have the gap between the excitation and the response but we do have the gap already in all the different sciences and different research projects which look at one of the factors. This looks even worse than one finds it to be at the first glance.

CHAIRMAN (Kuo):

That is interesting perhaps we can now move on to you again, Mr. Hormann, to talk

about judgement of stability in the light of that the point raised. Again, we may go back, if anybody has anything else to discuss later.

HORMANN:

Thank you, once again, Prof. Kuo. To bring you back to earth, this is one subject of my presentation here.

The item is "Judgement of Stability". And we all know that there is a statutory basis for that. This statutory basis is just one paragraph. It hasn't changed since 1929 when the first so-called SOLAS Convention came into being. And this paragraph I just quoted reads: "The master shall be supplied with such information on stability as is necessary to permit efficient handling of the ship." This is all. I mean the intent is right and there is a pointer that stability has to be borne in mind. But the evident complementary step is yet to be done. That is, what are the criteria against which the captain or the master has to judge his ship? Well, let's look at what we have.

Still, the only and almost only, more general thing we have is, I would call it still, the Rahola Criterion which was of course refined and brought about in a more sophisticated way by IMO in earlier years. But this set of criteria relates to a well defined and rather small group of ships, as we all know.

Now, especially in the last two decades, we have had quite a variety of new types of ships coming about. And again the question has to be raised: What is sufficient stability for such new types of ships? No one can reasonably agree from the onset that the Rahola Criterion or better the IMO Criteria would be suitable for such cases as well, and at least without being examined in a detailed way.

Now, let me take--as an example, could I please request the overhead graph which I prepared a few minutes ago? Please excuse me for the bad manner in which it is done. I take, as an example, a modern container ship. If you look at what I call No. 1, that is an ordinary type, medium-sized shelter deck ship, what we naval architects normally call a shelter deck ship, with a GM value which is great enough to make the ship comfortable and with a comfortable range of stability. You see that the curve is well above the GM line. You see that is a ship with a high, efficient freeboard.

The second one is what we normally call a full scantling characteristic. There you have a much higher GM value. And you see the GZ curve does not rise above that GM line. But still we end up with a nice range of stability.

Now, the modern container-ship which, by its very dimensions, the characteristics of its dimensions, has a bad combination of both these curves. You see we are back to

GM values which used to be only applicable so far for smaller shelter-deck ships, or shelter-deck ship at all. But now we have no longer the shelter-deck characteristics of the GM curve. I tried to indicate that in case No. 3 by saying that the range of stability is somewhere around 50 degrees and sometimes even less.

Well, this is just one example where, from the side of those having to judge what is sufficient stability, we have problems. So far you all will agree that no one can say 50° range of stability is sufficient or is not sufficient. And when it comes to a designer or an owner offering a condition of loading of that kind, he will not accept if we, as having to judge that, say that we do not know it quite right but we don't believe it is enough. He will demand a proof that it is not enough. Otherwise he wouldn't abstain from using that condition.

Well, what can be done--or better to ask, "What is being done?" We, if you look at what is going on in the world in this area, calculate more and more accurately the heeling moments of a ship, the heeling forces of a ship. For example, the wind moment. I won't go into the detail of this as you will find that in the paper. But even if we calculate the heeling moment in a very precise way, no one would take it seriously because the comparison is made with a very simplified curve on the other side. That is the uprighting moment in still water, which you all will agree is a highly theoretical curve when it comes to the real ship being in a seaway. In other words, there is no balance in the two sides of the equation.

Well, in my view, the decisive question still waiting for solution is: What have we to provide to counteract the forces imposed by waves? So far, we have just empirical answers. For example, the most far-reaching answer of that nature is again the Rahola IMCO set of criteria deducted from statistics for a small group, I repeat myself for a small group, out of all the ships which are ploughing the seas. But it is all clear that we must not wait for another statistic to come up to judge modern types of ships. I mean another statistic of that type.

The way we have to follow in my view is, therefore, a combination of tank tests and calculation. This item will be spoken to in many other papers during this week, I think, and I need not spell out what this in detail would mean.

Let me add few remarks at the end of my presentation here.

The first is, please look at the level of safety of the ships today as compared with those of 30-40 years ago. I don't mean stability, I mean the overall safety. And you will certainly agree that in many areas undoubtedly the level of safety has been raised during the decades. Look, for example, at the fire protection. No one



will doubt that the fire protection measures, both the passive and the active ones have had far-reaching improvement during the years. Or take the nautical equipment. It has been tremendously developed. But not so the matter of stability. I don't mean the provisions on board to judge stability. I mean the very subject. If you look at the earlier types of ships, you will find many more ships which are far beyond, far in excess in their stability data of what we even could question whether it was enough or not. I said "far beyond". Nobody needed to care in earlier designs. But now we are faced with quite a bunch of ships or with quite a bunch of ship types which we have had feelings with, again, of the nature that we are not sure that it is not enough, but we have neither a proof for one way or the other.

There is another point--I may add just this one sentence. That is, do not forget in all these discussions here this week that the main cause for the so-called stability accident is the shift of cargo; no ship in turn can be constructed and designed in a way that it can withstand each and every shift of cargo. Please exempt from this the tankers and similar ship types.

At the end I would quote from my paper just two or three lines, to tell you what a master of the German naval architecture, i.e. Prof. Horn, once said. Please be aware of the fact that this was said in 1943. That is almost exactly 40 years ago. I read it out.

"Under these circumstances, which did not yet allow a solution on a physical-scientific basis, so far perforce a route was taken to the extreme contrary, namely using pure experience."

And he continues in saying,

"One is dependent on the lessons of experience, which one probably will have to use in future also."

I think Prof. Horn would be very much surprised in learning that he is right still today 40 years after.

Thank you.

CHAIRMAN (Kuo):

Well, thank you very much, Mr. Hormann. We can see that we hear that we cannot define excitation very well. And here you have Mr. Hormann wishing to use the information, because he is doing the actual work. Now, let's move to our next subject, on slide 2, which is trying to see how we can link with the designers on stability we were talking about previously. And I have an indication that two persons would like to talk about this area and perhaps I could ask them first of all. Dr. Kastner, would you like to talk on this particular item?

KASTNER:

Thank you. I want to shorten my

written contribution to Dr. Hormann's paper, but want to raise two or three main points.

First of all, I think it would a tremendous step forward if authorities and designers would just apply some of the many methods suggested and available for the purpose to test in practice and to give feedback to the people who have designed these criteria. I think, by the way, we have progressed far enough to do so. We do not need further refinements just to start working with any of these criteria.

The second point is that, in my view, probably no one really expected ships of up to 100 meters designed to fulfill IMCO requirements to be safe under severe environmental conditions. In my view, this has been just the "minimum" value which people in the working group could agree upon.

So, my point again is that designers and the authorities just should apply in their designs more than just rules.

The last point which I think is very important is the burden of setting the practical limits, the burden put on authorities. I fear no researcher can release authorities from this burden. But certainly this is related with public acceptance of safety or unsafety. If we compare with the very sensitive safety aspects of the nuclear reactor technology, where I had an opportunity to work in for some time, all safety measures in Germany are taken according to a law. It is defined there by a law which says, I quote:

"All measures have to be taken according to the state of the art in science and technology."

As long as no rulebook is available, and there are no rulebooks on these aspects, I think if we take up this line, this would be very much in line with Dr. Hormann's plea for more detailed experimental and numerical testing, if not for every ship, at least for any ship type.

CHAIRMAN (Kuo):

Thank you very much, Dr. Kastner. Mr. Jens, would you like to come in at this point? I notice you raised your hand early on.

JENS:

Thank you, Mr. Chairman. Earlier, it was said that the master of the ship does not quite understand the criteria devised by IMO. That is the geometrical configuration of the righting arm curve. But he is responsible for the stability of the ship. And I think it was also pointed out by Mr. Hormann that the master should be supplied with enough information to judge the stability of his ship.

Now, the righting arm curve, the criteria for the righting arm curve and whatever other criteria you have these are designed, I know, in the years 1964-1966 to be provided as guidance to the designers of ships and to the administration to judge.

There is one full page of this very resolution on what then should be given to the master. That is: (1) The standard condition of loading, (2) perhaps GK limiting curve, and (3) other information which he could easily and readily understand and use, also to make correction what to do when you have deck load or to limit the center of gravity, and so on. There is a very long page. Now, it was also said when we made these criteria out of the casualty statistics, we had, for the ships on our hand, about 100 casualties for the coasters alone. And, therefore, we limited the criteria for ships up to 100 meter length. There were a couple of ships above 100 meter length. So at that time I thought we had got enough information from these casualties statistics and also on the intact stability calculations. We were aware, however, that these criteria were applicable only for fully loaded conditions and not for light ballast conditions. Unfortunately, that has not been mentioned in our statistics, but it has been always in our agenda. Prof. Prohaska at that time said that he will deal with that later. "We don't want to have some rules that will multiply." He set the GM values by the ratio of displacement. And he developed weather criteria. That was all in years 1964 and so on. I think we were nearer to this.

Now, times have changed. Technology has changed. The average freighter-coaster, what you have, is now replaced by sophisticated and specialized types of vessels. So, gradually, we tried to keep pace with it. You see in the international organization the pace is rather slow. We devised some additional criteria. One is, for instance, off-shore supply vessels for ships with a low freeboard, wetting of deckload in which water can be trapped, and so on. We have devised some additional, actually the better criteria for the drilling rigs. We have further for fishing vessels and so on, as there has been done a lot in this way.

CHAIRMAN (Kul):

Well, thank you very much, Mr. Jens. Would anyone else like to make some suggestions, comments, somehow to link with the designers on stability? Yes, right at the back. Would you be very brief, please?

DAHLE:

I agree that there is a lot of information available for designers. But for the time being the designers should not be too confused by a meeting like the one we are having here. There are a number of good advices to those designers.

Three examples. Cargo shifts, if this is the problem as seen by Mr. Hormann, the designers should be given better instructions. The reference is again to timber carriers where lashings are very well specified in the rules M. Jens mentioned. If

this is a big problem, the designer should be advised to improve upon this situation.

The second problem, water on deck was solved by Dr. Rakhmanin in his lecture earlier today. The solution is obviously a better GM or rather, as I see it, the extension of the GZ curve.

The third advice to the designer is quite obvious. To improve upon the damping of the vessels we will hear later in these lecture series that the damping is inadequate and very low for a lot of boats. So, there is obviously a lot of advice that can be given and we can still go on discussing the final points in conferences.

Thank you.

CHAIRMAN (Kuo):

Thank you very much, indeed, for the comments. Anyone else? I noticed a hand early on. Will you like a comment on this? Yes, very briefly, Dr. St. Denis.

ST. DENIS:

The designers at least in the United States endow the ship with the ability according to a certain criterion, which is law. There are two small criteria. One, you all know the energy required to restore --the restoring energy of a ship up to the second intercept might exceed by 40% the heeling energy when the ship is acted upon by a wind of 100 knots--of course how the 100 knot wind or what kind of wind it is--whether uniform or gusty, is not said. Nothing is said about it. It does make a great difference. But the important thing is this that when a wind of--that magnitude, but I have never been at sea when a wind of 100 knots blew, does blow the waves are pretty steep and and pretty high. Yet, the criterion does not take this effect into account anyway. In fact the first paper of the conference considered some tests on the Pentagon and it subjected the rig to a wind heeling moment but not to anything by waves. Well, I don't understand how you can endow a ship properly with the required stability when you neglect the strongest of all the upsetting forces of nature.

Here is something to think about.

CHAIRMAN (Kuo):

Thank you very much. Now, can I now pass to Dr. Rakhmanin, who will be presenting briefly on the possible approaches using the paper of Mr. Boroday.

RAKHMANIN:

The report of Dr. Boroday's deals with possible approaches to the estimation of the effect of wind heeling load on the ship which experiences motions in sea waves of arbitrary heading angle. It is pointed out that the question of practical merit of the approaches based upon the investigation of stability in the sense of steadiness of the



regular roll process is an arguable one.

It is found expedient to estimate dynamic stability of an oscillating ship on the basis of probability characteristics of the maximum angle of heeling on the leeward side, which occurs immediately after action of heeling roll. These characteristics may be calculated according to Dr. Boroday's method suggested and presented in the Stability Conference in Glasgow in 1975.

At first sight, it is quite natural to consider stability which is the quality determined by statics of a ship, when estimating safety of a ship sailing in stormy conditions. However, the usage of the motion stability of a ship itself is in relation with the more general case. When a ship experiences motions in waves, if it is also affected by wind is quite questionable. In the report the modification of rolling equation is presented. This equation takes into account an arbitrary wave heading, an arbitrary wind load, variations of righting moments in waves, transverse drift forces, as well as wave characteristics.

When the analysis of this equation is performed for small disturbances, the problem is reduced to the investigation of rolling stability approximately. In principle such investigation can be performed either on the basis of heel disturbance situations or by energy approach, in case for regular and irregular waves.

For large disturbances including consideration of ship's capsizing, it is apparent that stability or steadiness of rolling can be performed for given stability in case of regular waves, if the method of phase plane is employed.

A prospect of expanding this phase plane method for solution of a less schematic problem which deals with stability of regular ship motions in case of random value disturbances, is arguable. On the strength of that, in the report, there are given reasons showing that real large disturbance due to wind gust may be approximately described by rectangular function of time.

At this kind of wind load the maximum heeling angle of a rolling ship is observed at the first inclination following the wind disturbance. For such a case, an estimation of ship's safety can be made on the basis of the equation presented in the first section of the report, by means of transformation of this equation into a work equation, having in mind righting moment work including variations due to following waves, heeling moment work, and kinetic energy of a rolling ship as suggested in Reference 1 of the presentation. [Dr. Boroday and Dr. Nikolaev was submitted to the Stability Conference in Glasgow] This paper introduces the concept of exceedance probability function for dynamic heeling angle of a rolling ship, and gives the relationships for the mentioned function at given conditions of the ship operation and the fixed wind heeling moment.

The presented report contains the calculation results for stability characteristic of one ship example and exceedance probability of dynamic heeling angle versus sea state. The results are compared with corresponding characteristic values obtained for the case of disturbance random wind load in calm water. This comparison shows that there is nothing more absolute than to assume analyzing ship's safety at sea against capsizing, that the sea is calm and the stability curve is unchanged.

Thank you.

CHAIRMAN (Kuo):

Well, thank you very much, Dr. Rakhmanin for coming in such a short notice and making a good summary for us.

Can I now move onto the next item regarding how to link with practice? Because we already heard some views about ship operators' understanding of stability and application. Mr. Middleton, would you like to make some comments on this?

MIDDLETON:

Well, thank you, Mr. Chairman. Middleton, the United States.

We normally think in terms of a ship loading in a port and then proceeding to sea and using up its consumables and then arriving at some other port for discharge. But there are other operations that we should consider in developing criteria and its application. They are those that load in a port and discharge at sea, such as an offshore supply vessel which will be out discharging its cargo at a drill rig. And of course, fishing vessels are being considered many times. Well, that's the reverse, where they go out to sea empty and take on the cargo at sea and come back. It's rather difficult to judge the free board of a fishing vessel once it is loaded at sea, and determine whether it meets adequate stability, for just one particular item of it.

So, I think how to link with practice, and I am not offering any solutions but just adding to the questions, this should also be taken into account when we are developing stability criteria.

Thank you, sir.

CHAIRMAN (Kuo):

Thank you. I wonder if Dr. Abicht would like to come in with his comment now. Dr. Abicht, please.

ABICHT:

My comment to the paper of Mr. Hormann, I wish to make some remarks on the problem of shifting of cargo.

Recently, an investigation was made by Wagner who found out that 40% of all ships that capsized during the last ten years did not fulfill IMCO's stability criterion. I

think it is an alarming fact, that 60% of all ships involved in stability accidents capsized though they met the recommended criterion. In most of these cases capsizing was caused by the shifting of cargo.

Mr. Hormann told us that we will never have a dry cargo ship that has sufficient stability to sustain all possible shifting of cargo. This may be true. But, nevertheless, we should try to reduce the number of ships that will be lost every year by cargo shifting.

Perhaps a model test should be made in order to study how special kinds of cargo should be stored. By introducing the findings into practice, it will be possible to reduce the probability of capsizing by the movement of cargo in heavy seas.

In these studies, although the cross-section of the cargo hold should be varied and the effectiveness of temporary fittings should be investigated. More than 100 years ago, special rules for the carriage of grain were established. By these rules, the number of stability accidents caused by the shifting of the grain could be reduced. Why shall we not try to find solutions also for other kinds of cargo?

Thank you.

CHAIRMAN (Kuo):

Well, thank you very much, Dr. Abicht. I have actually two more discussers, one from Prof. Tamiya and one from Dr. Takahashi. But in case they think we have forgotten about them, their contributions will be called upon in the next session on Research.

Now, before I pass on to my co-chairman to summarize the first part of the discussion, I wonder if any of our host delegates would like to make some comments. Because there seems to be an absence of their participation in this session. So, any of our host participants wish to make a comment, general comments on this session?

Perhaps we can persuade them after the tea break. Now, I pass to my co-chairman who has been working very hard while I have been doing the talking. Now he is going to try and summarize the key points in the next five to eight minutes, for the first part of Philosophy. So, here you are, Prof. Paulling.

CHAIRMAN (Paulling):

Mr. Chairman, you have given me an extremely difficult task. I don't know whether it is going to be more difficult to summarize the discussion which has been carried on this afternoon or whether it is going to be more difficult for me to finish in six to eight minutes. However, I will attempt to bring out what I considered to be the most pertinent and succinct points of our discussion. And I will apologize in advance to those of you whose gems of wisdom that have been passed out before us today may have been overlooked. It is not an

error of judgement on my part, it is perhaps an error of difficulty of hearing or understanding.

Our Chairman has given us nine points to ponder in connection with the topics of philosophy and research. And each of these points has been addressed by members of the distinguished panel, the first three members of our panel at any rate. The other three will have a go at it after the break. A number of discussions have come forth from the floor.

I am afraid that I will not be able to present a very coherent summary, since I have had very little time to think this over, and I will have to merely skim through my notes, hitting on the high points.

The first question, of course, is one of the most pertinent which we had to grapple with. And that is the question of "What is stability?" A number of you presented your views on the answer to this question, and it was interesting to see the variety of definitions of stability which have been presented.

Essentially stability has been defined in several ways. One basis for the definition is the basis of economics for the owner without addressing the question of safety. The question has been defined in terms of operational excellence, again not speaking to safety. Mr. Cleary, undoubtedly speaking as a representative or speaking in terms of his background of membership in one of the regulatory agencies, noted that he often has to remind designers and operators of the necessity of considering safety as well as these other aspects of definition.

Two other aspects of definition of safety which have been brought out are those of good motion, pleasant motion, comfortable motion, or a motion simply of such a degree that the damage through the cargo and structure of the ship and its contents does not result. This produces the often experienced conflict between excessive stability, which results in severe motion, and the question of comfortable motion, which often goes along with or results from a sometimes insufficient stability.

Some more specific and scientific definitions of stability were brought forth, the classical definition of static stability is simply that the ship should return to its equilibrium upright position after a small disturbance from that position, without defining specifically what is meant by a small disturbance.

Then a number of discussants pointed out that often times the problem is not in defining stability itself. This definition is quite simple, whether we are discussing static stability, stability of equilibrium, or stability of motion in some dynamic sense. The real problem which we have to come to grips with is the problem of defining what we mean by adequate stability. That is the problem of defining the criterion by which stability is judged.

Dr. Morrall next presented a discussion on the aims of our research and in general what is the aim of research and studies concerning stability, its effects and its quantifications. He pointed out that the effects of stability or the lack thereof can always be ascertained after the accident. Of course, the effect is very obvious in the overturned or missing ship. What is often more difficult to determine is the cause. Sometimes the cause or the best evidence of the cause goes down with the ship. He quoted a very pertinent and timely example, the Mary Rose which we have all read about having recently been raised from her resting place in the water where the vessel capsized in 1545. The reason has never been definitely established. He also noted that IMCO criteria which might have been relevant in this case came approximately 400 years too late.

Some comments were directed towards the state of the art and the theory of ship's stability. Theory in its present form can help us in three different ways: understanding of phenomena; extrapolation of present experience and ideas to other cases, changing ship forms or ship types and modes of operations and so on; and finally an understanding of deficiencies of present technique.

With reference to Question No. 3: What are the forms of stability criteria? What form should they take? A number of different thoughts were expressed on this topic. They should be based on the physical behavior of the ship. The effects of the behavior of the ship on the people and contents should be taken into consideration. The question of demand versus capability undoubtedly should play an important role in devising criteria. We must always keep in mind the way the criteria are going to be used and the people, the background, the state of mind, the capabilities of the people who are going to use those criteria.

Question 4. The question of excitation versus restoration. Again, Dr. Morrall addressed this point and compared the case of demand versus capability in the ship stability problem, to the capability versus demand criterion as employed in the structure design. However, it was pointed out that serious gaps exist in how well we can represent demand and how well we can represent capability, and perhaps the state of the art with respect to the stability question is not so advanced as it is in the case of structural design.

Judgement of stability, Question No. 5: Mr. Hormann presented some interesting historical and developmental aspects of this question beginning with the quotation from 1929 SOLAS Convention, which I will not repeat because I probably would get it incorrect any way. But a vague requirement was contained at that time regarding the information which should be supplied to the ship masters and it was pointed out that we

still don't always know exactly what is needed and what should be presented to the master. The situation which is complicated by rapid changes in ship types, geometry, and methods of operation, to illustrate this last point, typical stability curves for several typical cargo ships: shelter deck vessel, the full scantling ship, the modern containership were displayed illustrating the drastic differences in stability curves of these different vessels, but which might all have been treated in more or less similar fashion under existing stability criteria.

Question 6: How to link research with designers. A number of interesting questions were brought out here. But I think one of the most relevant was that it would be extremely useful if designers and classification societies and other authorities were to get together and apply results of ongoing research to typical design cases, not necessarily because the new and advanced methods coming out of the research are superior to existing criteria or methods, but merely to gain experience and perhaps to reveal weaknesses and shortcomings in some of the results that may be coming out of research.

Possible approaches to the development of new criteria: I believe Dr. Rakhmanin gave a very astute observation in pointing out that it is possible to estimate stability by probabilistic methods. And this is probably the avenue that must be developed to a higher degree if we are truly to take into consideration the dynamics of ship motion and heavy seas in our stability criteria.

How to link research development with practice: We have no consider several factors here. The mode of operation of the ship, the specific example of the mode of cargo-handling was brought out by Mr. Middleton. There may be some very unusual and important aspects of stability which are determined by the way the vessel handles cargo and examples are vessels which take on or discharge cargo at sea, fishing vessels, offshore supply vessels, vessels which load/unload at offshore terminals often have great difficulty in determining the exact conditions of draft in loading. It is difficult to read the loadline marks when the waves are moving along the ship, for example.

Dr. Abicht, made a very interesting observation that during the past ten years, 60% of the ships which have capsized have met IMCO criteria. A major factor in these casualties seemed to be movement of cargo. The interior arrangements, therefore, or the interior geometry and construction would seem to be an important factor taken into consideration in any major future developments.

I believe, then, Mr. Chairman, this about summarizes my own, perhaps biased and prejudiced views of the proceedings which have gone on. I thank all of the partici-

pants for most interesting discussions.

CHAIRMAN (Kuo):

Thank you very much, Prof. Paulling. That was an excellent effort to summarize an hour and a half of discussion in such a short time. This is now time for tea. I am sorry we have kept you about 8 minutes late. But I would like to thank you the audience for actively participating in the discussion. If I have been hard on you with time, well, I was only doing my duty. I would like to take the opportunity to thank our panelists for the first half. There will be nine more questions to be tackled after tea on research. On behalf of my co-chairman and I, I would like to thank you very much for taking part in the first part of our panel discussion. Thank you.

(APPLAUSE)

(COFFEE BREAK)

CHAIRMAN (Kuo):

Welcome back to the second part of the panel discussion. On this occasion we will be talking about research. May I have the Slide 7 please?

PANEL DISCUSSION SESSION : PHILOSOPHY  
AND RESEARCH

PART 2: RESEARCH

1. Why do research?
2. What aims?
3. A major programme
4. What areas for research?
5. Some choices
6. How to use research results?
7. A practical example
8. How to develop research programme?
9. Summary

Slide 7

The principal panel members will be: Mr. Bird, Mr. Cleary and Mr. Helas. I would now pass to Prof. Paulling to introduce the panel members.

CHAIRMAN (Paulling):

The three members who will lead the discussion in the second session this afternoon--occupy the same positions which they had during the earlier session. I will take a great deal of pleasure in introducing these three gentlemen.

The first who occupies Chair No. 4 counting from this end of the table is Mr. Harry Bird who is with Marine Division of

the U.K. Department of Trade. Mr. Bird received his education in naval architecture at the Royal Technical College, which is now known as Strathclyde University in Glasgow. The U.K. universities seem to make a habit of this periodic change of names, I believe. Mr. Bird then went on to gain practical experience with a number of different ship-yards in Scotland, and since the early '50s he has been involved with the Department of Trade and with IMCO Subcommittees on Sub-division Stability and other Safety Problems.

The second member of the afternoon panel is Mr. Bill Cleary. He is with the Ship Characteristics Branch of the Office of Merchant Marine Safety, U.S. Coast Guard, Washington, D.C. Bill is a graduate of Webb Institute of Naval Architecture in Marine Engineering in New York. He is a member of the Society of Naval Architects and Marine Engineers, and the Royal Institution of Naval Architects. I have given his professional affiliation with the U.S. Coast Guard.

The third member of this panel is Mr. Gunter Helas who is with Ship Classification Society of the German Democratic Republic. He received his Dipl.-Ing. degree from the University of Rostock in Naval Architecture. His research and professional practice has been carried out in the Ship Classification Society where he has specialized in sub-division stability and loadlines. He is a member of IMCO Committee on Subdivision loadlines and Fishing Vessel Safety.

CHAIRMAN (Kuo):

Thank you very much, Prof. Paulling. Again, can we have slide 5 to indicate the areas which we are supposed to cover in this session?

Again the same pattern as we had used previously will be adopted. The question marks will be the ones which will be open to floor for discussion. So, the first question is: "Why do research?" The second one: "What should be the aims of research?" The next one will be a major research program which Mr. Bird will be presenting. "What areas for research?" will be the next topic we will discuss. And then some choices of research will be given by Mr. Cleary. This is followed by the next question of "How to use research results and application to practice." Then we again talk about a practical example by Mr. Helas. "How to develop research programs?" Will follow and finally, some general points. Again my able co-chairman will try to summarize the discussions.

Let us return to the first question which will run for about ten minutes. I would like to make two points. The first one is for those who are speaking. Would they try to keep the microphone close to the lips so that they can be heard in the rest of the hall, and the second one, we are determined to get some of our host participants to take

part in discussion on this occasion.

Again, I would just start with the question "why do research" by showing Slide 8 on the typical areas which we may use to try to generate the discussion. So, here are some reasons. We could do research to gain knowledge. We could contribute to safety. We would make use of available expertise. To advance theory, to improve practice. These are the typical examples which we have thrown open. And a lot of us here will be able to comment on this. So, who would like to start?

#### WHY DO RESEARCH?

Possible areas for discussion:-

- a) To gain knowledge
- b) To contribute to safety
- c) To use available expertise
- d) To advance theory
- e) To improve practice

#### Slide 8

#### VASSALOS:

Vassalos, University of Strathclyde. I noticed that in every one of the eight items presented in Part I the word 'criteria' has been used. And that can only indicate that this word represents one of the keys you set on. The opinion has also been presented that the criteria should be simple, simple enough to be understood by the operators. Although I share this view, I would also like to point out that the safety of our vessels is of such paramount importance that we should not stop there. There is nothing preventing us to have as sophisticated stability criteria as the state of the art would allow for the designers or the regulatory bodies on one hand, and then use simple or simplified criteria for the operators on the other hand based as much as possible on the existing knowledge. There is no question that the state of the art does not allow us at present to perform the calculations that we want, and we need to enhance it. Therefore, we need research.

#### CHAIRMAN (Kuo):

Right. Thank you. Any one else? Prof. Motora, what about you making comments on why do research?

#### MOTORA:

Thank you, Mr. Chairman. Well, this is a very difficult question, and my honest answer would be because it is interesting. However, there are three motivations perhaps I have. One is that when we hear from crew

who experienced the dangerous situation or who met casualty, they tell us about their experience. However, their experience is expressed not necessarily on a scientific base. So we need to interpret their experience into the scientific base.

The second one is, of course, to develop the basis for judging stability at the design stage. Our final goal must be such that we formulate the mathematical model both for the excitation and ship response, and do numerical simulation. And we can use the results of simulation for the judgement of the stability at the design stage or for regulatory purposes.

The third one I have in mind is the interpretation of the results of such research in very simple terms to let the crew know about their ship's stability for instance GM or KG. Because all that the crew can do is to just adjust the center of gravity. So, we have to let them know how they should do it and in what way they can do that.

Thank you.

#### CHAIRMAN (Kuo):

Thank you very much, Prof. Motora. Mr. Hormann, you would like to comment on why do research?

#### HORMANN:

Thank you, Prof. Kuo.

Both your terms put on the slide as well as the discussion by Prof. Motora, they induce me to make a plain forward statement of what I feel research is for. That is simply what you have on the second line, to contribute to safety. This is, I mean, the overall headline for all what we do. And so far I feel the five or six reasonings for why do research as you put it are not of the same order of importance. We, of course, can split it up. But in the first and prime instance, from my view, it should be the goal of increasing safety.

Thank you.

#### CHAIRMAN (Kuo):

Well, thank you very much for getting me corrected, but I did not really put the list in the order of importance. Anyone else who wishes to give a view on why we do research? Dr. Burcher.

#### BURCHER:

I think the first statement there, to gain knowledge, is the standard one. We do research just to find out what we don't already know. But I think in this case I would support the Chair and the Panel there that it is to contribute mainly to the safety of ships. The reason I believe that we should be doing research in this area, ship stability and capsizing, is that we just do not know enough about the mechanisms and means by which the ship may capsize.

Quite a number of pieces of work on research already have shown that there are quite a number of mechanisms, I believe quite different mechanisms. They cannot necessarily be swept up in one simple theory. And certainly our existing criteria do not cover all the parameters that are suggested by some of the mechanisms we have already discovered.

CHAIRMAN (Kuo):

Thank you for the points. Anyone else would like to make a comment on this particular item? If not, let's go onto the next one. Now, just suppose that we have chosen that contribution to safety as the prime reason for doing our research, what should be the aims to guide us in our research effort? Anyone will like to make a start on this one? Yes, Dr. St. Denis.

ST. DENIS:

The aim of research is to contribute to the safety of ships. Thank you.

CHAIRMAN (Kuo):

Right. That's a good, sharp, strong point. Now, can I ask perhaps Dr. Takaishi to give his views on what should be the aims of research?

TAKAISHI:

Thank you, Mr. Chairman. My comment is a proposal of comparative study of capsizing problems.

We have done recently in various countries a lot of experimental and theoretical studies on the capsizing problems or the safety of ships against capsizing, as well as the stability criterion, as we could find in this conference or at the first conference in Glasgow.

In these studies, each research has been always on a special ship form and using a special method, because the problem to be investigated was in most cases connected with the very practical need. And the facilities used have their limitations. Therefore, we can find various kinds of experimental methods. The results, although they are very useful to understand the capsizing phenomena of the ship in general, however, on the other hand, some confusions or divergence of the knowledge on capsizing problems were also recognized. Therefore, I have an idea that it is desirable to carry out a comparative study by the international cooperation, by using one or two common ship forms but by means of various methods according to the ability of each facility. I think such a study will be remarkably helpful to get a common understanding of capsizing phenomena and to realize a rational stability criterion based on the common understanding. I would like to see the aims to be directed at developing good theoretical and experimental correlations.

Thank you.

CHAIRMAN (Kuo):

Thank you very much. Now, we have got one specific aim. It is to link theoretical and experimental studies. That is one specific suggestion.

Any one else having any other aims which should be pointed out? Yes, Mr. Hormann.

HORMANN:

Thank you, Mr. Chairman.

Again from the practical viewpoint, I notice that many of the research works which have been carried out during the years are directed toward certain ships. Every serious scientist will put on his own work the limitations. This is only natural, and that is the only way to do a good job in this respect. But an aim of these research works should nevertheless be that we try to inter-connect the different types of vessels or the different particular vessels we do research on. This goes for both parts. For the experimental part of course it is more evident that you try to interpolate in between the few given points which you look at. But this goes as well for the theoretical or the calculation part of the problem. Thank you.

CHAIRMAN (Kuo):

Right. Anyone else? Yes, Mr. Middleton.

MIDDLETON:

Thank you, sir.

In several cases where I have been involved in research, we have tried to simulate under controlled conditions something that has happened under natural conditions and tried to emulate the conditions that occurred, particularly in the case of a casualty of a ship or a series of ships, and see if we can then understand what occurred when the actual casualty happened. So that under the controlled conditions we were then able to determine the cause of the casualty and then relate that to a former criterion. Thank you, sir.

CHAIRMAN (Kuo):

Well, that's a good point. Anyone else? Yes, Dr. Morrall.

MORRALL:

Mr. Chairman, perhaps it seems obvious but it seems to me that one of the aims of doing research is of course to improve our understanding on the subject of safety and stability. I would suggest that one aim be to lay the foundations for future stability criteria by pursuing these lines of research. So, this is my view, Mr. Chairman.

CHAIRMAN (Kuo):

Right. Any other views? Maybe this is a good time to call Mr. Bird to outline the major research program being carried out in the U.K.

BIRD:

Thank you, Mr. Chairman. Ladies and gentlemen. First of all, I think I would just like to say what a pleasure it is for me to be taking part in this conference, especially in Japan. Thank you, Prof. Motora, for making this possible.

This is a rather short paper we have written, Dr. Morrall and I, and for two very important reasons. Firstly, we are still at a very early stage in what is the major research program. Secondly, we wanted to make the paper concise and simple, so that our intentions are easily understood and read by as many people here and elsewhere as possible.

We had some discussion earlier about simple or complex requirements or criteria. Now, everyone would like to have simple regulations. That is if they want any regulations at all, and some people do not. Fishing boat skippers sometimes object.

Stability, however, is unfortunately, as we all know very well, not a very simple subject. And we, my colleague and I, believe that over-simplified regulations which do not evaluate forces acting on the ship in severe weather, neither encourage the designer to think about safety at all nor help him to explain what may have gone wrong in a casualty, so that he can improve future designs. Furthermore, such regulations give no indication as to what margins of safety may or may not exist. And margins of safety we would like to say quite a bit about in our paper, we believe to be necessary because of all the uncertainties in sea conditions and wind conditions and what may happen or may not happen to the cargo.

I suggest that public opinion would not readily accept if an aircraft or large suspension bridges or multi-storied buildings, or offshore structures--and we heard about one this morning--were to be designed and built without a proper force analysis. So, why not for ship stability? That is the question we have to ask ourselves.

Having said that, when travelling here last week, via Anchorage, as our plane came in to land the pilot for some reason I don't understand seemed to make a fairly sharp banking turn into the runway. I happened to be looking out of the window at the port wing in fact and rather horrified to see the amount of flexure in the wing structure. Now, one accepts that structures like aircraft, where weight is critical, tend to be designed, I suppose, close to safety limits and need therefore a very thorough investigation. Naval architects on the other hand are used to having larger safety margins and

may, therefore, tend in some cases to be a little complacent about stability. I think in case of some ships the margin of safety may be rather less than we like to think.

Advances in the present day knowledge and the use of computers can enable us to use a more intelligent approach to ship design without making life for the naval architect unbearable. In my time in the ship design office--and I am talking about 30 years ago now--we spent weeks and sometimes months doing manual calculations. In fact, I once 30 years ago spent six months doing tank calibrations for one tanker. We don't need to do that sort of thing nowadays. All these considerations led me to believe that the time was right for the new attitude to safety.

Experience of research on this subject in the United Kingdom during the previous decade has convinced me that in order to accelerate progress a major coordinated scientific effort is needed involving many specialists. Also my participation on the IMO working group, which I chaired for its first five sessions, confirms this view. I formed the Stability Working Group in the United Kingdom in 1976 which has debated this subject ever since and has had 25 meetings to date. We spent about two years planning what we now call the SAFESHIP project, which is briefly outlined in this paper.

I should like to draw your attention to Fig. 1 of our paper. I am not proposing to describe it in detail, because it is quite simple. But what we were trying to do here was to present a visual picture of all the various problem areas and to use it to devise a logical research procedure, which we believe is a necessary stage in working towards more effective safety regulations in the future. There is no other way to solve this problem. And the problem will not go away.

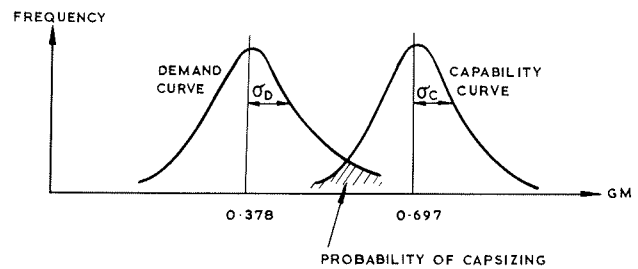
It also shows by SAFESHIP project numbers which are marked in little circles, where our efforts are presently being concentrated. There is a list of the SAFESHIP projects on page 56 of the paper. This Conference is rather early for us to be able to present many results or conclusions, although some preliminary results can be seen.

On the other hand, the Conference comes at an opportune time for us to reconsider our own priorities if there should be the opinion here this week. Our research is not motivated by academic interest, although that might provide stimulation for enthusiastic scientists. Our aim is to devise improved regulations and design criteria based on the findings of our research, which are more relevant to safety and hopefully cost effective at the same time.

The fact that stability related casualties are small suggests to me, at least, that most ships have adequate or even excessive stability, perhaps in some



margins, which is the term that I have used before. And if you look at the Slide 10, this you have seen in Dr. Morrall's other paper. It is a familiar diagram by now, I think.



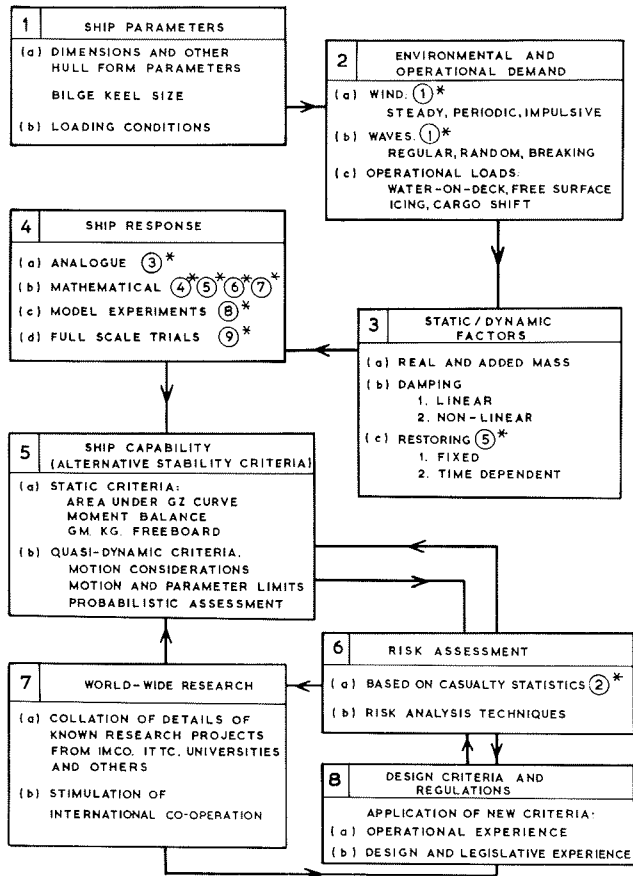
The third project, Analogue Computer. We believe this is rather useful tool in exploring different mathematical models and in obtaining roll angle distributions and so on. In fact there is a picture of one in the paper.

The fifth project is called the Mathematical Models, and is being undertaken by the University of Strathclyde under Prof. Kuo. The intention here is to devise a criterion in which wave moments are calculated directly. And I think, hopefully, this may answer the point Dr. St. Denis made a little bit earlier.

The seventh project, again Mathematical Modelling, by Bishop and Price. These two scientists believe that coupling between various modes of motion such as the roll, yaw, and sway, etc. have an important effect on the motion behavior. And, hence, they are using multi-degree linear equations.

Slide 8 shows a sort of schematic diagram of what we call a roll moment generator, which is used to excite the rolling motion, hopefully suppressing other unwanted motions like heel. Perhaps, Dr. Morrall can tell you how successful that was as has been used recently.

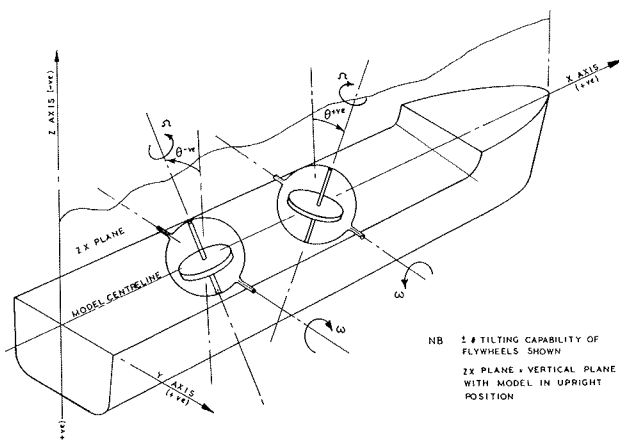
The second experimental program is to do with measuring roll-restoring moments on a wave, either on a wave crest or on the trough or somewhere else. If we look at



One, the first one, Environmental Demand. We believe that there is a need, at least in our own case at any rate, to have reliable data on wave height and wave period relationships and distributions. Also wind velocity, vertical distribution, gust duration, turbulence near to a rough sea surface, for example.

The second project, Risk Assessment. We need to rationalize questions of safety





Slide 11 Roll Moment Generator:  
Principle of Operation

Fig. 4 of the paper you can see the model situated in a circulating water channel or a flume tank if you like to call it that, sitting on top of a standing or stationary wave. I notice a rather similar diagram to this in one of the Japanese papers which we may be coming to later on this week. The objective here is to force the model to heel to a given angle in a wave and then measure the roll response on the model. So, you can see that in Fig. 4 and Fig. 7, we have plotted here some of the moments which were measured with this apparatus. And we calculated using a fairly well-known computer program, the SIKOB, the static restoring moments on the same wave profile which we measured by using probes in the tank.

The third experiment is dealing with motion responses in all six degrees of freedom. Fig. 5 shows the set of experiment and Fig. 8 is an interesting one. Here, we have a comparison in the paper, Fig. 8, of roll angle distributions which were measured using this model, which were also calculated using the analogue computer I mentioned earlier and at the same time using the MARKOV process. And it seems to me that the agreement is pretty reasonable. We also have done some full scaled measurements with the vessel which is called the Sulisker, a fishery protection vessel, of which you can see a picture at sea there.

Finally, ladies and gentlemen, I wish to stress that the purpose of the U.K. research is to develop better safety criteria and in a reasonable time scale. We feel that this is more likely to be achieved by international cooperation or collaboration if you like. If some framework to facilitate this could be begun this week, we, Dr. Morrall and I, believe that the success of Stability '82 would be greatly enhanced.

Thank you, Mr. Chairman.

CHAIRMAN (Kuo):

Thank you, Mr. Bird. Now, we have talked about a project, and perhaps it is a good idea to have some views about this area. You have heard Mr. Bird mentioning certain areas of research. Anyone has any views on other areas which should be studied or examined and deserve more attention? Who volunteer with some ideas of areas in which research should be done?

Well, that's interesting. No one does! Does everybody think that the program is completely covered? No, Mr. Jens, you don't believe that everybody has covered all the problems.

JENS:

Mr. Chairman, I would not respond to this question but the other way around. During the years, 20 years--and I am old--a lot of information on model tests had been passed to the Subcommittee. And perhaps somebody may go through this information, list them, and put them in a right order and find out the gaps where more information may be needed, which so far has not been covered in the various countries which conducted research of this kind.

Thank you, sir.

CHAIRMAN (Kuo):

Thank you very much for your kind offer. He has some data for doing research on capsizing. Any other ideas that anyone like to suggest? Dr. Rakhmanin, please.

RAKHMANIN:

Thank you, Mr. Kuo. I have an idea about what areas to research. I think that one we are speaking about is the dangerous situation at sea for the ship. Certainly, first of all, we should find out what are these dangerous situations. I think that at this Conference this problem will be discussed, because I have seen in the program some papers related to this. But when we found out on the experience what these dangerous situations are, then what to do then? Whether we prepare for the investigation of these dangerous situation, taking into account the necessity to investigate the large heeling, rolling with large amplitudes, etc. So the second goal of the investigation it is necessary to find out the instruments and methods for investigation of these new problems. Because, for instance, the equations and information which we are using for assessing the sea-keeping quality are based upon the equations and assumption which sometimes can be used for investigation of large heeling or large rolling. Then the next point to be investigated in view of capsizing phenomena, it is necessary to investigate the stability of roll process. Certainly today when I presented Dr. Boroday's paper, I expressed the view that it is difficult judge in a prac-

tical way the danger of capsizing by means of the investigation of the roll process stability. On the other hand, to my mind, the very interesting point or problem is to find out the method for investigating the roll process stability at the large perturbations or large disturbances. This is the point which should be investigated, specially taking into account the real situation. I mean the random roll process and the random disturbances at large values.

Thank you very much.

CHAIRMAN (Kuo):

Thank you. Anyone else? Yes.

NOMOTO:

Ken Nomoto, Osaka University. I would suggest stability research in the following sea. A ship can be unstable in a following sea in many ways, surfing, broaching to, parametric instability, and pure loss of transverse stability. One of the chairman, Prof. Paulling did research about that. Still, I feel that the mechanism of capsizing of a ship in a following sea remains something mysterious. So, I would suggest stability research in a following sea. Thank you.

CHAIRMAN (Kuo):

Thank you very much. Mr. Manum, did you wish to make a comment?

MANUM:

Thank you very much, Mr. Chairman. I am not sure if this is the right time to say what I am thinking. I will refer to Mr. Bird's and Dr. Morrall's paper. In their conclusion, they say--and I will read it--"The authors, therefore, welcome this opportunity to discuss United Kingdom research with those engaged on similar work in other countries and hope that closer international co-operation might be stimulated by this Conference. Research effort in different countries would thus be more effective and shorten the time-scale for significant progress". When you initiate a research program, it is very important to know what has been initiated elsewhere. I am not talking about the research now. I am talking about what is initiated. When you start, you should know what has been started elsewhere. You could show this by trying to establish some kind of a stability research data bank, where you could very quickly lift up-to-date information about research programs initiated. I am not saying this, because I am so afraid of some kind of duplication within the various different research institutes. Some duplication of work is needed in order to build up the knowledge everywhere. But I think, as said in the paper by Morrall and Bird, we could do the research more efficiently, and we could shorten the time scale, if we could

try to benefit from what others have been doing. Especially as regards the model tests which are extremely expensive, it could be very beneficial if we could really start cooperating. Mr. Jens referred to information within IMO. But I think such information then is perhaps, I'm sorry to say, assembled too slow. They won't be up to date. It is more collecting reserves. What I am about to propose is some kind of a data bank where you can get information about--I would say it now for the third time--information on research that has been initiated. We have such banks in other fields, in an offshore field. I know that we have it. So, it is not a new idea. So, I just put this forward for your consideration, Mr. Chairman and the Panel.

Thank you, sir.

CHAIRMAN (Kuo):

Thank you very much. Now, perhaps, we can now go to Mr. Cleary to examine the research, some choices of research area, and cover some of the grounds there. Mr. Cleary.

CLEARY:

Thank you, Mr. Chairman. When my co-author and myself submitted our paper we did not know that we would be on this distinguished panel and that I would be asked this specific question. I am glad I was asked the question. Because it gives me an opportunity to tell this group about the previous paper which I published last year and which has been recently republished in the Society of Naval Architects July Edition of their Marine Technology paper.

What I intend to do, therefore, is to talk about the paper which you have just only got yesterday, I suppose, because I didn't submit it on time. That has an advantage in that I won't get very many people arguing with me because you haven't read the paper yet.

The paper we submitted to this Conference was a summary of research on information to the master, which was produced in a research program done for the Coast Guard by Prof. Frankel of Massachusetts Institute of Technology about two years ago. The paper then by Mr. Perrini and myself is a short sketch of what was a 200-page research report and then our own comments.

The basic objective of that research report to ask Dr. Frankel and his team to go out to United States flag vessels which have the approved stability booklet on board as has been mentioned as required by the several conventions, SOLAS Convention and Loadline Convention in particular. As a private researcher, he was to find out how well the masters were using what we required them to use. We the government stayed out of it and we allowed him simply to find out what he could. He did a very good job and he found out many things which we wish he

did not find out.

He began simply by summarizing the type of information that is needed. The safety information that is needed, that is to meet minimum regulations; there is also operational information needed for the master; and there is load planning information needed. This is not necessarily needed because of the international convention, but it is necessary for the master to know these things.

This is a listing of the number of errors in the Trim and Stability booklet on actual cargo ship voyages of one single company during 36 voyages. The number of voyages and percentage of the total of 36, in which the error occurred, is listed. The last one, No. 7, permanent ballast on the weight form being correct, 50% of those voyage forms were incorrect. Now, when such a basic thing as permanent ballast is not correct on the form, how good have we been with our requirements and our review of the requirements as administrative authorities? And the master doesn't even know his permanent ballast--this is not temporary ballast--permanent ballast is not known correctly.

Further, loading errors were found. Poor quality of plans is one of the things that he found that the administration and perhaps other administrations are allowing plans to be submitted for our own use, which we understand as naval architects. But when they are used as a capacity diagram on the ship, for instance, and the reproduction method has not suited to it or the reproduction method fades, and after two years the master cannot read it any way. No wonder he doesn't use it. There is poor coordination between the loading ports, especially on container ships. We found or he found for us that the working group that the company uses to maintain high efficiency on getting the cargo to the destination, to the right person who bought the cargo, often means that the people on the ship either don't know what their load is or that decisions are made by the stevedoring group of the company, that they don't even meet the needs of the company, the company that has told that stevedoring unit to keep its cost down, so they make some decisions which don't even meet the needs of the company and also do not take in mind the safety of the ship.

He recommended several non-technical ways to improve the Trim and Stability booklet as it exists today. There were about ten of them using the large size papers. The usual things about printing and not having copies, copies fading, and graph paper should have only large grids on it for the master's use.

In reviewing this, Dr. Frankel didn't list it this way, but it took me six months of reading his report to get the 200 pages down to the size of our paper. And what you will see now is only four or five ideas out of it. The areas of limitation that the

ship may have are at least those either that you see listed. And another good one is, a ninth one will be training of the crew. By limitation I mean that any one of those might require a limit to either the trim, or the GM or the KG. And if some of them actually interact with one another because of international conventions, for instance the damage stability and survivability and the pollution, sometimes one works against the other in approving Trim and Stability booklet.

Next slide is a simple recommended T&S booklet index. There have been many of these, the IX group recommended one to IMCO, now IMO, some years ago. And this is very similar to it. It was also mentioned in the paper that the only thing that we have done differently is to take Section 2 as an operations sections, and pull it out of design approval, and tried to give the impression that in Section 2 the graphs should be only a simple curve, very bold, and not much information. One or two pages of information, and that's all. Because this is the part that the master is going to use hopefully.

The problem is that we have not communicated our problem either as designers or as researchers or as regulatory authorities. The step as I see it now, there are three very largest steps in ship design. No. 1 is the research which meets the needs of carrying things certain distance and certain extremes, and that can use up an awfully lot of computer paper and designer calculation. Step No. 2 is, having reduced all that paper to one simple formula or three simple formulas, we then have to make a criterion and see whether it is approved or not. And Step No. 3 is that, even the 20-30 pages by which we have to review something as one of the gentlemen said in an unsophisticated way, it is not going to be used by the master, and so it has to be reduced to a few simple graphs for pictures. The master has dozens of things to worry about, his crew, the cargo, his weather, the schedule of sailing, etc.

Some of the things on the next few slides have already been said and I kept quiet because I would have this opportunity to speak.

Research needs--these are from this paper of mine a year ago--there is a need to understand the total physical phenomenon. It has been said more than once today. We don't understand it. We need to express it in design terms and we need to find out which type of righting curve--I use the term righting moment because I think in smaller vessels we will have to go to a righting moment and not righting arm--and I need to express it on one page so that the operator can exercise his ability to enhance the total picture.

The overall research program---and I would support what previous speakers have said on overall research program--remains

necessary because we do not understand all the modes of capsizing. In this paper, I was able to list 34 different ways of intact ship being lost without flooding. There were 7 seaway action items, 2 wind items, 2 weather items, one structural, 2 cargo placement items, 2 cargos, 4 cargo security items, internal liquid movement, external forces from nets and mooring lines, draft management, water currents, and partial grounding. Those could be broken down into many more. The point I made in the paper was that as the administration people we cannot require any designer for his approval purposes to go through 34 equations. So we have to follow this business of trying to cover many things with one or two formula.

Again in this paper, there are 12 tasks suggested for the naval architect industry. The titles of them are shown there. I will show one or two examples.

This is what I call a roll diagram for ship motions and I find fault with it myself. But it is something that we should be thinking about in the future. It gives a ship master a rough idea of his period of roll. What he does with that rough idea of period of roll or something, we need to discuss.

(Chairman reminds the speaker that he should not be longer than two minutes)

There is my last item. There are twelve categories.

In the paper I show a lot of diagrams about how the ordinary righting arm curve can be changed by various heeling items. And I am not certain that we still want to use a standard righting arm curve. But that is what we have historically been working from. So, let's continue to use it at least as a start.

This is one I would recommend as the largest need is the one referred to by Dr. Takahashi earlier. I would agree with what he has said. It should be international cooperation. I don't think that the special ship form and special method would go away. But I do agree that we need to start on common ship forms and, for instance, find out in a small seaway and a large seaway how much this righting arm curve changes. And Mr. Helas, I think, has some very pertinent remarks in that same area.

Finally. About total safety. It has been said this morning that we cannot achieve total safety. And it has also been said that the operator can modify how much safety the designer can give and that changes in accordance with what the hazard is and also what the size of the vessel is. It would be good for us to engage in an exercise of trying to decide what the major stability hazards are for the various different sizes of vessels.

I will stop there. Thank you very much.

CHAIRMAN (Kuo):

Well, thank you very much indeed. Mr. Cleary. You made it exactly in 120 seconds.

Now, can we move onto the next item for the discussion because I think what Mr. Cleary raised has something to do with how to use the research finding. So, let's open this to the audience again to see views on how research should be used. Anybody has views on this one and like to express the opinions? Yes, Mr. Fulford.

FULFORD:

Seems ashamed to stand up for a very short answer I am going to give. How should research be used? Inevitably, "with caution."

CHAIRMAN (Kuo):

That is a good point. Dr. St. Denis, can you give one even shorter still?

ST. DENIS:

Yes, it won't be long. I should like to notice one thing, that ships and drilling rigs today are approaching the cost of 100 million dollar mark, and that is the price of about three or four jumbo aircraft. Yet, the master is neither schooled or trained to avoid casualties at seas and, when they occur, on how to and what to do about them. Now, this is very different from what happens in the aircraft industry. The pilots are indeed trained with the training devices which they operate, a computer, to fly all sorts of maneuvers in all sorts of atmospheric conditions. And when they have become sufficiently proficient, they are entrusted with flying an aircraft, without passengers to begin with. Well, I simply suggest that as part of the whole business to be improving the safety of ships at sea the masters should go through a similar course. Thank you.

CHAIRMAN (Kuo):

Anyone else? Dr. Kastner?

KASTNER:

Just a short comment on this problem. I think we naval architect should collaborate with the nautical people and the educators there. And I have often been asked by shipmasters what the main problems are. And I found that there is even a different language on the same terms. So, I think we should find out the basic things which we naval architects think we all know about but still among us there are differences on the ways to do, and find this out, and work together with the nautical lecturers. I am actually working at this school in Bremen, that is a nautical school. And we are planning to collaborate on this subject. Thank you.

CHAIRMAN (Kuo):

Thank you. One more comment on how to use research results? In that case, let me move on now to Mr. Helas' presentation and a practical example of how you use the research results.

HELAS:

Thank you, Mr. Chairman.

The usual method to evaluate the stability of the ship at present is to calculate the stability in smooth water only and to apply such stability criteria which require a certain amount of reserve stability. It is assumed that by this reserve stability in smooth water also the stability in the seaway is insured. But the experience has shown that all the ships having sufficient intact stability in smooth water can have insufficient stability in the seaway and that they can capsize in the following waves. This refers specially to small ships.

For this reason, we have tried in our Classification Society, and in cooperation with Polish Register of Shipping, to develop a method of calculation of the intact stability of ships in following waves and to fix some stability criteria.

It was our aim to take into account such important factors as irregularity of the waves, the speed of the ship, and the time of capsizing.

It was necessary to make some assumptions and simplifications for the seaway. It was assumed that the seaway is two-dimensional and irregular. The seaway is replaced by an effective wave. This wave has a sinusoidal form. Its length is equal to the length of the ship. Its height is given by a formula depending also on the length of the ship. The righting arms are calculated by a quasistatistical method and under following assumptions. The wave crest is in the middle of the length of the ship. In order to simplify the calculation, pitching and heeling oscillations, the influence of the ship's speed on the pressure distribution and the Smith's effect are neglected.

The method of effective wave was introduced by Prof. Grim. The amplitude of this wave has a random value which is determined from the condition that the changes of the righting arms caused by this wave and by the real regular waves are the same. This condition was fulfilled when the integral is taking over the ship's length of the square of the differences between the ordinates of the real base and the effective wave, which is the minimum. From this the formula for the wave height was derived.

In determining the formula for the wave height, a seaway was assumed which corresponds to the wind force of 8 Beaufort in the Atlantic Ocean. Furthermore, the ITTC's spectral density function and the 3% probability of exceedance were taken as the basis.

The effective wave changes its height with the speed, which depends on the speed of the ship. In the case of a high speed of the ship, the height of the effective wave alters only slowly and the ship is situated on the wave crest for relatively long time. This may lead to capsizing especially if the loss of stability is significant.

The influence of the speed on the safety of the ship in following waves is taken into account by introducing the average height of the effective wave within the time of capsizing. It was assumed that the time of capsizing is equal to 60% of the natural rolling period.

A further assumption concerns the speed of the ship. It is supposed that in the case of dangerous following waves the speed will be reduced so far that the Froude number 0.23 will not be exceeded.

This assumption led to the correction of the wave height according to the aforementioned formula.

The stability of the ship under wave crest is evaluated according to wind criterion and according to criteria for the righting arm curve. The value of the wind pressure is applied according to our National Stability Rules. Furthermore, it is assumed that the wind which has caused the seaway has changed its direction, while the waves move on in the original direction. The assumed change of direction is  $45^\circ$ , so that the wind meets the ship in oblique direction from behind.

The permissible dynamical heeling moment due to wind pressure results from the condition that the heeling angle or flooding angle may not be exceeded, whichever is less.

The criteria for heeling angle have been stipulated taking its pattern from the damage stability requirements. The initial metacentric height GM shall be positive. The angle of vanishing stability shall be at least  $30^\circ$ . And the maximum statical righting arm GZ shall be at least 8 centimeters.

The method of calculation presented in the report and also the proposed stability criteria have been applied to some ships different in size and purpose. The results presented in the paper and also other results show that the checking of stability in following waves is reasonable for small ships in all loading conditions. Even in cases of very good stability in smooth water, the stability in following waves can be insufficient.

For bigger ships, it can be assumed that the stability in following waves is sufficient, if in smooth water a considerable reserve stability is provided. When the stability in smooth water exceeds the required minimum values only to a small extent, the stability in following waves can be critical and its checking is advisable.

However, final conclusions can be drawn only when the method has been used in practice for a longer time and for many ships.

It is our first step to solve the problem of the stability of ships in following waves. Many assumptions have been made, in the respect of which the opinions can be different, and which needs further research.

It is our intention to use this method for additional checking of the stability, to use it in addition to the intact stability criteria for the ship in smooth water, especially in the case of small ships of unrestricted service.

Therefore, we tried to develop a method which is not too complicated, so that it can be used without too much expenditure when designing the ship.

Thank you.

CHAIRMAN (Kuo):

Well, thank you very much indeed. I have two discussions related to the points of this paper. Can I first of all call on Mr. Luis Perez-Rojas to make briefly a comment on some of the points?

PEREZ-ROJAS:

Thank you, Mr. Chairman.

The contribution is a small comment to the interesting paper of Mr. Helwig, that in the problem of the stability with following waves it is felt that there are two steps. Firstly, the calculation of the stability level with following seas and secondly, the evaluation of those stability levels with some specific criteria. In this second step aspect, it appears more adequate to apply a criterion that is based on experimental data of capsized model and on practical data of capsized vessel while sailing in following waves. This criterion should allow for the size of the ship, as it is well known that the influence of the ship's length in the stability varies with wave length in following seas and it would be not sufficient to allow for the wave length to be related only to the ship's length. To my mind, the size of the ship is important and must be included in some kind of criterion. But it is more important in this special case of stability in following waves.

Thank you, Mr. Chairman.

CHAIRMAN (Kuo):

Thank you very much, Mr. Perez-Rojas. Now, can I call on Prof. Shin Tamiya to make his comments please?

TAMIYA:

Shin Tamiya from Tokai University. The author states that the stability is always less than in smooth water if the ship is in wave crest. In my experience, this is not always true. For instance, a hypothetical cylindrical ship of V-shaped section will have even a larger righting level in the wave crest than in the smooth water, the calculation being made including the effect of orbital motion of water.

Moreover, Figures 7, 8, and 9 on the paper show that GM in wave crest exceeds that in the smooth water in the range of small heel angle. If we take care of righting level calculation in following waves, at least a computation including Smith's effect should be carried out. And also experimental research or confirmation should be made to cover the lack of information on the change of metacentric height due to advanced speed of the ship.

That is my opinion. Thank you.

CHAIRMAN (Kuo):

Thank you very much. And I think he points out how difficult it is sometimes to use research results.

Now, can we go to the next item, on which I have used the words 'How to develop research programs'. And I think I would like to make this much broader, so that if we are really looking for ways in which we can cooperate internationally, how we can benefit more internationally. And there are one or two people I know who have been much involved in this type of work.

HELWIG:

Helwig is my name. I think the presentation of the work done by Mr. Helwig is a good example of a possibility to link the results of research on ships on the irregular sea by replacing it by an effective wave since the normal information seamen at seas will receive consists of the seaway described by a significant wave height and the significant period. Now the only thing we have to do is to get some connections between the effective wave and the significant wave. And in this way you can get a lot of information that people on board the ships could use.

Thank you, Mr. Chairman.

CHAIRMAN (Kuo):

Right, thank you. Is there anyone else? Yes, Mr. Cleary.

CLEARY:

Thank you, Mr. Chairman.

On your question, I think we already have been given some several good suggestions. The paper that Mr. Bird and Dr. Morrall had presented had several suggestions in it. I would like to add my feeling that we need to have at least a moral commitment amongst ourselves to try to do this in less than seven years the next time. I don't want to see a conference every year since that will get very stale. But perhaps once in every five years or so we can watch ourselves on these research programs. And when good results are coming, we should get together like this. It is always astonishing to me that people say to us, those from outside of our industry, "Well, ships have been around for 3,000 years and you know all

about them." Well, we don't know anything about them.

Finally, no one has mentioned yet ship capsizing from or in bow waves. I have seen this occur, and we are trying to have some research done right now on the mechanism of small vessels, particularly fishing vessels, capsizing in bow waves. It is possible. I am not certain that it is mathematically possible to show why. But I know it does happen.

Thank you, sir.

CHAIRMAN:

Well, anyone else would like to comment? Yes, Prof. Johnson.

JOHNSON:

Yes, Bruce Johnson.

I would like to make a distinction between the probability of having a capsize in Pierson-Moskowitz sea which really should only occur due to a weight shift or adverse loading condition or something stupid that is done by the crew. I would suspect that most capsizings in extreme seas are not remotely related to the standard spectra that have been used, as just was done in the last paper. So that, if you are talking about probabilistic capsizings in standard sea spectra, you are really talking about unusual events in terms of loading conditions. And some of the real problems in terms of fishing boats and in terms of small boats frequently occur when, say, coming back to shore going through the surf zone, surrounding an inlet. There are many inlet capsizings. Those are the ones that I am familiar with in terms of the Coast Guard vessels that have recently capsized. Those are basically time domain capsizing. That is to say, there is no adequate spectral standard that could be used to describe that. That is what I was attempting to comment on previously. I did not do that very well. But I think certain types of large amplitude motions are strictly time domain analyses as far as attempting to analyze what's going on. I don't think you will ever satisfy any spectra formulation that we would obtain for our results. And we should attempt to make distinctions when we categorize research programs between the various kinds of capsizing. So, we really need more statistics on the typical or expected spectra that was present at the time of one of these capsizing.

CHAIRMAN (Kuo):

I think time is getting on and it has been a long day. So, again, instead of discussing any more points, I will now call my able co-chairman to summarize in next ten minutes or so the points we have discussed in the second session this afternoon.

CHAIRMAN (Pauling):

Well, again, I will attempt an instantaneous playback and this makes me feel somewhat like a television machine at a football game. But I found this afternoon session most interesting and I am sure that there are a number of quite pertinent points that have been brought forth concerning the general topic of why do research and what are the aims of research, and a number of related questions. Of course, we have quite varied audience here including academicians, researchers, ship designers, and so on, and when you ask a group so varied as this why do research, you certainly can expect a wide variety of answers. It seems that there are several quite important reasons that we hear over and over again. Of course, speaking as a professor, my instinctive reaction is to say that we do research in order to gain knowledge. But the really important purpose is to gain knowledge for some purpose and application, and, of course, the end goal of our research is to enable us to design better and safer ships.

The gaining of safety of ships is of paramount importance as it was pointed out by several of our discussers. Thus, while it is necessary to gain knowledge and information, that will enable us to thoroughly understand the behavior of ships, the manner in which ships might capsize and how to design ships against the various modes of capsize, it is probably useful and important to distinguish the two areas of application of this knowledge.

The first area of application is in the design of ships. Here it is essential that designers be able to bring to bear the full state of knowledge, including everything that is known about stability and capsizing, in order to design safer ships.

On the other hand, knowledge and information presented in a much simpler and terse form is necessary for operators of ships. This is not necessarily a reflection on a lower degree of intelligence or knowledge of such persons, but it reflects the fact that frequently the conditions and the means at their disposal for applying that knowledge are of a completely different nature than the means and circumstances under which a designer applies knowledge.

A number of other reasons for doing research were brought forth. One of those which was mentioned by a number of people was the necessity of conducting research in order to translate experience into useful terms. Specifically those who have experienced casualties or those who have survived the capsizing, have some very useful and vital information which needs to be translated into the forms applicable to our analytical, physical modelling and in terms which are useful to designers.

What is the goal of our research? Again it was emphasized that the goal is that of increasing the safety of our ships.



What are the aims of the research itself? Several discussers described the need for achieving a greater degree of correlation of experiment and theory and translation into practice. They suggested that it might be appropriate at this time to embark upon an international program of cooperative study, in which a common ship form or a small number of common ship forms would be subject to studies by a number of participating institutions and organizations which have developed the means of analyzing and predicting the stability and capsizing characteristics. Perhaps the organizers of this conference might be instrumental in stimulating such international cooperation.

There were two discussions of practical examples of on-going projects. The first of these was the SAFESHIP program under way in the U.K. at present time. And Mr. Bird, I believe, gave us a very interesting and informative description of this program, which is now being carried out through the medium of a number of individual research projects, all coordinated and brought together within a central organization.

Again the achievement of an effective program through a major collaborative effort involving a number of investigators and institutions, was brought forth through this description.

What areas of research should be pursued in order to achieve the goals and the aims which were addressed in the previous questions? It was pointed out that by Mr. Jens for instance, that during the past 20 years or so a great deal of information has been collected by IMCO and they are now in great need of someone to go through this accumulation of data, to classify it, systematize it, and help us define what has been covered, to what degree it has been covered, and what additional work is needed to fill in the gaps.

One gap or one area in which research is needed was discussed by Dr. Rakhmanon. There is critical need to find out what are the dangerous situations for the ship. And I believe that this is relevant to the point discussed by Bruce Johnson in the last few minutes.

There are very specific circumstances, combinations of waves, geographical or topographical situations, which create the wave and sea conditions that are especially dangerous to the ship. Not all of these situations are either thoroughly understood, or capable at the present time of being adequately described.

Again several of the discussers endorsed the theme brought forth by Mr. Manum who pointed out the need for accumulating a research data base in order that those who initiated a new research program have easy access to knowledge which has been accumulated in previous research. The suggestion of the need for an easily accessible research data base in the field of ship's stability and capsizing was pointed out.

Choices of research areas. Mr. Cleary described an extremely interesting project which was aimed at ascertaining the extent to which stability information is used on shipboard, and determining the weaknesses and errors contained in that information. The findings of the study, I think, could be a real eye-opener to those of us who are engaged in a more academic and theoretical aspects of this subject.

How should research findings be used? They should be used with caution. One of the goals of our research is to avoid casualties at sea. And in avoiding casualties at sea, proper training is one of the most effective means of achieving the goal.

How to develop research programs and make use of the results. Mr. Helas' example discussing the creation and development of the criteria to be applied by a regulatory body, in which the stability in following sea was computed and formed an essential aspect of the criterion, is a good example of the application of contemporary research results and should be followed up.

In order to effectively apply such a criterion, it is still essential that we understand thoroughly the dangerous situations in which capsizing may occur or in which the ship finds itself in danger of capsizing. We then need to know how to express these situations in terms of physically measurable or deterministic qualities such as significant wave height and other properties of the wave environment.

I believe, Mr. Chairman, this essentially completes my summing up of the discussions.

#### CHAIRMAN (Kuo):

Thank you very much, Prof. Paulling for another most useful summary of the second part of our panel discussion.

And I would like to just take two minutes to give my views on the panel session which has taken place this afternoon.

I think first of all what we have tried to do in the panel session is to whet your appetite for the rest of the conference. We hope we have taken up certain points and highlighted certain things, and although we have been talking for about three hours, I am sure lots of points were not covered. With so many papers, and an excellent location I am sure in the next three days you will have opportunity to discuss them in greater detail.

As far as the first session on philosophy is concerned, my own understanding from the discussions as well as my own views and that of my panel members is that we should get our "aims" right. If we do get our aims right, then we will know what we are trying to solve, whether this will be developing better tools for design or better way of making information available and use criteria for operational purposes.

I think one thing which was highlighted



very clearly is this contradiction between the information you need to have, as expressed by Prof. Johnson, about the difficulty in defining the excitation forces, and on the other hand you have the practical people who have to use information to do their jobs. And certainly in North Sea you have to design these rigs to perform the jobs even before you have all the answers. And I think a clear plea here is that while we appreciate that we need more knowledge, we need to have better links between design and practice. We must not let the gaps become too large, otherwise they will never be bridged.

As far as research is concerned, I think we touched on many points. And some comments made were very useful indeed. Perhaps, as far as I am concerned, maybe what we really want to do is to try first of all to obtain information and then to acquire better understanding, before developing the tools, and have the skill of communication before making application. All this with caution.

Now, this summarizes my views on the

discussion of today. It leaves me with the pleasant task of thanking my panel members, all six of them, for their contributions. They prepared papers and they were asked to condense their valuable contributions to a much shorter time. I hope they will forgive me for the way I have twisted their arms in order to get them to conform to certain time targets. I would like to thank my colleague, Dr. Vassalos, who has patiently handled the slides right way through.

Lastly, I think the discussions which we have managed to generate is achieved through your cooperation and your contribution. And I am sure I speak on behalf of all of us when I say we appreciate the active part you played in this discussion. So, on behalf of all of us, I would like to thank you for taking part in this discussion and for giving us your attention.

Thank you.

(APPLAUSE)

(END OF SESSION)

# ON QUESTION OF RATIONAL CRITERIA DEVELOPMENT FOR ESTIMATION OF SHIP'S DYNAMICAL STABILITY IN IRREGULAR WAVES

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## ABSTRACT

The report deals with possible approaches to the estimation of the effect of wind heeling load on the ship which experiences motions in waves. It is pointed out that the question of practical value of the approaches based on the investigation of steadiness of irregular rolling motions with finite disturbances is an arguable one. It is found expedient to estimate dynamic stability of an oscillating ship using probability characteristics of the maximum angle of heeling on the leeward side which occurs immediately after action of heeling load. It is suggested that these characteristics should be calculated according to the author's method. The method allows to determine probability of ship's capsizing that is to estimate security of her sailing in waves.

The results of probability calculations are given which allow to estimate quantitatively the fishing vessel stability in waves.

Solution given in the paper is a necessary stage of the development of ship stability standards on the probability basis.

- (1) At first sight it is quite natural to address stability which is a quality determined by statics of the ship when estimating security of ship sailing in storm conditions. However the usage of the notion "stability of ship" itself in relation with the most general case when the ship experiences motions in waves and is also affected by wind is quite arguable. In spite of such uncertainty in

terminology a quality of ship which is analogous to the classic notion of "stability" and which characterizes her resistance to heeling loads can be defined on the basis of the analysis of ship dynamics in waves with an additional external influence.

Let us consider motion of a ship oscillating in irregular waves under the influence of external load which causes her drift and heel. Let us write down the equation of ship's heeling about longitudinal horizontal axis crossing the ship's center of gravity this way [1, 2]:

$$(J_{xx} + \mu_{44})\ddot{\theta} + M_c + M_s^0 + \mathcal{M} = M_K, \quad (1)$$

$$\text{where } M_K = -P_a Z_H - R_p Z_R - \mu_{24} \dot{V}_2 - \mathcal{M}_{\eta}, \quad (2)$$

- heeling moment

Notations of formulas (1), (2):

$P_a = P_a(t)$  - transverse wind load

component;

$\theta = \theta(t)$  - ship's heeling angle;

$V_2 = V_2(t)$  - ship's drift velocity in still water;

$M_s^0$  - ship's righting moment in calm water;

$\mathcal{M} = \mathcal{M}(\theta, t)$  - additional righting moment in waves which is determined with consideration of ship's sway characteristics [1];

$M_c$  - moment of resistance forces;

$\mathcal{M}_{\eta}$  - righting moment component due to sailing of ship with a given heeling angle in calm water;

$J_{xx}, \mu_{44}, \mu_{24}$  - mass moment of inertia about longitudinal horizontal axis; added moment of inertia and static moment of sea

$Z_n, Z_R$  - water;  
 application points of  
 aerodynamic resultant  
 force  $P_a$  and trans-  
 verse horizontal  
 hydrodynamic force of  
 noninertial nature due  
 to ship's drift. The  
 vertical axis crossing  
 the ship's center of  
 gravity is directed  
 downwards;  
 $t$  - time.

It in the equation (1) we assume the  
 force dependency to be a certain function  
 which rapidly increases in time and then  
 drops in a certain time interval to its  
 initial value the initial conditions for  
 investigations of ship's motion according  
 to (1) will be as follows:

$$t = t_0, \theta(t_0) = \theta_0, \dot{\theta}(t_0) = 0. \quad (3)$$

$\theta_0$  is the largest inclination of ship on  
 the leeward side immediately after action  
 of heeling load and it is assumed for  
 simplicity that  $P_a(t) = 0$  with  
 $t > t_0$ , that is by the moment when  
 the ship reaches the largest inclination  
 aerodynamic load cease to act. A problem  
 of estimation of the value  $\theta_0$  will be  
 considered in more detail below that is  
 why in the following the case when the  
 ship avoid capsizing at the first incli-  
 nation is considered as the most general  
 one.

When the analysis is performed small  
 disturbance  $\varepsilon = \theta_0 - \theta_H$ ,  $\theta_H$  being  
 a heeling angle due to rolling at the  
 moment when load  $P_a$  start to act, the  
 problem is reduced to the investigation of  
 steadiness of the rolling motions in first  
 approximation. In principle such investi-  
 gation can be performed either on the basis  
 of studies of Hill disturbance equations or  
 by energy approach which allows to obtain  
 a simple condition of steadiness in the  
 explicit form [3]. However it is obvious  
 that formulation of the problem of finite

disturbance action estimation including  
 consideration of ship's capsizing  
 probability makes the analysis of  
 steadiness of her rolling motions in first  
 approximation unapplicable. Generally  
 speaking, it is quite correct to employ  
 equations of steadiness of motions [3] for  
 the investigation of stability at large  
 oscillations if one considers variation  
 of heeling angle after the moment when  
 $t = t_0$ , that is ship's oscillations  
 after the first largest inclination  $\theta_0$ .

Though difficulties of analytic  
 estimation of rolling steadiness at large  
 oscillations for random waves are quite  
 apparent it can be performed for given  
 stability diagrams in case of regular  
 waves by means of phase plane. In the  
 paper [4] steadiness of amplitude-frequency  
 curves of nonlinear rolling motions is  
 estimated by means of plotting separat-  
 rices which help to find areas of

"attraction" by branches of these curves at  
 different initial amplitudes and phases.  
 A prospect of expanding the phase plane  
 method for solution of a less schematic  
 problem which deals with steadiness of  
 irregular ship motions with random  
 disturbances using elements of reliability  
 theory is arguable and the problem itself  
 is still formulated in a very general  
 form [5].

(2) Equation (1) with  $P_a = 0$  describes  
 rolling motions of the ship keeping  
 an arbitrary course in waves. If the  
 wave conditions are steady-state  
 rolling motions will be determined  
 by a set of stable probability  
 characteristics, in particular, by  
 one-dimensional density  $f^0(\theta)$ ,  
 integral law  $F^0(\theta)$  corresponding  
 to it and probability of exceedance  $q^0(\theta)$ .  
 When  $t = t_H$  stable irregular  
 oscillations will be upset by  
 disturbances caused by the load  $P_a$ .  
 Initial conditions corresponding to  
 subsequent random inclinations will  
 be as follows:  $t = t_H$ ,  $\theta = \theta_H$ ,  $\dot{\theta} = \dot{\theta}_H$ .  
 We formally reduce equation (1) to  
 the linear form by adding equal  
 summands to its left and right part.  
 Let us write its solution this way:

$$\theta(t) = \sum_{i=1}^n \theta_i(t). \quad (6)$$

In formula (6) the following notations are  
 adopted:

$$\theta_1(t) = \theta_H e^{-\nu_\theta t} \cos n_1 t, \quad (7)$$

$$\theta_2(t) = \frac{e^{-\nu_\theta t}}{n_1} (\dot{\theta}_H + \nu_\theta \theta_H) \cos n_1 t, \quad (8)$$

$$\theta_3(t) = \frac{e^{-\nu_\theta t}}{n_1} \int_{t_H}^t e^{\nu_\theta u} (n_1^2 \theta - m_\theta^0) \sin n_1(t-u) du, \quad (9)$$

$$\theta_4(t) = \frac{e^{-\nu_\theta t}}{n_1} \int_{t_H}^t e^{\nu_\theta u} (2\nu_\theta \dot{\theta} - m_c) \sin n_1(t-u) du, \quad (10)$$

$$\theta_5(t) = \frac{e^{-\nu_\theta t}}{n_1} \int_{t_H}^t e^{\nu_\theta u} m_K \sin n_1(t-u) du, \quad (11)$$

$$\theta_6(t) = \frac{e^{-\nu_\theta t}}{n_1} \int_{t_H}^t e^{\nu_\theta u} m_1 \sin n_1(t-u) du, \quad (12)$$

where  $m_\theta^0 = \frac{M_\theta^0}{J_{xx} + J_{44}}$ ,  $m_c = \frac{M_c}{J_{xx} + J_{44}}$ ,  $m_K = \frac{M_K}{J_{xx} + J_{44}}$ ,

$$m_1 = \frac{M}{J_{xx} + J_{44}}, \quad n_1 = \sqrt{n_\theta^2 - \nu_\theta^2},$$

$M_c$  - moment of resistance forces;

$\nu_\theta$  - linear damping coefficient;

$n_\theta$  - frequency of ship's free undamped  
 oscillations, besides  $t > t_H$ .

The dependency  $\theta(t)$  defined by (6)  
 is unstationary random process and its  
 elements  $\theta_i(t)$  also belong to the  
 category of such processes. Being an  
 unstationary function of time the dependency  
 $\theta(t)$  which is considered according to  
 a set of its random value with  $t = \text{const}$ .  
 is described by current probability  
 characteristics from which the function of

probability of exceedance will be written as follows

$$Q(\theta, t) = 1 - F(\theta, t) = 1 - \int_{\theta \neq \theta} f(\theta_1, \theta_2, \dots, \theta_6, t) d\theta_1 \dots d\theta_6. \quad (13)$$

Here  $F(\theta, t)$  - current integral law of angle distribution  $\theta(t)$ ;  $f(\theta_1, \theta_2, \dots, \theta_6, t)$  - multi-dimensional current density of probability of components  $\theta_i(t)$ .

Time behaviour of functions  $Q(\theta, t)$ ,  $F(\theta, t)$  determines the character of ship's oscillations under action of load  $P_a$ . For example, if after the load has stopped to act it proves that

$$Q(\theta, t) \rightarrow Q^0(\theta), \quad F(\theta, t) \rightarrow F^0(\theta), \quad (14)$$

ship's rolling motions with a given value of  $P_a$  can be assumed to be asymptotically stable.

A decisive step in the process defining of probability of exceedance (13) is determination of the form of function

$f(\theta_1, \theta_2, \dots, \theta_6, t)$ . As to this function one cannot give a method suitable for application purposes of its construction accurate enough. It is also obvious that hydrothetical representation of the mentioned multi-dimensional distribution for the concerned problem will not give desired results. It is explained by the fact that even with a law for selected it is necessary to find correlation relations between the processes  $\theta_i(t)$  and also, which is more essential, by small validity of estimation of extreme terms of series by hydrothetical law if one doesn't set corresponding condition which is usually difficult to satisfy when selecting this law.

- (3) Under the mentioned circumstances it is impossible to construct the function  $Q(\theta, t)$  according to (13) correctly, and, consequently, to estimate excited rolling oscillations according to their current statistical characteristics analysed during a sufficiently long time interval. At the same time it is possible to describe the function of probability of exceedance for angles of ship's motions in the range of time moments values  $t$  adjacent to  $t_H$ , that is to the moment of external load action. This approach is realized if we consider as moments of time those moments which correspond to the first largest inclination of the ship after action of load  $P_a$ . Such analysis allows to construct the function of probability of exceedance for so called dynamic angle of heeling which is then considered as ship's response to heeling wind load. The approach which treats the first ship's inclination on leeward side as dangerous is a traditional one. Though, strictly speaking, one can't be sure that subsequent ship's angular oscillations will not exceed

the first one in amplitude, it is hardly probable in terms of real wind load conditions. In fact, the problem of stability ensuring in waves is the most important for comparatively small ships whose free rolling have periods with values usually in the range  $T_\theta = 6 \div 10$  sec. As the time interval in which the ship reaches the largest heeling on the leeward side under action of wind heeling does not differ greatly from  $T_\theta/2$ , the length of a wind gust dynamically applied to the ship can seldom exceed even the time interval in which this first inclination occurred. As it is seen from the data obtained during observations of steady gusty wind [2, 6] the mean length of gusts for strong wind is about 2,5 sec. Some single intensive gusts may be 6-7 seconds long or more. As to squall, observations [7] show a rather slow growth of wind velocity in terms of the problem involved. In some cases it takes as long as several minutes. For instance, even for the maximum velocity of squally wind of about  $V_{max} = 35 - 40$  sec. the time of its growth is  $\tau \approx 20$  sec. In this connection squally wind effect on the type of ships involved proves to be only slightly different from static one. For example, when  $T_\theta = 7$  sec. according to data of the paper [8] a difference between angles of dynamic and static heeling corresponding to the upper limit of squally wind velocity which equals 40 m/sec. makes up about 20%.

- (4) All these considerations about dynamic inclinations of the ship produced by squally and gusty wind were proved by the results of electronic modelling of ship's motion described by a system of equations of drift and heel:

$$a_1 \ddot{\theta}_g + a_2 \dot{\theta}_g = P_a(t), \quad (14^*)$$

$$\begin{aligned} & \beta_1 \ddot{\theta} + \beta_2 \dot{\theta} |\dot{\theta}| + M_\theta^0(\theta) + \mathcal{M}(\theta, t) = \\ & = c_1 P_a(t) + c_2 \dot{\theta}_g + c_3 \ddot{\theta}_g. \end{aligned} \quad (14^*)$$

The system (14<sup>\*</sup>) describes the effect of external load  $P_a$  on the ship applied at the moment of time  $t = t_H$  for a schematic regime of ship's motion in two-dimensional following waves. Due to the plane nature of waves the initial conditions with the presence of load  $P_a$  effect corresponded to the following case:

$$\theta_H = 0, \quad \dot{\theta}_H = 0.$$

Constant coefficients  $a_1, a_2, \beta_1, \beta_2, c_1, c_2, c_3$ , stability curve in calm water  $M_\theta^0(\theta)$  and additional righting moment  $\mathcal{M}(\theta, t)$  were defined for the fishing vessel by a displacement of 240t. The function  $\mathcal{M}(\theta, t)$  for a fixed heeling angle was assumed to be

$$m(\theta, t) = m_0(\theta) \cos(\omega_k t - \delta),$$

where the encounter frequency  $\omega_k = 0,55 \text{ 1/sec}$ , which corresponds to the motion of ship with a speed of 8 knots in following waves with a wave length  $\lambda = 0,95L$ . With a heeling angle of  $30^\circ$   $m_0$  makes up 0,33 of  $M_0^\circ$ . The period of ship's small rolling oscillations in calm water is  $T_\theta = 6,5 \text{ sec}$ . There types of possible variation in time of load  $P_a(t)$  were modelled (Fig.1):

- relatively slow growth of the load according to the linear law corresponding to a squall with a very high speed value;
- case when the load  $P_n$  instantaneously reaches its limiting value with the equation  $P_a = P_n = \text{const}$  holding true for the whole time interval concerned. The home stability standards employ just this relative character of wind load variation;

- $\pi$ -shaped variation simulating the real action of the stable gusty wind with a length of gust  $\Delta T = 5 \text{ sec}$ . which is about two times as long as an average gust length mentioned in item 3.

For each case of the function  $P_a(t)$  the modelling was made for different values of phase  $\delta$ , that is for different ship's positions relative to the wave profile at  $t = t_H$ . For instance, when  $t_H = 0$  the phase  $\delta = \pi$  approximately corresponds to ship's position on the wave top while the phase  $\delta = 0$  corresponds to the position on the wave trough.

In calm water with a load variation of the third type the third inclination proves to have the largest value which is explained by a slow growth of speed and low values of drift acceleration at the initial stage of ship's motion. As to the second and third types of load variation in calm water the ship reached the largest angles in the first inclination.

The presence of waves qualitatively changes the pattern of ship's behaviour.

Analysis of the results obtained by means of modelling for wave conditions which covers in total 144 dynamic

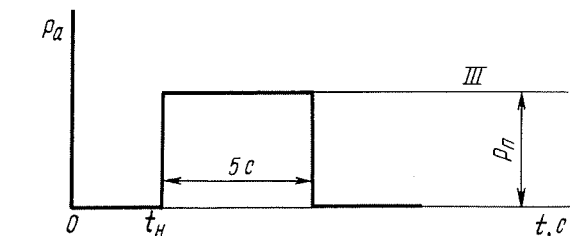
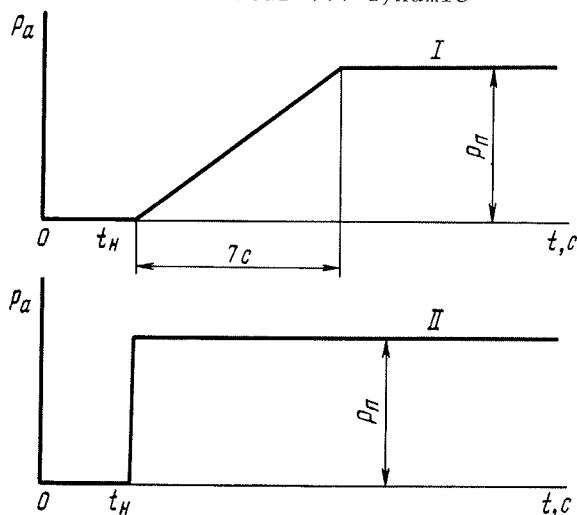


Fig.1. Type of  $P_a$  load variation

inclination regimes, described by different phases, variants of load effect,  $P_n$  values and combinations of the coefficients

$C_1, C_2, C_3$  allows to make the following conclusions:

- In all cases the first type action of the load is the most favourable one in the sense, that the ship has the smallest dynamic heeling angles;
- In case of the first and the second type action of the load the largest heeling angle of the ship on the leeward side depending on the phase  $\delta$  may occur in the first inclination as well as in the second or subsequent inclinations;
- The wind gust dynamically applied to the ship according to the scheme of the third type with any value of the initial phase  $\delta$  causes the largest inclination right at the first oscillation on the leeward side. Subsequent inclinations of the ship prove to be smaller than the first one;

Thus, in our opinion, when analysing ship's stability to wind one should consider the case of gusty wind as the most unfavourable one as regards dynamic nature of action. Also, as can be concluded from the above discourse, by the moment when after the first inclination on the leeward side the ship will heel on it for the second time one can exclude aerodynamic moment due to the load  $P_a$  from consideration, and ship's inclination will be determined only by forces due to her damped drift.

Estimation of ship's security by the analysis of the first inclination can be made on the basis of the relation (I) by means of employing equations of the paper [1].

- (5) In each separate case of heeling load action described by a specific realization of the random function  $m(\theta, t)$  and specific values of  $\theta_H, \delta_H$  the value of dynamic heeling angle  $\theta_0$  can be determined from the equation:

$$A_\theta(\theta_H, \theta_0) - K(\theta_H) \text{sign} \dot{\theta}_H = A_K(\theta_H, \theta_0), \quad (15)$$

$$\text{where } A_\theta(\theta_H, \theta_0) = \int_{\theta_H}^{\theta_0} [M_\theta^\circ(\theta) + m(\theta)] d\theta$$

- work of the righting moment without allowance for the effect of ship's sailing in calm-water conditions,

$$A_K(\theta_H, \theta_0) = \int_{\theta_H}^{\theta_0} M_K(\theta) d\theta$$

- work of the heeling moment in the range of heeling angles from  $\theta_H$  to  $\theta_0$ ,

$$K(\dot{\theta}_H) = (J_{xx} + J_{44}) \frac{\dot{\theta}_H^2}{2}$$

- kinetic energy of the oscillating ship at the moment of  $P_a$  load action. It is also assumed that at  $\theta = \theta_0$  the angular velocity  $\dot{\theta}_0 = 0$ .

The paper [1] introduces the concept of  $Q(\theta_0)$  - probability of exceedance function of dynamic angle of heeling  $\theta_0$  and gives relationships for calculation of the function at given conditions of ship's sailing and fixed moment  $M_K$ . These relationships were obtained on the basis of probability analysis of realization of the initial expression (15).

In our opinion, it is the method of the paper [1] that makes it possible to estimate objectively ship's dynamic stability with allowance for probability nature of factors acting on the ship in waves.

The methods given in the paper [1] dealing with a numerical estimation of ship's lateral stability in waves which is a characteristic that has properties of random functions are realized in a form of algorithm for calculation of righting moment distribution and the work performed by the righting moment. There was also developed an algorithm for calculation of probability of exceedance for dynamic heeling angle. With the help of computer programs based on the mentioned algorithms seiner stability characteristics in irregular two-dimensional waves were calculated. Fig.2 shows a diagram of seiner stability in calm water.

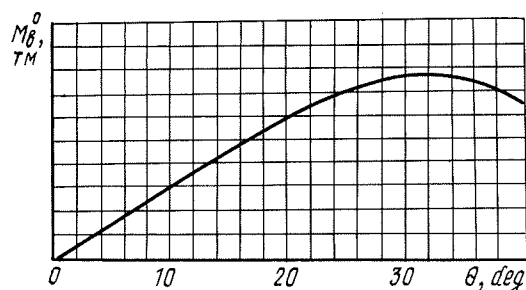


Fig.2. Stability curve of seiner

Calculation results given in Table I and Fig.3 quantitatively characterize the effect of waves on the value of ship's righting moment.

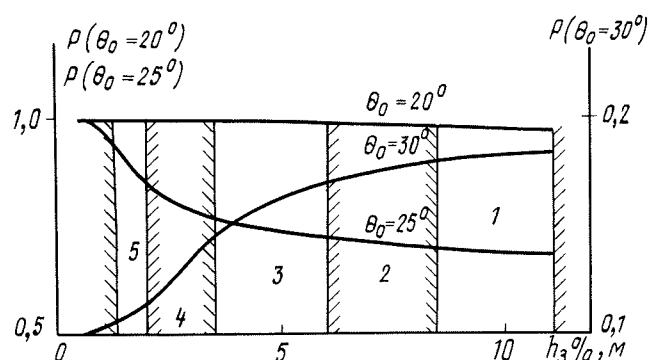


Fig.3. Probability of exceedance of dynamic heeling angle vs. wave height, load type I. Intensity of waves: 1 - sea state 8; 2 - sea state 7; 3 - sea state 6; 4 - sea state 5; 5 - sea state 4.

Results of stability characteristics calculation

Table I

| Wave intensity,<br>sea state                                     | 3  | 4                          | 5                          | 6                           | 7                            | 8                            |   |
|--|--|----------------------------|----------------------------|-----------------------------|------------------------------|------------------------------|---|
| Wave height with 3%<br>probability of<br>exceedance $h_{3\%}, m$ | 0,75-1,25  | 1,25-2,0                   | 2,0-3,5                    | 3,5-6,0                     | 6,0-8,5                      | 8,5-11,0                     |   |
|  | 1  | 2                          | 3                          | 4                           | 5                            | 6                            | 7 |
| Average value of<br>additional<br>righting moment<br>$M, tm$     | $\theta = 10^\circ$ 1,026<br>$\theta = 20^\circ$ -0,773<br>$\theta = 30^\circ$ 0,509 | 1,473<br>-0,657<br>0,707   | 1,904<br>-0,391<br>0,858   | 2,032<br>-0,312<br>0,875    | 2,119<br>-0,306<br>0,843     | 2,219<br>-0,335<br>0,766     |   |
| Dispersion of<br>value $M,$<br>$D_m. (tm)^2$                     | $10^\circ$ 5,010<br>$20^\circ$ 19,088<br>$30^\circ$ 20,350                           | 10,944<br>39,867<br>41,049 | 19,450<br>70,851<br>74,565 | 26,494<br>94,735<br>102,458 | 30,925<br>109,060<br>120,038 | 33,987<br>119,974<br>134,433 |   |

|  |     |       |       |       |       |       |       |
|--|-----|-------|-------|-------|-------|-------|-------|
| Average value of work performed by the moment $M, \text{tm}$ | 10° | 0,090 | 0,129 | 0,167 | 0,178 | 0,185 | 0,194 |
|  | 20° | 0,112 | 0,200 | 0,300 | 0,328 | 0,344 | 0,359 |
|  | 30° | 0,089 | 0,205 | 0,340 | 0,378 | 0,391 | 0,397 |
| Dispersion of value $A, (\text{tm})^2$                       | 10° | 0,038 | 0,084 | 0,149 | 0,203 | 0,237 | 0,260 |
|  | 20° | 0,962 | 1,257 | 1,774 | 2,564 | 3,074 | 3,473 |
|  | 30° | 1,570 | 2,497 | 3,945 | 5,481 | 6,451 | 7,212 |

As it follows from the table the growth of wave intensity increases this effect. For example, with waves of state 8 intensity at the heeling angle  $\theta = 30^\circ$  the maximum value of righting moment variation calculated as tripled mean square deviation will make more than 45% of righting moment value in calm water.

Fig.4 gives probability of exceedance curves corresponding to the given calculation. One of them corresponds to the wave intensity of state 8 and the other is the limiting form of probability of exceedance curve for sailing in calm water. In the domain restricted by these curves there are functions  $Q(\theta_0)$  for the wave intensity in the range states from 3 to 7.

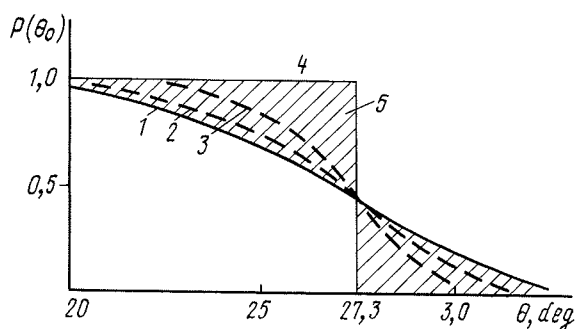


Fig.4. Probability of exceedance curves of dynamic heeling angle, load type I. Intensity of waves: 1 - sea state 8; 2 - sea state 6; 3 - sea state 4; 4 - calm water; 5 - region of probability of exceedance curves corresponding to waves conditions.

A discussion of possible approaches to the estimation of ship's dynamic stability given in the paper allows, in our opinion, to make the following conclusion. It is rational to employ the approach mentioned above which is based on the estimation of probability of exceedance for the dynamic heeling angle and ship's capsizing probability. One ought to stress that probability

estimation of ship's security in storm conditions with given environmental conditions is a necessary element based on the probability approach of stability standardization system discussed in [9].

Dependencies shown in fig.4 prove very well the truth of the following opinion expressed in a sharp polemical way [10]: "There is nothing more absurd than to assume when analysing ship's stability that the sea is calm".

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JUDGEMENT OF STABILITY - QUESTIONS TO BE SOLVED -  
A CONTRIBUTION FROM THE POINT OF VIEW OF AN APPROVING AUTHORITY

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ABSTRACT

The history leading to the present situation with respect to stability requirements and stability criteria is briefly outlined. The specific features of today's ship designs are compared with the prerequisites for the use of the established system of criteria. The development of IMCO's work towards improving the stability criteria (as given in Resolution A.167) is critically assessed. The need for resolving as quickly as possible the question of what is sufficient stability is illustrated by looking at the variety of modern ship types. The only way for achieving this goal appears to be the combination of model tests with mathematical methods of describing the ship's motion in a seaway. - The weather criterion presently being developed is discussed in the light of the fact that a sophisticated determination of all heeling moments can only be used with appropriate results, when the other side of the equation, i.e. the uprighting moments of the ship in its real environment, is assessed with the same degree of approximation of the reality.

From its inception 1929 up to now, the SOLAS Convention does not contain any stability requirements but just the fundamental (and not at all quantified or even qualified) phrase: "The master shall be supplied with such information (on stability) as is necessary to permit efficient handling of the ship." The substance of this sole stability related paragraph in the International Convention changed very little between 1929 and 1974/78.

This - in the early years - was already some achievement in as much as there was an international pointer to the fact that stability had to be borne in mind, when running a ship. It was, however, and is still like closing the eyes after having seen something what has to be done in order not to feel obliged to do it: the complimentary step after requiring stability information necessarily is to tell the master what criteria he has to meet to be safe. - The solution of this problem was tacitly left to the national Administration, and it is for quite a number of ships yet unsolved.

Let us go back some 50 years. By then naval architects very well had realized that it was not all to calculate stability values for ships and to assess them, while the ship was in service; however, all they did over and above was, to state (as it appears now) arbitrarily certain values for a particular ship as providing sufficient stability. - If one reads the literature it becomes evident that almost all specialists dealt in detail only with calculation methods, inventing more accurate, simpler, better methods. This trend - with very few exceptions - kept on going for some 20 years. It was only the introduction of computer techniques which finally made all further



efforts in this respect superfluous - each and every indicative value could be calculated with as-you-please-accuracy. This finally turned the attention to the real problem: what is sufficient stability? I spoke about the early exceptions: the most important was Jaakko Rahola, who in 1939 used the casualty data of some coastal trade ships to define values representing safe stability for this group of ships by using the statistical approach. This was the first time that somebody could prove for more than one ship that the values he proposed were really sufficient.

In Germany the findings of Rahola were used very early with consequence: immediately when the marine activities after the war started again under German flag, Germanischer Lloyd's safety department, acting as technical advisor to the German Maritime Administration required ships to comply in all loading conditions with the main one of Rahola's criteria: at least 0,20 m righting lever at 30°.

In the early sixties IMCO developed their set of stability criteria (known as A.167-criteria), using also statistics from casualties. The results were more or less a recapitulation of Rahola's findings, now better defined, more sophisticated but in essence nothing new. This might sound somewhat rude against those then involved in that exercise, but the proof that Rahola was right, is a value in itself. - This we had in 1968: reasonable criteria for ordinary ships up to 100 m in length, and today? Except for a few special ship-type-related values, we have nothing more, nothing! All that appears to be new, in my view is of the same quality as the statements to this issue in the thirties: just a guess, without any real proof!

The fact that no criteria exist for ships over 100 m in length was not an urgent problem as long as the geometric parameters of such larger vessels (ratios B/D, L/B etc.) were such as to achieve stability values significantly above those taken as limits for smaller vessels. But modern ships often show values no longer in excess of the criteria given in A.167.

Being one of those who have been, since about 20 years, directly involved in or at least related to the day-to-day work of judging whether stability data offered for a certain ship are sufficient or not, I regret this situation very much. I have no solution for the problem, but - perhaps - my questions could help to raise thoughts which induce useful steps forward. Let me formulate my problems in an example.

A yard has designed a 140 m-Container-ship under a set of usual parameters, i.e. the data of service and economy as given by the customer. The designer after completion of the project has made a preliminary assessment of stability and comes up with a series of righting arm curves for varying service conditions. A few of these curves

show maximum righting levers of about 30 to 35 cm, good GM values but an angle of vanishing stability of 47° or so. And I know by experience that the theoretical wave crest righting arm curve for such condition will show negative values in its entirety. The IMCO-A.167-criteria (applied all over the world more or less also for ships over 100 m in length) are complied with, without any doubts. The only thing I can say is: I believe, this is not enough. Next question is: what is then sufficient?  $\theta_v$  50°, 55° or what? I know: there is no tool to make a reasonable reply to this question, and every one degree more I require for  $\theta_v$  means quite a number of containers less on deck. With these obvious and severe economic implications it is simply impossible to end the discussion in stating a certain requirement reasoned only with "I believe"! As long as I cannot prove that below a certain limit the ship is endangered, this remains unsatisfactory. Of course, nobody wants to solve the question with the same statistical approach as Rahola! On the contrary: this is, what we want to avoid under all circumstances: contributions to such a statistic.

As I said earlier: the introduction of computers made it clear that from now on all stability related efforts of naval architects and scientists could be directed exclusively to solving the question: what is sufficient stability. But looking at what happened in this respect in all the years I have the suspicion that still some of the undertakings, meant to contribute to solve the key question, exhaust themselves by creating mathematical methods to cover one or the other external force affecting a ship, without relating the result to the stability capability of a ship, i.e. without proving from what point onwards the safety is endangered. The wind criteria is a good example for what I mean by this; I will return to it later.

(I just said: all efforts can now be put on the question: what is sufficient stability. I should like to mention another open question, i.e.: what stability has a ship in a certain condition? This also is not sufficiently solved, but this question does not represent theoretical difficulties; this is just a question of practical means to be developed. I do not underestimate the problems involved in creating practical, economic and accurate equipment for stability assessment on board. But between this and our main problem we have a difference of the order of the distinction between "development" and "invention").

It deserves appreciation that IMCO already at the time by which A.167 was finalized, initiated further work in the field; it was made very clear that much remained to be done. After a specialists group on "external forces" during a few years had put together a comprehensive material collection on forces impairing the

stability of a ship at sea, the special Ad Hoc Group on Intact Stability was formed in 1975. This group in the meantime worked successfully on quite a number of particular questions, and they also, already in its very first meeting, formulated as one of their goals the term: "quantify sufficiency of stability", in my view the focus of the whole exercise.

I hope, I will not be misunderstood, when I now give some critical comments on the course of the work of that group. I do, of course, not in any way intend to discredit the in every respect valuable work of that group. I may, however, get permission for the purpose of illustrating my thoughts to only reflect on that part of the group's work in which I find something to criticize:

When the group started, they made an assessment of factors affecting stability and stated that the following were sufficiently dealt with by A.167:

- free surfaces
- Loading and ballasting
- Wetting of deck cargo
- Icing, Crowding of passengers
- Rudder action

"only" 3 effects remained to be looked at:

- water trapped on deck
- waves
- wind

Since then, mainly the wind was looked after. This, in my view, is not an adequate approach. Certainly the term "waves" has a prime importance for the attempt of advancing the stability criteria. But the factor "waves" has quite a different quality than others like: rudder action, free surface, wind. - To assess the effect which waves have on stability means - at least for me - to learn the decisive lesson on what really is sufficient stability, since - let us be honest - the view that (for small ships) A.167 means sufficient stability was achieved empirically, by experience - by no means through comparison of adverse moments compared with the uprighting moments actually available. - We have to realize, it was - and still so far is - a comparison with a theoretical value (still water uprighting moments) for which just experience gives justification. And we have a lack of respective experience for all types of ships except those "smaller ordinary cargo ships" forming the basis for Rahola and for A.167.

The problem ahead of us is really open so far, not a single aspect of it is solved. I should like to quote a Nestor among the German naval architects, Prof.E.Horn, who in 1943 said:

"Under these circumstances, which did not yet allow a solution on a physical-scientific basis, so far perforce a route was taken to the extreme contrary, namely using pure experience";

and he continued a little later:

"one is dependent on the lessons of experience, which one probably will have to use in future also".

Prof.Horn certainly would be surprised to learn that his words from 1943 are still valid today, 40 years later!

And while I am quoting; I can nothing but agree to what the IMCO Ad Hoc Group in 1978 wrote into one of their reports:

"It was noted that the topic of capsizing is very complex and it would be unrealistic to expect rational criteria incorporating ship motions and external excitation to be obtained within short time scale - such as three to five years.

Research studies on this topic should be performed from various view points as it is appreciated that no single approach can yield the complete solution, i.e. there must be theoretical studies as well as model experiments and full scale measurements to augment practical experience".

All experts are united in seeing three means of approaching the problem in focus:

- analysis of capsizing events (luckily there are only very few cases, but for those which occurred the knowledge of what really happened is very limited)
- model simulation
- mathematical methods

I would make a plea for a combination of the latter two. Model simulation appeals to me as promising, however, this way is very costly, because very soon the specialists had to realize that it was not possible to achieve general limiting values, applicable to a whole group of ships, perhaps distinguished by size or type or service particularities. The comparison of the material available from model experiments in my view clearly reveals that it is not sufficient to use main dimensions and parameters of that order to compare stability characteristics. I am afraid, it will be necessary to take into account the hull form to such an extent that sophisticated parameters need be introduced.

That this is so, can be seen already by simple comparative calculations: If you calculate for a general cargo ship built in the early sixties righting arms for still water and for the ship on a wave crest and compare that with the same calculation for a ship of equal length but 10 years younger, you will find significant differences. If you then even take as modern example a containership designed in its hull form to carry as many as possible containers under deck you can be pretty sure to find the following situation: The still water curve of the older ship will raise significantly above the GM-tangent, the GM is not very high, say 40 to 60 cm but the range of the curve is nicely in between 60 and 70 degrees. The curve of the modern ship - although her actual freeboard is more or less the same - does not at all raise above the GM-tangent, but the GM being of the same order; this has

the effect that the range of stability is poor, about 50 to 55° for the angle of vanishing stability. If we express that in a general form, we can say in comparing present days' designs with those at the time, when A.167 was established:

- generally lower stability data for larger vessels

- reduction of the range of stability

The look at the wave-crest-curves in our example is alarming: the old ship shows a comfortable positive range, while for the modern ship there is no positive lever at all. This means that, being on the crest of such a wave (wave length equal to ship's length and wave height  $L/15$ ), the ship is in an unstable condition. Such a situation is not purely theoretical, similar situations may happen, while the ship sails in following seas. This temporary instability has in my view a more severe influence on the behaviour of the ship and on her safety than, for example, the wind pressure. - Even between two modern ships of equal main dimensions, the wave crest curves may differ significantly. This shows that more sensitive parameters have to be looked for to describe the differences between ships. One could argue: these differences are of no interest as long as we define external forces to be applied to all ships, we superimpose them in a suitable way, and we look whether the uprighting moments e.g. on a wave crest compare favourably with them, (a safe margin yet to be defined). I have to object: my above theoretical examples of two modern ships show that very well one might show positive results after such a calculation, but the other negative ones. The crux is: both types of ships are built many times, each of them made many voyages; the fact that - with a pretty high probability - several times the worst anticipated conditions were met by both sample ships, can be reasonably interpreted as meaning: both types are sufficiently safe; we are not entirely sure, but we have no reasons to believe in the contrary. It is here, where more sophisticated parameters of form might help us to understand. And I can say from own experience: it is frightening to look at the wave crest characteristics of the majority of modern medium size ships; even if the fact is taken into account in the calculation that the theoretical wave form and its height is significantly flattened by the influence of the ship plying in it.

I should like to convey another thought which highlights the fact that the non-availability of sound stability criteria for a large number of ships is out of step: it is generally felt, that today's level of safety in ships is higher than some 30 or 40 years ago. This is certainly correct for most of the areas contributing to safety, e.g. fire protection, navigational equip-

ment and others. This is not true for the area of intact stability for example for such ships I mentioned earlier, which have generally lower stability-data compared with those of equal size designed a few decades ago. These lower stability-values were not introduced on purpose, because we were convinced (or could prove) that they are still sufficient! No, we were more or less forced to accept them, since they resulted necessarily from other design parameters and since we were not able to prove that they were not sufficient.

I pointed to the differences even between ships of the same age and the same size. This is also new in recent years: up to the - I would say - mid-sixties the ships could be grouped into a few types which were uniform to a far reaching extent; today we observe a vast diversification in the parameters of form and their combination, so that we have a huge number of different groups, while the number of ships falling in such group is normally very small. This development is primarily due to the trend towards specialized ships and also due to the fact that naval architects often disregard almost all "traditions" and design their ships around the "desired block of cargo" without any compromise (here also the computer renders the decisive help to cope with the problems of such constructions; without the computer the designer would much more be forced to stick to traditions!).

To put it in a nutshell, what I tried to convey with the forgoing related to our problem is: we must not expect in future general stability criteria of the order of those in A.167. At least the limiting values to be applied will differ with the diversity of ships.

These thoughts, certainly, are not very encouraging for those working in the fields of model experiments and of mathematical approach; but the work is necessary; it is the only way, if we agree not to wait for a sufficient number of fatal accidents!

The result must be: a ship-specific criterion or criteria. The parameters possibly could be:  $B/D$ ,  $B/d$ ,  $L/B$ , block coefficient, waterline coefficient, bow and stern form but also absolute size, position of deck houses and others. We have to care that the search for those ship-specific criteria is not blocked by or forgotten over a success in developing a sophisticated wind criterion.

As an example for the successful development and use of a ship-specific criterion I would mention the criterion for harbour tugs, which is applied since more than ten years for German tugs. Here the important and unique external force is obvious: the pull athwartships which acts on a tug attached to the stern of a big ship to be moored. Systematic model experiments brought about a set of diagrams,

by which it is possible to calculate, dependent on the underwater-hull form, the capzising force, with the speed of the tug athwartships through the water as parameter. Since the compulsory introduction of these additional criteria no German harbour tug has suffered a capzising accident. - I know, this example is not convincing, if we try to deduct something from it regarding our problem. The tug in the still harbour water is ideal with its defined forces affecting stability. Nevertheless, it demonstrates that it is possible to find means to distinguish between the capabilities of ships having similar main dimensions, but differ in the details of their form.

The reasoning for using additional parameters for defining the particularities of a given hull form, cannot be understood through the hydrostatic approach (or pseudo-static one). It is the hydrodynamic effect, which details of the hull form have, which we must look after. All the refined details of a hull are naturally taken into account in a summarizing, all embracing way, when performing model tests. The accompanying mathematical approach has to try to separate the more important from the less important factors. I repeat: the great variety of ships makes a comprehensive research along these lines a very costly exercise; but I see no alternative.

Let me now tackle the so-called weather criteria as under development in IMCO since a number of years:

Much efforts were put into the problem of calculation of rolling angle, the value of which has decisive influence on the result when applying the proposed formula. Still, the experts have different views on this and, I agree, the problem of rolling angle deserves attention. We must, however, be careful not to take the weather criteria, once finally blessed by IMCO after so much work put into it, as the improvement of the A.167-criteria. We must see that this criterion does not add anything for most loading conditions of most types of ships. It will only have effect, if a ship is in a lightly loaded condition and obviously, when it comes to ships with comparatively large windage area. It might have effect on ships specially susceptible to rolling, but I wonder, whether the formula is fine enough in this respect to detect special susceptibility to rolling, except when the bilge keels are missing.

In earlier sessions of the IMCO stability group it was stated and agreed that the weather criterion should be an interim measure used until the fundamental theoretical research will carry a crop. This approach, in my view, is right.

It should be realized that the problem is not to add an additional criterion to what exists so far; the problem is to find the limit line between safe and unsafe. If we apply the weather criterion, whatever

its final factors might be, we will get a GM-value required in the given condition for the ship to theoretically sustain the action of wind as anticipated. I cannot see a proof that the ship is actually safe by meeting this requirement: the rolling angle calculation will certainly be sufficiently accurate, the assumptions for wind force and wind gust also, but the underlying assumption for the GM-required-calculation is that the heeling energy so accurately calculated be compared with the ship's energy to upright itself (area under the righting lever curve). This, however, is a highly theoretical curve, which only physically exists in still water. In the real environment this curve is changing quickly and in wide ranges. I do hesitate to accept that the curve used is a reasonable approximation of reality. Simplifications and approximations are necessary and justified, but if applied, one has to realize that better accuracy only in parts of the whole set of calculations, or on one side of the equation does not add anything to the accuracy of the final result.

Let me once more return to what I earlier suggested: in my view the only possible approach to learn more about stability is the combination of quite extensive model tests with mathematical methods. Since I personally have very little expertise in the mathematical part, I think the number of models tested should be as high as possible. I know of course that the amount of money needed is enormous (Cleary once spoke of 30 to 50 hulls required to be tested to come to sound results). I hope, reasonable figures can already be achieved for quite a number of ships by evaluating tests with 4 to 5 models and applying mathematical methods in parallel for generalisation of the test results. - The interim evaluation of the running test series performed in the HSVA, Hamburg which are presented in other papers to this symposium are promising.

Finally I would add a remark which we all - as being engaged in the science of ship stability - must recapitulate from time to time, to put the object of our thoughts into the right order: We will never have a cargo ship (tankers excluded) that has sufficient stability to sustain all possible shifting of cargo. We have to keep as a condition that the cargo remains in place, also and especially under extreme weather conditions. Naturally, the greater the external forces acting on a ship, the greater is the danger of cargoes beginning to shift or to fall over board. This thought highlights the fact that assessing the stability and even knowing what sufficient stability is for a given ship, still is not all. Operational precaution and, closely related to this, the professional skill of people on board is also decisive for achieving safety at sea.

This paper, of course, is not thought

to discourage scientific efforts. On the contrary, this should help to keep our eyes open for the main question:  
what is sufficient stability.

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## PHILOSOPHICAL ASPECTS OF ASSESSING SHIP STABILITY

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### ABSTRACT

The term 'safety' implies that no accidents are acceptable, but this is in contrast to the maritime field where the reality of the situation is that a substantial accident risk has always been present. It seems inevitable that losses will continue to occur at a significant level, especially for fishing boats, but is there an acceptable level of risk? It has to be recognised that there is no absolute level of acceptable risk at present and the best that can be done is to attempt to assess priorities for action which will give optimum returns for improved safety.

There are two basic prerequisites to assess overall priorities for ship safety: first, identification of the main problem areas, and second, knowledge of remedial actions which will be effective. It is this latter aspect that has concentrated the mind of the naval architect for some considerable time, particularly on matters such as intact stability. Although the operational side is recognised to be of some importance in relation to ship safety in general, it is the seakeeping qualities of a vessel that are of vital concern in assessing an adequate level of intact stability.

Unfortunately, in recent years there has been an over emphasis on the theoretical aspects of ship stability. As a result little or no thought has been given to the fundamental requirement of ship stability assessment from a designers point of view. The best way to overcome this difficulty is to address the problem of safety in a more rational way using recent concepts of reliability analysis and long-term prediction techniques to provide the designer, owner and regulatory authority with a more

realistic approach to stability and safety assessment.

### 1. INTRODUCTION

If a discovery in engineering or naval architecture has merit it is soon taken by others and is thereby understood, extended and applied. A report on a formal investigation of a casualty such as the loss of the TITANIC [1] or the capsizing of HMS CAPTAIN in 1870 [2], is often enough to stimulate important advances. In the latter case an important consequence was to concentrate the mind of the naval architect on large angle stability and the theory of capsizing. Although in 1870 the evidence from the GZ curve was too novel and the understanding of dynamic effects too limited for the stability data to be interpreted, this debate continues today!

It does not seem to matter much whether the original idea was immediately widely understood or whether it was at first understood by only a few. In safety matters, new ideas and new principles do not often appear. The basic principles are mostly well understood: the problems are to keep them widely known and to get them applied.

The naval architect will normally design a vessel that is safe, functional and acceptable aesthetically, on time, and to a price. Although safety is very important it is not the only consideration. Only in those special cases where safety is paramount can technology and materials be used almost regardless of cost but even in these circumstances safety cannot be guaranteed.

One of the cardinal principles of safety analysis is that when the possibility of an accident is being studied, the possibility of that accident leading to a second consequential accident must always be

included in the study. Education is also required to improve the general understanding by engineers and naval architects of safety related to dynamic behaviour of a vessel, its variability and the associated problems of design and operation.

The survival of a vessel in heavy seas as a result of extreme motions, and roll motions in particular, is one of the most fundamental requirements considered by the naval architect when designing a ship. It is therefore unfortunate that ship rolling response to these extreme conditions is one of the least understood phenomena in ship motion theory and there is still a lack of either an adequate mathematical model or experimental data upon which to base criteria for judging the survivability of a particular design.

It is this latter aspect that is the main concern of this paper. Although there have been many noteworthy papers on intact stability in recent years, most have concentrated on the theoretical aspects of an apparently intractable problem with little or direct reference to safety and to ship performance in rough seas. This over-emphasis on the mathematical modelling of rolling motion has overshadowed the equally important fundamental aspect of providing practical design data for the naval architect. There are however, notable exceptions to this concentration on the theoretical approach, Krappinger [3], [4] and Kure [5] for example, deal with the more philosophical aspects of ship safety, and recognise the need for using modern methods for assessing safety in extreme seas such as risk analysis concepts.

This paper discusses some of the philosophical aspects of assessing ship stability, and then proceeds to consider some of the theoretical aspects of roll prediction, including short and long-term prediction techniques. An environmental risk model is advocated as a possible means of assessing intact stability in the design stages of a vessel. Finally, general conclusions are drawn which suggest that every effort should be made to harness current stability theory and statistical techniques to assist in the design process.

## 2. STABILITY ASSESSMENT

In making a realistic assessment of stability the behaviour of the ship under environmental load should be paramount, in so far as that its behaviour affects the safety of the ship itself and the persons using it. Safety in this context is its relation to the primary effects of loads upon the ship. In practice this can mean any kind of load - such as the distribution of cargo, the natural wind and waves or the effect of control surfaces, such as rudders or stabilizers - which can be expected to act upon the vessel during its working life. The expectation may vary greatly in degree from loads that are always present, to the loads that may arise only a few times in

that life.

The meaning of safety in this context is that a ship is considered to be safe if it withstands the loads that come upon it during its working life, that it continues to serve the functions for which it was designed to the satisfaction of the owners and users without causing danger or undue disquiet to the general public. The satisfaction of the owners and users of course depends on the first cost of the ship and its maintenance costs. The aim of the designer is thus to achieve the required degree of safety economically and in the course of this he may run some risks of extreme rolling motion or uncomfortable motions and in some cases capsize.

The assessment of the chances of such untoward extreme motions or capsize to meet the design aim is part of the process of assessing ship safety the concern, as in most practical cases of engineering design, is not perfect safety but with degrees of safety, measured in some cases by the probability and costs of trouble during the life of the ship. A corollary to this is that perfect safety in an economic sense is unobtainable.

Because of the desire and need to be efficient in the economical sense, a designer will on occasion step across the hazy boarderline between safety on the one hand and disaster on the other. Figure 1, based on [6] illustrates the different types of total losses and serious casualties between 1976-1980 for certain United Kingdom fishing vessels; only a small percentage are designated "capsizings". There is then behind us a long history of trial and error and efforts to minimise accidents by appeal to systematised experience, whether empirical or scientific.

## 3. WEATHER CRITERION

The merchant ship Weather Criterion, for assessing the effect of severe wind and rolling on a vessels intact stability, has been the subject of much debate at IMCO and elsewhere for some considerable time. Guidelines for its use were included in the 1977 Fishing Vessel Safety Convention [7]. In more recent years the IMCO Working Group on Improved Intact Stability Criteria have been developing an International Weather Criterion [8], for merchant vessels in all conditions of loading and especially for lightly loaded conditions. It is important to realise that the Weather Criterion as formulated is essentially a "design point" problem: that is, the criterion is configured to indicate an adequate level of intact stability for one particular value of ship speed, heading, sea state and wind speed and yet the vessel must operate satisfactory for an infinite variety of off-design conditions, including those more severe than actually specified.

Although the use of a "Weather Criterion" approach for the assessment of stability for mobile offshore drilling rigs [9] has proved reasonably satisfactory to

date, this is likely to be for completely the wrong reasons. As far as its application to ships is concerned it must be regarded as a specific "design point" criterion and complementary to the more general IMCO stability criteria in current use.

#### 4. APPROACHES TO ADEQUATE INTACT STABILITY ASSESSMENT

There are two basically different approaches to the solution of adequate intact stability for a ship, which are analogous to those of structural design as advocated by the ISSC Committee 10 on "Design Procedure" [10]. In the first, or 'evolutionary' approach, satisfactory rules and procedures are evolved gradually by a process of trial and error, experience and modification. Using the sea as a laboratory, and ships as full-size models, a selection of service data is accumulated, including information on ships that have been lost, which can then be interpreted in the form of acceptable levels of stability for future similar ships. In its purest form, the 'evolutionary' method makes little use of theory. This method has disadvantages which are well understood, even when applied only to conventional ships and it is one which cannot easily distinguish between adequacy or over-adequacy of intact stability.

The other approach might be called 'deterministic', since it proceeds by determining quantitatively as many as possible of the factors affecting the behaviour and seaworthiness throughout its life, and using this information to prepare a design with sufficient degree of intact stability with a minimum reference to previous experience. Thus, the designer attempts to quantify the wave loadings on the ship, the relevant seakeeping properties, the detailed response of the ship to each condition of loading, etc. and hence to determine largely by calculation, its probable behaviour during the lifetime.

It appears that intact stability assessment is on the threshold of undergoing a transition from 'evolutionary' to 'deterministic' methods as found in the proposed IMCO "Weather Criterion". The basic reason for this lies in the chief disadvantage of the evolutionary method, that in the absence of relevant previous experience, it is largely useless or at best dubious. So that where there is no 'previous ship' (e.g. as in the first SWATH ship design) on which to base a new design, the designer must attempt to develop a deterministic approach. Often previous experience with conventional vessels is used, unless a supporting research programme is providing the necessary guidelines. So long as the designer or regulatory body is confronted with only minor changes from past experience, the evolutionary method should ensure a reasonable degree of intact stability, although the margin of safety or the probability of

'capsize' would remain unknown, and the design would not necessarily be either economically efficient or suitable from an operations point of view.

It is when some significant departure from previous practice occurs that the need for a more deterministic approach to defining adequate levels of intact stability for design becomes imperative. In naval architecture these significant departures have been arising increasingly often in recent years. The most striking example had been the introduction of the offshore supply vessel, which has demanded a bold extrapolation from past experience by the introduction of an 'equivalent' area under the vessels righting arm curve to correspond to that required of more conventional hull forms [11]. However, with the aid of theoretical considerations, comparative calculations and data on sea states and ship roll response in extreme seas it will soon become possible to make such extrapolations with reasonable confidence. This is supported by the ever increasing volume of data from research which should enable advances to be made in the deterministic approach for all ship types.

The significant departures which create the most difficult problems for design in general are those in which extrapolation is not possible, because some new technology is involved. Examples of this in the maritime field include novel ship forms, such as twin-hulled ships or semi-submerged ships (SWATH ships) oil drilling rigs, stable platforms, ocean research vehicles etc. In all of these developments the designer, and no less important, the regulatory body - must have some means of determining, with little or no recourse to previous experience, whether his proposed design is likely to be satisfactory, and this must include an assessment of adequate intact stability.

Faced with this situation it is essential to develop a rational philosophy, and a logical procedure, of assessing intact stability in particular, wherein the essential steps and decisions are clearly indicated. Such a procedure when applied to a conventional or novel vessel will not only ensure a consistent approach to design, but will show clearly where the uncertainties lie, and where further research is most needed.

##### 4.1 Design Procedure and Criteria

Following the steps considered by the ISSC Committee 10 on "Design Procedure" to be essential in a rational design process for structures, similar ones are suggested for stability assessment as indicated below and illustrated in a more general framework in figure 2, which is based on [12].

1. Choice of ship configuration and the fixed main dimensions.
2. Selection of area of operation and functions to be performed.
3. Determination of environmental conditions and or critical operating conditions affecting the roll response of the vessel.



4. Choice of intact stability criteria, such as static loads that cause a list (passengers crowding to one side) or the dynamic instability in following seas.
5. Preliminary design of vessel and analysis of its response to excitation assumed in 3.
6. Comparison of results in 5 with criteria in 4.
7. Choice of criterion for optimization, such as most economical vessel i.e. payload against first cost etc.
8. Redesign of vessel with different configuration, with lower centre of gravity, wider beam or larger freeboard, and selection of optimum designs using 7.

The designer, confronted with a new design, for which there is little guidance from experience, needs to consider all these items, though not necessarily in exactly the order given above. It is worth emphasizing the important difference between intact stability criteria (step 4) and the optimization criteria (step 7). The former is concerned with only ensuring adequacy of stability of any proposed design. The latter is a criterion by which the best of all these adequately stable designs shall be selected; it is an "owners criterion" by which he can select the design of maximum efficiency for his requirements.

It follows that although both criteria are of importance to the owner and designer, the approving authority is more concerned with the intact stability criterion. Inadequate stability can have economic consequences, and it may involve loss of life. The intact stability criterion, and the margins of safety which it implies, are thus of crucial importance to all concerned with the design. Moreover, in considering this criterion, the right compromise must be found between providing excessive safety by unnecessarily large margins and thus render the design less competitive, and using too low margins of safety and risking loss of life and revenue. It is because of this important element in the design process that the author considers the stability design criteria to be worthy of special study.

#### 4.2 The Need for Margins of Safety

It is suggested that the overall criterion of intact stability is that the design should have an adequate margin of safety against capsize or an acceptably small risk of excessive roll motion or capsize during its working life. This requires a definition of "excessive roll motion", by which the design is rendered "inadequate". At least two categories of limiting roll motion or dynamic stability (as opposed to statical considerations) may be postulated:

##### (i) Excessive Rolling

The roll motion of a vessel could be considered excessive if the roll amplitude velocity or acceleration exceeds certain

bounds, or acceptable limits, as a result of excitation at either general or particular harmonics or sub-harmonics of the vessel's natural roll frequency, or due to pure loss of quasi-static stability when poised on a wave crest.

The limiting values of the roll related responses will determine the boundary of operation and the resulting safety margins. However, most forms of unstable and sub-harmonic response are notoriously difficult to predict, as in the case of Mathieu instabilities of tethered buoyant platforms [21, 22]; they are often sensitive to the degree of randomness in the waves, and to damping.

##### (ii) Dynamic Directional Instability

In this case the roll motion is considered potentially unacceptable if the coupled motions of roll, yaw and sway give rise to an excessively "directional unstable" vessel, judged by certain acceptable limits of rudder control. These limits could be similar to the manoeuvring criteria suggested by Clarke [15] or the prediction of loss of adequate control advocated by Bishop, Neaves and Price [16]; both of these approaches are in fact based on linear theory.

Having postulated the limiting states corresponding to excessive rolling and dynamic directional instability, it is then necessary to find the environmental conditions which will bring about these conditions of unacceptability or failure. This is primarily an analytical problem of determining the appropriate motion response to environmental loading. It is assumed that with the aid of the mathematical models or experimental results, the limiting environmental loading causing concern can be determined at the design stage. Here it should be noted that the measures of ship performance that should be used in the design stage are those of seakeeping and manoeuvring in quartering seas where the greatest danger from extreme rolling motion usually occur.

#### 4.3 Margins Between Working and Limiting Conditions

Let us assume that the designer can define a 'working' or 'normal' level of stability for a vessel. In some simple cases this is straightforward where static loads are dominant, such as in lifting heavy loads onboard the vessel. However, where the stability is varying continuously, some realistic condition, representative of the anticipated operating conditions, must be adopted as a design working condition. Here we are considering simple working levels of stability defined as a unique measure as opposed to levels of stability deduced from probabilistic considerations.

The need for some margin between the numerical values of 'working' and limiting levels of stability is obvious due primarily to uncertainties of various kinds. If we define the safety margin in the general way:

Safety margin =  $\frac{\text{Limiting level of stability}}{\text{Working level of stability}}$

where the working level could be the static heel or the significant roll motion etc. In this case, once the working condition is defined and a safety margin chosen, the design problem is to simply ensure that the ship has the characteristic necessary to exceed the limiting condition. The problem here may be to decide on the limiting value of roll motion which would probably be greater than the angle of deck edge immersion and less than the angle of vanishing stability.

In structural engineering there has been a progressive movement towards the control of safety in design by the use of "load factors", based on strength and load data of a statistical kind. It is therefore suggested that the load factor is the most general and meaningful way of expressing a safety margin for the safety of a vessel in a seaway. However, it is also known that in structural engineering the determination of the demand, and the capability, and therefore the safety factor, is subject to many uncertainties.

It is in the treatment of these uncertainties that the statistical approach differs from the conventional procedure. In place of unique values of demand and capability the method uses statistical distributions of those quantities. The essence of the procedures is given in figure 3 where the frequency distributions of both demand (D) and capability (C) are expressed in terms of quantities such as load or number of cycles. In the conventional safety factor method of design a single value of capability is compared with the working value of demand; this is the risk of failure or capsizing. In the statistical approach the probability that the capability exceeds the demand is the reliability of the ship remaining upright.

In estimating risks of extreme roll motion or 'capsizing' using a 'probabilistic' approach as indicated above, it is the tails of the distribution that are of vital concern and this is where the data is most scarce and extrapolation most unreliable.

Even when data and statistical techniques are available there will remain the important question "what is an acceptable risk of extreme roll motion?". Two possible ways of arriving at answers to these questions, similar to those suggested by structural engineers are postulated. The first is to try to express quantitatively the risks of extreme rolling motion which the operator, or the designer, consider acceptable, albeit unknowingly. From casualty data on losses and accidents to ships and their cargo and from measured operational data risks of 'capsizing' might be interpreted as design criteria. The alternative method is to use statistical techniques to relate the risk of 'capsizing' to the mean values of 'capability' and 'demand' as already indicated. In any event it would be desirable for a ship

designer to estimate the extreme value of rolling motion for a particular design and then relate this value to an acceptable risk of 'capsizing' or extreme rolling motion.

## 5. THEORETICAL MODELS

### 5.1 Roll Prediction

The problem of predicting the rolling motion of a vessel due to wave action has been a matter of considerable interest to naval architects for a long time. This is because for many vessels the natural frequency of rolling motion is of a similar magnitude to the predominant frequencies of wave energy. Furthermore, the hydrodynamic damping associated with rolling motion is usually relatively small. It is not therefore surprising that vessels may experience a large resonant response leading to extreme rolling motion and possible capsizing. Many theoretical studies of this problem have been undertaken since the pioneering work of William Froude in 1861 [17]. Froude clearly recognised that ship rolling was essentially a problem of dynamics which must involve the inertia of the ship, the effect of damping and the hydrodynamic restoring moment. Moreover, he demonstrated that both the damping and restoring moments varied in a distinctly non-linear fashion with roll angle.

Although extensive efforts have been made over recent years to develop a more realistic theory for rolling motion by treating the wave input as a stochastic process, the linear spectral theory introduced into ship motion studies by St. Dennis and Pierson [18] is not particularly appropriate because of the non-linear nature of ship rolling. Unfortunately a general theory for non-linear system response to stochastic processes, of the same scope as the linear theory, is not yet available and thus progress in developing a satisfactory stochastic theory for extreme angles of ship rolling has been slow. On the other hand if linear theory is used coupling terms can be readily included, the theory is simple and gives both short-term and long-term predictions, and allows some stability phenomena to be examined (e.g. Bishop and Price et al [16]).

Most of the theoretical work which has been undertaken so far is derived from linear concepts. Consequently these approaches are only capable of yielding limited information on roll response statistics, such as estimates of the mean square of the roll angle or in some cases, the effect of non-linearities on the response spectrum. However, the linear approach can yield useful information on the probability distribution of roll angle but not on extreme angles. It is precisely this kind of information which is of vital importance for design purposes since it enables the probability of the roll angle exceeding specified bounds to be quantified.

There appears to be at least one method which is not derived from linear theory and which, in principle, is capable of predicting the form of the response distribution for non-linear response. This is the so-called Fokker-Planck-Kolmogorov (FPK) method [19] with an averaging approximation introduced by Roberts [20, 21], which is related to the general theory of Markov processes. The Markov approach does however make certain very restrictive assumptions, for example the conventional roll equation is assumed to be valid up to large roll angles and certain important coupling effects cannot be included at present. This approach does enable the effects of certain non-linear terms (e.g. damping and restoring moment) to be examined in isolation.

The greatest danger from extreme rolling motion usually occurs in following or quartering seas. For such cases it is essential to use coupled roll-sway-yaw equations of motion as the neglect of couplings can have a significant effect on roll motions, especially roll-sway couplings [22], [23]. Although many five and six degree of freedom, ship-motion prediction computer programs are in use throughout the world [24], the state of the art for the theoretical prediction of lateral motions, that is, the coupled sway, roll and yaw motions is not completely satisfactory. In particular, empirical or experimental data have to be provided to determine roll damping moments.

The naval architect therefore cannot rely on strictly theoretical procedures for the prediction of lateral motions, particularly for roll. Although the use of quasi-linear methods can be quite satisfactory for moderate sea condition when augmented by experimental data on damping it is not suitable for predicting roll motion in extreme seas. Moreover, the linearized equations of lateral motion are not suitable for use near zero wave encounter frequency where the accurate prediction of ship roll, sway, and yaw motion requires complex non-linear hydrodynamics. Furthermore, rudder-induced yaw, roll and sway motions would have to be recognized when such a situation arises, that is, for certain ranges of ship speed in quartering and following waves. Most conventional ship motion prediction methods implicitly contain the assumption [25] that the ship is directionally stable without the presence of a steering system and ignore the slowly varying second-order wave excitation with very long natural roll periods.

## 5.2 Spectral and Probabilistic Approaches

Two alternative spectral and probabilistic approaches to design of offshore structures are being more widely adopted to take into account the whole range of sea conditions which will be encountered during the structure's lifetime, rather than a single severe wave [26]. Both methods are

largely complimentary, describing different features of an irregular sea. For ease of application, the response, wave forces and surface elevation must be related linearly, which has doubtful validity for extreme cases. The widely-used technique of equivalent linearisation allows non-linear terms to be replaced by linear terms in an optimum way. The spectral approach is in terms of its frequency content, and demonstrates clearly the effects of natural frequency response. The probabilistic approach is concerned with the number of times given stress or response levels are exceeded and is thus relevant to fatigue life.

A linearised model also makes a probabilistic treatment more straightforward. Sea waves, for example, are usually represented as a Gaussian random process [27]. If wave forces and responses are related linearly to the surface elevation, then they too are Gaussian, and probabilities of exceedance of extreme values can be calculated directly.

For example the probability that continuous roll angle of a ship will exceed  $\pm\phi^*$  assuming a Gaussian distribution is approximately [25]:

$$P(\phi > \pm\phi^*) = 1 - \text{erf} \left( \frac{\phi^*}{\sigma\sqrt{2}} \right) \quad (1)$$

where erf is the error function and  $\sigma$  is the rms roll amplitude value.

Unfortunately, however, non-linear effects tend to become most apparent in extreme conditions, and long-term probabilities of exceedance based on a linear theory may be misleading. There are two ways in which to approach this problem. The most obvious is to use numerical simulation, thus including all relevant non-linear terms; the problems are of course exactly the same as with other methods i.e. the equation of motion for large amplitude waves and motions are unknown. The simulated history is analysed as if it were an experimental record, and the resulting histogram approximates the probability density function (pdf). Dalzell [28] estimated the peak rolling motions of a ship by this means. Numerical simulation is basically an experiment on a computer, but has two advantages over the model experiments: the parameters can be varied in a highly controlled way and it is possible to consider more extreme conditions than are possible within the laboratory. It is however very expensive to carry out simulations to represent 50 or 100 years of sea conditions.

There are further considerable difficulties if the dynamic equations contain non-linear terms. Haddara [29] tackled the problem of ship rolling by using the theory of Markov processes. Roberts [20] extended this technique by an averaging process, and investigated extreme rolling motion on a ship. The Markov technique is limited in its possible range of applications, though this range is rather

wider than is suggested by St. Denis [30] particularly if Robert's averaging technique is employed. It sometimes offers an elegant, economic and powerful means of extracting long-term statistical information.

### 5.3 Choice of Probability Density Function

It is usual to assume that for 'short term' prediction of ship motion the peak values of ship responses in a seaway follow the Rayleigh probability law. This presumes a random, linear narrow banded process. However, when estimating the extreme values of a vessel's responses in a seaway there will be a serious problem regarding the linearity assumption relating ship response to waves, especially for the rolling response. That is, because of the nonlinearity in rolling, the response may not be considered as a Gaussian process, even though the sea may approximate to a Gaussian process. It follows that for rolling motion of a ship the Rayleigh law cannot be applied in general for prediction.

With our present knowledge the Rayleigh probability function is not appropriate for predicting the rolling motion response and a different function is required, possibly a generalisation of the Rayleigh or gamma probability function. Jasper [31] has shown that the statistical theory of extremes, as developed by Gumbel [32] is applicable to the analysis of records of rolling. Ship motion and extreme stress variations analysed by Jasper fitted the measured data using the method of Gumbel under the assumption that the underlying distribution is of the log-normal type.

One of the main difficulties in predicting extreme response by any such approach is in knowing the limitations of the theory i.e. when the assumptions fail. This applies equally to the log-normal-type fitting procedure as to the more theoretical approaches.

### 5.4 Probable Extreme and Design Value

The probabilistic extreme value 'on average' of rolling motion of a ship is defined as the largest value of the maxima (peaks and peak-to-trough excursions) which will occur in a specified number of observations or in a specified period of time. Extreme values can be estimated, using the long term prediction approach given by Ochi [33] from the probability function  $g(x)$  which provides information concerning the largest value. Here,  $g(x)$  is derived by applying order statistics to the probability function,  $f(x)$  which govern the rolling response in a seaway. The extreme value which is likely to occur (the probable extreme value denoted by  $y_n$ ) is the modal value of the probability density function of  $g(y_n)$ .

If the probability distribution is assumed to be an exponential-type distribution the probability that the largest value will exceed the most probable

extreme value,  $y_n$ , is theoretically  $1 - e^{-1} = 0.632$  for large  $n$ , where  $n$  is the number of observations. Since this probability is rather high, it is not appropriate to consider this value for engineering design. For design considerations, a certain margin is required, and this can be obtained by estimating the extreme value which is unlikely to be exceeded with a preassigned small probability. This implies that, for large  $n$ , we may determine the design value from the cumulative distribution function of the response such that the probability of being exceeded is  $\alpha/n$ . Then, this design value is equivalent to the value which the extreme response may exceed with probability  $\alpha$  - see figure 4.

The spectral analysis technique for predicting ship responses in a seaway is now a well established technique. This approach is based on the linearity assumption between waves and ship and is therefore not really suitable for predicting non-linear responses, such as roll motion in heavy seas. Unfortunately there is no well established non-linear spectral analysis available at present and from the prediction of non-linear responses may at best be based on random sample data from experiments in the form of a histogram of a response. The problem of using histograms to predict non-linear response is essentially the same as with numerical simulation. That is to say unless there is sufficient data to define the tails of the distribution the estimates of extreme response will be unreliable.

A prediction can then be made from an appropriate probability function which governs the response. The objective in establishing this function is to derive useful information for design considerations. Estimates of extreme roll motion may be of particular interest, especially if the vessel is to withstand the severest environmental loads expected in her lifetime, and this can be achieved by applying order statistics to the probability function.

Two extreme values of roll motion are of most interest: one, the "most probable extreme value" and the other the "design extreme value". The former is the mode value and may reasonably be compared with the largest value observed in model tests or full-scale trials. The latter is the extreme value for which the probability of being exceeded is a small value,  $X$ , which is specified by the designer.

If the designer wishes to evaluate the extreme value of roll motion for a design the approach indicated above will provide an indication of the vessel's capability. An alternative method of determining the capability is based on probabilistic reliability analysis. In this case the probability function for both environmental loading (demand) and a vessel's capability in terms of roll response are considered. Then the assurance of safety or reliability is expressed in terms of the probability of non-failure. Inversely, the risk (probability) of failure,

the first passage failure in this case, is given by the overlapping areas. (In the first method it is understood that the probability function representing capability is considered to be a point probability function in contrast to a probability function with some dispersion in the second approach).

As far as the estimation of extreme values is concerned, it is appropriate to consider only severe seas which a vessel is expected to encounter in the service area and apply the short-term prediction method. The long-term prediction method has a significant disadvantage in estimating the extreme values [34], [35]. This is because a considerable percentage of the vessels responses in the estimation are of small magnitude in relatively mild seas and these do not contribute very much to the extreme value estimation.

### 5.5 Physical Consideration and Limitations

A statistical treatment of the kind suggested will tend to ignore certain physical considerations. Perhaps the ideal method to apply to such a situation would be the development of a complex mathematical model of the physical situation on the basis of which predictions could be made, but this represents a daunting problem of applied mathematics. Meanwhile the best possible use must be made of the statistical methods that are available and perhaps in time certain physical considerations could be incorporated into this approach.

Questions can be raised as to whether the suggested treatment makes allowance, for example, for the occurrence of freak weather conditions that may occur perhaps once in ten years and would be the very conditions in which there is a high risk of capsizing. Similarly it may well be that the sort of sea condition that would make a capsizing likely are normally avoided and a capsizing may occur on isolated occasions when trawlers are out in such conditions, perhaps for economic reasons or due to sudden unexpected changes in conditions. Accidents could also be partly due to human failings - tiredness, bad loading of vessels etc. It is difficult to see if, or how, such factors are, or can be, adequately taken into account in a mathematical model, but these items could be accounted for in a semi-probabilistic approach.

### 5.6 Method of Estimating Extreme Value

In predicting the design extreme value of rolling motion for a ship, two particular approaches may be considered as advocated in the design of an offshore structure. The first approach takes into account all sea states expected in the vessel's lifetime weighted by the frequency of occurrence of each sea state using a family of wave spectra in each sea severity [23]; referred to as the long term prediction. The second approach for evaluating the design extreme

value considers individual sea states taking into account their persistence, and referred to as the short term prediction method [36].

In order to derive a probability density function which is applicable to the lifetime response of a vessel, various factors such as frequency of encounter with each sea severity, various shapes of wave spectra expected to be encountered, ship speed, loading, heading to waves etc. have to be considered. For this purpose it is recommended to use a statistically established family of wave spectra for design considerations rather than a single wave spectrum for each sea state.

Two alternative families of wave spectra are available which are suitable for deriving the required probability density function. The first family is a series of spectra consisting of eleven members for the six-parameter spectrum [37] and the second series consists of nine members for the two-parameter spectrum [38]. Both of these series have been statistically established with a confidence coefficient of 0.95. Figure 5 is a parametric modelling of a joint probability of wave height and period such as the sample data set shown in figure 6. This joint probability can be translated into a corresponding population of representative two-parameter spectra entered by respective wave height and period.

Alternatively, a different approach may be adopted for modelling wave climate in terms of parametric directional spectra using simple wave and wind statistics as input. This method essentially translates the joint probability statistics of wave height, wave period, wind speed and direction into a population of parametric directional spectra with matching statistical properties, [41], [42]. A technique of joint probability modelling used for predicting extreme wave heights is illustrated in figure 7 based on [43]. It shows that at each wind speed  $W_r$  having a probability density  $P(W_r)$  the joint probability density at any Height  $H_s$  denoted by  $P(H_s, W_r)$  can be expressed in terms of a scatter with a distribution function  $P(H_s | H_r, \sigma_r)$  where  $H_r$  and  $\sigma_r$  are the mean height and its standard deviation respectively and can be expressed as empirical functions of wind speeds.

From the response spectrum for each of the wave spectra in a given sea state, we can derive the probability density function of the response. It should be noted that all the above functions are assumed to be Rayleigh probability density functions but have different parameter values. However, the probability density function of the rolling response for the vessel's lifetime, is a weighted accumulation and is no longer a Rayleigh distribution. The design extreme value based on the long term prediction approach is then evaluated from the cumulative distribution function by integration. This concept of the probabilistic extreme value estimation

for rolling motion is worthy of consideration and could be implemented in the design stage of ship design.

Using this approach it is possible to assess the expected roll motion by applying recent technology developed on the probability concept. The assurance of extreme rolling motion against wave-induced loads may be expressed in terms of probability of possible 'capsize'.

If the probability density function of the vessels capability is not known, as in the early design stage of a vessel, then one way to determine the design value is to estimate the extreme loading which a vessel may experience in a lifetime and then design the level of intact stability to withstand this extreme loading. Here it would be necessary to consider the design extreme value for the assessment of the design value.

#### 6. A FRAMEWORK FOR A NEW STABILITY AND SEAWORTHINESS CRITERIA

In the long term intact stability criteria and the related safety of a ship in extreme seas should ideally be rationally quantifiable in terms of risk or loss or of exceeding certain bounds of roll motion and ideally taking account of operational factors such as poor seamanship. Although the analysis of ship losses can provide some information on the safety of those vessels in a conventional risk analysis sense, the causes can not always be established with sufficient certainty to enable design improvements to be made [44, 45, 46]. An alternative and practical solution to this problem is therefore to develop a design method for predicting the probability of occurrence of extreme roll motions that are judged to be dangerous or undesirable for the safe operation of the ship in question. Although a detailed statistical description of the environment will be required in such a method this will only be of academic interest unless it can be translated via the appropriate physical understanding into statistics relevant to the vessel's roll response.

A detailed description of the environment will indicate how often conditions occur in relevant sea areas, together with additional information on wave group/statistics and wind and wave spectra. Whereas the definition of the appropriate conditions for 'instability' or dangerous motions can be helped by physical models relating to wind and wave conditions and from research work on mathematical models, current design practice and operational experience. The degree of success of failure of any new stability criteria will therefore depend upon selecting the key design and environmental parameters together with the analysis procedures that can be translated in good design practice.

It is suggested that a framework for a rational model of 'capsize' risk could be

developed, figure 8, incorporating existing information on wave spectra as indicated in section 5.6 and appropriate mathematical and physical models relating the wind and wave conditions to extreme roll motion. The statistics of such a 'capsize' risk model are unlikely to be particularly sophisticated and the degree of success or failure will depend crucially on the appropriate choice of wave spectra or design wave parameters. From this information it would be essential to evaluate the joint probability of sufficient conditions for extreme roll exceeding certain bounds for a probability to be assigned to the risk of 'dangerous motion', or more simply a design extreme value could be obtained as discussed in section 5.4. Comparisons of predictions of this probabilistic model or the design extreme value should then be made in the first instance with time-domain simulation and ultimately model experiments, sea trials as well as assessments of risk from casualty data. In either case a large quantity of data would be involved and this might prove expensive. There are also problems with the model experiments concerning scale effect. The estimates of risk obtained could also be assessed if necessary by season and area of operation.

#### 7. DISCUSSION

In recent years there has been an over emphasis on the apparently intractable problem of modelling ship capsize. In contrast, little attention has been directed towards developing a framework for future design and stability criteria. Little more can be expected from any further concentrations of effort on either linear or non-linear single degree of freedom equations. Mathematical theory in its present form is not suited to the problem of capsizing. This is because mathematical stability theory is mostly concerned with conditions under which an initial perturbation becomes unbounded, whereas in the study of capsizing we are concerned with conditions under which motion exceeds a prescribed practical bound.

As a corollary to the above, theoretical stability theory should be concerned with large amplitude motions. This will require the development of more advanced theory for fluid active and reactive forces that vary with amplitude and the use of mathematical models that are sufficiently accurate to describe coupled roll-sway-yaw motions and associated antisymmetric motions. These equations may be of a completely different form from those used at present - which are basically the linear equations with simple additional terms. Unfortunately, such advanced theories in hydrodynamics do not exist. In this context the development of any new mathematical theories, although desirable are unlikely to be of any immediate help in assessing levels of intact stability



in the near future.

The main purpose of studying the simplified equations is to examine the effects of various non-linear terms in isolation. Model experiments are unlikely to provide a full answer to the stability problem since experiments can only be carried out in a relatively limited range of conditions, and it is necessary to extrapolate the results to cover a vessel's lifetime; this is precisely where you need a theoretical model.

It should be appreciated that if existing theoretical models are used it is not possible to 'predict' capsize or even estimate a probability of capsize, because the problem is too poorly defined. However, there are a number of ways in which existing theory can be of assistance;

- 1) To help understand some of the important parameters, and the way in which they interrelate.
- 2) To extrapolate the best available data (theoretical or experimental or both) to more extreme conditions or to cover the vessel's entire lifetime.
- 3) To understand the deficiencies of the present techniques, and to formulate more complete models (both theoretical and experimental).

While some areas of seakeeping need further research, such as the effect of the above water shape and manoeuvring in following seas, the primary interest today appears to be moving towards the use of existing technology in the ship design process. Against this background there is a need to establish a rational code of practice of operational limits of roll motion based on current practice and experience. The value of such a code is that it would enable the extreme roll motion of a vessel to be assessed in the design stage using state of the art coupled roll-sway-yaw equations; ultimately this could be assigned a probability of risk from dangerous motion.

A further refinement would be to take into account the absolute seaworthiness of the design and a measure of poor seamanship in order to make an overall assessment of capsize safety. Poor seamanship in this respect may be the result of inexperience in extreme weather conditions or the effect of motion fatigue causing a loss of mental and physical coordination; such effects may well help to explain some fishing vessel losses in recent years.

## 8. CONCLUSIONS

The general conclusions that have emerged from considering the philosophical aspects of assessing intact stability and safety are as follows:

- (1) In the interests of ship safety it is considered essential to establish a code of practice on operational limits of roll motion which should be a statement of current practice and should prescribe systematic procedures

and principles rather than limiting or standard values.

- (2) The greatest danger from extreme rolling motion usually occurs in following on quartering seas. For such cases it is essential to use coupled roll-sway-yaw equations of motion as the neglect of couplings can have a significant effect on roll motion, especially roll-sway couplings. Furthermore, rudder-induced yaw, roll and sway motion need to be recognised when manoeuvring in following/quartering seas.
- (3) The assessment of the risk of extreme roll motion (or a static list) to fulfill the design aim should be part of the process of assessing ship safety. The concern is with degrees of safety rather than perfect safety.
- (4) For significant departures in ship design where extrapolation is not possible, it is essential to develop a rational design philosophy and a logical procedure of assessing intact stability and thereby safety. Such a procedure when applied to a conventional or novel vessel will not only ensure a consistent approach to design, but will show where the uncertainties lie and where further research is most needed.
- (5) Risks of extreme roll motion or 'capsize' can be estimated using a probabilistic approach, although the important tails of the distribution are difficult and expensive to define. The important question of "what is an acceptable risk of extreme roll motion" can either be determined from casualty or operational data or from statistical techniques relating values of "capability" and "demand".
- (6) The estimation of the "design extreme value" of rolling motion using statistical techniques and a statistically established family of wave spectra can provide the designer with an indication of the vessel's capability.
- (7) Future intact stability criteria and the related safety of ships in extreme seas should ideally be rationally quantified in terms of risk of loss or of exceeding certain bounds of motion as a result of environmental forces. An environmental risk model is advocated as a logical framework for assessing safety in the design stages of a vessel.
- (8) The 'Weather Criterion' on the other hand is essentially a 'design point' problem formulated from a deterministic approach and as such it is much less general than the IMCO stability criteria. It must also be appreciated that its application will not necessarily indicate that a vessel will capsize or that it will enjoy a reasonable margin of safety or low probability of 'capsize'. Similar arguments could also apply to the

predictions of extreme roll angles using statistical techniques. However, in this latter case a more general solution is envisaged that would be of use in the design stages of a vessel.

## 9. ACKNOWLEDGEMENTS

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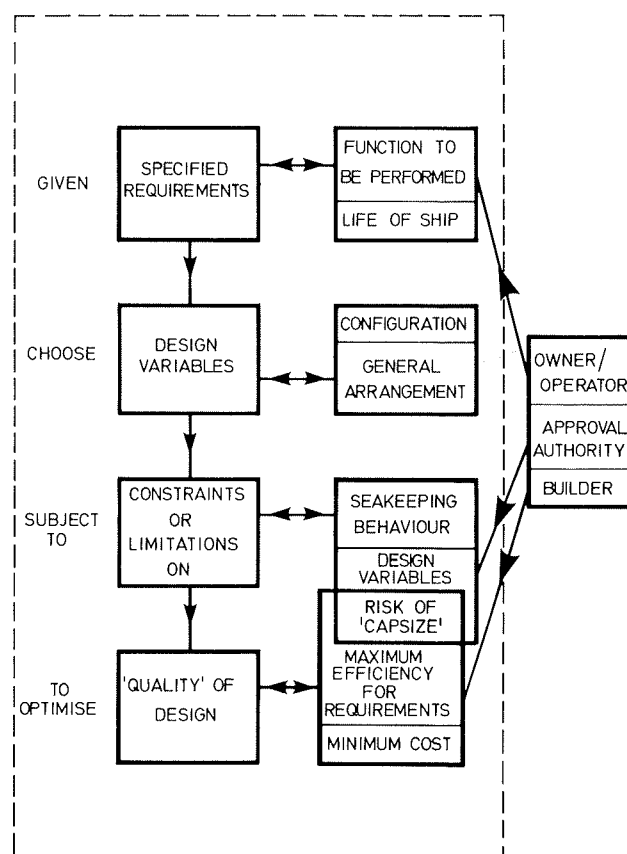
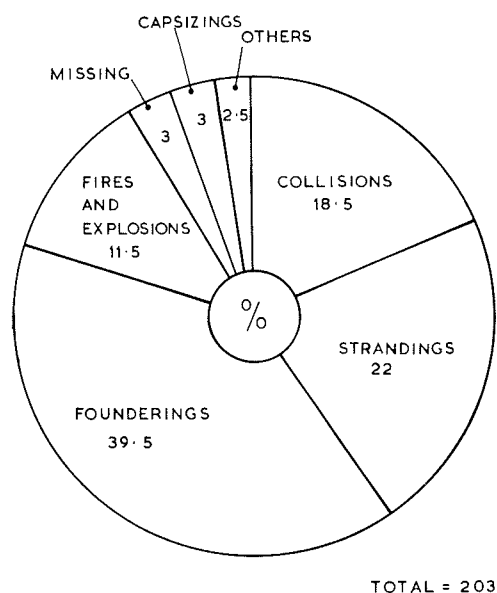


Fig. 1 Total Losses and Casualties of United Kingdom Fishing Vessels (24m in length and above): By Nature of Casualty - 1976-80, based on [6].

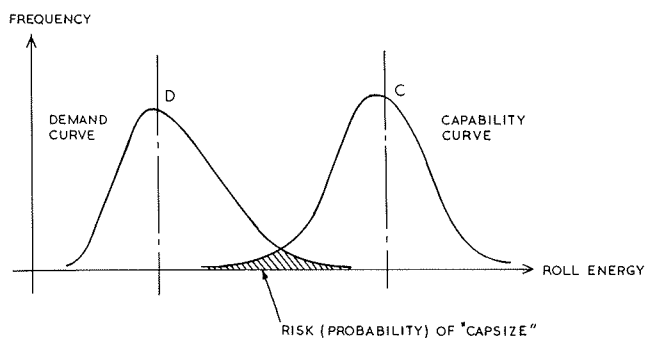


Fig. 3 Reliability Analysis of Extreme Roll Motion.

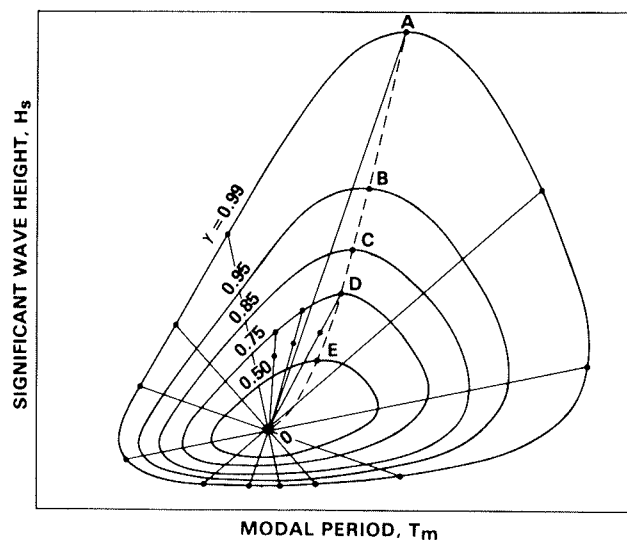


Fig. 5 Diagram illustrating a method to establish a series of wave spectra for long-term prediction, reprinted from [39].

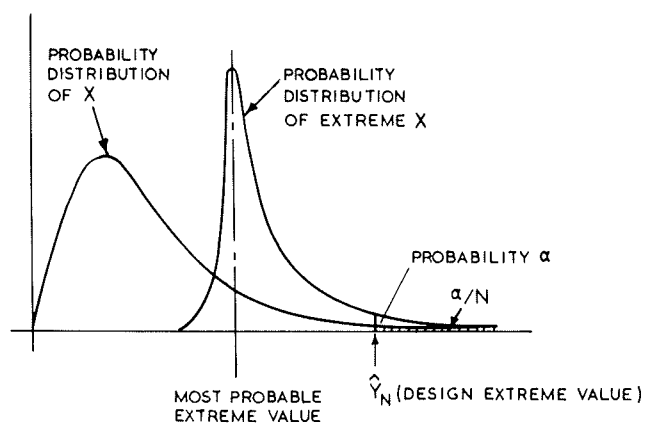


Fig. 4 Explanatory sketch of Design Extreme Value, based on [33].

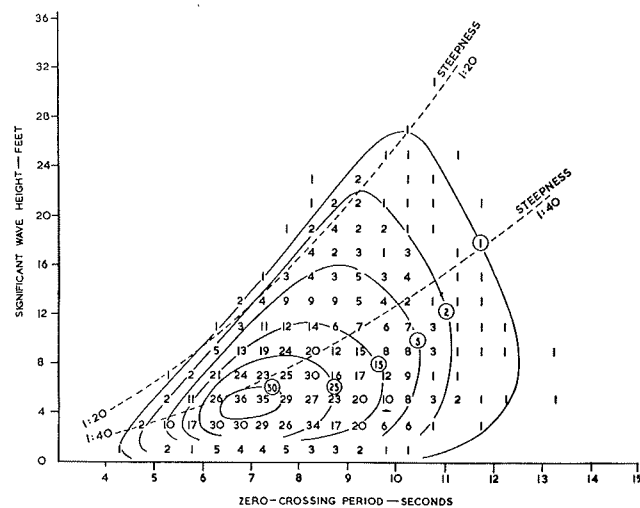
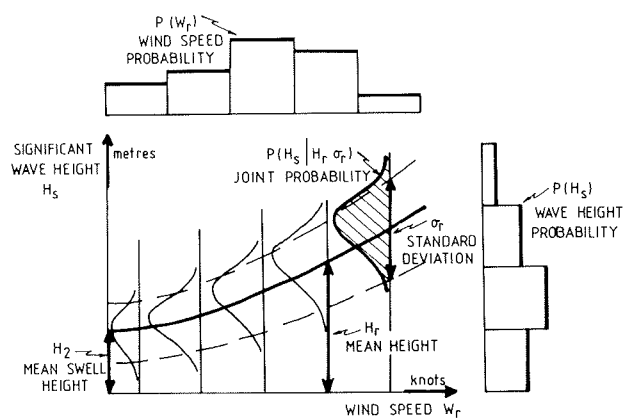


Fig. 6 Sample Joint Probability of Significant Wave Height and Zero Crossing, reprinted from [40].

PARAMETRIC JOINT PROBABILITY MODEL  
WAVE HEIGHT AND WIND SPEED



$$P(H_s) = \int_0^\infty P(H_s | H_r, \sigma_r) \times P(W_r) dW_r$$

WHERE:  $H_r = [(aW_r^n)^2 + H_2^2]^{1/2}$  m  
 $\sigma_r = H_2 [b + cW_r^n]$  m  
 $P(H_s | H_r, \sigma_r) = \frac{q^{p+1}}{\Gamma(p+1)} H_s^p \exp(-qH_s)$   
 $p = H_r^2 / \sigma_r^2 - 1$   
 $q = H_r / \sigma_r^2$

RECOMMENDED VALUES OF COEFFICIENTS

|               | $H_2$ m | a     | b    | c      | n    |
|---------------|---------|-------|------|--------|------|
| OPEN OCEAN    | 2       | 0.033 | 0.5  | 0.0125 | 1.46 |
| LIMITED FETCH | 0.5     | 0.023 | 0.75 | 0.0188 | 1.38 |

Fig. 7 Wave Climate Synthesis: Derivation of Wave Statistics from Wind Statistics using Parametric Model of Joint Probability, based on [48].

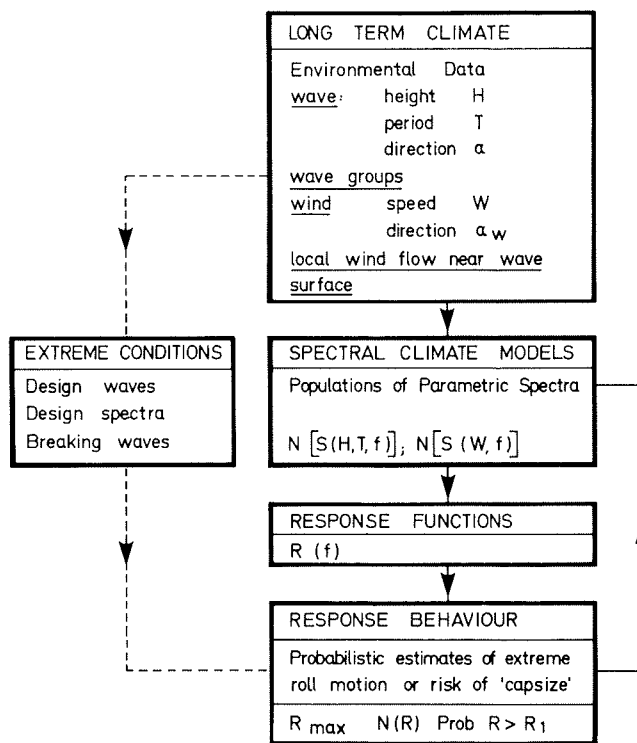


Fig. 8 An Environmental Risk Model of 'Capsize' or of Extreme Roll Motion.

## SHIP STABILITY - A RESEARCH STRATEGY

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### ABSTRACT

The authors believe that in any engineering enterprise, particularly where human life is exposed to dangerous conditions, it is the responsibility of the designer as well as the statutory authorities concerned to ensure that the structure, vehicle, etc. is safe judged by the scientific knowledge of the day.

Investigations into accidents sometimes reveal that the relationships between safety, a word often used in an abstract way, and design, constructional and operational aspects are not always fully appreciated by those who are responsible. Furthermore, over-simplified regulations or codes of practice may inhibit sufficient original thought being directed to safety.

Ship stability is, unfortunately, a property which is not amenable to simple definition. From the protracted debates at IMCO over the last 20 years it is evident that there has been developing world wide a desire to seek a better solution than is reflected in existing stability criteria.

It was out of such considerations in the United Kingdom that the SAFESHIP stability project was developed. It has become clear that a major scientific effort would be needed, employing the best resources available in terms of hydrodynamic, mathematical and computational techniques.

The SAFESHIP project represents one strategy for tackling this most intractable problem and the paper discusses the separate parts which form it and the need for national and international co-operation.

### 1. INTRODUCTION

The question of what are legally acceptable minimum standards of intact stability

of ships is not new but because of the subject's complexity it is one which has not so far been satisfactorily resolved.

Traditionally, the stability is deemed adequate depending on compliance with certain parameter values of the still water static stability curve, based on 'experience' (eg Rahola, IMCO Resolutions A167, A168, etc.).

Since these parameter values are the same for all ships regardless of size, type operating and weather conditions the margin of safety must vary considerably and is unknown.

With the advent of computer techniques and improvements in knowledge of hydrodynamics it seems reasonable to expect that at some time in the foreseeable future a better "engineering approach" to this problem will be found.

The SAFESHIP project was formulated to analyse the problem from a dynamic approach using the most up-to-date techniques available and if possible to establish safety criteria which contain proper and recognizable "margins of safety".

### 2. BACKGROUND TO THE SAFESHIP PROJECT

It has been our experience in the United Kingdom when analysing stability casualties that existing static stability criteria are not very helpful in establishing the reasons for dangerous rolling or capsize in relation to the reported weather and sea conditions. Consequently any design deficiency in the ship cannot be identified and the means taken therefore to prevent recurrence of such accidents. Furthermore the degree of safety provided by these regulations for specific sea conditions is unknown and is certainly not consistent in relation to different types and sizes of ships.

In 1974 IMCO formed a Working Group to develop improved stability criteria [1]. To support the U.K. delegation in this work and to co-ordinate stability research already going on in our country at the time, the U.K. Intact Stability Working Group was formed in May 1976.

This U.K. Working Group, after a careful review of the situation, decided to formulate a co-ordinated programme of research to be carried out at various Universities and other research centres where ship stability expertise was available. This eventually became known as the SAFESHIP project. In the process of formulating this research programme account was taken of known research in UK and abroad, eg. model experiments at NMI, mainly related to casualty investigations, papers read at the First International Conference on Ship Stability in Glasgow in 1975 [2], research carried out on behalf of the United States Coast Guard [3-4] and opinions expressed during a Seminar at NMI in May 1978 [5]. The somewhat similar Norwegian project S.I.S. was also studied [6].

### 3. AIMS AND OBJECTIVES

The aim of the SAFESHIP project is to advance existing knowledge of large amplitude rolling motions and capsize mechanisms and so to develop better design criteria and stability regulations. This will be achieved by exploiting current developments in the theory of ship motions, computer techniques and model experimental procedures. Ideally the result will be to establish relationships between survivability in terms of relevant ship dimensional and non-dimensional parameters and typical environmental parameters.

The main objectives are as follows:

- (1) To ascertain the present state of knowledge regarding ship motion stability in a world-wide context.
- (2) To sponsor extension of current theoretical and experimental research in the United Kingdom.
- (3) To establish by such research how the influence of the principal hull parameters should be reflected in future criteria.
- (4) Prepare the basis for future stability criteria which are related to acceptable survival probabilities and which recognise the influence of the parameters in paragraph (3).
- (5) To provide guidance to ship operators to assist them in avoiding dangerous situations when manoeuvring ships at sea.

### 4. FORMULATION OF RESEARCH PROGRAMME

The intention was to establish a coherent programme that would address all the relevant problem areas within a reasonable budget and time-scale. The procedure adopted was as follows:

- (1) A flow chart was prepared (Figure 1) indicating the types of forces,

internal and external, acting on the ship; the ship properties affecting response behaviour; theoretical and experimental techniques for predicting motion behaviour under these forces and finally to devise a practical criterion by a method which combines casualty experience with risk analysis.

- (2) To identify specific projects, eg. Environmental Demand, Risk Analysis, Mathematical Models, Experimental Programmes, Full Scale Measurements and Formulation of Criteria.
- (3) Having listed specific projects, to assess the approximate cost and time-scale of these individual projects and establish the likely overall cost.
- (4) From this a basic list of what were considered essential projects was made and specifications were drawn up. A document describing the SAFESHIP project and its aims was prepared and this together with the specifications were distributed to various Universities and research establishments.
- (5) On receipt of tenders a short list was made of contractors who were interviewed by a Steering Committee and a final selection made.

There was an enthusiastic response which suggested that there is a good deal of interest at least in U.K. to make advances in the present state of knowledge of this subject.

Several different approaches to mathematical models were suggested and since all of these appeared to have merit we have, at present, four different projects of this type.

A list of all the projects comprising the SAFESHIP project is as follows:

- (1) Environmental Demand, National Maritime Institute (NMI)
- (2) Risk Assessment, Newcastle University
- (3) Analogue Computer Simulation, British Ship Research Association
- (4) Mathematical Modelling, National Maritime Institute
- (5) Mathematical Modelling, Strathclyde University
- (6) Mathematical Modelling, British Ship Research Association
- (7) Mathematical Modelling, Bishop, Price & Partners
- (8) Model Experiments, National Maritime Institute
- (9) Full Scale Experiments, National Maritime Institute

### 5. A BRIEF DESCRIPTION OF EACH PROJECT IS AS FOLLOWS:-

#### 5.1 Environmental Aspects

This project is intended to provide information on the wind, wave and icing parameters which affect the motions of a vessel. Guidance will also be given on the prediction of the magnitude and frequency of occurrence of wind, waves and icing severity in selected areas in a form suitable for use

in modelling ship response. It is the authors view that the principal causes of a vessel's roll motion and occasional capsizes, namely wind and waves, should be allowed for in future stability criteria.

The work includes a literature review on the subject of waves, especially wave spectra, definition of high risk areas, eg. from casualty statistics with the intention of providing data in a form suitable for other SAFESHIP projects. Information on the wind strength at various heights above sea level will be obtained from measurements taken on the North Sea West Sole Offshore platform together with wind energy spectra from other available sources. Tests in a wind/wave facility will also be used in which extreme motions of a dynamically scaled model will be observed in a variety of wind and wave conditions.

## 5.2 Risk Assessment

The main purpose of the project is to explore the feasibility of developing and applying risk analysis as a basis for assessment of ship safety from capsizing, which may lead to improved design criteria.

It is well known that the factors influencing capsizing of ships exhibit variability from nominal or assumed values. This applies not only to the "external" influences of the environment which impose a "demand" on the survival capability of the ship, but also to the "capability" of the ship to meet the demand safely, being dependent on "internal" ship parameters which cannot be regarded as invariant. The probabilistic approach to risk assessment, and to the development of criteria for the acceptability of a design, has been making good progress in other areas of engineering and indeed in certain aspects of naval architecture (figure 2).

The probabilistic demand-capability approach to safety works reasonably well provided that:-

- (1) There is no significant coupling between the demand D (or load) and capability C (or strength).
- (2) Adequate data are available on which to base statistical predictions of extreme values of D and C.
- (3) A valid "model" of the interaction between D and C is available.
- (4) The resulting assessment of the risk of failure (i.e. the probability that  $D > C$ ) is meaningful, that is, levels of (non-zero) acceptable risks can be defined.

In the context of ship capsizing, each of these conditions presents difficulties which combine to make it uncertain whether this approach to survivability assessment will ultimately prove practical and advantageous. It is the intention of the project to investigate such difficulties.

## 5.3 Analogue Simulation of Ships' Rolling and Capsizing

The rolling behaviour of ships is being

simulated on a hybrid analogue computer, using non-linear time domain equations. Initially, a single degree of freedom rolling equation was used to explore the stability boundaries and methods were developed to produce on-line displays of roll amplitude histograms. These techniques are also being used as the system is enhanced to simulate rolling behaviour with three degrees of freedom, roll yaw and sway. It is intended eventually, to simulate a full six degree of freedom system.

By choosing different coefficients in the motion equations, different ship types and loading conditions may be simulated. A digital seaway generator allows the simulation to be forced by wave excitation equivalent to any desired amplitude spectrum.

Examination of the roll angle histograms allows estimates to be made of the probability of capsizing in any particular condition.

Although at present the simulation incorporates equations of motion which accord with the known state of the art, the intention is to utilise other forms of mathematical models as they become developed during the SAFESHIP project.

## 5.4 Mathematical Modelling

This theoretical approach is based on the theory of MARKOV processes which has been used to predict the probability of extreme events in non-linear systems. The probability is obtained analytically rather than by physical models or numerical simulation. Initially the emphasis will be placed on a comparison of roll amplitude distributions for a single degree of freedom equation obtained by this method and by conventional simulation. A comparison has also been made with experimental data obtained from model experiments on the FPV "Sulisker". Investigations into the effect of coupling between roll and other motions will also be made.

The mathematical model currently used assumes that the vessel's roll motion can be described by a single degree of freedom equation of the form:

$$A\ddot{\phi} + B(\dot{\phi}, \phi)\dot{\phi} + C(\phi, t)\phi = F(t)$$

where the damping and restoring stiffness may be non-linear. Both the total mass and the damping are assumed independent of the wave frequency: in practice values at the natural roll frequency are used. The excitation moment  $F(t)$  is assumed to be a stationary random process (linearly related to wave elevation). The ship's response, however is non-Gaussian by virtue of the non-linearities in damping and stiffness.

The aim of this work is to develop a theoretical model describing the probability distribution of roll amplitude.

## 5.5 Mathematical Modelling (Calculation of Wave Moments)

Roll moments which correspond approxi-

mately to restoring and excitation in the conventional model roll equation are computed.

The roll moment calculated is that which arises by virtue of the static equilibrium position of the ship in the incident waves. It includes effects such as direct wave excitation and restoring (the latter including time-dependent variations which give rise to parametric excitation). It does not include moments which depend on instantaneous roll velocity or acceleration, nor moments due to wave fields generated by the previous motion of the ship. These excluded effects are normally thought of as added mass and damping, and may be incorporated as simplified model terms.

The calculations are performed for ships travelling in waves of any given length, height and direction. Preliminary calculations have been carried out and stability criteria, based on energy balance over a predetermined critical roll cycle, will be suggested.

### 5.6 Mathematical Modelling

This study aims at deriving a realistic mathematical model of coupled large amplitude rolling motion to provide:

- (1) A time-domain simulation of roll time histories (deterministic and random) to provide an insight into the mechanisms of large amplitude roll motion build-up.
- (2) To devise stability criteria by making use of the derived equations of motion and the direct stability assessment method (eg. Lyapunov, etc.) [7].

In addition a study to rationalise the current IMCO weather criterion is also being undertaken.

### 5.7 Mathematical Modelling

Experimental evidence supports the view that ships travelling at relatively high speeds in following seas are at risk. In this project capsize is considered as an "antisymmetric motion" involving the coupled sway, yaw and roll motions of the ship.

Theoretical predictions using linear theory are not reliable for ship capsize but can indicate conditions under which the ship is likely to be difficult to control, suffering both directional stability and roll resonance simultaneously. These predictions rely on data derived from limited experimental testing and calculations. The latter are based on a hydrodynamics theory which models the complete immersed hull form and includes the influences of forward speed, wave radiation and diffraction, etc. From work done so far different ship types have all revealed different behaviour depending on the inter-relationship between instability and resonance.

Included in this project are the following specific investigations:-

- (1) Rudder-fixed stability
- (2) Manoeuvring characteristics in a sinusoidal seaway.

- (3) The influence of bias due to heel on the hydrodynamic coefficients.

### 5.8 Model Experiments

The model experiments are intended to provide measured data for the other related topics within the SAFESHIP project, such as the mathematical modelling projects and the analogue simulation project. Although model experiments in the United Kingdom have previously been carried out to study the behaviour of particular ships which were assumed to have capsized, the use of model experiments within the SAFESHIP project are being directed towards obtaining more basic data.

The work may be summarised under three main headings:

#### (1) Forced Rolling Experiments

The ship model is force-rolled by a sinusoidal roll moment generator that produces pure rolling moments. The roll moment generator, specially manufactured for the SAFESHIP project, consists of two spinning and tumbling flywheels that cause a sinusoidally varying roll moment to be applied to the model. The roll moments produce a rolling motion of large amplitude at resonance. Both linear and nonlinear damping coefficients will be obtained from the analysis of the forced rolling experiments (figure 3).

#### (2) Roll Restoring Moments

In order to establish experimentally the influence of following waves on transverse stability experiments are being conducted in a circulating water channel. The model is poised on a stationary wave which is generated by a fixed beam in contact with the moving water surface ahead of the model. Restoring moments are measured, using a modified GZ apparatus, at a number of restrained angles of heel (figure 4).

#### (3) Ship Motion Responses

In order to provide experimental data on ship motion responses in waves for validating the theoretical models developed within the SAFESHIP project, experiments are being conducted in unidirectional waves for both small and large amplitudes of motion. Measurements of all six ship motion responses in both beam and following waves will provide comparative data for the mathematical models. Initially the experiments were made with the model moored with light lines attached to the bow and stern and the model subjected to a range of wave spectra (figure 5). Subsequently the model will be partially restrained for experiments in following waves.

### 5.9 Full-Scale Experiments

Full-scale trials are being carried out in the North Sea on a 64m Fishery Protection Vessel 'Sulisker' (figure 6) to provide comparative data on the measured motion responses in a number of moderate and severe sea states in relation to theoretical and



model predictions.

Initially a pilot trial was made to obtain motion responses for the vessel while stationary in beam seas. A wave buoy was deployed during the trials to measure wave heights in the vicinity of the vessel.

#### 6. PRELIMINARY RESULTS OF THEORETICAL, AND FULL SCALE EXPERIMENTS

In accordance with our intentions mentioned in sections 5.8 and 5.9 to correlate theoretical predictions with model and full-scale experiments some preliminary results so far obtained are shown as follows:-

- (1) Comparison of righting levers obtained from experiments described in section 5.8, para. 2, with those calculated by computer (SIKOB program) using the wave profile as measured in the circulating water channel (figure 7).
- (2) Histograms of roll amplitudes obtained from the experiments described in section 5.8, para.3, are compared with those obtained by analogue computer as described in section 5.3 and with the MARKOV process in section 5.4 (figures 8). For this comparison a JONSWAP spectrum was used (figure 9).
- (3) Histogram of roll amplitudes obtained from full-scale trials (figure 10). It might be noted that the wave spectrum differs from that adopted for the theoretical and model predictions.

#### 7. PHILOSOPHY FOR NEW STABILITY CRITERIA

Ultimately it is desirable that safety of ships from dangerous rolling or capsizing in relation to particular sea conditions should be quantifiable in terms of risk, ie. probability, with reference to ship and environmental parameters which are found significantly to affect rolling behaviour. Until substantial progress is made along these lines, design criteria and regulations will not truly reflect sufficient safety.

At the present time analysis of ship losses can provide some information about safety in the manner that was used by Rahola or later by IMCO when developing Resolutions A167 and A168. However completely rational stability criteria can not be achieved by this means alone for various reasons such as smallness of sample, variability of ship types, loading conditions and sea conditions at time of loss of these sample ships (2).

The authors believe that a better alternative is to develop analytical methods for studying the behaviour in relation to defined sea states and using these to assess the significance of important hull parameters which can therefore be accounted for in future criteria.

The SAFESHIP project has been conceived with the primary objective of developing improved mathematical models and solution techniques for predicting large amplitude motions. Some attempts are being made to validate these predictions with models and with full scale measurements. The environ-

mental studies will provide data to enable probabilities to be assigned to such estimated motions. In this way it is hoped that the degree of safety could be defined in probabilistic terms.

When considering what form future stability criteria might take it is useful to look for analogies in other branches of engineering. For example, in structural engineering concepts of reliability analysis have grown up over the years from notions of "factors of safety" [8]. In the early days of structural engineering, eg. cast iron railway bridges, lessons learned from failures indicated the need for such factors of safety to allow for unexpected loads, live loads, corrosion, fatigue and other unreliabilities in materials.

Concepts of risk analysis evolved as randomness in both loading and structural response became better understood and such methods have also been applied to aircraft and ship structures [9-10].

(It may not be possible to carry this analogy too far in the safety assessment of ship stability not only because the response is more difficult to determine in a demand/capability relation but also because the 'capability' is not independent of the 'demand', especially in head and following seas where the environment changes the capability).

#### 8. CONCLUSION

Although some preliminary results of the SAFESHIP project have been given in this paper, and other papers submitted to this Conference, the project is far from complete and from our efforts to date it is clear that we are dealing with a subject of great complexity which may well be beyond the resources of any one country.

The authors, therefore, welcome this opportunity to discuss United Kingdom research with those engaged on similar work in other countries and hope that closer international co-operation might be stimulated by this Conference. Research effort in different countries would thus be more effective and shorten the time-scale for significant progress.

#### 9. ACKNOWLEDGEMENTS

The SAFESHIP project is being funded by the Department of Trade, Marine Division, and we are grateful to them for this financial support without which none of this research would have been possible.

We are also grateful for the encouragement given both by that Department and by the National Maritime Institute and for their permission to publish this paper.

The active support of all the members of the UK Intact Stability Working Group throughout the development of the SAFESHIP project is also gratefully acknowledged.

Any opinions expressed are the authors' own and should not be attributed to the above mentioned bodies.

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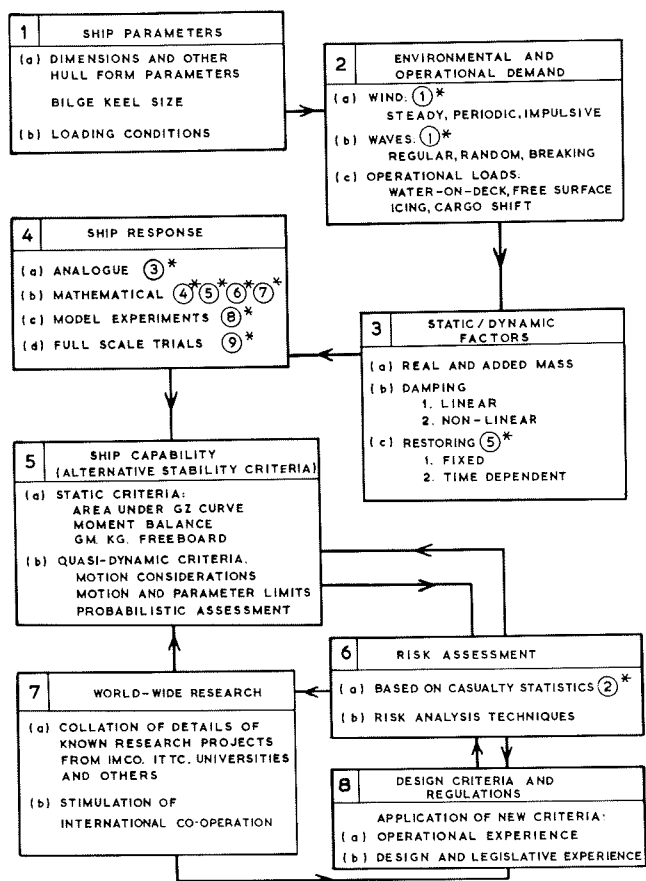


Fig. 1. SAFESHIP Project Flow Chart

\*Numbers in circles are SAFESHIP project numbers

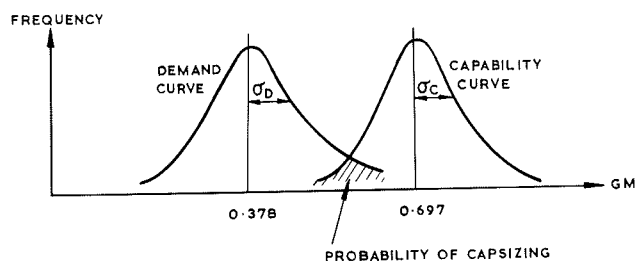


Fig. 2. Schematic Diagram Illustrating Margin of Safety

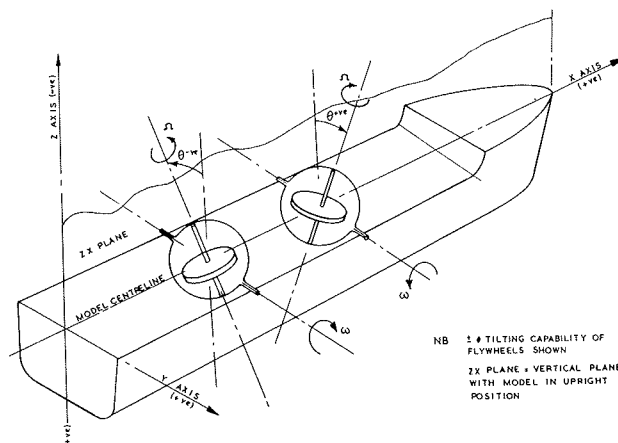


Fig. 3. Roll Moment Generator: Principle of Operation

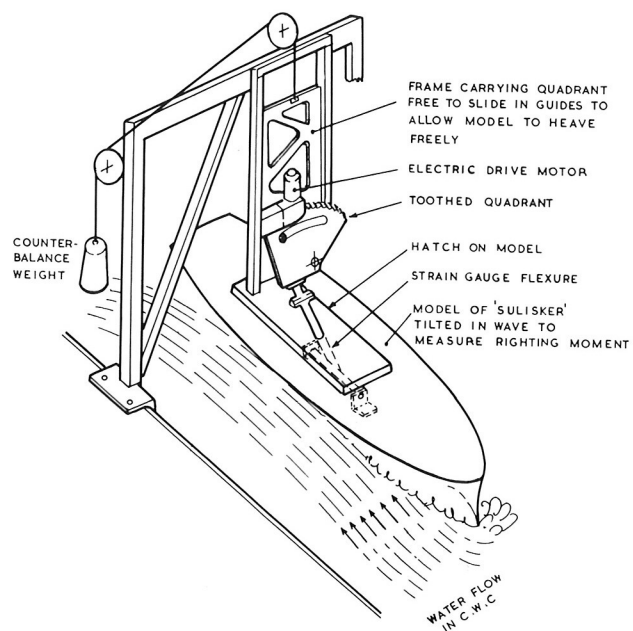


Fig. 4. Model under GZ Apparatus to Measure Restoring Moment while heeled on a stationary wave in circulating water channel

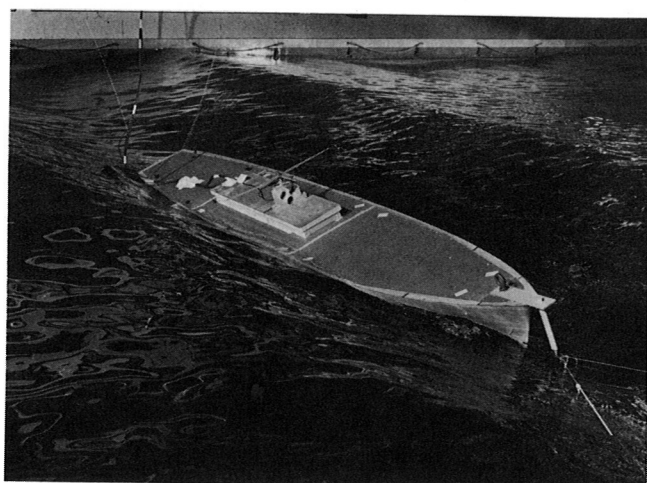


Fig. 5. 'Sulisker' model in beam waves



Fig. 6. Fishery Protection Vessel 'Sulisker'

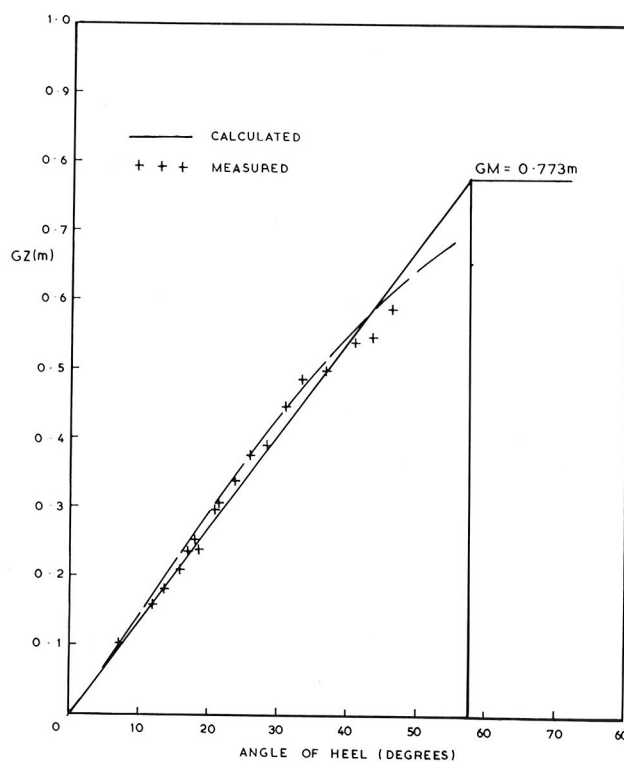


Fig. 7. Comparison of Righting Levers obtained from experiment and calculation: wave crest amidships,  $\lambda = L$ , wave height  $L/20$ .

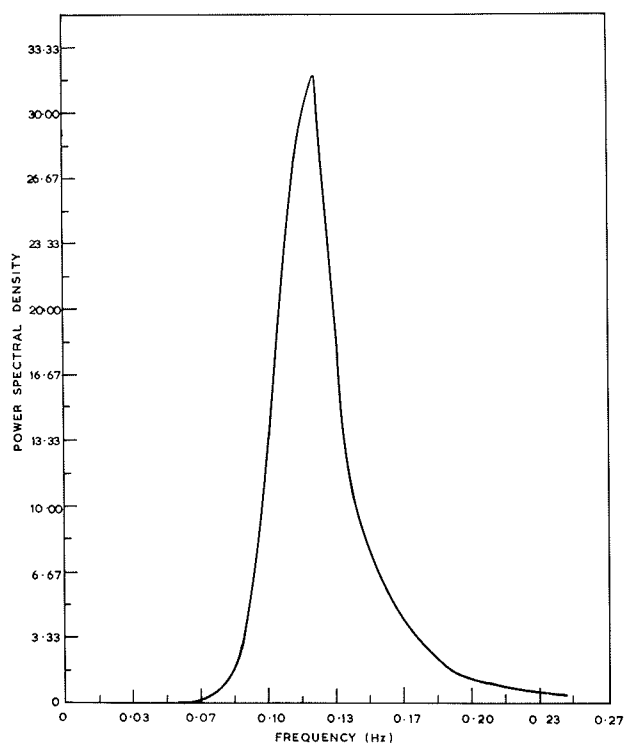


Fig. 9. Measured JONSWAP Wave Spectrum:  
 $H_S = 4.4\text{m}$   $T_Z = 6.6\text{s}$

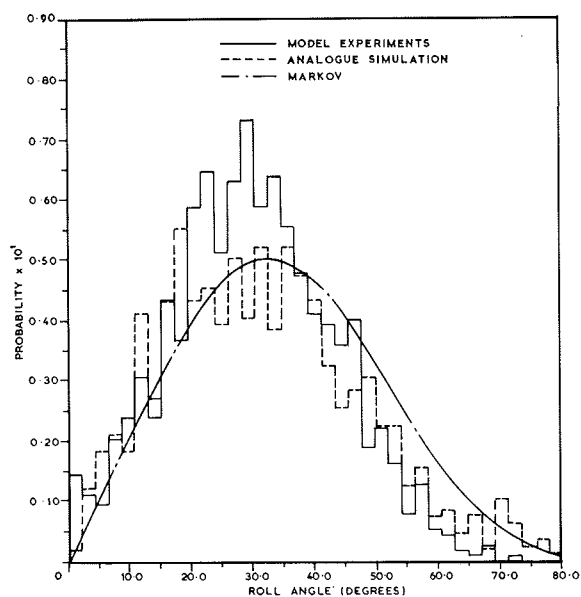


Fig. 8. Probability Distribution of Roll Angle (peak to peak) obtained by three methods: JONSWAP wave spectrum:  $H_S = 4.4\text{m}$ ,  $T_Z = 6.6\text{s}$

- (1) Model experiments  
 (1:20 scale)
- (2) Analogue simulation
- (3) Markov process

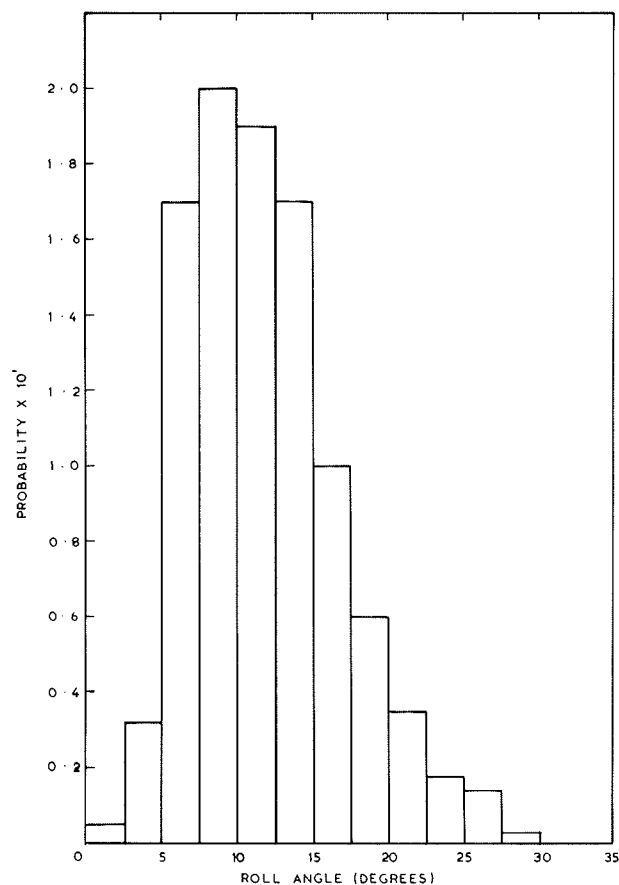


Fig. 10. Probability Distribution of Roll Angle (peak to peak) obtained from full-scale trials:  $H_S = 2.6\text{m}$

## IMPROVEMENT OF INFORMATION TO THE MASTER

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### INTRODUCTION

In the past quarter century, the amount of official information to the Master provided or required to be provided by Administrations has increased many fold and has, at the same time, become more intricate and more sensitive to workload and legal problems, both national and international in scope.

The particular subject of this paper is the stability information needed by the Master to lead his ship safely across oceans and through unusual loading patterns or maneuvers. Is the information supplied adequate?; Are there several reasons for supplying this information?; Are the different reasons compatible or do they indicate a need for different types of information to be supplied?.

The statements in this paper are either the authors opinions or that of the research team. No part of this paper may be considered official policy of the U S Coast Guard, Dept. of Transportation, or of the U S Government.

### BACKGROUND Part I

The Master of a commercial ship has traditionally required information on a wide variety of subjects. Among these are :

- Trade Routes
- Culture and Traditions of countries visited
- Traditions of the Sea
- International & national Law
- His Ship
- the Oceans

Significantly, since IMCO began its discussions on marine safety, much information on ocean weather and wave systems has been discovered. So much so, that Masters of ships on full ocean voyages (especially high latitude voyages on which the seaway is expected to be severe a great percentage of the year) are often expected to use the new techniques of "weather routing". This information has only recently been utilized by designers, and is not yet fully utilized in Approved information to the Master unless it is necessary as a restriction on his operation of the ship.

### Growth of Marine Systems

Additionally, the variety of the ships and floating systems on the world's oceans have multiplied. This point is of great significance. It means that the rules, regulations, customs and laws that fit the slower changing maritime world of the first half of this century, probably will not adequately fit the great many new marine systems much longer. New rules, regulations and laws will be necessary. The confidence begotten by several decades of experience on traditional types of ships cannot be assumed to be completely valid for all of the newer marine systems which have been noted to change every five to seven years..

Thus, the Master of an oceangoing ship or marine system such as a mobile drilling or mining unit must be supplied with information of many kinds; must make a sequential list of decisions, some of them in a very short time and each decision may affect the next several decisions in his voyage.

#### Safety Research, Standards, Operation

Although the three areas are closely allied to each other, they are not synonymous and they do not have the same specific objectives nor the same material result.

Safety Research is often quite specific, time consuming, and expensive. Much printed material including ships plans and graphics of all kinds must be reviewed before a decision as to the proper safety standard can be reached. We must acknowledge that, during the research phase of a stability analysis on any ocean system, any calculation method which is comfortable to the researcher and any amount of calculation must be accepted as proper.

Safety Standards necessarily include a generalization of the characteristics of groups of vessels in a given situation. When a safety standard is created out of safety research and generalized for a wide group of ships it becomes a minimum acceptable level of achievement for the designer.

This paper is largely concerned with safety operation, the last of the above three functional areas. The information to the Master through assists safety of operation through several international documents as well as many national administrations.

#### Stability Requirements

There are now three international conventions and two international codes of safety which require the Master to be supplied with 'Approved' information. The Administration according to each of these conventions or codes has accepted full responsibility for the certificate. Many Administrations have placed these requirements into national regulations.

These are:

International Convention on Load Lines, 66

Safety of Life at Sea Convention 1974

Marine Pollution Convention 1963 \*\*

Code for Safety of Chemical Ships \*\*

Code for Safety of Gas Ships \*\*

\*\*not yet in force

#### APPROVED Information

The requirement to place 'approved' information on board each ship makes different impressions on various listeners. Those groups that may be concerned with information to the Master from time to time include Research people, Designers, Standards writers, Owners, Masters and Crew, Port Authorities, Shipbuilders, Harbor and River pilots as well as the general public and their political organizations.

The Master of the ship might assume that the 'approved' information is required only to help him load his ship. Actually, 'approved' information quite often requires more careful loading of ship because it may restrict the amount of cargo, the permissible location of the cargo, its position in the ship or the time of loading.

The Designer may assume that the standards makers are able to produce standards which will prevent capsizing in all situations or, at the least, in any situation in which the Master remains in control of his ship. He might then also presume that any stability accident to be due to the Master's failure to properly control the

ship or, failing that, must be an act of Providence. Actually the published stability standards today represent only a minimum technical level of acceptance which is set as high as possible within reasonable economic limits. They do not represent TOTAL safety (Ref 1) nor do they achieve a single similar level of safety on all ships because ships differ in size, loading conditions and operational areas.

The Administration, regulator or standards maker often either desires or is required to assume that any standard to be set must be able to fit both existing floating systems and all future floating systems of similar type. Rarely can this be fully achieved.

In addition to differences in individual perception or organizational approaches to stability information, international legal documents have different reasons for requiring stability information.

The preamble to the Safety of Life at Sea Convention tells us that Safety of Life is the objective. The preamble to the Convention on Load Lines maintains its purpose is the protection of life and property at sea. The Marine Pollution Convention mentions preserving the marine environment

Thus it must be recognized that there are many different approaches to the 'approved' stability standards.

#### 'Rapid and Simple Process'

Since those involved in stability have many different purposes, several observations can be made:

In the research efforts of examining unique stability problems, huge volumes of computation must be processed and examined. This great volume of material,

while fully necessary for the research effort, is not acceptable as the 'rapid and simple guidance' for the Master.

In the Standards making effort, the research effort must be distilled. Further, it is necessary that the standards maker must generalize the specific research on the limits of performance of a unique ship to groups of ships in order to create an acceptable general stability rule.

It has been generally accepted that the 'rapid and simple process' is necessary because the Master will use the Approved stability information, will use it for each loading, and must be able to use it quickly since the master may have a few other things to do while preparing for a voyage. Hence the need for a compact presentation of operational safety limits in the approved Information to the Master.

In order to determine how accurately Administrations have judged the Master's situation, the U S Coast Guard funded a research study of actual usage of stability booklets on real ships. The results are both interesting and thought provoking. It is hoped that readers from other nations will realize that this survey of U S shipping may have some application to other national fleets as well.

#### Part II

##### U.S.A. RESEARCH PROJECT

##### "STABILITY INFORMATION FOR MASTER"

In August 1980, the final report was published of a U S Coast Guard sponsored research project which had the objective of achieving a realistic evaluation of the actual usage of the TRIM and STABILITY (T/S) Booklet required by International Convention. Copies of this research have been provided to the IMO (IMCO) library and they can also be obtained from the National Technical Information Service (Ref 2).

The views expressed in the research report are those of the contract research team. The U S Coast Guard does not accept or reject any of the ideas proposed by the research team in the process of accepting the research report. The research report, at this time, is being used for review of current regulations to help judge whether they are effective. In time, those research findings that are considered highly significant may be accepted into regulation. The authors of this paper will offer some evaluation of the research findings in their own judgement in Part III.

The research paper has seven sections the most important of which are;  
Section 3- Effect of Training/Licensing ,  
Section 4-Review of Existing T/S Booklets  
Section 6-Recommendations.

#### SYNOPSIS

The following selected synopsis borrows from the entire research report.

##### General

The study initially points out that there is a distinct difference between the shipping and the airline industry in that the airline flight operational procedures are provided in great detail but very little explicit operational guidance is available to officers of ships. The information needed by the Masters and officers for the loading and the safe maneuvering of the vessel has several different aspects which may be viewed as separate ideas if the differences are made distinct enough. The researchers make the point that, although, at present, loading within the limits of the T/S booklet proves that the vessel is being safely operated, separate operational guidance for trim and stability at sea is also needed and it is important that this information be presented as simply as possible. As an example, one major shipping company found that different masters, each ballasting the same type of vessels in his own manner, achieved widely varying results for the ballasting of a vessel. After a review of actual operating experience, that particular company decided upon a standardization of ballast procedures which considerably improved the predictability of fuel weights and usage for the same voyage.

The researchers also observed that the actual loading calculations used by the ship's crew, were often made from the capacity plan and not from the official approved stability booklet, showing that the capacity plan had the operational information needed in a better form than the Trim and Stability Booklet. The researchers conclude that very often those persons producing the Trim and Stability Booklet have a different goal than the Master.. The designer is often more concerned that the Trim and Stability Booklet will please the Administration and is not as fully concerned that the arrangement will please the ship's crew

##### Past Stability Info

The research team noted that, in past decades, cargo stowage was often done on what could be called common sense basis. The team points out that the common sense basis was easily possible until the 1950's with the old break bulk freighters and smaller tankers of that time which had perhaps twice as many sets of triple cargo tanks as today's much larger tankers. With fewer compartments in the newer tankers and specialized ships like parcel & ULCC carriers and containerships and barge carriers which have exacting stability and special strength requirements, --- the margin for error in both strength and stability is much less than it was 20 or 30 years ago.



The team further points out that the operational requirements which have become necessary in the last decade due to high speed, deckloading, or offloading containers at many different ports, have created an extremely complex stability auditing problem for various cargoes systems.

The team then goes on to point out that the operational changes of the past two decades have brought more organizational units into the task of determining the stability calculation for each voyage. Instead of stability being done entirely by the masters, there are now terminal groups essential to cargo coordination, and surveyors and special container personnel far inland, all of whom may affect loading of the vessel.

Several times the research team made a reference to an industry observation that there is a split between the stability requirements and the strength requirements. Namely, that classification societies require strength and loading manuals while the administration requires the Trim and Stability Booklet. For international readers, it may be helpful to point out that this particular system is simply the way it is done in the United States. The Administration in each country (the Coast Guard in the United States) is the organization ultimately responsible for both strength and stability or any other aspect of safety. The fact that the administration accepts the strength evaluation done by the classification society does not mean that the Administration is not responsible for it.

#### Present T/S Booklet Usage

Table 1 shows the difference between the information needed for safety and the information needed by the master when the vessel is under way and that which is required for planning and monitoring the loading or offloading of a vessel.

Table 1

|  |
|--|
| Safety Information Needed                        |
| Meets minimum national regulations               |
| Meets international regulations                  |
| Meets designer's professional solutions to needs |
| Operational Information Needed                   |
| Seakeeping Guidance                              |
| Vessel Maneuvering Characteristics               |
| Damage Stability                                 |
| Load Planning Information                        |
| Type of Cargo                                    |
| Separation or Segregation                        |
| Weights and Centers of Gravity                   |
| Delivery Ports                                   |
| Limits of Draft, Trim, Stability and Strength.   |

Also, not all stability related tasks now are accomplished by the Master. For many types of cargo carrying vessels various stability tasks are related to other persons or groups. In recent years these tasks have been divided up so that some of them are the responsibility of the ship, some are undertaken by the shipping company and others are the responsibility of a terminal operation. The report states that Masters of containerships quite often are not given the information needed to actually control loading. Loading has already been precalculated by a land based team and the Master is sometimes required to accept the precalculated load which the land organization has prepared for him. This is a significant finding. Unless the Master is given proof, in a form he can check, that the precalculated loading does in fact meet the limits of safety required by the applicable international convention, it is not possible for the Master to be truthfully in control of the loading. If he is, in truth, not in control of the loading then someone else must be responsible to the administration. The amount of and sequence of loading becomes even more important. Container vessels often cannot compensate for more than seven to eight degrees of heel by their own ballast capacity. If the container loading is not done well enough by the terminal crew the ship may actually leave port with several degrees of uncompensated list. This reduces the amount of flexibility that the master has with fuel and ballast during the voyage. In Table 2, the research project team showed that a number of a particular type of ships sailed with various discrepancies.

Table 2  
T/S ERRORS-Actual cargo ship voyages  
(single fleet during 36 voyages)

| Problem  | # voyages | %  |
|--|-----------|----|
| 1. Sailed with excess bending moment                 | 8         | 22 |
| 2. GM higher than expected                           | 30        | 83 |
| 3. GM less " "                                       | 5         | 14 |
| 4. Departure draft calc. wrong by 3 inch or more     | 17        | 47 |
| 5. Calculated wrong Trim                             | 1         | 3  |
| 6. Calc. FORM stated more FUEL in TANK than possible | 11        | 31 |
| 7. Permanent Ballast Weight on form correct .        | 18        | 50 |

These discrepancies could have been a subject of disciplinary action by the administration or by the classification society or by the company itself. The information was treated as proprietary in the research report and has been honored as proprietary by the administration in the interest of getting a truthful report. The significance of such a table is that the terminal operation does not

often or automatically produce a loading calculation within the limits of either stability or strength. Therefore consideration ought to be given to special additional control of loading when it is done by land based organizations and such groups should be included as responsible parties. Table (3) is a summary of operational errors in loading and assessing stability within the experience noted by the research team.

TABLE 3  
CONTAINERSHIP LOADING ERRORS

Poor Quality Plans  
up to 30% not stowed per plan  
cargo loaded at sailing not on plan  
shore weights in different units  
limitations on stack weight ignored

Poor Coordination between Loading Ports  
cargo from earlier ports left on bottom  
each port group more conscious of its own stevedoreing expense than either the company need OR the SAFETY OF THE SHIP

Stability Calculations-Poor Quality  
weights in volume terms, not weight  
arithmetic errors  
LOADICATOR not correctly installed  
calculated incorrect GM and bending moment  
HYDROSTATIC properties on Capacity  
Plan different from T/S book  
Free surface used in calc had no relation to ship's condition

#### License Examinations

The researchers also note that in the process of giving examinations for a masters license, there are so many subjects covered that stability questions are only four out of 50 asked. The researchers have also examined the course materials for several academies and find that sometimes the stability material is taught in an automatic format such that the officer does not become familiar with basic assumptions and the essential concepts but relies upon a specific method which may not apply to the vessel he later assumes command of.

The researchers also found that at least one of the tests being used by the training establishments leading to a master's license had statements in it which tended to cause an experienced master to challenge the test as untrue. Therefore, the Master would not feel confident that he was learning something useful. One direct example was given by the team. The training book said that transverse inclinations greater than 20 degrees very rarely occur. In fact, some of the newer specialized vessels encounter such heavy rolling for many hours at a time. This would tend to make a first mate or Master trainee

going for a masters license disagree with the training book and lose confidence in it. The researchers conclude the section on the attitudes and training of the master and first mates by noting that some formal teaching establishments have stereotyped methods of presentation; that peer influence of one seagoing officer on another is the principle factor in shaping a mental approach (either positive or negative) to the judgement of stability and also to vessel handling. The researchers suggest that serious consideration needs to be given to new types of teaching aids such as video cassette television which could be used both in the academies and during off duty hours while at sea.

#### T/S Booklet Evaluation

Section 4 in the research report is an evaluation of existing T/S Booklets. In order to do this the researchers asked questions like--- Does the masters need justify each item in the booklet? Would more data be useful to change the stability conditions? Are there unnecessary conditions in the booklet? Can the average mate or master use the information for the operational purpose of the vessel easily? Is special training necessary needed to use the T & S Booklets?

The researchers note that often there is no Table of Contents or Index to the Trim and Stability Booklets. This could contribute to errors in usage of the approved information.

The teams comment on the lack of operational data included in the T/S Booklet, such as the highest bridge clearance, which is one of the useful operational items needed by the Master.

Additionally the researchers found that when there was a restriction in the stability booklet often the restriction was not explained to the masters and mates, in the Trim and Stability Booklets. It merely was stated that the ship can't load over certain amount etc. Thus there was no opportunity for the T & S Booklets to become a teaching function for the master and mate. It also might cause the master and mate to disregard the book depending on what they have been told regarding the need for the restriction. The researchers also noted that there is no agreement as to what the sequence of each stability calculation in the T/S Booklet should be. It is left up to each individual designer as to 'how' he will present the information to the master. Many Masters interviewed expressed no satisfaction with a minimum (GM) metacentric height presentation. They apparently do accept the converse, a maximum KG calculation, especially if it is presented in an expanded Tabular form. The KG calculation may also represent the maximum KG to be allowed in order to keep a certain area under the statical

righting arm curve. FIG 1 shows the standard T/S Index recommended by the International Association of Classification Societies. FIG 2 shows a similar index recommended by the research team.

FIG 1

CONTENTS OF STANDARDIZED T&S BOOKLET SUGGESTED  
BY THE INTERNATIONAL ASSOCIATION OF CLASSIFICATION  
SOCIETIES

Section I

A. Compulsory Information

1. Ship particulars
2. Any scaled drawing
3. The estimated total weight and center of gravity of items such as passengers and crew and their effects (per unit), vehicles, etc.
4. Tables of capacities and centers of gravity for every compartment
5. Tables or curves of the effect of free surface
6. The results of the inclining experiment, the lightship condition
7. A diagram showing the load line, corresponding freeboards, and also displacement, metric tons per cm. immersion, and deadweight, corresponding to a range of mean drafts
8. Complete hydrostatic particulars
9. Cross curves of stability

B. Conditions to be described in the Stability Manual

1. Light condition
2. Docking condition
3. The standard conditions stipulated in Item 1 of Appendix II to IMCO Resolution A. 187 (ES. IV)
4. Any other condition of loading appropriate

C. Presentation of Stability

Each standard condition of loading shall include certain details

1. Profile, plan view diagrams showing the distribution of all components of the deadweight
2. A calculation giving the displacement and positions of centers of gravity, free surface effects to be clearly indicated.
3. A summary of condition showing the mean draft, displacement, longitudinal centers of buoyancy and gravity, trimming moment, trim calculation, draft at after and forward perpendiculars and, if necessary, draft in way of draft marks.
4. A summary of initial stability including the calculation of GM with and without free surface effects; a curve of righting levers (GZ) corrected for free surface effects.

Section II.

A. Data to assist the master in evaluating corrective heeling moments based on subdivision and damage stability

B. Information, a precalculated diagram, enabling the Master to determine the stability in any condition of loading

Non-Technical items

The research team made 10 specific non-technical recommendations regarding construction of approved T/S booklets:

(1) Use an oversize page in order to place all significant data about a particular condition on one page (SIZE A-3 or 11x17)

(2) Orient all pages in same direction. Place tables, graphs, ship profile etc in same position on each page.

(3) Sums, Products, Moment Summations etc to be transcribed to another sheet should be bold type

(4) Numbers should be typed or printed but not handwritten

(5) Do not use those copying processes which fade

(6) Heading, Titles should be liberally used throughout and repeated

(7) Vessel identification should be on every page

(8) Use a binding to avoid loss of pages from frequent use.

FIG 2

RECOMMENDED TABLE OF CONTENTS OF T&S BOOKLETS

Cover

Introduction

TABLE OF CONTENTS

Table of Contents

Instructions

General Instructions and Conversion Factors

Operational Guidance

Flume Tank Data and Instructions

Cargo Space and Capacity Data

Summary of Cargo Spaces, Centers, and Weights

Plan, Profiles, and Elevation of Vessel Showing

Cargo Spaces

Consolidated Tankage Table: Capacities, Centers, Operational Effects

Cargo Tanks

Ballast Tanks

Fuel and Diesel Oil Tanks

Miscellaneous Tanks

Flume and/or Other Stabilizing Tanks

Combined Table of Hydrostatics, Deadweight, and Required GM Reduced-scale "Long" Form, with Instructions

& Supporting Worksheet

Table of Maximum Allowable Virtual Vertical Moments

Van der Ham Diagram

Draft Particulars

Draft Diagrams and Stern Profile

Draft and Freeboard Particulars for Fresh and Salt Water

Cross Curves of Stability

Instructions for Plotting Statical Stability Curve

Interpreting the Curve for Seaway Conditions

Longitudinal Stress Determination

Instructions for Assessing Longitudinal Strength

Table of Bending Moments

Appendices

A - Table of Vessel's Principal Characteristics and

Details of Effective Lightship Condition

Name, Official Number, Date and Place

of Building, Principal Dimensions

Definition of Lightship Used in Booklet

Lightship Weights and Centers of Gravity

References to Plans and Drawings

B - Sample Loading Conditions

Cargo Conditions

Suggested Light Ballast

Suggested Gale Ballast

C - Damage Stability Information

Source and Governing Criteria

(9) Use conventionalized grids and color codes to minimize search time.

(10) Graph paper when used should have only large grids strongly identified and not background grids.

Electronic Stability Aids

The contract researcher was asked to evaluate the Trim & Stability Booklet as it now exists in its approved form. That is, a paper booklet reviewed and approved by the administration. The research team was not asked to develop an approach for electronic stability measurement.

However the ships that the research team visited were quite often using various forms of electronic aids. Some were using a hand held electric calculator. Other ships had a full microcomputer on board. However, there are many other ships (mostly smaller non-scheduled ships) on which the only mathematics utilized is for payroll and purchase of fuel and stores.

Another thought expressed by the research organization was that the instructions in the Trim and Stability Booklet ought to refer to any stability aids on the vessels (i.e. graphs, hand held calculators, mini-computers, models, etc).

#### Tables, Curves or Graphs

One interesting comment by the research team is that the graph format (for curves of form displ etc) which has been popular for many years is now being either supplemented by tables or actually displaced in some booklets by tables which can be easily interpolated with the hand held calculator.

In discussing graphic versus tabular presentation the research team noted that either or both may be appropriate depending upon the usage of the booklet. The graph vividly illustrates the effect of the shape of the vessel on all of the main particulars while the tabular presentation is faster to use from an operational standpoint.

The research team also recommends that if tables are used, the draft interval should be approximately 1 quarter of a foot or approximately 3 inches. In the metric system this could be either 75 or 100 millimeters. It is noted that the conventional table is sometimes considered inaccurate by the operational seafarer because the longitudinal center of flotation and the moment to trim 1 inch changes so much with trim on the newer geometric forms.

The researcher notes that the instructions for the use of graphs ought to be either on the graph or table or on the facing page. In commenting on the use of the 'required GM curve' or graph the research team noted that where a single curve has been acceptable and would cover all the requirements, now a family of curves is becoming more common since the cargo varies to a great extent. Cargo loading heights and centers of gravity and also ballasting are much larger operations than 20 years ago.

The research team noted the desirability of labeling the safe area and the unsafe area on the two sides of the required curve. The team points out that graphs with a curve that have simple and prominent instructions printed right on the same page have a high degree of acceptance with operational people. The research team also noted a weakness of today's T&S booklets is that the tank capacity tables usually do not have centers of gravity of partially filled tanks. For operational use, it is recommended that every compartment available for the carrying variable items such as liquid cargo or fuels, stores, water or ballast should have a complete table of capacities including the center of gravity at partially loaded conditions.

#### Sample Loading Calculations

Another major topic is the group of sample loading conditions. Here the researchers note that there have been a great increase in the number of sample loading conditions. The team notes that although this may be helpful to some officers, it maybe an impediment to officers with weak stability backgrounds who may choose to assume one loading condition is close enough when in fact they should be calculating a completely new stability calculation. Of course there can be little argument that if the sample loading condition does in fact reflect the precise loading condition, then the Master has no need to go through a recalculation.

The research team takes exception to the practice of some naval architects making note that loading conditions meet or exceed the stability requirements of some national or international regulations without stating those requirements, because the reader may have no knowledge of what those requirements are. The research team prefers that the information be self explanatory.

One of the more important findings of the research team is that on board ship many mates and masters create their own operational form in order to calculate either loading, strength or stability.

#### Special Forms

Some of the special individual forms that were noted by the research team include a table of hydrostatic properties with differences between consecutive entries noted to assist in interpolation and a single voyage summary form with clearly marked spaces for stability in both the departure and arrival conditions which enables the operational seafarer to identify the most crucial point of the voyage immediately. Space for calculated and observed drafts, vessel name, date and voyage, origin and destination should also be provided.

The sample loading conditions are also recognized as having a teaching value.

The research team notes that the GM criteria for the United States is much more conservative than the GM required by other nations. Finally the team recommends the statical stability curve approach as more likely to cover the most frequently encountered perils.

#### Visits and Interviews

In the section of the research report which covered the visits and interviews with actual officers, several interesting ideas were brought to light. Among these are the understanding that dry cargo ship officers tend to be more familiar with a

stability concept since they have a varied cargo while tanker officers tend to be more familiar with the strength or loading calculations analysis because strength is the major concern for liquid carrying vessels (because of their size). The research team noted that many of the seafarers responding to the discussions at the academies and industry training schools were not satisfied with the basic format of existing trim and stability booklets. One item of dissatisfaction centered around the fact that the material was often presented with simplifying assumptions in order to make it simple and easy to use but often those simplifying assumptions were not completely appropriate for the vessel concerned because of the variation in cargo. A second major point of disappointment of the seafaring officers was insufficient guidance in basic principles which would have allowed them to understand better what the simplifying assumptions meant and what the limits of those assumptions might be. Overall, dissatisfaction tended to create a distrust of the booklet.

In discussing what sort of items the seafaring officers would like to see in a revised trim and stability booklet the research team recorded the three items gaining the most favor among the interviewed ships officers; (1) that the format of the trim and stability booklets needed simplification, (2) Tabular information was desired where possible rather than the graphical form, (3) the cargo capacity and centers of gravity should be supplied for intermediate levels of capacity.

#### RECOMMENDATIONS-by team

Perhaps the most important section in the research on trim and stability booklets is the recommendation section. It has a number of ideas which are worthy of consideration, although not every one is automatically the best thing to do. The recommendations section of the report reminds us that the reason for the trim and stability booklet authorized and approved by the administration is its safety function and states that the safety function requires instruction and data to develop safe operation sequences which comply with both inspection and load line requirements. The information booklet should have methods to enable a master to check ship's stability just prior to the voyage and at any time during the voyage to allow a change in stability when needed. The booklet must contain all the governing information. If it does not contain all the governing information then it has not covered safety. The governing information may include longitudinal strength information even if that is also in a classification society loading manual.

As a second objective of trim and stability booklets, the research team mentions assisting in the training of the new officers or to increase the efficiency of vessel operations. Here the researchers state that the emphasis should be on loading options with full instructions. It notes that the present form of sample conditions contains some of this information but does not really satisfy the seagoing officers need for specific guidance.

The researchers then state that if additional information is needed because of a special trade or unusual cargo it should be in a different location, especially if it is a large amount of information.

The researchers also mention that whatever stability related material is included in the T & S book should be indexed. The researchers state that the evidence from the many vessels they reviewed would allow a single trim and stability format for all commercial hull forms. Therefore, there is support for a standard T & S booklet. An Approved list of pages of information is recommended. The study recommends that the trim and stability booklet can be both shortened and reorganized. It proposes the use of more graphical and tabular techniques

such as a special diagram (FIG 3) for trim and tables of maximum vertical moment in order to help the operational as well as the safety side. Of course, the information given in the operational portion of a T & S Booklet would have to lead to a result which would be within the safety limits approved by the administration.

#### Electronic Aids

The researchers then make the statement that nearly all masters and mates were using some kind of electronic or electric aid. The discussions on electronic aids had many interesting facets. Both analog and digital loading computers were discussed. Ship officers would like to see the input preserved so that input errors can be checked. They would like to have a facility for conversion of cargo weights and volumes and input ability for the density of dock water so that they could estimate the change of draft easily. There were some that wanted to see the bending moment and shear force and stability curves actually displayed which would of course require a sophisticated electronic system.

The cargo space and capacity data is the thing most on the mind of the master and mates. If the cargo space and capacity data is not so complete that it can be used for operational usage they tend to lose confidence in the entire trim and stability booklet. The researcher recommends that a separate non approved, pocket size guide, with very

Change in Draft with 4 Tons Loaded  
at Any Distance from  $\text{C}$ , in Inches

h



Similar comments about tankage for liquid cargo vessels are made.. The researchers report that in their workshops with masters and mates, 92% of the masters and mates favored tabular presentation rather than graphic. Masters prefer to interpolate from a table provided that the interpolation steps are small enough to minimize errors.

A Recommendation is made for a maximum vertical center of gravity chart adjusted for free surface.

COMBINED HYDROSTATICS AND REQUIRED GM  
TABULATION

[illegible]

Although wave dynamics has been mentioned before it has not yet drawn a specific recommendation. The research team did place a figure in their paper calculated as a nomogram, which would allow a ship's master to use parallel rulers to find Zones of Heavy Rolling.

### PART III-REVIEW AND COMMENTS

When trying to evaluate the many observations of the research team, it becomes evident that the official Information to the Master is only one of several expressions of stability information that is necessary to achieve one designated level of safety. It is required that the stability in each condition be within safe limits. However, the exact limits of stability are not specified in any international convention to date. Even the TORREMOLINOS Convention (Ref 3) sets a base criterion upon which the designer must build to create a safe ship. It is also evident that the required information must be in a format the Master can use.

#### T/S Booklet Accuracy

There are several problems with the actual methods in use to provide the data necessary to implement the above agreements.

First, and most important there has been discovered the fact that some nations do not check their ships to ascertain that proper stability (and other) information which has been agreed to be on board actually has been provided. As a result, one of the delegations to the IMCO 'SubCommittee on Subdivision Stability and Load Lines petitioned the SubCommittee to make the requirements in SOLAS 74 (Reg 19) much more definitive. The SubCommittee agreed and in April 1982 the Maritime Safety Committee agreed.

A second problem has been caused by the need to be complete in the information given the Master. The complexity of modern shipping practice in some areas of the world is such that many different cargoes are carried in many different configurations.

In addition to the Official Approved Information, there appears to be the following needs for stability information, some of which utilizes the same information as the Approved Booklet;

a-On large ships, the stability information needs to be coordinated with strength information for loading manuals so that the Master does not have to work two different sets of information.

b-When damage stability limits are in effect, the information should be coordinated with stability and strength so that the Master does not have to work three different sets of information simultaneously.

c-When pollution limits are in effect, the information should be coordinated so that the Master does not have to work four simultaneous sets of information.

d-There may be limitations on trim or heel set by the designer in order to achieve desired speed or other operational qualities.

e-There may be limitations on maneuvering such as-a designated maximum trim by the bow or the influence of wind on the turning of a high-sided ship

f-The Master and Mates need modified stability information for controlling the seakeeping qualities of the ship during the voyage.

g-Some companies have cargo management groups at several ports in order to assist in the distribution and flow of cargo. The information needed by this group is primarily concerned with proper service to customers-preventing overcarriage of cargo, assuring timely delivery of cargo, etc.

h-The Master and Mates need stability information for training -for the particular ship and for their general ship knowledge.

#### Interaction by

##### Designer/Owner/Admin/Master

There are at least 7 interrelationship areas between owner, administration, designer, Master and Port Loading Group. The following table lists these and notes the type of interaction between parties.

Table 4

| Interaction                    | Necessary for Proof                                  |
|--------------------------------|--|
| Designer/Owner                 | (1) Deadweight & Volumes                             |
| Designer/Admini-<br>stration   | (2) Curves (Stat, KG, GM                             |
| Designer/Master                | (3) ?  |
| Master/Admini-<br>stration     | (4) Sample of Blank<br>conditions                    |
| Master/Owner                   | (5) New conditions calcu-<br>lated                   |
| Master/Port                    | (6) Criticize loading<br>calculate alternate<br>plan |
| Loading Group<br>Master/Master | (7) Seaway conditions and<br>stability reserve       |

#### What Proof is Needed

(1) The Administration must be satisfied to approve the calculation (the owner requires the Designer to satisfy Administration on regulatory criteria.) Additionally the owner must be satisfied that it will carry the desired load.

(2) If special conditions or exceptional usage of the system are contemplated -- The Administration may require special conditions. Adm. must be satisfied conditions meet minimum rules so they can be "approved".

(3) In our experience, Designers seldom have the opportunity of creating an operating T/S booklet from the direct recommendations of the Master. Instead the Master must use the booklet presented by the owner for Administration approval.

(4) The Master must be able to show the Adm. that a non-specified condition is within prescribed limits of Strength & Stability and that he can accurately perform the calculation.



(5) The Master must be able to calculate all new loadings desired and show that condition satisfies or exceeds the limit.

(6) The Port Loading Group must satisfy the Master that all necessary safety parameters have been met.

(7) The T/S Booklet must include enough information that, if necessary, the Master can safely modify the present condition at sea (ballasting and/or seaway orientation etc.)

In pursuing the safe operation of a vessel, the Master needs first to know that the ship is within the boundaries of stability, strength and other standards set for the particular ship. Then, while examining several alternatives for each loading condition depending on the destination of cargo, the placement of weights in the ship, the need for separation of one type of cargo from another, there is a need to maintain the vessel within prescribed limits.

Accordingly, what the Master needs is a weights and centers of gravity calculation Guide which will state in YES/NO terms whether any ship limitations have been exceeded. It must also assist the Master to correct the problem without exceeding any other parameter. It would also be helpful if the guide would explain the margin of exceedance and amount of reserve available for an emergency.

Several national delegations to the IMO SubCommittee on Stability, Load Lines and Safety of Fishing Vessels have expressed concern over the amount of material which has to be reviewed for approval. We have just recited the several criteria and noted that they are not mutually independent. In order to assure the stability/loading information remains helpful throughout the ship's life, it must be reworked by the designer when it is

found to contain significant errors, modified when some new cargoes are carried which may affect stability (or other safety parameter) and, the ship must be checked periodically during its operational life to be assured that the stability booklet on board still accurately reflects the current usage of the ship. This is especially true when a ship changes trade routes or is utilized for a new trade entirely.

Some administrations authorize a classification society to act on behalf of the administration for the purpose of approval. This is an acceptable method of reducing the routine work of approval but the administrations remain the responsible party to the several conventions. The administrations must be prepared to discuss exceptions or unique designs at I.M.O. The administrations must therefore remain fully knowledgeable concerning the limitations of all ships

and ocean vehicles approved by them and must lead the way in discussions leading to new parameters or standards (whether national or international).

In our opinion, the Master/Mate can be expected to utilize the stability (or other information) if it is in a format which helps them do their routine tasks or helps them to understand some unique occurrence in the life of the ship.

On the other hand, if the information on board requires them to move about the ship gathering information each time it is desired to use the booklet, they probably will not use the T/S information. Only in cases of hardship (stranding etc) will they attempt to investigate each tank (ullages) or hold.

The stability/strength/seaworthiness information prepared for the Master should assist him by:

a. guidance with loading of stores and cargo.

b. guidance for seaworthiness during voyage

c. guidance for offloading/reloading cargo at one or more destination ports

d. guidance for teaching mates the stability process and qualities of the ship.

Although the Master does not want to be given unsafe information nor any information which will permit him to take the ship beyond the limits of design, still he can only use information which is directly related to loading, maneuvering in harbor, high seas voyaging & seakeeping, and offloading (including restowing cargo when necessary). At sea, he cannot use the righting arm/moment curve and know that a specific reserve of stability exists.

Accordingly the information must be prepared so that the limits of operation are included in the loading conditions shown. If a limit is exceeded, the Master needs to be able to spot the excess value immediately.

In a sense this same criticism of T/S booklets is included in the research team's complaint that today's approved booklets do not explain why there is a restriction on the vessel. The team felt that if explanations were provided, the ship's officers would show more confidence in the standard and would begin to show greater understanding of the stability reserve for each condition.

While the authors of this paper recognize that explanations for any restrictions are highly desirable, from 2 points of view - namely training and acceptability of the T & S Booklets, still it must be recognized that explanations might tend to be rather extensive lessons in naval architecture which would not fulfill the need to streamline the stability information.



## Teaching & Relearning Stability at Sea

The preceeding discussion leads to a discussion of the need to teach stability in maritime schools, in the officers license examinations and in refresher material given at sea.

The fact that stability is only one of dozens of subjects which the prospective Master must learn, makes it difficult to explain the full variety of stability lessons in the short time usually allocated to stability.

The Master must remain able to prove the stability calculation for his vessel at any time but the necessity of performing dozens of other tasks makes it unlikely that he will remain an expert in stability.

Thus it would seem a better approach to teach correctional stability rather than theoretical stability from the outset of Deck Officer training and to emphasize phases of operational stability refresher training after the deck officers have left formal training facilities and are actually engaged in daily ship control and administration at sea.

### Standard T/S Index

Regarding the sequence of items to be included in the T/S Booklet and the total list of items to be included in the T/S Booklet - we refer to Fig 1 which represents the index list of the International Association of Classification Societies (IACS). With modifications in two items the recommendations of IACS seem well suited to provide the operational stability information many Masters have requested.

The exceptions are:(referring to Fig 1 Section I)

A.4 and 5 -- Tables of capacity, vcq, and free surface should include a complete range of loading in each tank.

A.8 Hydrostatic particulars should be in tabular form based on draught range.

Also it should be noted that the item B 4 - "Any other condition of loading appropriate" is an important consideration. It does not limit extra conditions. There may be none or hundreds of conditions. Referring to Item B-3, the standard conditions of loading are only one each homogeneously loaded and ballast voyages at departure and arrival. This places the absolute minimum number of loading conditions at four. There are often many other voyages for which the ship must be considered separately.

Section II of the IACS document is important in that it recommends information on damage & flooding situations and secondly because it asks for a precalculated diagram, which will enable the Master to determine stability in any loading condition. The latter is a good objective but the authors of this

paper do not know of any single stability diagram which enables the master to determine stability in any loading condition. That is, normally some calculations must be done for the non-standard conditions. Additionally, stability at any instant of a voyage must be approached by concentrating efforts of the Master on maintaining the Center of Gravity of the entire marine system within acceptable limits, recognizing that the Center of Bouyancy constantly moves depending on the seaway. The righting forces vary every moment depending on the exact nature of the seaway influencing the ship.

At present the approach nearest to this objective combining all presently known loading conditions and any future loading condition would be a combined righting curve such as Fig. 5. With the approved minimum shown as a hatched area and all known approved loading conditions shown as overlays, such a presentation would benefit interactions between owner/designer and designer/administration but it would not necessarily be the quickest check for the Master.

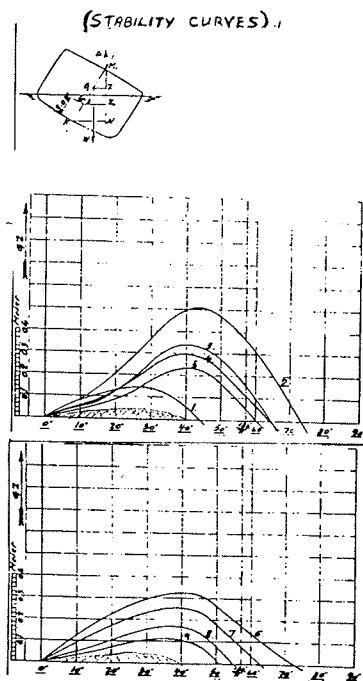


FIG 5

The quickest check for the Master prior to departure would be a simple KG curve, Fig 6, or perhaps a table as per Fig. 7. The table would show only the lessor of the allowable maximum KG as per the shaded lines on Figs. 5 & 6. The table would be the simplest form for the Master (crew) to use while loading and reloading. It has the advantage of simplicity and ease of transposition of numbers. However, the graph is the better training presentation since it can show a summary of a family of required maximum KG's.

FIG 7  
MAXIMUM ALLOWABLE VCG for:  
Minimum Stability  
Damage Survivability \*  
Pollution Reduction \*  
Other Standards \*  
(\* if applicable)

| DISPL.<br>tonnes | VCG<br>m. | DISPL<br>tonnes | VCG<br>m. |
|------------------|-----------|-----------------|-----------|
| 20,000           | 8.---     | 13,000          | 8.---     |

NOTE: If the final VCG(after free surface correction) is not greater than the applicable VCG, the vessel can be accepted as 'in compliance' with the standards set forth on page \_\_\_ of this booklet.

#### Seaway Stability

The authors of this paper do not support a nomograph for operational seaway ship control information for the reason that it can so easily be misread. It depends very heavily for accuracy on an exact calculation of ship's natural roll period which changes, sometimes significantly, with modest changes in loading or ballasting. A tabular approach that showed the master enough to avoid certain conditions with speed or course changes would seem to be a better approach.

#### Non-Technical Items

The authors of this paper accept most of the non-tech recommendations of the research team. However we do not feel that full binding (No. 8) is the best idea although we agree that mere stapling is a poor method which can lead to loss of end pages. There are newer forms of binding such as spiral plastic which allows the book to lie flat at any page, which could be a better approach.

#### Electronic Stability Aids

Coast Guard in U.S.A. has maintained a neutral policy thus far regarding electronic aids for either stability or strength calculations. No ship is required to be provided with an electronic calculator.

#### MAXIMUM ALLOWABLE KG VS. FREEBOARD

(Freeboard is measured at Fr. 23 from waterline to bridge deck)

| Loaded Condition        | Freeboard Measured from Bridge D'k Fr. 23 | Draft (M'd) at Fr. 23 | Displ in S.W. Tons | Actual K.G. above B | Maximum K.G. Permissible Above B |
|-------------------------|---|-----------------------|--------------------|---------------------|----------------------------------|
| Light Ship              | 11.98'                                    | 8.25'                 | 223                | 10.36'              | 11.93'                           |
| Arrival at Fishing Grds | 10.28'                                    | 9.95'                 | 302                | 9.96'               | 11.65'                           |
| Ready for Sea           | 10.13'                                    | 10.10'                | 308                | 9.97'               | 11.90'                           |
| Return to Port          | 9.71'                                     | 10.52'                | 330                | 9.74'               | 11.65'                           |
| Ready to Return         | 9.63'                                     | 10.60'                | 335                | 9.80'               | 11.53'                           |

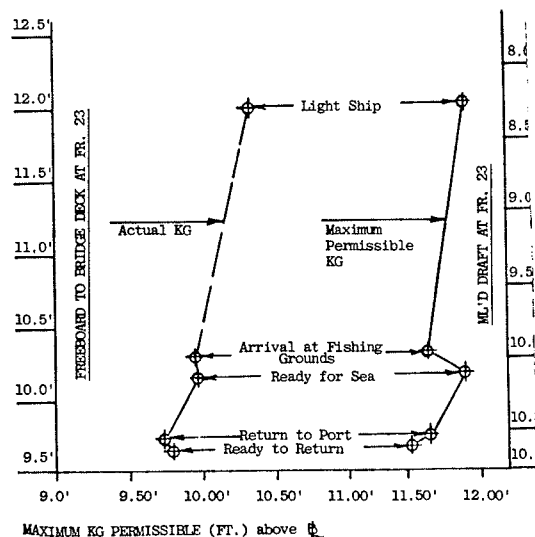


FIG 6

However, when a ship has been provided with a special calculator at the option of the owner or Master, Coast Guard has required that the specifics of the calculator be made known to CG technical and that the CG technical personnel check the electronic calculator using information from the approved T & S Booklet in order to prove that the on board calculator or computer will provide the same result as would be reached by using the approved information booklet. There have been actual cases in which electronic calculators/micro-computers placed on board were in error and resulted in unknown overstressing which caused structural defects.

One item often used as a selling point for the electronic box is the claim that stability (i.e. GM) can be constantly and instantly recalculated while at sea. This is only possible in those electronic machines that have a roll period memory with a storage capability for over 200 rolls while making a simultaneous summation of roll periods. If the electronic GM calculator does not use roll period to estimate gyradius (but assumes a constant gyradius) the GM calculation will constantly be in error by an unknown amount.

Since several of the problems of the current stability information system are connected with the speed of information retrieval; the need to minimize calculation; and the need to make quick adjustments in the stability summation condition, it is obvious that an electronic system properly installed could answer these problems.

In order to assure a proper electronics stability calculation, the ship designer and the Master need to have similar but not identical inputs and/or interface with the machine. For instance:

(a) The ship Designer needs to prove to the Administration that the ship or ocean vehicle meets the minimum applicable standards while the Master only needs to know that his ship is within specified applicable

limits with any proposed loading.

(b) The Master also needs to know that any variation in loading will still be within acceptable limits.

(c) The Master further needs to know what adjustments to the loading condition can be made for seaway/weather conditions.

(d) Ultimately the Master needs to know what weather/seaway condition should be cause for unusual action on his part to save lives/ship/cargo etc.

#### Summary of Discussion

In order for the Information to the Master to be effective it must:

\*prove that all known conditions meet the applicable Standards;

\*be willingly used by Masters to check loading conditions;

\*be able to prove a new (proposed) loading condition is within applicable standards.

\*show the reserve of intact stability in seaways of various magnitude or actually state limiting seaways. Additionally, it is desirable that the booklet:

\*show the reserve of stability in each loading condition.

\*enable the crew to review it during voyages in order to gain familiarity and confidence with it's use.

It would seem desirable and necessary that the Approved Trim & Stability Booklet should have a page, just after the Index page, which will set forth the limitations of all national or international regulations, codes etc applicable to the particular ship. Some of the limitations in these documents are:

|                  |                    |
|------------------|--------------------|
| stability        | strength           |
| trim & heel      | Cargo density      |
| cargo volume     | cargo placement    |
| segregated cargo | segregated ballast |
| visibility       | maneuvering        |
| local strength   | damage stability   |
| subdivision      | damage strength    |

#### NEW T/S BOOKLET FORMAT

The authors feel that most of the recommendations for improvement in the research paper and most of the objectives of this paper can be fulfilled by adopting a special format for the Approved Information for the Master. The Index for this proposed new format is set out in Table 5.

#### RECOMMENDED T/S BOOKLET INDEX

TABLE 5

#### SECTION I INTRODUCTION

Table of Contents

General Ship Characteristics

Standards of Design and Operation

Owner requirements

Classification requirements

National requirements

International requirements

Booklet Instructions

(step by step instructions on the use of forms, tables, graphs, etc. in the booklet)

#### SECTION II OPERATIONS

Tankage and capacity data

Access data

(drains, discharges, manholes, etc.)

Trim Table

Hydrostatics Table

VCG limit Table

VCG calculation Form

Seaway/Wave guidance Table-(optional)

#### SECTION III DESIGN APPROVAL

Cross Curves of Stability

Righting Curves (R. Arm or K. Moment)

Minimum Stability Plot which

meets all restrictions

plus all precalculated

condition curves

Show reserve of stability

in all loading conditions

All pre-calculated loading conditions

Tabular loading Conditions

shown in detail

Longitudinal strength limitation

Damage Stability /Flooding Info

Pollution Info(if necessary)

Other desired information

The NEW format should assist the Master to do his job properly, and should gain acceptance of more seagoing Masters by gathering in one compact section of the book those pages needed for operational stability decisions. All presentations should be TABULAR. Administrations could give

permission for small pocket-sized copies of this section to be provided to each officer for daily usage.

The DESIGN section should retained the more traditional pages and format because of its importance to the approval process between the naval architect and the Administration. The design section should also contain all explanatory notes on Stability restrictions or limitations on loading.

Appendices should be used to such extra items as Stability Test information and refresher training for deck officers while at sea.

## CONCLUSIONS

Conclusions can be drawn in several areas:

1. Perhaps the most immediate and far reaching conclusion that should be discussed is the observation that the Information to the Master required by so many international documents and reaffirmed by the IMO just this year is not achieving the desired goal. The principal reason appears to be difficulty in using the provided information during actual operations.
2. The Master often does not retain the stability expertise throughout his career that he had when just graduating from training school.
3. Despite the limited operational acceptance of such information, there is both theoretical and actual evidence that such information is more necessary now than ever before. The complex marine systems of the 1980's require that Information to the Master be more specific, more detailed, and more quickly utilized. The last requirement is due to the greater speed and vessel turnaround.
4. The operator needs the same overall stability information that the Administration needs but he needs it in a different format.
5. Electronic calculators can assume a major role in providing Information to the Master IF they are truly representative of the ship loading conditions, ships form, and ships girder. The speed of operation, storage capacity, and ease of data manipulation make it possible to give the Master immediate information. However administrations, companies and masters must not be misled into believing that electronic information is automatically valid information. Erroneous information by electronic loading calculators was suspected in at least one large vessel needing structural repairs.

6. There may be as many as 8 interconnected areas of limitation in the preparation of information to the Master. These possibilities are:

- (1) Stability
- (2) Strength
- (3) Trim/Heel
- (4) Maneuvering
- (5) Damage & Survivability
- (6) Pollution
- (7) Seakeeping
- (8) Port Cargo Management

Since any variation in loading may violate the limits of one or more of these subjects all applicable rules must be considered by the design naval architect as the T/S booklet is being completed.

An optional ninth functional area for Information to the Master is that of crew training.

## RECOMMENDATIONS

1. IMO (IMCO) should agree on a minimum index of information to be included in the information required by Regulation 10 of ICLL '66 and Reg 19, Chap. II, SOLAS.
2. An Operational guide should be developed with the approved stability information with a simple tabular format. (see REC #3 and Table 5)
3. A new PROPOSED Table of Contents is listed in Table 5. The NEW feature is the separation into OPERATIONAL and DESIGN Approval sections. The special section on operations is for the exclusive use of the Master. It should be in TABULAR form. It has only 3 or 4 pages. It is separate and specifically identified so that COPIES can be made or it can be used as a separate appendix if the ships officers so desire. It is and should remain part of the approved information in order to avoid disseminating improper information.
4. Bearing in mind that the worklist for the IMO technical subcommittees is already overcrowded so that several levels of priority have been utilized in deciding what to work on, the naval architectural professional societies should recognize that the Master needs improved information and should move toward providing it without the impetus of international legal incentives..
5. Professional naval architects and the professional societies which represent them should take the lead in developing a universal format for on-board stability information. Direct communication between ship masters and naval architects is strongly recommended as the best approach.
6. IMC should also agree on recommended stability refresher training material to assist newer ship's officers to learn the stability limits of the ship during voyage time.

7. Dynamic Wave Stability limitations should be introduced into the Information to the Master as soon as possible to assist the crew in avoiding storm damage or loss. At present, seaway roll information is available only for single hull ships with gyradius of roll based on homogeneously stowed internal cargo. Such information can be developed for other ships and marine systems with the aid of model tests.

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## INTACT STABILITY OF SHIPS IN FOLLOWING WAVES

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ABSTRACT

This paper is based on results of scientific-technical co-operation between DDR-Schiffs-Revision und -Klassifikation (DSRK) and Polish Register of Shipping (PRS)[1].

In the paper a method of calculation of the intact stability of ships in following waves is presented.

In the method it is assumed that the ship is running in following irregular waves. The waves are replaced by an effective wave. This wave has a height which is so chosen that it causes a similar change of stability as the real waves themselves. The height is reduced taking into account the speed of the ship and the time of capsizing.

The stability in following waves is evaluated according to a wind criterion and criteria for the righting arm curve.

Calculation results for five ships of different types are shown.

The presented method is intended for additional checking the intact stability of ships and applies especially to small ships of unrestricted service.

NOMENCLATURE

|             |       |   |
|-------------|-------|---|
| B           | m     | breadth of the ship                           |
| c           |       | constant for calculation of time of capsizing |
| D           | m     | depth of the ship                             |
| $d^2 \xi_E$ | $m^2$ | variance of the amplitude of the effective    |

|                      |         |  |
|----------------------|---------|--|
|                      |         | wave   |
| $d^2 \xi_E$          | $m^2$   | normalized variance of the amplitude of the effective wave   |
| $d^2 \xi_E \Delta T$ | $m^2$   | normalized variance of the average amplitude of the effective wave   |
| $F_n$                |         | Froude number  |
| GM                   | m       | initial metacentric height   |
| $GZ_{max}$           | m       | maximum statical righting arm  |
| g                    | $m/s^2$ | acceleration due to gravity  |
| $H_E$                | m       | height of the effective wave with 3 % probability of excess corresponding to wind force 8 <sup>0</sup> B in the Atlantic Ocean |
| $H_E 3\%$            | m       | height of the effective wave with 3 % probability of excess  |
| $H_E \Delta T 3\%$   | m       | average height of the effective wave with 3 % probability of excess  |
| $h_V$                | m       | lever arm of the heeling moment due to wind pressure   |
| KG                   | m       | height of centre of gravity above base line  |
| L                    | m       | length of the ship between perpendiculars  |
| $S_{\xi_E}(\omega)$  | $m^2s$  | spectral density function of the effective wave  |

|                      |          |   |
|----------------------|----------|---|
| $S_{\xi\xi}(\omega)$ | $m^2s$   | spectral density function of irregular sea                        |
| $T'$                 | s        | characteristic wave period  |
| $T_L$                |          | non-dimensional characteristic wave period                        |
| $T_{\phi_0}$         | s        | natural rolling period  |
| $\Delta T$           | s        | time of capsizing   |
| $t$                  | s        | time  |
| $v$                  | m/s      | speed of the ship   |
| $\Delta$             | t        | displacement of the ship  |
| $\xi(x_0, t)$        | m        | irregular wave profile  |
| $\xi_{AE}$           | m        | amplitude of the effective wave                                   |
| $\xi_{AE \Delta T}$  | m        | amplitude of the effective wave averaged in the time of capsizing |
| $\bar{\xi}_w$        | 1/3 m    | significant wave height   |
| $\theta_v$           | o        | angle of vanishing stability                                      |
| $x$                  |          | coefficient for calculation of the average                        |
|                      |          | height of the effective wave                                      |
| $\omega$             | $s^{-1}$ | circle frequency  |

## 1. INTRODUCTION

### 1.1 The capsizing of ships in following waves

The analysis of intact stability casualty records shows that most of the casualties occur in a rough sea [2]. More than a half of these casualties occur in following ( $\sim 28\%$ ) and quartering ( $\sim 22\%$ ) wave conditions [3]. These conditions are especially dangerous when a ship is additionally exposed to the action of heeling moments due to beam wind, water on deck, shifted cargo etc. This is because following or quartering waves can strongly affect the ship's stability, especially when the wave's length is approximately equal to the ship's length. In this case the maximum increase of righting arms for the ship in the wave trough and the maximum decrease of these arms for the ship on the wave crest can be observed.

Capsizing experiments with models in irregular following and quartering waves conditions [4] show that the decrease of stability by the waves plays a very important role in nearly all cases of capsizing. Furthermore, these experiments show that it is possible to distinguish three different modes of capsizing:

- low cycle resonance,
- pure loss of stability,
- broaching.

In this paper the pure loss of stability will be dealt with. Pure loss of stability usually occurs in a following sea at high speed. The ship encounters one or more very steep and high waves of about the same length as the length of the ship moving with a wave phase velocity nearly equal to the ship's speed. The ship may remain almost stationary relative to the crest for a period sufficient for capsizing. This, of course, needs a high speed of the ship. From results of experiments it appears that a speed between the group velocity (half of phase velocity) and phase velocity can result in this mode of capsizing.

Statistics of intact stability casualties show that the danger of capsizing increases when the length of the ship decreases and the Froude number

increases. That means that capsizing due to pure loss of stability occurs more frequently than the other two modes.

### 1.2 Stability criteria in following waves

The establishment of stability criteria is a task which can be solved only approximately. The oldest stability criteria are based on minimum requirements for the parameters of statical or dynamical righting arms. Minimum values were achieved by analysis of casualty records and comparison of the righting arms of ships which capsized with the righting arms of ships which were found safe enough in service. Another approach is based on the calculation of the heeling arms in comparison with the righting arms. This leads to the well known different weather criteria. However, the statistics of intact stability casualties show that some ships, which fulfilled the requirements of the above mentioned criteria, capsized in following sea conditions. This was the main reason for trying to establish some stability criteria concerning ships operating in following sea conditions.

### 2. Righting arm curve of the ship in following regular waves

#### 2.1 General

The righting arms of the ship in following regular waves are calculated by a quasi-statical method, although the stability in a seaway is a dynamical

problem. However, the available knowledge on the dynamical behaviour of a ship in a seaway is not sufficient for laying down a method for calculation and valuation of the safety against capsizing which is applicable in practice. Furthermore, the quasi-static approach ensures a reasonable volume of calculation.

The stability of the ship in following waves is the least one, if the wave length is equal to the length of the ship and a wave crest is in the middle of the length of the ship. In this case the stability is always less than in smooth water. These facts are known from experience and have been confirmed by calculations.

The wave height is of substantial importance for the decrease of the stability of the ship on the wave crest compared to the stability in smooth water. The higher the ratio of wave height to wave length, that means the higher the wave height for a constant wave length, the less the stability

will be. The initial stability, the maximum righting arm, the angle of vanishing stability and the area under the righting arm curve become smaller. Furthermore, the calculations, by which this have been proved, have shown that the decrease of the righting arms is approximately linear for the increasing wave height. The linear dependence means that the righting arms also for other wave heights can be calculated very easy, if the righting arms in smooth water and for a certain wave height are known.

## 2.2 Influence of oscillatory motions

For investigation of the influence of oscillatory motions on stability in a seaway, pitching and heaving are especially of interest. However, at present no suitable results from theoretical investigations or model tests are available. Therefore, it is assumed that a ship in following waves pitches and heaves in such a way, that there is always equilibrium between buoyancy and weight forces of the ship when the ship is overrun by the waves. From this assumption follows that the influence of oscillatory motions may be neglected.

## 2.3 Influence of ship's speed

When a ship is under way, its stability changes due to dynamical effects. Available test results show different tendencies. In general an improvement of stability is to be noticed for increasing speed in case of ships in smooth water. For normal types of ships

and speeds, the righting arms increase up to 15 %. The increase depends on the ratios of main dimensions and on the form of the hull. At present no method of calculation is available. The influence of the speed can be determined only by model tests. The question, whether results gained from tests in smooth water can be transferred to the ship on the wave crest, is also not yet answered.

Contrary to that also results from model tests are known which show that in case of increasing speed the stability decreases at first, reaches a minimum at a certain Froude's number, increases again after that and becomes higher than the stability of the ship making no way. The minimum is at Froude's numbers between 0,30 and 0,35. Here the great influence of the bow form and the form of the bow wave system involved by it is to be seen clearly.

Because of the missing method of calculation and the not very great change of the righting arms (mostly towards to the safe side), the influence

of the ship's speed is neglected. However, the ship's speed is taken into account in the calculation of the average effective wave height.

## 2.4 Smith's effect

When calculating the buoyancy of the ship, a hydrostatic pressure distribution in the wave is assumed. The decrease of pressure in the wave crest and the increase of pressure in the wave trough from the orbital motion of the water particles are neglected. It is actually possible to take into account these pressure changes, but the calculation will become much more voluminous by that. It is known that the mistake arising from the neglect is not very great and lies at the safe side.

## 2.5 Practical method of calculation

It is assumed for the calculation that the seaway complies with the following conditions:

- .1 The seaway is two-dimensional and irregular.
- .2 The seaway is replaced by an effective wave having the following form and dimensions:
  - .2.1 The wave is sinusoidal.
  - .2.2 The wave length is equal to the length  $L_{pp}$  of the ship between perpendiculars.
  - .2.3 The wave height is equal to the height given in Section 3.



For the calculation of the righting arms the following is assumed:

- .3 The wave crest is in the middle of the length of the ship between perpendiculars.
- .4 The pitching and heaving oscillations of the ship are neglected.
- .5 The influence of the ship's speed on the pressure distribution is neglected.
- .6 The pressure distribution in the wave is hydrostatically (neglect of Smith's effect).
- .7 The waves are not disturbed by the ship (neglect of disturbance of the water particle motion in the wave).

### 3. The concept of the average effective wave for checking the ship's stability in following irregular waves

#### 3.1 The effective wave as a random function and its characteristics

Considering the stability of a ship running in a following irregular sea it is assumed that the irregular sea is two-dimensional and stationary with a Gauss distribution of the wave ordinates and a Rayleigh distribution of the wave heights. Such a random sea can be characterized fully by a spectral density function indicating the distribution of energy in the infinite number of component waves. It is also assumed that the spectral density function is given by the following expression recommended by ITTC:

$$S_{\xi\xi}(\omega) = \frac{A}{\omega^5} \exp\left(-\frac{B}{\omega^4}\right) \text{ m}^2 \text{ s} \quad (1)$$

The constants A and B are given as follows:

$$A = \frac{173 \bar{\xi}_w^{1/3}}{\bar{T}^{1.4}} \text{ m}^2 \text{ s}^4 \quad (2)$$

$$B = \frac{691}{\bar{T}^{1.4}} \text{ s}^{-4} \quad (3)$$

If it is assumed that the horizontal co-ordinate  $x_0$  of the absolute system  $O_0x_0y_0z_0$  with its origin taken at the still water level is positive in the direction of wave propagation and the vertical co-ordinate  $z_0$  is positive downward, the surface irregular wave profile can be written as follows:

$$\begin{aligned} \xi(x_0, t) &= \int_0^\infty \cos\left[\omega t - \frac{\omega^2}{g} x_0 - \varepsilon(\omega)\right] \times \\ &\quad \times \sqrt{2 S_{\xi\xi}(\omega)} d\omega \\ &\approx \sum_{n=1}^N \cos\left[\omega_n t - \frac{\omega_n^2}{g} x_0 - \varepsilon_n\right] \times \\ &\quad \times \sqrt{2 S_{\xi\xi}(\omega_n) \Delta\omega_n} \quad \text{m} \quad (4) \end{aligned}$$

When the ship is running in the same direction as the wave, it heaves and pitches and the wave affects its stability. These phenomena are random as the sea itself. It is, of course, always possible to calculate the righting arms of the ship for its different locations with regard to the irregular wave profile and to find some statistical characteristics of so indicated random righting arms. But this needs a lot of work and can't be used in practice. The best expedient from this situation is Grim's theory [5] which introduces the so-called "effective wave". This is a hypothetical regular wave, sinusoidal in form, of a length equal to the ship's length, the crest (or trough) of which is situated amidships and which has a random amplitude. The amplitude of the effective wave is determined from the condition that the change of the righting arms caused by this wave and by the irregular waves is similar. Grim has assumed that this condition is fulfilled when the integral, taken over the ship's length, of the square of the difference between the ordinates of the real and the effective waves reaches the minimum.

From this the following formula for the amplitude of the effective wave was derived:

$$\begin{aligned} \xi_{AE} &= \sum_{n=1}^N \cos\left[\omega_n t - \frac{\omega_n^2}{g} x_0 + \varepsilon_n\right] \times \\ &\quad \times \sqrt{2 S_{\xi E}(\omega_n) \Delta\omega_n} \quad \text{m} \quad (5) \end{aligned}$$

The spectral density function of the effective wave is given by the following formula:

$$S_{\xi E}(\omega) = S_{\xi \xi}(\omega) \times \left[ \frac{\frac{\omega^2 L}{g} \sin\left(\frac{\omega^2 L}{2g}\right)}{\omega^2 - \left(\frac{\omega^2 L}{2g}\right)^2} \right]^2 \quad \text{m}^2 \text{s} \quad (6)$$

The above introduced effective wave doesn't take into account oscillatory motions and the translatory motion of the ship. It is quite clear that the effective wave is a random phenomenon as well as the irregular sea itself.

Calculations for checking the hypothesis of Grim have shown that the differences between the righting

arms calculated for the ship running in irregular waves and righting arms for the ship in the effective wave are small enough to be neglected.

Additional calculations of the transfer function of the effective wave, in which the oscillatory motions of the ship were taken into account, have shown that in such calculations heave and pitch may be omitted. So the transfer function of the effective wave depends only on the wave frequency and the length of the ship.

The variance of the effective wave is a function of the length of the ship and the state of the sea, i.e. the significant wave height and the characteristic wave period. Using the ITTC spectral density function, the normalized variance can be obtained as a function of the ship's length and the characteristic wave period only. The following formula was derived:

$$d_{\xi E}^2 = \frac{\int_0^\infty \left[ \frac{x^2 \sin x^2}{x^2 - x^4} \right]^2 \frac{1}{x^5} \exp\left(-\frac{691}{x^4} \cdot \frac{1}{T_L^4}\right) dx}{\int_0^\infty \frac{1}{x^5} \exp\left(-\frac{691}{x^4} \cdot \frac{1}{T_L^4}\right) dx}$$

(7)

where:  $x = \omega \sqrt{\frac{L}{2g}}$  and

$$\bar{T}_L = \frac{\bar{T}}{\sqrt{\frac{L}{2g}}}$$

The normalized variance depends only on the non-dimensional characteristic wave period.

The maximum value is reached for

$$\bar{T}_L = 3, \text{ which allows to determine the}$$

most unfavourable (from the stability point of view) length of the ship for different sea states and sea areas.

As a basis for calculations of the righting arms of ships in following irregular waves the wave height with 3 % probability of excess is used, which is calculated for the effective wave in the same way as for the irregular sea itself:

$$H_{E \ 3 \%} = 5.27 \sqrt{D^2 \xi_E} \quad \text{m} \quad (8)$$

### 3.2 Influence of the ship's speed - the effective wave averaged in the time of capsizing

Being a random value the effective wave changes its amplitude with a velocity depending among others on the ship's speed. In the same way the righting arms change their values. When the ship is fast, the amplitude of the effective wave changes slowly, and the ship is situated on the wave crest for a long time. This can lead to capsizing, especially if the loss of stability is great. On the other hand, a slow ship is overtaken quickly by following waves and, therefore, is situated on the wave crest for a rather short time, which reduces the danger of capsizing. The influence of the speed on the ship's safety in following waves is taken into account by introducing the amplitude of the effective wave averaged in the time  $\Delta T$  which is needed for the ship's capsizing:

$$\xi_{AE \Delta T}(t) = \frac{1}{\Delta T} \int_{t - \frac{1}{2} \Delta T}^{t + \frac{1}{2} \Delta T} \xi_{AE}(t) dt \quad \text{m} \quad (9)$$

So, the amplitude of the average effective wave depends not only on the length of the ship and the sea state, but on the ship's speed as well. When the time of capsizing is known, this amplitude can be calculated. For further

simplification of the end formula it would be very useful to determine the time of capsizing as a value dependent on the ship's length as follows:

$$\Delta T \approx c \sqrt{\frac{L}{2g}} \quad \text{s}, \quad (10)$$

where  $c$  is a constant, which is to be determined. It is assumed that the following relation between the time of capsizing  $\Delta T$  and the natural rolling period  $T_{\phi 0}$  exists:

$$\Delta T \approx 0,6 T_{\phi 0} \quad \text{s} \quad (11)$$

From this the following formula for constant  $c$  follows:

$$c \approx 0,6 T_{\phi 0} \sqrt{\frac{2g}{L}} \quad (12)$$

Statistical data for more than 200 ships show that the values of this constant differ considerably, but  $c = 3,3$  can be chosen as a rough approximation. It offers the safe solution, i.e. a rather short time of capsizing. With that the normalized variance of the average amplitude of the effective wave can be calculated according to the following formula:

$$\sigma_{\xi_{E \Delta T}}^2 = \frac{\int_0^\infty S_{\xi_{E \Delta T}}(\omega) \left[ \frac{\sin 1,65 \left( \omega \sqrt{\frac{L}{2g}} - \sqrt{2} \frac{\omega^2 L}{2g} F_n \right)}{1,65 \left( \omega \sqrt{\frac{L}{2g}} - \sqrt{2} \frac{\omega^2 L}{2g} F_n \right)} \right]^2 d\omega}{\int_0^\infty S_{\xi \xi}(\omega) d\omega} \quad (13)$$

This variance depends only on the non-dimensional characteristic wave period and the Froude number. Calculations carried out for different ship's lengths and different Froude numbers have shown that it can be simplified as follows:

$$d^2 \xi_{E \Delta T} \approx \kappa^2 d^2 \xi_E \quad (14)$$

$\kappa$  is a coefficient which depends first of all on the Froude number. It

can be approximated as follows:

$$\kappa = \frac{\sin [2,89 (1 - 2,51 F_n)]}{2,89 (1 - 2,51 F_n)} \quad (15)$$

Now the average height of the effective wave with 3 % probability of excess can be calculated:

$$H_{E \Delta T 3\%} \approx \kappa H_{E 3\%} = 5,27 \kappa \sqrt{D^2} \xi_E \quad \text{m} \quad (16)$$

### 3.3 The height of the effective wave for different sea areas and different sea states

The above presented approach gives the possibility to determine the height of the effective wave with 3 % probability of excess for different sea areas and different sea states. Such calculation has been carried out for the sea states corresponding to wind forces of 8, 10 and 12 Beaufort numbers in the Atlantic Ocean, North Sea and East Baltic Sea using the statistical data containing the significant wave height and the characteristic wave period given by Hattendorff [6]. Results of these calculations are shown in Fig. 1. They lead to following conclusions:

1. The effective wave is much higher for open sea areas, such as Atlantic Ocean and North Sea, than for restricted areas, for example the Baltic Sea. When introducing categories of ships depending on their area of navigation, not only the different heeling moments due to wind pressure, but also the different heights of the following wave are to be taken into account.
2. For restricted areas the height of the effective wave increases not very much (10 or 12°B) or remains almost constant (8°B) when the length of the ship increases. However, for open areas it increases substantially with the length of the ship.

### 3.4 The height of the average effective wave for stability calculations

The height of the effective wave with 3 % probability of excess corresponding to wind force 8°B in the Atlantic Ocean was chosen as a basis for checking the stability of ships in following waves. The dependence of this height

from the length of the ship was approximated by the following formula:

$$H_E = \frac{L}{4,14 + 0,14 L} \quad \text{m} \quad (17)$$

When calculating the average effective wave according to formulae (15) and (16), the Froude number was limited to the value 0,23. Therefore, in formula (15)  $F_n$  denotes:

$$F_n = \min \left\{ \frac{v}{\sqrt{gL}} ; 0,23 \right\} \quad (18)$$

This limitation follows from the assumption that in a critical situation in following waves the Master will reduce the speed of his ship, so that  $F_n = 0,23$  will not be exceeded.

#### 4. Heeling moment due to wind pressure in following waves

The stability in a seaway shall be sized in the same way as the stability in smooth water according to criteria which take into account the external forces affecting the ship. It would not be right to require only parameters of the righting arm curve which apply to all ships. The most dangerous heeling moment which can affect a ship in a seaway is the heeling moment due to wind pressure. Therefore, it is taken as a basis for sizing the stability.

The value of the wind pressure is assumed according to the DSRK and PRS Rules for the Classification and Construction of Sea-Going Ships, Part IV "Stability". Furthermore, it is assumed that the wind which has caused the seaway has changed its direction, while the waves move on in the original direction. The assumed change of direction is  $45^\circ$ , so that the wind meets the ship in oblique direction from behind.

The permissible dynamical heeling moment due to wind pressure results from the condition that the heeling angle or flooding angle may not be exceeded, whichever is less.

#### 5. Weather criterion for ships in following waves

The righting moment of a ship in following waves changes permanently. It changes between a maximum value, when a wave trough is in the middle of the length of the ship, and a minimum value, when a wave crest is in the middle of the length of the ship. The variations lead to rolling motions of the ship.

In regular waves very high roll amplitudes can occur. In irregular waves, when the variations of the righting moment are also irregular, substantial smaller roll amplitudes are to be expected. A method for calculation of the amplitudes in following irregular waves is not available. The known methods for calculation of roll amplitudes used for checking the weather criterion in smooth water are not suitable. These methods have been developed under the assumption that the waves meet the ship from the side.

For the mentioned reasons the rolling oscillations are not taken into account when checking the stability in following waves. The check is restricted to the heeling moment due to wind pressure. The weather criterion is only a wind pressure criterion.

#### 6. Critical parameters of the righting arm curve in following waves

Following criteria for valuation of stability in a seaway have been stipulated taking its pattern from the damage stability requirements:

- 1 The initial metacentric height  $GM$  shall be positive.
- 2 The angle of vanishing stability shall be at least  $30^\circ$ .
- 3 The maximum statical righting arm  $GZ_{\max}$  shall be at least 0,08 m.

Theoretical arguments for these requirements do not exist. A checking and eventual correction is only possible after a longer application in practice and analysis of casualties.

#### 7. Results of test calculations

The method for calculation of the intact stability of ships in following waves has been applied to several ships. Some results of these test calculations are given in the following.

##### 7.1 Research ship (Fig. 2 and 3)

|                       |                                 |
|-----------------------|---------------------------------|
| $L = 54,00 \text{ m}$ | $F_n = 0,2682$                  |
| $B = 10,60 \text{ m}$ | $H_E = 4,615 \text{ m}$         |
| $D = 6,80 \text{ m}$  | $\lambda_E = 0,7685$            |
| $v = 12,0 \text{ kn}$ | $\lambda H_E = 3,547 \text{ m}$ |

The ship has in all investigated loading conditions a stability in smooth water, which much exceeds the values required in the Rules. Nevertheless, the stability in following waves is insufficient.

Only in loading condition 1 the requirements on the righting arm curve are complied with partly. The results of other loading conditions are in between conditions 1 and 2.

## 7.2 Motor coaster (Fig. 4)

L = 68,00 m       $F_n = 0,2689$   
 B = 12,00 m       $H_E^n = 4,978$  m  
 D = 6,80 m       $\kappa = 0,7685$   
 v = 13,5 kn       $\kappa H_E = 3,826$  m

The requirements on the stability in following waves are complied with in all loading conditions. Loading condition 1 is the most unfavourable one.

## 7.3 Lifting vessel (Fig. 5)

L = 68,00 m       $F_n = 0,2490$   
 B = 14,80 m       $H_E^n = 4,978$  m  
 D = 7,394 m       $\kappa = 0,7685$   
 v = 12,5 kn       $\kappa H_E = 3,826$  m

The requirements on the stability in following waves are complied with in all loading conditions. The reserve stability is considerably.

## 7.4 Fishing and factory vessel (Fig. 6)

L = 91,80 m       $F_n = 0,2503$   
 B = 15,20 m       $H_E^n = 5,403$  m  
 D = 9,70 m       $\kappa = 0,7685$   
 v = 14,6 kn       $\kappa H_E = 4,152$  m

The requirements on the stability in following waves are complied with in all loading conditions.

## 7.5 General cargo ship (Fig. 7 to 11)

L = 114,355 m       $F_n = 0,2688$   
 B = 18,40 m       $H_E^n = 5,674$  m  
 D = 9,90 m       $\kappa = 0,7685$   
 v = 17,5 kn       $\kappa H_E = 4,361$  m

In conditions 1, 2 and 3 the ship has a stability, not only in smooth water but also in following waves, which exceeds considerably the required values. However, in loading conditions 4 and 5 the stability in smooth water exceeds the lower permissible limit only a little, and the stability in following waves is sufficient only in respect of the initial metacentric height. The other criteria are not

complied with.

## 7.6 Conclusions

The presented and also other results show that the checking of stability in following waves is reasonable for small ships in all loading conditions. Even in cases of very good stability in smooth water, the stability in following waves can be insufficient.

For bigger ships can be assumed that the stability in following waves is sufficient, if in smooth water a considerable reserve stability is provided. When the stability in smooth water exceeds the required minimum values only to a small extent, the stability in following waves can be critical, and its checking is advisable.

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# FIGURES

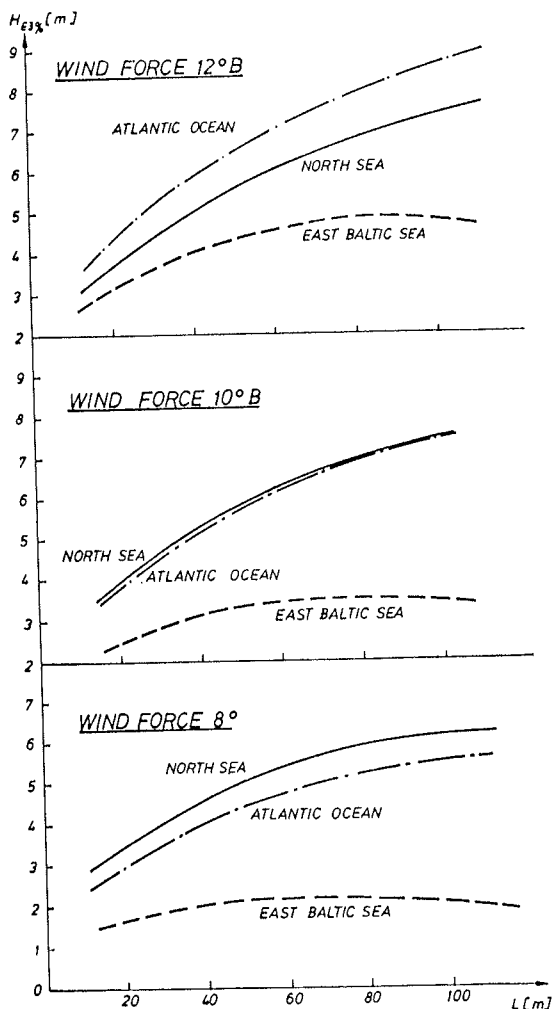


Fig. 1  
The height of the effective wave for different sea areas and different sea states

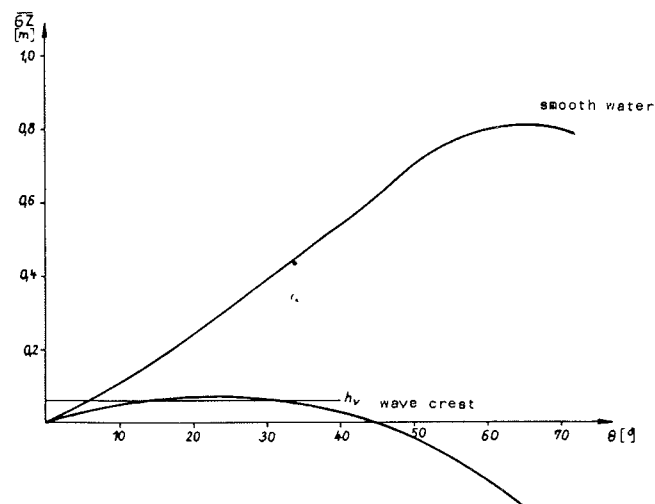


Fig. 2  
Research ship, loading condition 1  
Ship with 22 t of expedition goods,

100 % stores.  
 $\Delta = 1652$  t;  $KG = 4,81$  m;  $h_v = 0,06$  m  
 Smooth water:  $GM = 0,66$  m  
 Wave crest:  $GM = 0,29$  m;  $\theta_v = 44,5^\circ$ ;  
 $GZ_{max} = 0,07$  m; wind criterion: no

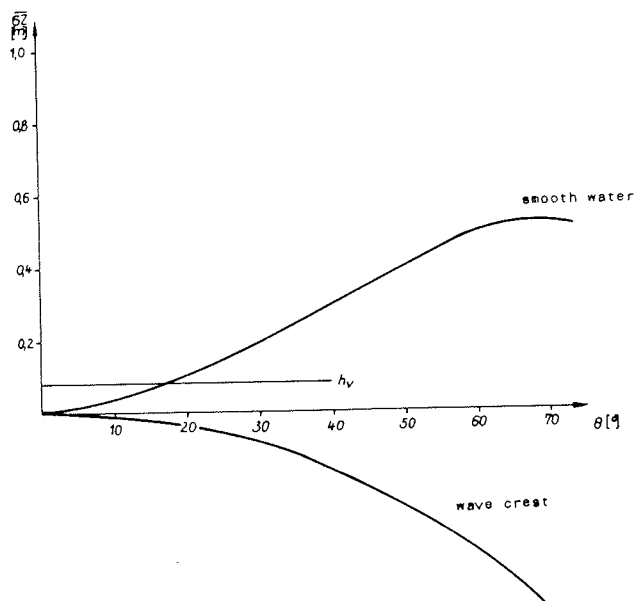


Fig. 3  
Research ship, loading condition 2  
Ship with 8 t of expedition goods, 10 % stores, icing.  
 $\Delta = 1400$  t;  $KG = 5,18$  m;  $h_v = 0,08$  m  
 Smooth water:  $GM = 0,21$  m  
 Wave crest:  $GM = -0,12$  m;  $\theta_v = 0$ ;  
 $GZ_{max} = 0$ ; wind criterion: no

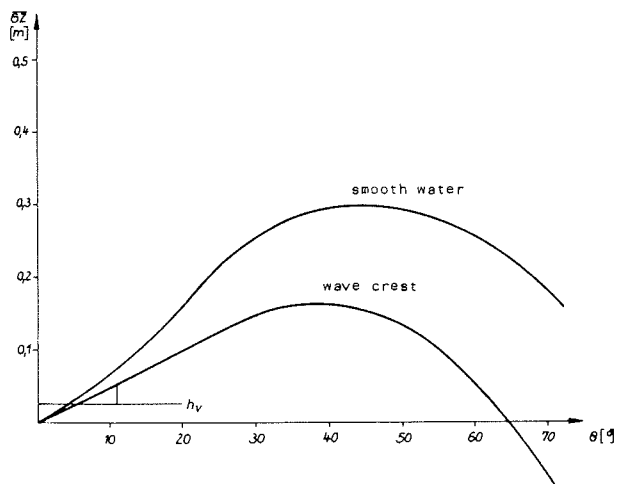


Fig. 4  
Motor coaster, loading condition 1  
Ship in fully loaded condition, 100 % stores.  
 $\Delta = 2080$  t;  $KG = 4,66$  m;  $h_v = 0,05$  m  
 Smooth water:  $GM = 0,71$  m  
 Wave crest:  $GM = 0,52$  m;  $\theta_v = 64,5^\circ$   
 $GZ_{max} = 0,32$  m; wind criterion: yes

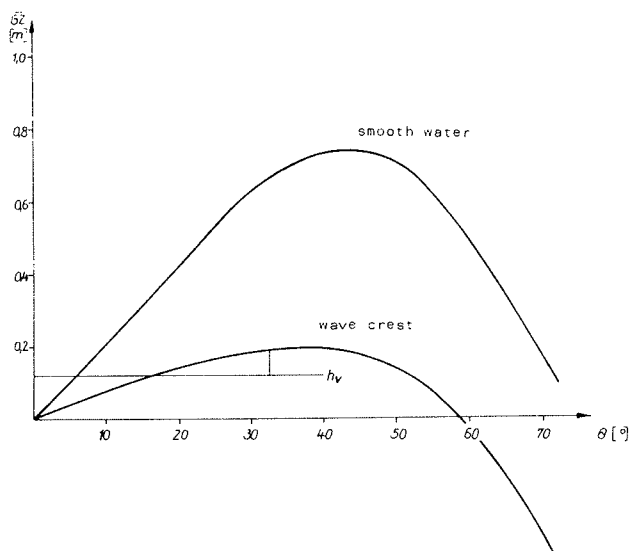


Fig. 5  
Lifting vessel, loading condition 1  
Vessel without cargo, 100 % stores.  
 $\Delta = 2796$  t;  $KG = 6,23$  m;  $h_v = 0,12$  m  
Smooth water:  $GM = 1,03$  m  
Wave crest:  $GM = 0,44$  m;  $\theta_v = 58,5^\circ$   
 $GZ_{max} = 0,19$  m; wind criterion: yes

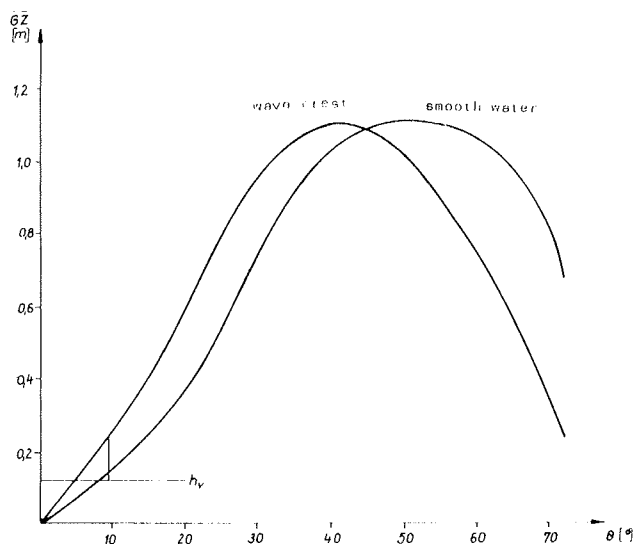


Fig. 7  
General cargo ship, loading condition 1  
Ship without cargo, 10 % stores.  
 $\Delta = 6303$  t;  $KG = 6,50$  m;  $h_v = 0,12$  m  
Smooth water:  $GM = 0,78$  m  
Wave crest:  $GM = 1,46$  m;  $\theta_v > 70^\circ$   
 $GZ_{max} = 1,10$  m; wind criterion: yes

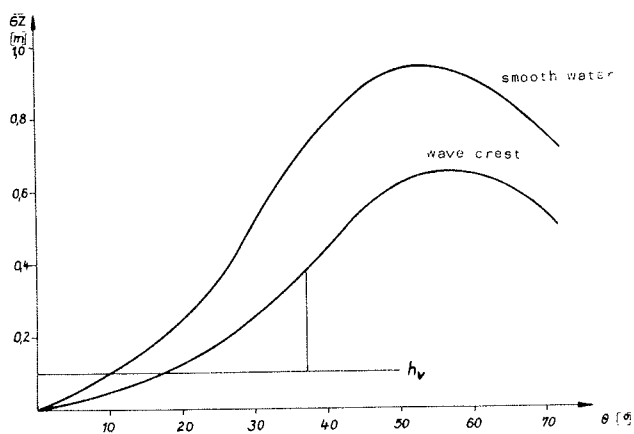


Fig. 6  
Fishing and factory vessel, loading condition 1  
Vessel with full catch, 10 % stores, icing.  
 $\Delta = 5985$  t;  $KG = 6,27$  m;  $h_v = 0,10$  m  
Smooth water:  $GM = 0,52$  m  
Wave crest:  $GM = 0,20$  m;  $\theta_v > 70^\circ$   
 $GZ_{max} = 0,65$  m; wind criterion: yes

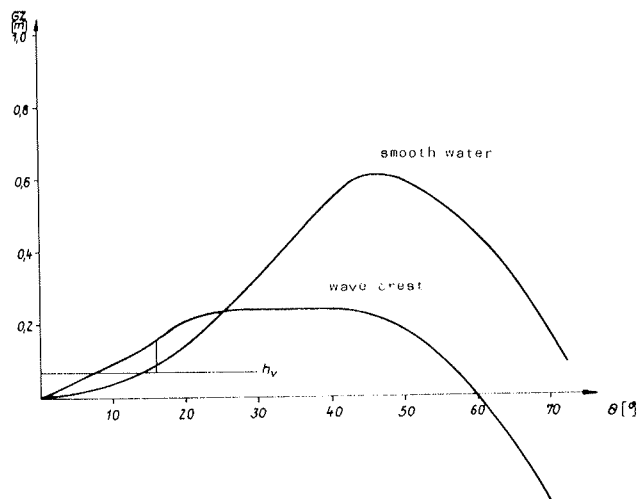


Fig. 8  
General cargo ship, loading condition 2  
Ship loaded with general cargo, 100 % stores, tonnage mark draught.  
 $\Delta = 8770$  t;  $KG = 7,00$  m;  $h_v = 0,07$  m  
Smooth water:  $GM = 0,15$  m  
Wave crest:  $GM = 0,51$  m;  $\theta_v = 60^\circ$   
 $GZ_{max} = 0,24$  m; wind criterion: yes

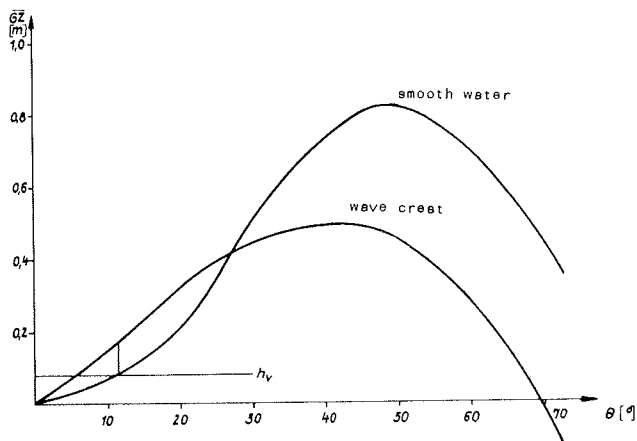


Fig. 9  
General cargo ship, loading condition 3  
Ship loaded with containers, 100 %  
stores, tonnage mark draught.  
 $\Delta = 8319$  t;  $KG = 6,74$  m;  $h_v = 0,08$  m  
Smooth water:  $GM = 0,38$  m  
Wave crest:  $GM = 0,81$  m;  $\phi_v = 69,5^\circ$   
 $GZ_{max} = 0,49$  m; wind criterion: yes

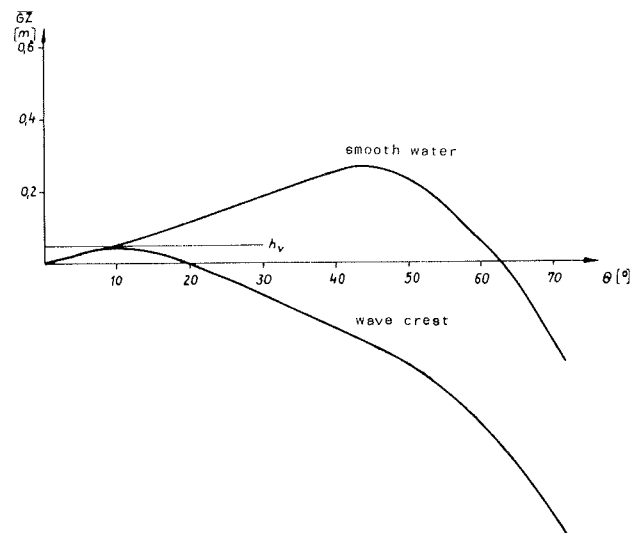


Fig. 11  
General cargo ship, loading condition 5  
Ship loaded with general cargo, chilled  
cargo and containers, 10 % stores.  
 $\Delta = 10732$  t;  $KG = 7,21$  m;  $h_v = 0,05$  m  
Smooth water:  $GM = 0,19$  m  
Wave crest:  $GM = 0,24$  m;  $\phi_v = 20^\circ$   
 $GZ_{max} = 0,04$  m; wind criterion: no

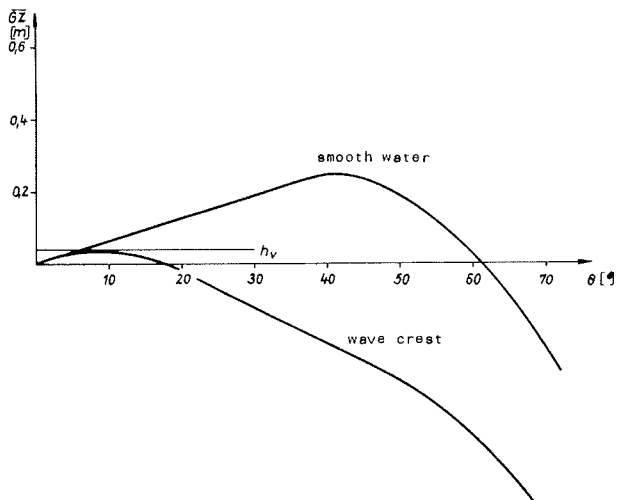


Fig. 10  
General cargo ship, loading condition 4  
Ship loaded with general cargo, chilled  
cargo and containers, 100 % stores.  
 $\Delta = 11150$  t;  $KG = 7,19$  m;  $h_v = 0,04$  m  
Smooth water:  $GM = 0,29$  m  
Wave crest:  $GM = 0,21$  m;  $\phi_v = 17^\circ$   
 $GZ_{max} = 0,03$  m; wind criterion: no



## PANEL DISCUSSION II: CRITERIA AND REGULATIONS

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Federal Republic of Germany

### CHAIRMAN (Middleton):

This is Panel II, the closing panel or session of the conference except for a summary which will follow.

I would like to first introduce myself. I am Edward Middleton, formerly of the U.S. Coast Guard and now working for a business organization called Maritime Institute for Research and Industrial Development. Maritime Institute for Research and Industrial Development doesn't mean that we do research, but we look for areas in which to have research done and, of course, that is probably of interest to some of the researchers here.

I would now like our Co-Chairman to introduce himself.

### CHAIRMAN (Rakhmanin):

Thank you. Ladies and gentlemen, my name is Nicolay Rakhmanin. I was educated at the Leningrad Shipbuilding Institute in Ship Hydrodynamics. I was graduated from this institute in 1955, and from that time I have been working for the Krylov Shipbuilding Research Institute in Leningrad. Now I am responsible for investigation and seakeeping as the head of the laboratory, corresponding laboratory, of this institute.

From 1966 I am participating in different IMO bodies, subcommittees, and at present I am Chair of the Working Group on Stability. Thank you.

### CHAIRMAN (Middleton):

Thank you, Dr. Rakhmanin. You reminded me, of course, I, too, have attended IMO, formerly IMCO, sessions for the last 16 years, in the Subcommittee on Subdivision, Stability, later called Subdivision, Stability and Load Lines, and now called the Subcommittee on Stability and Load Lines and on Fishing Vessel Safety.

I also have attended the meetings of the IMO Subcommittee on Containers and Cargoes, its Maritime Safety Committee, various conferences dealing with safety of life at sea, fishing vessel safety, container safety, tanker safety and pollution prevention and the Conference on Standards of Training and Watchkeeping of Seafarers. At present I am the Chairman of the IMO Subcommittee, on Stability and Load Line and on Safety of Fishing Vessels, now known as SLF.

Our panel today consists of the gentlemen to my right, and I will begin at this end of the table. On my right and also to your right we have Rear-Admiral O'Dogherty, who is a member of the Executive Committee

of ITTC, and Director of the El Pardo Tank from 1970.

Next to him we have Mr. Ji who is head of the Sea Keeping Section Laboratory of MARIC, Vice Chairman of the Sea Keeping Panel of the Chinese Society of Naval Architects. He was graduated from Shanghai Shipbuilding Institute in 1954, he worked as a Research Assistant on Sea Keeping Section in the CSSRC until 1971, and since 1977 has been in charge of the recent position in MARIC which he is now the head of that Sea Keeping Section.

Next to Mr. Ji is Mr. Gu, who was graduated from the Henry Lester Polytechnic Institute in Shanghai. He studied at the University of Michigan in the Department of Naval Architecture and received his Master's Degree in 1949. He has been a lecturer, assistant professor and professor in the Harbin Shipbuilding and Engineering Institute from 1952 till 1979. He is now the Director of the China Ship Scientific Research Center, from 1979 to the present time.

Proceeding along the table, we have Dr. Dahle, Associate Professor, University of Fisheries of Trondheim, Norway. He has a Doctor of Engineering Degree on the theme of "Rolling and Damping for Small Fishing Vessels", and for several years he has been the leader of the University Fisheries Department at the University of Technology in Trondheim. He was a member of the official investigating group for fishing vessel casualties, and many years as a Norwegian official delegate to IMO.

The next author is Mr. J. Jens, who before joining IMO in 1966, was a member of the Ship Safety Department of Germanischer Lloyd, and in this capacity was in the delegation of the Federal Republic of Germany to the 1960 Safety of Life at Sea Conference and in the Subcommittee on Subdivision Stability, and the Panel on Fishing Vessels. He is now in the IMO Secretariat, and holds the position of Head of the Subdivision for Technology. In this position, he has participated in nearly all of the various conferences of the organization. He is also responsible within the Secretariat for a number of subcommittees and also for the relevant work in the Maritime Safety Committee and the IMO Assembly.

Last but not least, we have Mr. Arndt, who holds a Diploma in Engineering and works for SCHIFFKO, Consulting Engineers, Hamburg, West Germany; a Degree of Certified Engineer in Naval Architecture in 1957 at the Technical University of Hannover; more than ten years Research Assistant in the team of Professor Dr. of Engineering Mr. Wendel, whom I'm sure all of us know. He is the Chairman of Ship Design and Ship's Theory. He was involved in stability research, elaboration of stability regulations for the West German Navy, capsizing experiments with remote control models, shifting of bulk cargo and stability of lifeboats. Since 1969, SCHIFFKO Department for Special Investigations, e.g., Loading Computers, and so forth. He is a member of the STG since

1962, and he is Chairman of ISO TC8, Subcommittee 15.

That, ladies and gentlemen, is the introduction of our distinguished panel. As you can see, we have some expert authors to discuss our work in Panel II today.

Before we begin, I thought perhaps it might proper to have a little recap of how we got to Panel II. As you all know, the conference began with Panel I discussing the philosophy of ship stability in relation to research. In trying to define the term "stability" it was determined that its definition is divided into two parts: first, the description of what is meant by "adequate stability" and, second, the criteria needed to ensure adequate stability.

During the various conference sessions in the week, there have been many papers and reports presented which represent a great deal of work that has taken place leading up to this conference, and much of this work has been directed toward a better understanding of ship and ocean vehicle stability. This work, of course, has gone on and it is justified by increasing the state of the art, but of itself it may not justify all the effort. The outcome of this work may better be directed toward some meaningful and utilitarian purposes, and that brings us to the title of our present panel which is "Criteria and Regulation". Perhaps the word "regulation" seems a little harsh. Maybe we could talk about "requirements" rather than "regulation", but our title is "Regulation".

In this panel today, we will deal with the analyses of research and experience upon which to base stability criteria, and on the development of associated requirements to ensure adequate stability.

At this point, we have some view graphs, there are three of them, and we are going to show all three of them one after the other, and then later on we will go back to take them one at a time, but to familiarize you with the view panels we will describe them all at once, and I will ask Dr. Rakhmanin to give this description. Dr. Rakhmanin.

#### CHAIRMAN (Rakhmanin):

Thank you. Ladies and gentlemen, in order to facilitate the organization of our discussions, on the part of the Chairmen we have prepared these three tables which, in principle, may show how the development procedure of stability requirements and regulations might look. So we call this first table "Criteria Development".

Next slide, please. The second is "Criteria Determination", and the third one "Development of Regulation".

What we are going to do is to suggest to help our discussions according to these three steps, and when we show the corresponding tables I will explain in more detail what point for discussion will be suggested, and during the demonstration of the tables the corresponding papers to this panel discussion will be submitted. So we would like to show the first table.

This table contains the following information: In the horizontal line, as you can see, are areas of investigation. I mean those ship qualities which are under consideration: stability in still water, stability in waves, and ship's motion in waves.

On the vertical line to the lefthand side of the picture, we present the physical data and forces acting on the ship or ocean vehicles.

So those particulars and features of external forces, environmental conditions, are indicated here which in some way are taken into account when research and investigation are made.

In the center of this table and to the righthand side of the table under a certain number, it is suggested the topics which are of interest to discuss during this first part of our discussions.

So at this stage I think we may ask Admiral O'Dogherty to submit his paper.

REAR-ADMIRAL O'DOGHERTY: (Presentation)

CHAIRMAN (Middleton):

Thank you, Admiral O'Dogherty. Sorry to hasten you on. We were a little late starting and we have to make up some time, or else we'll be here well beyond our allotted time. I might mention to all of the participants that, first, we have some comments that have been submitted, and then there will be an opportunity for some questions and answers, and we hope to get participation from participants other than those who have submitted questions, submitted comments, and we want to generate from all participants their views on the basic ideas that were shown on our chart at the beginning of the session, this chart.

We have a comment from Mr. Shin Tamiya of Japan, Tokai University.

TAMIYA:

In principle, I agree with the author's treatment for assessing the ability of small fishing boats (maybe enormous in number). Following the presented criterion, however, a constant  $GM/B$  will be needed for a given  $L$  and  $C_s$  ( $L$ ) which leads to give:

- 1) a small  $GZ_{mx}$  for ships of small  $P$ , and
- 2) limited heel-angle  $\theta_m$  corresponding to  $GZ_{mx}$  for ships of large  $B$ .

Both results do not seem favorable to prevent capsizing, and it is considered better to take  $GZ_{mx}$  into the criterion because  $GZ_{mx}$  is naturally included in the calculation of  $e_k$ , a known quantity. The authors would certainly investigate such an idea and it will be instructive to have your opinion.

O'DOGHERTY:

Thank you very much, Professor Tamiya. I agree in principle, but I think you have misinterpreted our criterion. It is not true that using the criterion will imply to

have constant  $GM/B$  for a given length, so the criterion is flexible so that any  $GM/B$  will be all right, provided the dynamic stability is complying with the criterion. So when the ship has a smaller  $GM/B$ , we will require to have more dynamic stability and, in principle, the  $GZ_{mx}$  will be safe. So that this is not true, really.

At the same time, fixing a maximum  $GZ$ , establishing a rigid maximum  $GZ$ , to my own view is not a good idea because it is better to have dynamic stability satisfactory no matter how it is distributed. So we may have stability curves of A or B type, that is stability curves with high initial stability or with low initial stability. If initial stability is high, then our criterion is enforcing the need for a minimum well determined value of dynamic stability, depending on the length.

Alternatively, if  $GM/B$  is very small, then dynamic stability must make up for this deficiency and have a larger value, so that we are requiring larger value for dynamic stability for ships with a small initial stability, and the larger dynamic stability will be concentrated mostly at large angle stability, thus making the ship safe. We are feeling safe in our assumptions because the correlation with practical data of ships to help (from capsizing) is quite good. Thank you.

CHAIRMAN (Middleton):

Our next commenter is Mr. Fulford of the United Kingdom, British Shipbuilders. Mr. Fulford.

FULFORD:

Firstly, I found the paper very interesting and thank the authors for it. I have a small quibble. It's small but it comes somewhere near the root of the problem with all criteria.

In their introduction the authors imply that stability assessment has changed from a simple use of experience to something different. In terms of philosophy this is hardly so as all the current criteria for stability represent only codified experience of the type carried by the builders that were mentioned. The arbitrary choice of a small number of regulated parameters, bearing only a limited relation to the problem addressed, ensures this. They are all historical data-based empiricisms.

But I don't mean to disparage any of them or this one by saying that. The one here proposed I think represents an interesting reshuffle of the parameters, but the essential problem of the basis on which adequacy is to be judged remains. Casualties may not necessarily be a good basis. Is the cause of the loss certain? And even if it is known, is it relevant? That is, is it directly concerned with intact stability? And even if it is, is the demand such that it is unreasonable to expect the capability to meet that demand? Again, the base needs to be defined.

And, lastly, how do the authors propose to test the validity of their proposal and determine whether it is an improvement on, say, the IMCO ones at the moment?

O'DOHERTY:

Thank you very much. Well, I feel that the paper must have been not well meant or well expressed but, really, we did not mean that we are thinking to do something different from everybody else. I think that this paper is empirical as all ... are. The only thing is that we are aiming to establish a very easy empirical criterion by analyzing the external forces and the actions that are more dangerous to ship safety, and so we have both taken into consideration the action of the waves, the waves and wind, and also the stability loss due to the following waves.

Of course, there are many more actions to be taken into consideration and many more influences in ship stability, particularly those connected to ship operation but these are, of course, left out of any possible mathematical evaluation.

As far as the validity of the criterion is concerned, we are happy that the validity is proven by the comparison of the application of the criterion to ships that have been lost at sea, or, alternatively, to those that have been deemed to be satisfactory by well known experts. Thank you.

CHAIRMAN (Middleton):

Thank you very much. Are there any other questions that were not submitted in written form? I see none. Then perhaps the best thing for us to do is now go on to the second author, Mr. Ji, who will present his paper. Mr. Ji.

MR. JI:

(Presentation)

CHAIRMAN (Middleton):

Thank you very much, Mr. Ji. We have one comment on Mr. Ji's paper, again from Prof. Shin Tamiya of the Tokai University.

TAMIYA:

Thank you, Mr. Chairman. The authors make use of a servo-controlled moving weight mechanism to give heel movement of wind. It is surely a convenient and simple method. However, it should be noted that wind moment is usually composed of two horizontal forces, that is wind drag and water resisting force, or sway intertial force. The authors' device may be better considered as simulating the shift of cargo, not wind moment.

And the second, as to the paradox  $E_1^S > A_1^S$ , two possibilities of explanation are considered to remain.

One is the effect of restraint to the model, and another is the possible difference between the actual average heel angle

in waves and that determined by static calculation.

If the model has no restraint and is drifting freely by wave action, the time averaged non-zero heel angle will be realized. If, contrary, some restraint is given to the model, supporting mechanism and/or the location of the restraint might affect the motion of the model in an unexpected manner. Thank you.

CHAIRMAN (Middleton):

Thank you, Prof. Tamiya. Would the author like to respond to the comment, please?

JI:

I thank Dr. Tamiya for his contribution. In our experiment, the moving weight is designed to simulate the heeling moment produced by the wind force acting on the ship above the waterline and the hydrodynamic reactive force acting on the underwater hull. The center of action of the hydrodynamic force due to sway velocity has been extensively investigated by Yen Jiaji et al in China.

On the whole, the center of action varies with ship form. So in our experiment we adopted a center of action suigable to the specific ship form, and simulated the wind moment by taking the couple formed by wind and hydrodynamic forces into account.

As to the second comment, our experiment was done with a model floating freely in water without restraint, and I appreciate the suggestion of Dr. Tamiya which will serve as a reminder in further investigations. Thank you.

CHAIRMAN (Middleton):

Thank you, Mr. Ji. Those were the only written comments. Are there any questions of the author from the floor? Anyone has a question he might like to have answered? I see no volunteers.

At this point we will go back to Step I and criteria development which is shown on the screen, and also at this point we would like to invite comments from various participants in the conference with regard to the items listed, 1 through 6 on this chart, bearing in mind that when we speak of stability we are not limited to intact stability, but the philosophy behind items 1 through 6 might equally be applied to damage stability. We have put this up as a shopping list, sort of a first step in the process of criteria development, and perhaps there are other suggestions that may fill in any missing blanks if someone feels there are some missing blanks.

I would now ask Dr. Rakhmanin if he would further elaborate somewhat.

CHAIRMAN (Rakhmanin):

Thank you, Mr. Middleton. I would like just briefly to make a small comment for this table taking into account the paper

presented. From this table you can see that the development of criteria has several stages. From the paper presented, you can see the practical model of criteria made of different forms, and so this force might further deviate, have a very large deviation from the real situation but, nevertheless, you know that in the development of stability requirements the simplification of the situation is a necessary step. So I would like to invite discussion on all these topics indicated on the screen. I would like also to bring your attention to the gap between theoretical model of the real situation and the practical model of criteria. It's of interest to my mind. Thank you.

CHAIRMAN (Middleton):

Anyone? Yes, Mr. Helwig.

HELWIG:

If you want to add damage stability to the list as point 7, that is in my opinion a good idea. But the present situation is that a lot of ships especially RO-ROs, won't comply with damage stability situations. When I look at the older line ships with, let's say, five holds, they are usually one compartment ships, and when I look at a lot of container ships they may sustain a lot of damage.

But as soon as you are going to put on the list damage stability requirements, that will change ship types and it will demand a lot of investigation on the present ships. But I think it's unreasonable to require damage stability calculations, for example, chemical tankers, gas ships, and not for normal cargo ships. That is what I wanted to say. Thank you.

CHAIRMAN (Middleton):

Thank you, Mr. Helwig. Any other comments? Yes, please.

TSUCHIYA:

Thank you, Mr. Chairman. What I wanted to express here is the consigning of the model experiment problem. Our institute has been continuing many kinds of model experiments in rough seas. But I am going to think about some appraisal. Our model is only Japanese type fishing boats, and the proportion and the fullness are much different from the European fishing boats. For such kind of European fishing boats the results of the model experiments will be quite different from our results.

So here I want to propose to other organizations or institutes that the comparative model experiments proposed by Dr. Takaishi in the First Panel Session, that is comparative model experiments should be planned for several popular types of fishing vessels, for instance the moderate type, trawlers in Europe and in Japan and, if possible, high speed small coastal fishing boats having hard chine and overhanging decks.

I believe such kinds of experiments, comparative model experiments, will contribute very much to promote the research work and the mutual understanding of the stability criteria and stability information which will be used or improved in the future.

CHAIRMAN (Middleton):

Thank you very much. Indeed, we appreciate your comment, and as you have indicated you would expect some comparative model experiments and, of course, that is an invitation to others to conduct these experiments, and that falls within the line of item 2 on our chart which calls for both theoretical studies and model experiments. Again, these are basic philosophies of the development of criteria, and we would invite other comments. Yes, Professor Motora.

MOTORA:

Thank you, Mr. Chairman. Item No. 3 at the righthand side, theoretical model of a criterion, has to be, in my opinion, a theoretical model to describe phenomena. Then we proceed to simplification of the theoretical model and items 5 and 6. Thank you.

CHAIRMAN (Middleton):

Any other comments? Yes, Mr. Manum.

MANUM:

Thank you, Mr. Chairman. I was actually looking for Professor Gerritsma because I was eating with him at lunch and he had some observations, but he is not raising his card, so I have to say it myself.

First, I think that the wording here confuses me a little. I'm not sure exactly what it means, but the way I read item 5 I think there is a gap between theoretical model and practical criterion. Is that the understanding? Could I have a short reply on that? My understanding of 5 is that gap between theoretical model and practical criterion. Could I have a reply from Rakhmanin?

CHAIRMAN (Rakhmanin):

Yes, again this was a gap. You see, when you are trying to describe the phenomena under investigation, for instance the ship motion and following seas, then you see the real theoretical description of this investigation or theoretical model might be very complicated and it might be used for computer computations. It might be used for explaining the different facts of this phenomenon. But it might be difficult to use for practical criteria.

So when one thinks about the practical criteria, one can suggest some simplified model of this criteria. I may refer to the paper where proposal for criteria for checking stability in following waves which was just mentioned in Mr. Ji's paper, when ship proceeding in following waves and subject

to the gust wind.

So this model of practical criteria differs from those phenomena which one can observe in following seas, parametric... stability, etc. So when we indicated in item 5 the gap between theoretical model and practical model, it is the suggestion to comment how the theoretical model and practical one which can be used in regulations may be different.

MANUM:

Mr. Chairman, then I would understand, Rakhmanin, that we mean in my wording practical criterion, and then I would say item 5 is the one of the highest priority and it is also covering several of the others because in order to reduce the gap of course you have to simplify, and I would just like to underline that when you have found theoretical models that are feasible, of course you should as soon as possible try to put that into practical usable criterion. Thank you very much, Mr. Chairman.

CHAIRMAN (Middleton):

Thank you, Mr. Manum. Mr. Jens.

JENS:

Mr. Chairman, I think I have a small suggestion to improve the lefthand side here. Special external forces are mentioned here right at the end. I understand that conventional external forces up there, wind and wave effects. Now, special external forces up there, wind and wave effects. Now, special external forces for me, first of all it is not only the passengers on deck but the passengers crowded on one side of the ship, ... and it said of course etc. You have certainly for the high altitude countries icing effect. I think that is the influence of ... force, may also better be under "special external forces." That's just a short remark.

CHAIRMAN (Middleton):

Thank you, Mr. Jens. As I said before, this was not an all-inclusive list, but certainly if there is an item that is of importance that it might be highlighted by being included in the list, then indeed the one on icing is an important one, especially for high latitudes. Professor Motora?

MOTORA:

Mr. Chairman, in special extra forces, the shifting of cargo should be highlighted. Thank you.

CHAIRMAN (Middleton):

Thank you, Professor Motora. Are there any other comments or suggestions with regard to this table? May I take it then that we can hopefully, in the outcome of this panel session, add those comments to the table and make this the shopping list

for research in the development of criteria as a first step in the procedure for stability requirements and/or regulations. That now brings us to step 2, and I would ask for a brief introduction of this step by Dr. Rakhmanin.

CHAIRMAN (Rakhmanin):

Thank you. Ladies and gentlemen, you can see this second step as we have conditionally called criteria determination. We mean that the forms of representation of criteria might be a different ones, and we tried very briefly to indicate here there is a static one, a physical one and a probabilistic one. In square brackets you can see the brief explanation for each. I will not read these explanations because you can see them. The topics for discussion are suggested under items 7, 8 and 9, but before this discussion it might be expedient to suggest that Dr. Dahle present his paper which reflects most of these items indicated in this table.

E.A. DAHLE:

(Presentation)

CHAIRMAN (Middleton):

Thank you very much. Dr. Dahle. We have one written comment by Mr. Fulford of the British Shipbuilders, and I would ask Mr. Fulford if he would give his comment.

FULFORD:

I find the approach to a stability criterion founded on the probability of non-capsize to be logical, but a number of questions occur.

I note you intend to include other dangerous motions such as you just described. How would you propose to determine the upper acceptable levels of pitch, rolling and acceleration?

Secondly, what sort of level of probability of capsizing is considered acceptable or alternatively bar the areas for heavy weather, and on what basis would either of these judgments be made?

As a designer, I would ask what attempts have been made or will be made to ascertain both significant parameters, the identification of significant parameters, and thereafter to map out a field of "safe" parameters which could be of value in the initial stages of a design before it was subjected to the full rigors of the regime you've been suggesting? Thank you.

CHAIRMAN (Middleton):

Thank you, Mr. Fulford. Dr. Dahle, would you respond, please?

DAHLE:

Thank you. The first question, the answer there would be that you have to look at each vessel individually, and of course

the cargo that it is carrying. In the case of fishing boats these days, they only take a few hundred kilos. The matter is not very interesting from that point of view. As regards heavy motion, the cargo is not ... around, and if it is, it doesn't matter. The same will go for other boats. If there is no, I would say 100 percent, safe way of securing the cargo, you have to look at the accelerations and the movement, and connect that to the type of cargo that you have in each individual case. There is a lot of knowledge of this type in IMO regarding bulk cargo. For other cargoes you should be certain that the cargo will not shift up to large angles of heel and the accelerations you would measure, possibly in model experiments.

The second question, the level of probability of capsize, it is considered acceptable. As it is now, based on the data that we have and roughly calculated, one boat, small, type fishing boat, will capsize each 1,829 year. That seems to be not acceptable. That's why so few skippers experience a capsize, by the way. They never live that long, and those who do take on the capsize never survive. So I think it should be improved upon. I think the level should be much lower than it is today, also have some ideas of how to do it, mainly by improving the extension of the GZ-curve, and also, as I said, put more serious emphasis on the operation of boats that do not have the design features that are safer than today. I think you should be much more strict on the operational side, on the operations of the boat in difficult weather. You should not allow it as an authority.

The third question is actually answered already because I could accept any design you come up with. I would restrict you severely if it was not being safe by my approbation procedure. You could come up with anything. Finally, you will not be allowed to go out to the harbor. So that is my answer. Come up with anything. More or less this will take care of it. The safer you make it the less I will punish you on this side. Thank you.

CHAIRMAN (Middleton):

Thank you very much. Is there any other question from other participants to ask Dr. Dahle with regard to his paper? Dr. Odabashi?

ODABASHI:

I was going to ask you a few questions on table 2 which I think is without caption actually, as the practical approach to safety in the seaway. That is the proposed way. Dr. Dahle, there we have seen in items 1, 2, 5, 6 and 9, 10, six out of twelve, advice to use two simulation programs. One is called CAPSI, the other is ROLLSI, which I believe were developed during the SIS project. But as I understand from what I know on ROLLSI, for example, it is a two-dimensional method

to calculate two-dimensional ship oscillating in a two-dimensional seaway without any effect of viscosity, without any proper effect of wind or gusting effect, so I wonder how are we going to manage to get anything out of that.

Furthermore, how many runs of how long duration does Dr. Dahle anticipate for each case to verify the required probability loads? The same question obviously applies to the case of program CAPSI, and how does he obtain the coefficient for that program to run? Thank you.

CHAIRMAN (Middleton):

Thank you, Dr. Odabashi. Perhaps, since it is now time for tea, and I believe you have some notes, it would be fair for Dr. Dahle to have these notes so he can prepare his answers during tea time, and then we will come back at 3:30 after we have had tea. Thank you very much.

(Break)

(After tea break)

CHAIRMAN (Middleton):

It is now 3:30, and since there are some participants who would like to get away tonight, we don't want to run our session over, so we'll begin again.

Just before we took the tea break, there was a question from the floor to Dr. Dahle, and I gave him the benefit of a nine-count, so to speak, to reply, so the first order of business after tea is a reply from Dr. Dahle.

DAHLE:

Thank you. Yes, the real answer to this question is that Table 2 should have the heading "Examples of Evaluation Procedures in Future". What Dr. Odabashi is pointing at, rightly, is that the ROLLSI program is a two-dimensional model, and of course you have to proceed for a long time to obtain what you need in statistics, for statistic treatment of the boat.

However, the idea of using such a model is to find out if this boat, which you have looked upon in the administration and which looks somewhat peculiar, if this boat has a rolling characteristic which is peculiar as well, that means high angles of heel and large accelerations horizontally and vertically. I think it can be used for that purpose to compare vessels, what you could call normal with unnormal. Especially with bilge keels which show clearly here using such a model that the bilge keel would, for instance, add substantially to improving the situation. I'm talking about cargo ships mainly.

Second question CAPSI, this is a deterministic model, and we are working on it at the time being, and this is absolutely a future model. It is working for small



movements, but it doesn't work for what we wanted it to work for, for extreme waves. It is certainly also a one-dimensional model, as you know, so please mind that Table 2 is Examples of Evaluation Procedures in the Future. Thank you.

CHAIRMAN (Middleton):

Thank you, Dr. Dahle, and I see Dr. Odabashi also is shaking his head in the affirmative. (laughter) Are there any other questions with regard to Dr. Dahle's paper? I see no hands raised, and may I take it then that we can go on.

In our timetable this is now the period to discuss Step 2, criteria determination. Again, as I mentioned earlier, in all of these steps these are lists that occurred to the panel and the two chairmen, and we are open for comments as to whether they are adequate, accurate, and of course fill in any blanks that might occur to someone else. On that basis, I would like to have some discussers from the floor who may contribute to this. Dr. Krappinger, please.

KRAPPINGER:

I would like to make a small formal suggestion with regard to the headline of this list. Contrary to probabilistic, it is not physical but deterministic. Probabilistic approach also has to be based on a physical approach, and if you look at the left two headings, static and physical, I would say the static approach is also physical or what is called here "static", it is also physical. I would say "implicit physical and explicit physical." That's the difference between the first column and the second column. I would suggest that this be a little bit adjusted. Thank you.

CHAIRMAN (Middleton):

Thank you, Dr. Krappinger. I might add at this point that those who have made oral presentations you are reminded that you should fill in the comment forms so that your comments may be included in the proceedings, and of course the authors are doing the same with their replies. Anyone else have any comment to make? Yes, Mr. Jens.

JENS:

Mr. Chairman, I have perhaps similar to Professor Krappinger some improvement to say under "physical", ship stability against given external forces, it should perhaps better be "assumed external forces". You don't know, you are not present in the high seas.

In item 8 in the second line it says "to a given ship". Is it not better to say "to a particular type of ship", not one single ship, and 9 maybe all right, but "agreement on appropriate values for the criteria".

CHAIRMAN (Middleton):

Thank you, Mr. Jens. Yes, Professor Gu.

GU:

I don't want to be sitting in this distinguished panel not doing anything, but actually I'm here to assist my colleague here. By the way, with the criteria determination, I think I also have a suggestion to make, that is we have the criteria development following the six procedures. I was giving my comment with regard to Mr. Jens' comment just now on the verification of the requirements for existing ships because you're not in the high seas, so it is not worded properly that way.

My comment would be that in the light of this conference, many good theoretical models have been proposed, and especially some Japanese authors have developed a sort of capsizing alarm to be installed on ships.

So I was proposing an ambitious step to this panel here that why shouldn't we try to verify the criteria developed actually on the high seas, if possible, and that is we should encourage the development of monitoring devices in the ships just like we have in the planes. I heard in airplanes when you have a crash, there is a small black box which you pick up and you know all about what was happening to the plane before the crash. Why don't we have something on the ship that could use some kind of microcomputing technique with the practical model of the criterion developed in the previous phase and let it monitor the ship all the time? That is, we have the ship motions recorded and also the computer model trying to verify the theoretical computations with the practical records, and in case some accident happens we should have some means to save this monitoring device on the ship, that is it should be self-floating or something and can be retrieved, and in that way I think we should have good advances toward verification of the requirements on the existing ships.

The probabilistic results that we get are actually results obtained from the high seas and from the actual recorded movements and actions of the ship, instead of what we guess what happened after the accident with a lot of investigations and that kind. In fact, we do the investigation and we don't know what actually happened at the time of capsizing. Thank you.

CHAIRMAN (Middleton):

Thank you very much, Professor Gu. That certainly is some food for thought, and maybe this will generate another whole new industry with black boxes for ships. I am reminded that in the past there have been monitoring devices, stress gauges, to determine the effect of ship motion on structures, so why not stability?

Admiral O'Dogherty.



O'DOGHERTY:

I quite agree with Professor Gu's proposal. I think that it would be possible at the present stage of knowledge to think of preventive criteria, not only ultimate criteria of safety for the ships to comply with and to be calculated in the design stages or after the ship has been built, but also to have the ship operator some sort of guidance for action to be taken when the situation of the ship is developing into a dangerous possibility, that is to say possibility of loading or ballast in the tanks or doing something modifying the course, or putting the ship into a better situation, something that may be well known beforehand, and maybe even provided by a microcomputer on board.

CHAIRMAN (Middleton):

Thank you very much, Admiral O'Dogherty. Mr. Hormann.

HORMANN:

Thank you, Mr. Chairman. I have a more general comment. I have, I can say, my problems with these three lists, I mean some 14 items which you have listed. The problems relate to the logics behind that order of lists.

As far as I understand that, you come down by outlining first in very general terms areas of research, and then you narrow that down in a certain way to end up with criteria or regulations, as you call that, in Step No. 3.

Now, what I was looking for and what I am actually missing, is the question what have we really to look after. I refer to my paper at the beginning of this conference. I pinpointed one of the, call them "white spots" on the chart of ships. In other words, I am missing a question. I think it should fit very high up in the rank of logical order of these things, a question as to where is additional knowledge needed. It's a very simple question. We ask ourselves for what types of ships have we got criteria and which types are not catered for or which types are catered for in a very general way, and this general way might be of a kind which appears to us not quite convincing for special cases.

Again, as just one example, take the larger container ships, or take it in a more general way. I don't think that anywhere in the world we have a clear statement what is necessary in terms of stability criteria for all general cargo ships and specialized cargo ships over 100 meters in length.

In former years, the idea was promulgated that one could more or less apply A-167, but I think more and more the idea was implanted that this obviously is not enough, at least not if we look at the present designs.

Therefore, to me it appears to be a very big white spot on the chart, and if you compare that with geographics you could even

say there is a whole continent missing, and not the smallest one. If you could take that into account in the further discussions here, I would be happy. Thank you.

CHAIRMAN (Middleton):

Thank you, Mr. Hormann. Indeed, when we put together this chart, we assumed that you would enter the chart at any point along its length, depending on the state of knowledge and state of research at the time. I believe this is a point that was made by Mr. Manum when he indicated that Step No. 5 was probably the most important in that much research has gone on in the theoretical sense, but there is a gap between the theoretical model and the practical criterion, and that we should not begin all over again by going back to Step 1 for all of the research that has already been done.

So the point is to enter wherever you are with regard to research, whether it's in development or in determination or, finally, after it has been determined whether or not there is a need for requirements.

It is a good point that the minutes of the meeting or the report should reflect that we should be looking where additional knowledge is needed in many areas, and as has been pointed out, and we'll find later on in one of our later presentations, that there is indeed a large realm of requirements under IMO or the areas where some work has been done, but they are not the know-all see-all today and that there are some other areas where the present requirements don't cover types of ships, but I shouldn't be talking, you should.

Any other discussion? Yes, Mr. Helwig.

HELWIG:

I would like to make a small note on the use of statistical material. I would like to suggest that when we are going to continue statistical material as a basis for criteria determination, that we should separate the material gathered by fishing boats from the other ones because there is an absolutely different way they operate. Thank you.

CHAIRMAN (Middleton):

Thank you, Mr. Helwig. Indeed, this is the case. When we talk about ship types, we mean that certain requirements for one ship type are inadequate or unnecessary, one way or the other for other ship types. Any other discussion? Yes, Dr. Kastner.

KASTNER:

Mr. Chairman, I am not sure about point 3, theoretical model of a criterion. In my opinion, there is no need to look for just one criterion. I don't know if that is the reason for that point, and then even after that, I mean that's oversimplified. And then after that again this theoretical model is that meant in point 4 for the model of a

criterion, so this again will be simplified. So, in my view, we don't need a criterion, and if we look for that we won't ever find it. We only need a procedure we might agree upon, find out, say, criteria for different ship types and work on the same basis with a common procedure, but not with just one criterion. Thank you.

CHAIRMAN (Middleton):

Thank you, Dr. Kastner. I believe that it was not our intention. If you look at item 8, it says "choice of specific criteria applicable to a given ship", and when we spoke of a model of "a criterion" we mean taking each or every, whatever is necessary. If it is better expressed by some other form of words, we will take that into account so that what's intended will be expressed. Thank you, Sir.

Mr. Cleary.

CLEARY:

Mr. Chairman, gentlemen: I, too, would like to return to Monday afternoon. The paper that I gave at this conference did not give these gentlemen the opportunity to read the one I referred to, which was the situation that I found myself in speaking about research. So I had to speak to two papers, one of which was in the Society of Naval Architects Bulletin "Marine Technology", July of this year, in which I listed 34 ways an intact ship may capsize. Without trying to be a mathematician or a scientist, I call myself a "political naval architect" nowadays because we have to stand in the middle between the research man and the captain of the ship and try to explain what the research man says to the captain of the ship.

Speaking to some of the previous discussion, we obviously will not be able to take care of so many different hazards for an intact ship with one criterion, so the language that has been used by our committee has to be thought of not as fully specific but as general for us to think about. I think especially in the newer types of ships where the older formulas which we really have confidence with experience are no longer applicable, we then have to go down the list of 34 hazards if that is applicable and find out which hazard is greater, and then try to make a limiting criterion or two or three limiting criteria.

I think we should think very flexibly because it will vary from one industry to the next. It will vary from one type of ship to the next. It will vary with geometry. The important thing is to keep going what has been started. Thank you.

CHAIRMAN (Middleton):

Thank you, Mr. Cleary. I believe at this point, since there will be additional time for some further discussion after we've heard the other two papers, that this would

be a logical time now to move on to papers submitted first by Mr. Jens of IMO and, at the same time, we may look at Step 3 on the slide whenever it's free from his presentation or figures. Mr. Jens, please.

JENS:

(Presentation)

CHAIRMAN (Middleton):

I think I have to interrupt you now because we have some questions that I will need your answers to, and I think you covered this, and of course I hope that most of the participants have had an opportunity to read your paper, so if I may I would now like to ask. We have a written comment by Dr. Odabashi, and I would like to have him present his comment. Dr. Odabashi?

ODABASHI:

Mr. Chairman, I'm addressing the questions to Mr. Jens. Actually Mr. Jens on behalf of IMO, so he may not be the right person to answer them, I don't know.

The questions actually are rather funny, have shortcomings in both derivation and implementation of certain criteria. We have always heard that Resolution 167 is a statistically derived criterion, but when you look at it you just wonder what part of it is statistics actually. For example, has anyone carried out a significance test on the variables chosen, like  $GZ_{max}$  or .... area under the curve up to 30 degrees and 40 degrees? Has anybody looked at it whether these variables are independent or they are interrelated? Has anybody looked at them coupled with hull form shape parameters like in one of the lectures we have received during the symposium?

So to call something "statistical", one has to carry out statistics in a proper fashion, and to me the only thing we have seen so far in so many years about Resolution A-167 is that we have just seen a lot of histograms based on frequency values.

The second part is related to the weather criteria which we are seeing here. I have some indirect involvement with it. In that we are very meticulous. We require extreme accuracy in the determination of righting levers, and yet for the calculation of quantities of wind and gust moments or rolling gust, very ad hoc and devious methods are generally being employed. One wonders the wisdom, then what is the necessity to be so accurate in the calculation of righting levers.

JENS:

Thank you, Dr. Odabashi. In respect to the first question, I can tell you we have made these. As you said, you have seen them in the paper, just the parameters in itself, not any coupling. They were done independently. They were ship form parameters. You are right. We have had at that time coast-

ers mainly. Eighty to 90 percent were coastal ships and they were, I think, the form parameters, very similar, on the one hand, and then the fishing vessels, on the other hand. I personally always, when I discuss the matter say there may be a second step after 20 years to make analysis because now you have fast ships, etc., and that should be evaluated. I completely agree with you and there may be a time to do it.

In respect to the second question, I fully agree by not wanting to put IMO in an awkward position. Perhaps Dr. Rakhmanin as the Chairman of the group is more competent to talk about this than I myself.

RAKHAMANIN:

Thank you, Mr. Jens, for conveying this question to me. (laughter) I would like certainly to answer this question. First of all, I would like to say that the decision of the *ad hoc* group in IMO was to start deliberating this weather criteria. This decision was based upon the national practices existing where this was a criteria during the long time period, and in these national requirements, for instance, requirements which existed in Japan, there is a full and proper guidance about treating this matter concerning the accuracy. Certainly in these calculations we do not use the accuracy. Certainly in these calculations we do not use the accuracy, minutes or seconds, for rolling angles, etc., and so all values involved in this criteria are certainly assessed with the proper accuracy which are approximately equal to each other. Thank you.

CHAIRMAN (Middleton):

Thank you very much. May we then move on to the paper by Mr. Arndt dealing with some studies of stability regulations, over 20 years' experience in the West German Navy. Mr. Arndt, Please.

MR. ARNDT:

(Presentation)

CHAIRMAN (Middleton):

Thank you, Mr. Arndt, and in particular for your brevity. We do have a written comment. However, it is quite long and the person commenting is not here, so this will be one of the surprises from the conference. The comments will be included I believe in the proceedings with Mr. Arndt's answer. Our floor is open if anyone has any specific questions to address to Mr. Arndt. I see no volunteers.

I believe that brings us then to Step 3, the development of regulation. Sorry, Professor Kastner.

KASTNER:

Thank you very much. Mr. Chairman, may I present my written contribution to Mr.

Arndt? Thank you. How much time do you give me, Mr. Chairman?

CHAIRMAN (Middleton):

Three minutes.

KASTNER:

That's fine. Thank you very much. Looking at all the forces acting on the body and summing up is a basic requirement in mechanics. This is the idea behind Wendel's moment balancing approach as it was put into practice already 20 years ago. However, for the practical way of doing so, this is to devise criteria on that basis, one has to agree on certain assumptions since regulation needs are well defined. This had been done by Wendel and his group according to the knowledge at that time. He devised also research work to give a better foundation.

In fact, since then, those rules do not only serve as stability regulations but also as guidelines for designers.

Why not ask IMO and national authorities to go ahead and just devise recommendations in the same way Wendel did 20 years ago, and I do not mean with just the same method, but with all the present knowledge included or applied.

I want to add I do not really expect Mr. Arndt to answer that question, but I want to put it to the panel and the audience. Thank you very much.

CHAIRMAN (Middleton):

Thank you, Dr. Kastner. Indeed, you fell within the time or you would have heard my buzzer. That is a good question to open up a discussion. Would Mr. Jens like to address that? Mr. Jens.

JENS:

As IMO was mentioned I thought I was to respond to this. Of course, as I said earlier, in the subcommittee 25 to 30 countries collaborate, and the method is this, that proposals are made not from the public or from the street but from the member governments or organizations with consultative status. So if Dr. Kastner would like to pursue this idea, perhaps you should address yourself to the Ministry of Transport, which then the representatives bring it forward in IMO, and this is of course the same way for any other country, and they may take with them the scientists and the people who are behind the idea.

I was cut short. I wanted to say that as advisors in the various national delegations, a lot of people from universities and specialized areas are participating and giving their expert advice through the committees. This is true also for the Subcommittee on Subdivision Stability and Load Lines.

As I said, we have a host of information of model tests submitted by various national delegations and I think that should go on. We should have a lively exchange of

views. A number of administrations devise funny requirements which nobody can work with. So we have to take care of, on the one side, the state of the art presented by a forum like this and also the state of knowledge and knowledge on the people on board.

This is I think the situation, so perhaps the proposal of Dr. Kastner and others should be fed through the national delegations to be taken up by the organization.

CHAIRMAN (Middleton):

Thank you very much, Mr. Jens. Any other comments? If there are no comments in regard to the two papers, now we go back to Step 3, development of regulation. These were examples of how regulations have been developed over the years. While these regulations, in particular those of IMO, are now part of requirements, this does not mean that they as requirements keep abreast of what is happening in a development of new types of ships, and even they were not the last word for the types that they address. So we can't say the job is finished but, rather, that we have only reached one of the levels of requirements for meeting safety and that in the future work in development of requirements and regulations, there may be some improvement on those that are now a matter of record in various international conventions and resolutions.

As I said before, we still have five or rather four other points, and that if anyone has any suggestions to improve, to amplify, or to add to these points, these are welcome. Either the hour is late or else this is complete. If that is the case, we do have one other comment and that is one by Mr. Cleary with regard to, I think, the overall philosophy. He made some comments earlier but perhaps Mr. Cleary now would like the opportunity to address his written comment. Mr. Cleary.

CLEARY:

Mr. Chairman, gentlemen: I wrote this comment yesterday, an enthusiasm idea, because I saw that we were on the fourth day of a very productive five-day conference, and if it were possible we could say it has even exceeded what we did in 1975, which makes it a great success in my mind, and I began to think, well, where do we go from here?

You have just heard a discussion about IMO. Those of us who attend the subcommittee meetings at IMO, discuss research often, we discuss the results of the research. What we are trying to do, though, is to reduce the great reports that are put together in research form and try to make a very wide criterion which can be used for a large group of ships.

When we speak in terms of research, we often have to take unique aspects of a ship into consideration that only comes into play at IMO very seldom because we are trying to

find a broad formula.

So my remarks are toward the objective that we should continue what has been so beautifully started in the first and second International Stability Conferences. It has to be at minimum cost, as we all know. We need some sort of a method of communication-writing letters or having further meetings.

It has also been discussed informally this week that if we set a specific date, two years or four years away, we run the risk that perhaps the research will not have gone far enough and we will spend a lot of money making another conference at the wrong time. Yet, we want not to lose this momentum that we now have.

So I wrote two pages which perhaps is too much to read. First of all, on the first page I listed the fact that problems of stability can be categorized in many different ways. If we wish to think of them as functional, we could pick out four ways: research, design, criteria, operation.

Another large group, looking at it in a different way from the technical category, is broken into also four major points: philosophy; fundamentals of stability; unique aspects which would be from single hulls, multi-hulls and surface effect ships, that sort of thing, unique aspects is the third one; and, finally, measurement of stability which can be both initial and continuous.

A third major way one could look at stability is to divide it into the various conditions. We can look at intact stability on any ship or floating system, we can look at damage stability, and we can look at all the various external forces such as waves, wind, icing, anchors, etc.

The suggestion I make, subject to the comments by all the members as well as the panel, has four parts:

First, this conference is not an international legal conference. We are quite free to make our own recommendations based on our own thoughts. I would hope that we could have general agreement by all the members of this conference that we do need to continue what has been started in, say, three to five years at some location which has to be determined of course by who is willing to undertake the cost, etc.

The second point is that we should make a strong recommendation to ourselves for correspondence between interested members in similar fields of research. We have noted that we have similar fields of research and yet many different fields of research, as we have shown by the agenda for this week.

The third point, to try to get an idea of the progress over the next couple of years to decide whether another conference might be in three years or five years or four. We need to get together and talk, or some few of us need to get together and talk..

I put down a suggestion that after looking at the list of participants here, almost all of the nations represented have a delegation that comes to the IMO subcommit-

tee which is now called Stability, Load Lines and Safety of Fishing Vessels. We have not forgotten about subdivision, though.

We have to be careful in trying anything with IMO, as Mr. Jens would be quick to remind us. We cannot do anything officially because that is an international legal setup under the United Nations. But my suggestion is that we can take advantage of the fact that we as human beings are physically in London, or if somebody is representing you or your nation is there, we might meet informally as an ad hoc group to review the progress of research either for a half a day or a few hours after one of these weekly meetings, or perhaps before. That is certainly subject to all of the thoughts that the delegates would have.

Finally, my plea, the last one, is that the areas of stability research to be considered should be selected from any one of the categories that I listed in my first page of my document. Whether they are functional, technical or conditions of stability, let each research delegate select his own way of approaching this and then let the International Committee decide how they should put them in together for the next conference. Thank you.

CHAIRMAN (Middleton):

Thank you very much, Mr. Cleary. Would anyone else like to comment, in particular on the suggestions made by Mr. Cleary and, of course, hoping to tie his suggestions in with what we have attempted to show in our charts regarding the procedure for research? Yes, Prof. Ursell.

URSELL:

Mr. Chairman, the proposition which we have heard may have considerable merits, but I am never very happy when we are expected to reach an agreement on something which has not been seen by all the members, which has not even been shown on the board but which has just been read out to us.

Now I am in a privileged position as a member of the International Technical Committee. I had seen these propositions at one o'clock today and this has given me some time at least to think about them.

Now, having looked at these, I would say the time, maybe three years or five years, we shall have to see how the situation develops, but, doubtless, it's that order of magnitude, and so we will just have to wait for that.

Similarly, Proposition B, which again I have in front of me and you do not, is correspondence between interested parties, again seems to be unobjectionable.

I am not nearly so happy about points C and D because I do not see that the IMO is the appropriate kind of body to deal with this sort of thing. The members of the IMO subcommittee need to be aware of the results of past research, and some members or

advisors may be leaders in research. But, on the other hand, there are some delegations that do not include research workers and are not necessarily well informed about research trends.

I should, therefore, be much happier if the ad hoc committee were independent of the IMO, and we may say that as far as the past two conferences are concerned, of course they have not been connected with the IMO, I think they have been very successful.

Now, Section D suggests that the topics should be chosen from three fields. Here, again, I would like to say the results of future research are unpredictable and, therefore, I personally would not wish the ad hoc committee to be restricted in the way suggested by Mr. Cleary. Thank you, Mr. Chairman.

CHAIRMAN (Middleton):

Thank you, Prof. Ursell. Any other comments? Perhaps since this is a proposal or perhaps even, sorry, Mr. Manum.

MANUM:

Thank you very much, Mr. Chairman. This is such an important issue that I think we should have more than one comment. My comment is that in item B, Cleary says that agreement for correspondence between "interested parties". Then I will question who are the "interested parties". Of course, this week we have been together mostly researchers, but also they are varying very much. Some research institutes deal solely with basic research and they handle that research by themselves. You have more commercialized research companies dealing more with development. We could call it more correctly, but in our country we are very broad in the interpretation of the word "research", so we also call that research.

In our country we also have other bodies. We have bodies that support research. They try to outline perhaps beforehand in what region the research should go or in what area the money is allocated, and then they apply for research programs, and then they come in, evaluate the problems and decide and spread the money or put all the money into one or two research institutes.

So who are the "interested parties"? It's a difficult question. Who should know what is going on? Who should participate in this correspondence? It is very difficult to decide beforehand. That is why I proposed on the first day of this conference that there should be some kind of a data bank or data base where you immediately could be informed or brought up to date on all research that has been initiated. I am not so much concerned about results because those you can get from various sources, but today to my knowledge there is no place where you can be informed about research initiated in another country.

So I feel that we have to think very

carefully about how we create international cooperation in the future regarding research in order to gain progress, in order to make sure that all the interested parties in this field are properly brought up to date at all times when they have to make decisions, and that time can be varying in various countries. There are specific dates in Norway when we always make those very, very important decisions for the researchers and also, of course, in the future for the seamen because that is the main purpose of all our research in this field. Thank you very much.

CHAIRMAN (Middleton):

Thank you, Mr. Manum, for your comments with regard to this. May I suggest that of course we will not be taking decisions on these at this meeting or in this panel but that the comments that have been received will be taken into account by the members of the ITC in response to what has been said at the various sessions. We have to remember that the ITC is not an organization that has funds of its own but it is all individual volunteers who contribute to it, so we can't set up a program of having periodicals going out or other things unless someone volunteers the funds. But I'm sure that the members of the ITC, two of whom have just spoken, will perhaps take into account what has been said here and, in turn, take action as they deem fit in order to advance research in the future and also for the scheduling of conferences, and if they deem it wise to set up ad hoc groups. I'm sure Mr. Cleary pointed out that any such ad hoc group being held in London in connection with an IMO meeting was not intended to link the two together but only by convenience of having people there who would be available to meet under the banner of this organization rather than that of IMO.

Any other comments from the floor?  
Yes, Dr. Dahle.

DAHLE:

Mr. Chairman, I'm slightly provoked by this discussion because it sounds like there is no collaboration and exchange of information among the scientists, which is completely wrong.

Again, as always, to mention Norway, we have really tried hard at this recent stability research, and also the U.K. has responded in a very favorable way and we have had very good and close collaboration with the U.K. with their state ship project and the research in Norway.

We also invited other scientists to Norway, and they come to Norway, and also the scientists from Norway go abroad. However, I am not so sure that the scientists of any country would spend one week in London just to go to the ad hoc group that is proposed by Mr. Cleary. There are a number of scientists in Norway who would not dream of spending one week with IMCO in

London, sorry to say. They have other important matters to attend to.

Also the scientists tend to publish their research in journals, as you know, so there is not a complete lack of communication.

May I add that this kind of international communication is very time consuming. It takes a lot of work to really make it work, and it is also very expensive to travel.

However, may I add a final comment? When big research projects are being planned in other countries, I propose that these people who are involved invite on their own cost some foreign scientists to participate in these discussions. For instance, in Norway, we can provide the money for that, travel money for someone to come to Norway, and I suppose also other countries can do that, so in the preparation of their research they could do that.

Further, Mr. Manum's proposal is supported by me about a data bank where you can go and get information on what is being planned. This has special address to the Eastern countries of the world where it is not so easy to get information, from my experience. Thank you.

CHAIRMAN (Middleton):

Thank you, Dr. Dahle. As I perceived it, I didn't believe Mr. Cleary's proposal was that the scientists would get together to discuss scientific matters, but perhaps this would be some sort of a steering group of the ITC to determine or attempt to determine where the scientists who were doing research had completed some work, or what the present state of research would be at that time in order to help the committee in determining where and when there might be another conference, and also to perhaps provide the information that's suggested by data bank. But, then, that's my perception at the moment and not necessarily the views of everyone here.

Again, I think any comments being made at this point are for the benefit of the ITC and the decisions will come from them. Yes, Dr. Odabashi.

ODABASHI:

Mr. Chairman, I may sound rather rude. I think most of the discussion is rather futile in the sense that one may form a committee whether it's related to ITC or whatever, and any kind of steering can take place with the present conference or without relating to it, nor is it the privilege of a few to organize a conference on stability. So I think the whole argument is essentially futile. Thank you.

CHAIRMAN (Middleton):

Thank you very much. In the absence of any more comment, it is about time. Normally there would be a summary of what has been accomplished or what has been said during

this panel, but I believe it would be better that we each go away with our own impressions of what has taken place and await the distribution of the proceedings of the conference in which all things will be covered rather than me to now repeat or my colleague to repeat what has been said by whom I think at this point. Unless there are no further comments, the panel work has been finished, and this perhaps gives Prof. Kuo a little bit more time to make the closing remarks of the conference. So at this point I would like to thank all of the members of the panel for their presentation of papers, and sorry I had to goad them into going more rapidly, and I hope they didn't omit something that would be useful.

But, again, we are grateful to the work of the distinguished panel that provided us with something to talk about.

More importantly, as a chairman I am happy to have such a large gathering on the last day of the conference, and for the participation of the many members who have spoken from the floor.

Thank you very much, and I think we owe our panel a round of applause. Thank you.

(APPLAUSE)

Panel Session II is now closed.

(END OF PANEL SESSION II)



## FISHING BOATS STABILITY CRITERION, OBTAINED FROM STATISTICAL ANALYSIS OF SHIP LOSSES

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### ABSTRACT

Intact Stability of Fishing Boats is a factor of primary importance for these vessels that have to stay at sea, with all weather conditions. In this paper, a Criterion for Fishing Boats Stability is introduced, based upon the consideration of the service conditions of these ships and the analysis of Stability data of many vessels, including many that have been lost at sea.

### NOMENCLATURE

- B = Ship breadth.  
C<sub>S</sub> = Non-dimensional stability coefficient  
 $e_{\theta} = \int_0^{\theta} GZ.d\theta$  = dynamical stability lever, up to angle  $\theta$ .  
 $e_k$  = dynamical stability lever, up to the angle of vanishing stability.  
GM = Transverse metacentric height.  
GZ = Righting arm.  
KG = Height of center of gravity, above the base line.  
LBP = Length between perpendiculars.  
S<sub>1</sub> = Initial stability coefficient.  
S<sub>2</sub> = Dynamical stability coefficient.  
 $\Delta$  = Ship displacement.  
 $\theta$  = Angle of inclination.  
 $\theta_k$  = Angle of vanishing stability.

### INTRODUCTION

The assessment of Ship stability has evolved in the present century, departing from a situation where stability was taken for granted, due to the inherent stability conditions linked to the choice of ship dimensions and coefficients, to a more sound approach where stability is judged in terms of the values that correspond to certain stability parameters.

In this century, many attempts have been made to establish stability criteria, either for universal application, or for specific ship types. Generally those criteria are based either on fixing minimum values for different stability parameters, or on the evaluation of the ship response to different heeling actions, mainly produced by external forces, such as the combined action of wind and waves, under the assumption of certain extreme conditions.

Aiming to establish a simplified Stability Criterion for deep sea Fishing Boats, the authors have analysed, during a long period of time, the stability conditions of hundreds of ships of all types, including many fishing boats, paying a special attention to those vessels having reliable stability data.

Two types of vessels have been considered as most interesting :

- Ships involved in sea accidents, specially capsized vessels, lost ships or those which were abandoned by their crews, in view of the risk of sinking.



- Ships of good stability and seaworthiness, confirmed by their satisfactory seakeeping conditions.

The analysis of the most important stability parameters, as well as the consideration of the design conditions of these vessels, provide a sound basis for establishing stability criteria with sufficient reliability. On Table 1, the main stability particulars of many fishing boats are included, together with stability information of many ships, either capsized, sunk or lost at sea. The stability data for fishing boats comprise both ships involved in sea accidents, and other vessels considered of safe stability by experts.

In (Ref. 1), an schematic analysis of the effects of the various external forces that influence the environmental service conditions of fishing boats, such as wind, waves and icing, is made, studying at the same time the various heeling effects that may endanger ship safety, such as the loss of stability in waves, the combined heeling effect of wind and rolling, the rudder action, shifting of the cargo, shipping of water on decks, etc.

#### CONSIDERATIONS ON EXISTING STABILITY CRITERIA.

By studying the evolution of the Stability Criteria that have been established in the last decades, a certain progress may be appreciated, as shown by the tendency to cover the following stages.

- Ship stability obtained by building a new ship, without a proper stability study, but based on the dimensions and proportions of a previous vessel of sound stability, as was customary in the past.
- Establishing Stability Criteria relative to setting a minimum value for the transverse metacentric height,  $GM$ , or fixing a maximum permissible value for  $KG$ . One such criterion for fishing boats was proposed by Nickum (Ref. 2).
- Fixing minimum values for various  $GZ$ 's, as in the well known criterion of Rahola (Ref. 3).
- Consideration of the dynamical stability levers, as in the criteria of Rahola and IMCO (Ref. 4).
- Study of the heeling action of a side wind, combined with strong rolling motions. These criteria assume that this effect is the most unfavourable one for Ship Safety.

- Setting of various criteria that apply to different parameters of the stability curve for the ship, in all service conditions, as proposed in (Ref. 5).
- Detailed analysis of the various heeling moments that may take place, in order to make sure that the ship stability provides sufficient margin to withstand the effects that may reasonably appear in service, as proposed by Wendel (Ref. 6)

This evolution of the approach to the evaluation of Ship Stability was brought about by the realization that it is not correct to establish fixed minimum values for various stability parameters in ships of different size, as the possible heeling actions depend on ship size, so that criteria that are based on uniform minimum values, such as those of Rahola and IMCO, are not fully satisfactory.

Many authors, Nickum among others, consider as the most important stability parameters, in assessing the stability of fishing boats, the freeboard and initial stability,  $GM$ , as those parameters are decisive for the safety of the ship.

On fig. 1, the values of  $GM/B$ , as a function of LBP, for Spanish fishing boats, are represented. It may be pointed out that  $GM/B$  tends to decrease with increases of LBP, what may be attributed to the larger values of effective freeboard that correspond to large fishing boats, in comparison with average small flushdecked fishing boats. It may be stated that ships with ample freeboard are able to achieve a satisfactory stability, with moderate or small  $GM$ , but good dynamical stability at large angles, whereas small ships, with a small freeboard, need a rather large  $GM$  in order to achieve a reasonable reserve of dynamical stability.

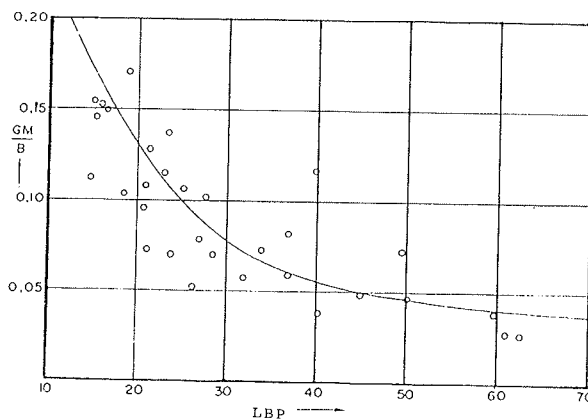


FIG. 1

## A NEW STABILITY CRITERION FOR FISHING

### BOATS

The above considerations lead to the desirability to find a Stability Criterion based on the assessment of the two most important stability parameters, i.e.

- Initial stability, represented by the transverse metacentric height, GM.
- Dynamical stability.

The following considerations were taken into account:

- It is not advisable to establish fixed minimum values for any stability parameter, as those values could be very variable, for ships of different size and service conditions.
- The dynamical stability must consider the dynamical stability lever at large angles, specially the value of  $e_k$ , dynamical lever that corresponds to the angle of vanishing stability.
- It was not considered convenient to set up independent minimum values for the two main parameters, GM and  $e_k$ , as those two variables are mutually interdependent, so that a ship with small freeboard (and  $e_k$ ) needs to have a higher GM, whereas a vessel with ample dynamical stability does not require to have a moderate or high GM.
- From the study carried out in this research, it has been found that, for vessels of the same type, as it is the case in deep sea trawlers, the value of GM/B gives a good indication of the accelerations produced in the vessel at sea.
- It was also found that in most cases the main danger to ship safety does not lie on the combined effect of severe wind and rolling, as the majority of cases of fishing vessels, experiencing capsize, take place either in rough weather, with quartering or following seas, due to the stability loss brought about by those circumstances, or by the effects of other causes, such as shifting of the cargo, icing conditions, water trapped on deck, etc.
- Many of these heeling effects depend on the ship dimensions, so that the stability requirements must be related also to the main dimensions of the ship.
- The stability parameters used in this Criterion are based on GM and  $e_k$ , by using two nondimensional stability coefficients:

$S_1$  = initial stability coefficient.

$S_2$  = dynamical stability coefficient.

The values of  $S_1$  and  $S_2$  are defined as follows:

$$S_1 = 50 \cdot \text{GM/B} \quad (1)$$

$$S_2 = 1000 \cdot e_k / \text{LBP} \quad (2)$$

The constants 50 and 1000 have been chosen empirically, so that the values of  $S_1$  and  $S_2$  are convenient, and at the same time provide a reasonable relative importance to each parameter, in the assessment of the overall ship stability.

The Criterion uses the Stability Coefficient,

$$C_S = S_1 + S_2 \quad (3)$$

where  $S_1$  must be taken as the smallest of two alternative values, i.e.

$$S_1 = 50 \cdot \text{GM/B}$$

or

$$S_1 \text{ limit} = 7.7 \cdot -0.115 \cdot L + 0.0005 \cdot L^2 \quad (4)$$

where  $L$  = LBP, meters.

The nondimensional Stability Coefficient,  $C_S$ , is the sum of two parts, which depend, respectively, on initial and dynamical stability. In the assessment of the ship stability, by means of  $C_S$ , there may occur two cases:

- $S_1$ , as given by  $50 \cdot \text{GM/B}$ , is smaller than  $S_1$  limit (4). In this case the value of GM is not excessive. The requirement of dynamical stability, in order to comply with the Criterion, will be more severe for ships with smaller GM, what shall result in good stability conditions in all cases.
- $S_1 = 50 \cdot \text{GM/B}$  is larger than  $S_1$  limit. In those cases, the demand for dynamical stability, established by the Criterion, will represent a constant minimum value for each ship length. As in these cases the initial stability of the ship may be considered sufficient, it is reasonable to accept a qualified minimum value for the ship dynamical stability.

## APPLICATION OF THE STABILITY CRITERION

- For a deep sea fishing boat to have adequate stability, the Stability Coefficient,  $C_S = S_1 + S_2$ , must satisfy the condition:

$$C_S \geq 3.7 + \frac{230}{LBP} - 0.02 \cdot LBP$$

where LBP is expressed in meters.

- This condition must be fulfilled in all service conditions of the ship. If the ship is intended to operate in Arctic areas, the effect of icing on decks, superstructures and rigging must be taken into consideration.
- In the calculation of the values of  $S_1$  and  $S_2$ , the values of GM and the righting levers, GZ's, must be corrected by the effect of free surfaces.
- If the angle of vanishing stability,  $\theta_k$ , exceeds 80 degrees, the dynamical stability lever,  $e_k$ , to apply for the evaluation of  $S_2$ , shall be:

$$e_k = \int_0^{80^\circ} GZ \cdot d\theta$$

- If the vessel has an abnormal proportion of superstructures, or very small draught, it could be convenient to check the ship stability by an additional criterion, such as those included on (Ref.21) and (Ref.22), in order to allow for the effects of severe wind and rolling.

## CONCLUSIONS

1. Fig. 2 shows the values of  $C_S$ , as a function of LBP, for all the vessels included on the Appendix, whose main dimensions and stability parameters are shown on Table 1. It may be seen that the proposed Stability Criterion correlates well with the evaluation of the stability of the great majority of these ships, providing a good guidance for the assessment of Ship Stability.
2. The minimum values of  $C_S$ , as a function of LBP, have been chosen by statistical analysis of the stability data of ships, either capsized or satisfactory, making sure at the same time that the requirements of this Criterion enable the ship to withstand the loss of stability with following waves, which constitutes the maximum danger for fishing boats, in most cases.

3. Generally speaking, it may be stated that the optimum stability conditions for fishing boats may correspond to those ships that satisfy this Criterion with GM values that correspond to values of  $S_1$  that are somehow smaller than  $S_1$  limit, so that the corresponding requirement of  $e_k$  ensures a reasonable dynamical stability, while the ship has a moderate GM value, leading to normal rolling motions and good seakeeping qualities.
4. The application of this Criterion must be followed by more detailed stability studies, when the special circumstances of ship service indicate the convenience to do so.

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- ◇ CS VALUES FISHING BOATS
- △ CS STABILITY DEEMED INSUFFICIENT BY EXPERTS
- CS STABILITY ACCIDENTS
- ☆ CS AT TIME OF LOSS

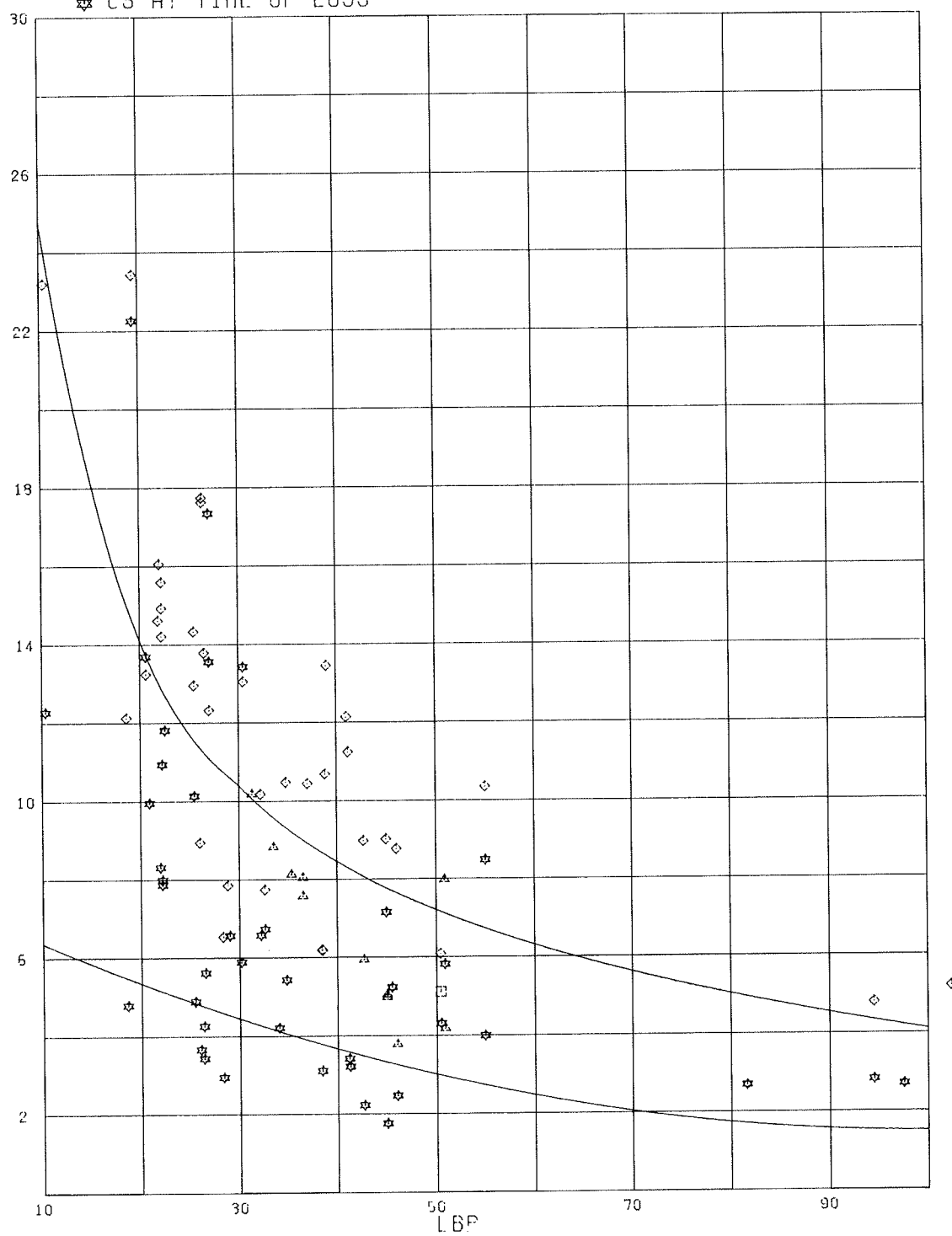


FIG. 2

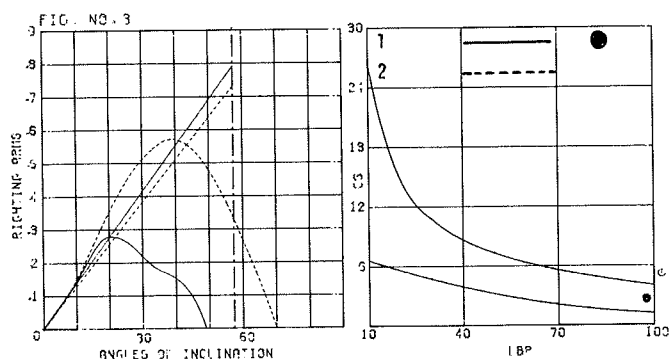
TABLE 1

| SHIP | LBP(M)  | B(M)   | D(M)   | TM(M) | DISP(T) | GM(M) | EK    | S1 LIM | S1    | S2     | CS     | CS MIN | FIG. | STABILITY    |
|------|---------|--------|--------|-------|---------|-------|-------|--------|-------|--------|--------|--------|------|--------------|
| 1    | 97.530  | 16.230 | 9.600  | 7.620 | 7915.0  | 0.790 | 0.142 | 1.240  | 2.434 | 1.459  | 2.699  | 4.103  | 3    | LOSS         |
| 2    | 102.400 | 17.520 | 12.190 | 7.920 | 8439.0  | 0.730 | 0.413 | 1.167  | 2.083 | 4.032  | 5.199  | 3.898  | 3    | SATISFACTORY |
| 3    | 30.180  | 7.620  | 0.0    | 3.650 | 482.0   | 0.630 | 0.053 | 4.685  | 4.134 | 1.755  | 5.839  | 10.717 | 4    | LOSS         |
| 4    | 38.810  | 9.300  | 0.0    | 3.920 | 704.0   | 1.000 | 0.260 | 3.990  | 5.376 | 6.694  | 10.634 | 8.850  | 4    | SATISFACTORY |
| 5-1  | 41.150  | 6.400  | 3.610  | 0.0   | 0.0     | 0.230 | 0.067 | 3.614  | 1.797 | 1.621  | 3.413  | 8.466  | 5    | LOSS         |
| 5-2  | 41.150  | 6.400  | 3.610  | 0.0   | 0.0     | 0.550 | 0.306 | 3.814  | 4.297 | 7.430  | 11.245 | 8.466  | 5    | SATISFACTORY |
| 6-1  | 50.400  | 8.000  | 4.400  | 3.870 | 1342.0  | 0.610 | 0.147 | 3.174  | 3.812 | 2.915  | 6.039  | 7.255  | 6    | SATISFACTORY |
| 6-2  | 50.400  | 8.000  | 4.400  | 4.020 | 1375.0  | 0.656 | 0.098 | 3.174  | 4.100 | 1.939  | 5.113  | 7.255  | 6    | ACCIDENT     |
| 7    | 35.330  | 6.700  | 4.320  | 3.600 | 275.0   | 0.780 | 0.136 | 4.251  | 5.821 | 3.850  | 8.122  | 9.503  | 7    | INSUFFICIENT |
| 8    | 36.510  | 6.850  | 4.420  | 3.750 | 370.0   | 0.800 | 0.141 | 4.163  | 5.839 | 3.874  | 8.042  | 9.269  | 7    | INSUFFICIENT |
| 9    | 31.320  | 6.310  | 3.940  | 3.290 | 255.0   | 0.860 | 0.175 | 4.537  | 6.815 | 5.591  | 10.180 | 10.417 | 7    | INSUFFICIENT |
| 10   | 39.000  | 6.550  | 4.430  | 3.630 | 373.0   | 0.885 | 0.370 | 3.976  | 6.756 | 9.477  | 13.452 | 8.817  | 8    | SATISFACTORY |
| 11   | 37.000  | 7.000  | 4.590  | 4.000 | 440.0   | 0.760 | 0.233 | 4.130  | 5.429 | 6.311  | 10.440 | 9.176  | 8    | SATISFACTORY |
| 12   | 41.000  | 6.850  | 4.750  | 4.150 | 595.0   | 0.750 | 0.341 | 3.826  | 5.474 | 8.311  | 12.137 | 8.490  | 8    | SATISFACTORY |
| 13   | 36.500  | 6.850  | 0.0    | 0.0   | 353.0   | 0.715 | 0.124 | 4.169  | 5.219 | 3.397  | 7.556  | 9.271  | 9    | INSUFFICIENT |
| 14   | 33.500  | 6.700  | 0.0    | 0.0   | 267.0   | 0.767 | 0.148 | 4.409  | 5.587 | 4.413  | 8.821  | 9.896  | 9    | INSUFFICIENT |
| 15-1 | 42.670  | 7.620  | 3.580  | 3.320 | 608.0   | 0.270 | 0.019 | 3.703  | 1.772 | 0.451  | 2.223  | 8.237  | 10   | LOSS         |
| 15-2 | 42.670  | 7.620  | 3.580  | 0.0   | 427.0   | 0.610 | 0.225 | 3.703  | 4.003 | 5.275  | 8.979  | 8.237  | 10   | SATISFACTORY |
| 15-3 | 42.670  | 7.620  | 3.580  | 0.0   | 445.0   | 0.430 | 0.133 | 3.703  | 2.822 | 3.117  | 5.939  | 8.237  | 10   | INSUFFICIENT |
| 16   | 38.400  | 8.000  | 4.600  | 3.610 | 522.0   | 0.239 | 0.062 | 4.021  | 1.494 | 1.614  | 3.108  | 9.922  | 11   | LOSS         |
| 17   | 38.400  | 8.000  | 4.600  | 0.0   | 513.0   | 0.413 | 0.139 | 4.021  | 2.581 | 3.613  | 6.194  | 8.922  | 11   | SATISFACTORY |
| 18   | 54.900  | 8.850  | 4.110  | 3.980 | 1400.0  | 0.395 | 0.096 | 2.894  | 2.237 | 1.750  | 3.987  | 6.791  | 11   | LOSS         |
| 19-1 | 20.600  | 6.400  | 2.860  | 2.280 | 145.0   | 0.920 | 0.159 | 5.543  | 7.187 | 7.703  | 13.245 | 14.453 | 12   | SATISFACTORY |
| 19-2 | 20.600  | 6.400  | 2.860  | 2.040 | 122.0   | 0.830 | 0.168 | 5.543  | 6.484 | 8.145  | 13.638 | 14.453 | 12   | LOSS         |
| 20   | 25.450  | 6.900  | 3.650  | 2.790 | 218.0   | 0.870 | 0.235 | 5.097  | 6.304 | 9.232  | 14.329 | 12.228 | 12   | SATISFACTORY |
| 21   | 20.900  | 5.050  | 2.750  | 2.620 | 165.4   | 1.007 | 0.093 | 5.515  | 9.951 | 4.478  | 9.953  | 14.287 | 13   | LOSS         |
| 22   | 45.500  | 8.170  | 5.250  | 3.190 | 860.0   | 0.150 | 0.196 | 3.503  | 0.918 | 4.314  | 5.232  | 7.845  | 13   | LOSS         |
| 23   | 22.500  | 6.800  | 3.250  | 2.590 | 176.0   | 0.674 | 0.154 | 5.366  | 4.956 | 6.954  | 11.810 | 13.472 | 14   | LOSS         |
| 24   | 29.000  | 5.670  | 2.590  | 1.980 | 165.0   | 0.584 | 0.052 | 4.786  | 5.150 | 1.790  | 6.555  | 11.051 | 14   | LOSS         |
| 25   | 50.440  | 8.380  | 4.570  | 3.758 | 917.0   | 0.330 | 0.118 | 3.172  | 1.969 | 2.334  | 4.303  | 7.251  | 14   | LOSS         |
| 26   | 25.450  | 6.900  | 3.650  | 2.330 | 233.0   | 0.790 | 0.128 | 5.097  | 5.725 | 5.037  | 10.134 | 12.228 | 15   | LOSS         |
| 27   | 34.000  | 7.300  | 3.490  | 3.060 | 473.0   | 0.420 | 0.045 | 4.363  | 2.877 | 1.335  | 4.212  | 9.785  | 15   | LOSS         |
| 28-1 | 46.000  | 8.800  | 3.900  | 3.530 | 1026.0  | 0.750 | 0.244 | 3.458  | 4.261 | 5.306  | 8.774  | 7.780  | 16   | SATISFACTORY |
| 28-2 | 46.000  | 8.800  | 3.900  | 3.530 | 1026.0  | 0.290 | 0.098 | 3.458  | 1.648 | 2.127  | 3.775  | 7.780  | 16   | INSUFFICIENT |
| 28-3 | 46.000  | 8.800  | 3.900  | 3.530 | 1026.0  | 0.180 | 0.066 | 3.458  | 1.023 | 1.441  | 2.464  | 7.780  | 16   | LOSS         |
| 29-1 | 27.000  | 6.400  | 3.200  | 2.420 | 209.0   | 0.690 | 0.232 | 4.960  | 5.391 | 8.600  | 13.560 | 11.679 | 17   | LOSS         |
| 29-2 | 27.000  | 6.400  | 3.200  | 2.790 | 262.0   | 0.710 | 0.199 | 4.960  | 5.547 | 7.361  | 12.320 | 11.679 | 17   | SATISFACTORY |
| 30-1 | 55.000  | 9.600  | 3.960  | 3.560 | 1332.0  | 1.010 | 0.410 | 2.888  | 5.260 | 7.457  | 10.345 | 6.782  | 18   | SATISFACTORY |
| 30-2 | 55.000  | 9.600  | 3.960  | 3.560 | 1332.0  | 0.900 | 0.307 | 2.888  | 4.687 | 5.538  | 8.475  | 6.782  | 18   | LOSS         |
| 31-1 | 34.800  | 7.200  | 3.490  | 2.510 | 363.0   | 0.560 | 0.229 | 4.304  | 3.839 | 6.589  | 10.473 | 9.613  | 19   | SATISFACTORY |
| 31-2 | 34.800  | 7.200  | 3.490  | 2.750 | 403.0   | 0.400 | 0.092 | 4.304  | 2.778 | 2.651  | 5.429  | 9.613  | 19   | LOSS         |
| 32-1 | 44.960  | 7.700  | 5.290  | 3.500 | 700.0   | 0.840 | 0.246 | 3.540  | 5.455 | 5.479  | 9.019  | 7.916  | 20   | SATISFACTORY |
| 32-2 | 44.960  | 7.700  | 5.290  | 3.500 | 700.0   | 0.630 | 0.162 | 3.540  | 4.091 | 3.610  | 7.150  | 7.916  | 20   | LOSS         |
| 33-1 | 45.000  | 8.200  | 3.300  | 3.300 | 870.0   | 0.180 | 0.030 | 3.538  | 1.098 | 0.650  | 1.757  | 7.911  | 21   | LOSS         |
| 33-2 | 45.000  | 8.200  | 3.300  | 1.820 | 426.0   | 0.350 | 0.128 | 3.538  | 2.134 | 2.849  | 4.933  | 7.911  | 21   | INSUFFICIENT |
| 33-3 | 45.000  | 8.200  | 3.300  | 3.250 | 854.0   | 0.390 | 0.120 | 3.538  | 2.378 | 2.671  | 5.049  | 7.911  | 21   | INSUFFICIENT |
| 34-1 | 26.320  | 6.750  | 3.400  | 2.600 | 220.0   | 0.190 | 0.053 | 5.020  | 1.407 | 2.028  | 3.435  | 11.912 | 22   | LOSS         |
| 34-2 | 26.320  | 6.750  | 3.400  | 3.130 | 296.0   | 0.730 | 0.335 | 5.020  | 5.407 | 12.719 | 17.739 | 11.912 | 22   | SATISFACTORY |
| 35-1 | 50.840  | 9.000  | 5.350  | 3.870 | 1218.0  | 0.260 | 0.139 | 3.145  | 1.444 | 2.730  | 4.175  | 7.207  | 23   | INSUFFICIENT |
| 35-2 | 50.840  | 9.000  | 5.350  | 2.750 | 832.0   | 0.280 | 0.326 | 3.145  | 1.556 | 6.417  | 7.973  | 7.207  | 23   | INSUFFICIENT |

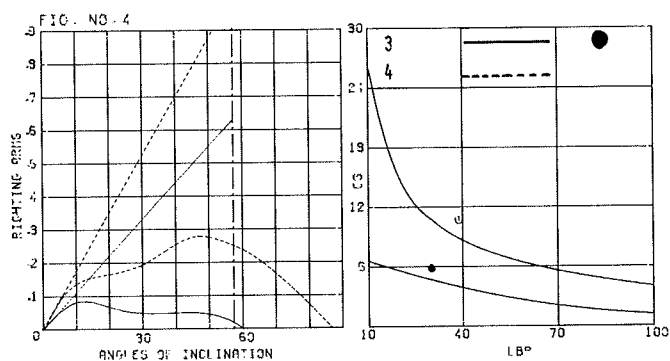
TABLE 1

| SHIP | LBP(M) | B(M)   | D(M)  | TM(M) | DISP(T) | GM(M) | EK    | S1 LIM | S1     | S2     | CS     | CS MIN | FIG. | STABILITY    |
|------|--------|--------|-------|-------|---------|-------|-------|--------|--------|--------|--------|--------|------|--------------|
| 35-3 | 50.840 | 9.000  | 5.350 | 3.500 | 1091.0  | 0.290 | 0.214 | 3.146  | 1.611  | 4.206  | 5.817  | 7.207  | 23   | LOSS         |
| 36   | 81.600 | 13.200 | 7.900 | 5.430 | 4575.0  | 0.270 | 0.148 | 1.645  | 0.871  | 1.814  | 2.696  | 4.887  | 24   | LOSS         |
| 37-1 | 94.480 | 14.020 | 8.480 | 7.070 | 6420.0  | 0.580 | 0.330 | 1.298  | 2.068  | 3.497  | 4.795  | 4.245  | 24   | SATISFACTORY |
| 37-2 | 94.480 | 14.020 | 8.480 | 7.070 | 6420.0  | 0.580 | 0.144 | 1.298  | 2.068  | 1.523  | 2.821  | 4.245  | 24   | LOSS         |
| 38-1 | 22.200 | 6.000  | 3.100 | 2.250 | 148.0   | 0.810 | 0.123 | 5.393  | 6.750  | 5.550  | 10.943 | 13.616 | 25   | LOSS         |
| 38-2 | 22.200 | 6.000  | 3.100 | 2.530 | 195.0   | 0.850 | 0.196 | 5.393  | 7.083  | 8.816  | 14.210 | 13.616 | 25   | SATISFACTORY |
| 39-1 | 22.200 | 5.800  | 2.750 | 1.950 | 135.0   | 0.520 | 0.078 | 5.393  | 4.483  | 3.506  | 7.988  | 13.616 | 26   | LOSS         |
| 39-2 | 22.200 | 5.800  | 2.750 | 2.540 | 186.0   | 0.740 | 0.212 | 5.393  | 6.379  | 9.532  | 14.926 | 13.616 | 26   | SATISFACTORY |
| 40-1 | 19.400 | 4.400  | 2.560 | 1.570 | 64.0    | 0.890 | 0.345 | 5.657  | 10.114 | 17.776 | 23.433 | 15.168 | 27   | SATISFACTORY |
| 40-2 | 19.400 | 4.400  | 2.560 | 1.280 | 45.8    | 0.850 | 0.322 | 5.657  | 9.773  | 16.596 | 22.253 | 15.168 | 27   | LOSS         |
| 41   | 28.600 | 6.200  | 3.000 | 2.100 | 187.0   | 0.480 | 0.115 | 4.803  | 3.871  | 3.983  | 7.854  | 11.110 | 28   | INSUFFICIENT |
| 42   | 38.500 | 7.200  | 3.500 | 2.560 | 380.0   | 0.410 | 0.129 | 4.014  | 2.847  | 3.340  | 6.187  | 8.904  | 29   | SATISFACTORY |
| 43-1 | 10.400 | 3.700  | 1.600 | 1.130 | 19.4    | 0.650 | 0.173 | 6.558  | 8.784  | 16.642 | 23.200 | 25.607 | 30   | SATISFACTORY |
| 43-2 | 10.400 | 3.700  | 1.600 | 1.020 | 16.4    | 0.380 | 0.074 | 6.558  | 5.135  | 7.137  | 12.272 | 25.607 | 30   | LOSS         |
| 44-1 | 32.600 | 6.740  | 3.100 | 2.600 | 329.0   | 0.560 | 0.117 | 4.482  | 4.154  | 3.585  | 7.739  | 10.103 | 31   | INSUFFICIENT |
| 44-2 | 32.600 | 6.740  | 3.100 | 2.350 | 288.0   | 0.400 | 0.101 | 4.492  | 3.635  | 3.095  | 6.730  | 10.103 | 31   | LOSS         |
| 45-1 | 18.600 | 6.100  | 2.800 | 2.360 | 156.2   | 0.730 | 0.119 | 5.734  | 5.934  | 6.394  | 12.128 | 15.694 | 32   | SATISFACTORY |
| 45-2 | 18.600 | 6.100  | 2.800 | 2.230 | 142.1   | 0.340 | 0.037 | 5.734  | 2.787  | 2.008  | 4.795  | 15.694 | 32   | LOSS         |
| 46   | 27.000 | 8.000  | 4.200 | 3.882 | 510.3   | 0.932 | 0.334 | 4.960  | 5.825  | 12.378 | 17.338 | 11.679 | 33   | LOSS         |
| 47   | 41.220 | 7.750  | 4.162 | 3.750 | 703.6   | 0.382 | 0.031 | 3.809  | 2.465  | 0.747  | 3.212  | 8.455  | 34   | LOSS         |
| 48-1 | 22.200 | 5.800  | 2.750 | 2.540 | 186.1   | 0.740 | 0.226 | 5.393  | 6.379  | 10.196 | 15.589 | 13.616 | 35   | SATISFACTORY |
| 48-2 | 22.200 | 5.800  | 2.750 | 1.950 | 134.5   | 0.520 | 0.075 | 5.393  | 4.483  | 3.381  | 7.864  | 13.616 | 35   | LOSS         |
| 49-1 | 22.000 | 6.000  | 3.180 | 2.900 | 200.5   | 0.620 | 0.240 | 5.412  | 5.167  | 10.891 | 16.058 | 13.715 | 36   | SATISFACTORY |
| 49-2 | 22.000 | 6.000  | 3.180 | 3.140 | 233.0   | 0.530 | 0.086 | 5.412  | 4.417  | 3.894  | 8.311  | 13.715 | 36   | LOSS         |
| 50-1 | 26.320 | 6.750  | 3.400 | 3.130 | 296.0   | 0.730 | 0.332 | 5.020  | 5.407  | 12.603 | 17.623 | 11.912 | 37   | SATISFACTORY |
| 50-2 | 26.320 | 6.750  | 3.400 | 2.600 | 220.0   | 0.270 | 0.060 | 5.020  | 2.000  | 2.263  | 4.263  | 11.912 | 37   | LOSS         |
| 51-1 | 26.500 | 5.300  | 2.700 | 2.048 | 174.5   | 0.560 | 0.233 | 5.004  | 5.283  | 8.784  | 13.798 | 11.849 | 38   | SATISFACTORY |
| 51-2 | 26.500 | 5.300  | 2.700 | 2.444 | 220.1   | 0.400 | 0.049 | 5.004  | 3.774  | 1.842  | 5.616  | 11.849 | 38   | LOSS         |
| 52-1 | 26.000 | 5.400  | 2.550 | 2.180 | 200.8   | 0.585 | 0.101 | 5.048  | 5.417  | 3.894  | 8.942  | 12.026 | 39   | INSUFFICIENT |
| 52-2 | 26.000 | 5.400  | 2.550 | 1.931 | 174.8   | 0.284 | 0.027 | 5.048  | 2.530  | 1.033  | 3.652  | 12.026 | 39   | LOSS         |
| 53-1 | 28.300 | 5.480  | 2.660 | 2.132 | 198.7   | 0.431 | 0.074 | 4.846  | 3.932  | 2.605  | 6.537  | 11.261 | 40   | INSUFFICIENT |
| 53-2 | 28.300 | 5.480  | 2.660 | 2.340 | 226.6   | 0.278 | 0.012 | 4.846  | 2.536  | 0.414  | 2.951  | 11.261 | 40   | LOSS         |
| 54-1 | 25.450 | 5.600  | 2.600 | 1.996 | 200.3   | 0.516 | 0.213 | 5.097  | 4.607  | 8.754  | 12.961 | 12.228 | 41   | SATISFACTORY |
| 54-2 | 25.450 | 5.600  | 2.600 | 2.180 | 210.5   | 0.333 | 0.049 | 5.097  | 2.973  | 1.918  | 4.891  | 12.228 | 41   | LOSS         |
| 55-1 | 32.200 | 6.000  | 3.100 | 2.710 | 379.5   | 0.540 | 0.183 | 4.515  | 4.500  | 5.680  | 10.180 | 10.199 | 42   | SATISFACTORY |
| 55-2 | 32.200 | 6.000  | 3.100 | 2.883 | 413.8   | 0.442 | 0.093 | 4.515  | 3.683  | 2.895  | 6.579  | 10.199 | 42   | LOSS         |
| 56-1 | 30.470 | 6.300  | 3.150 | 2.683 | 376.5   | 0.280 | 0.330 | 4.660  | 2.222  | 10.833 | 13.055 | 10.639 | 43   | SATISFACTORY |
| 56-2 | 30.470 | 6.300  | 3.150 | 2.574 | 356.7   | 0.283 | 0.341 | 4.660  | 2.246  | 11.182 | 13.428 | 10.639 | 43   | LOSS         |
| 57   | 21.850 | 6.570  | 3.390 | 2.310 | 145.2   | 0.860 | 0.201 | 5.426  | 6.545  | 9.188  | 14.614 | 13.789 | 44   | SATISFACTORY |

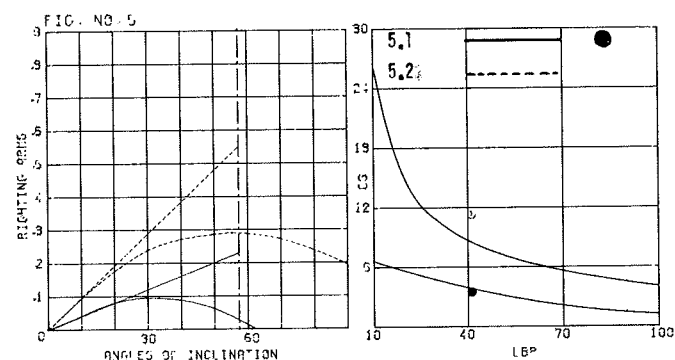
# APPENDIX



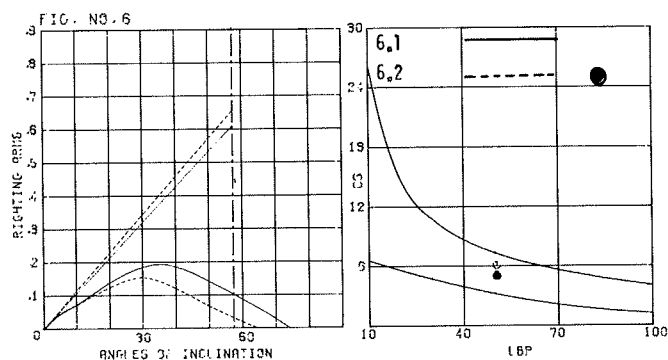
REMARKS. The capsizing of the Monitor CAPTAIN (1), in 1870, during a storm, in the Gulf of Biscay, had a great impact in the study of Ship Stability, as it demonstrated the importance of dynamical stability. The Monitor MONARCH (2), of smaller GM, but larger freeboard and dynamical stability, withstood safely the same storm, making clear the need to pay attention to dynamical stability, as well as GM. (Ref. 3)



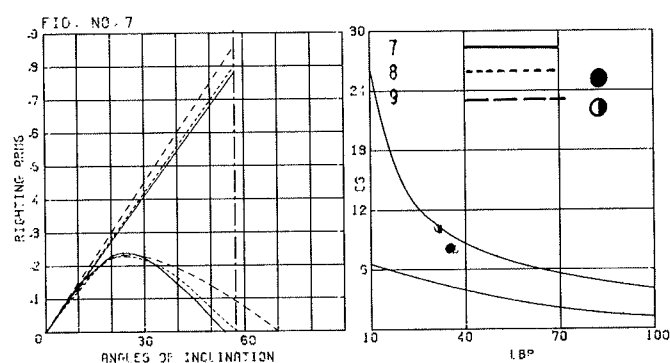
REMARKS. In Ref. 7, Paulling compares the stability conditions of the Tuna Clippers 3 and 4. Ship 3, of unsatisfactory stability, capsized in her maiden voyage, while Ship 4 is considered to have satisfactory stability and seaworthiness.



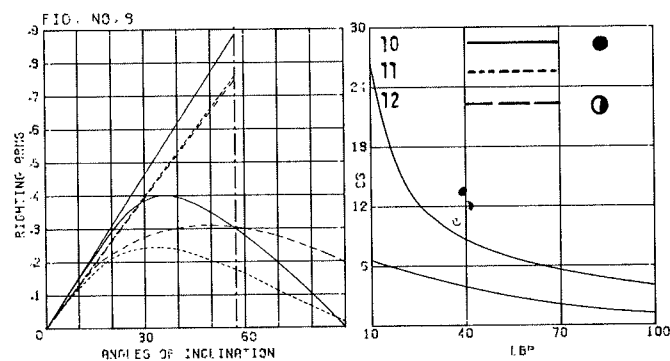
REMARKS. 5.1 corresponds to the stability conditions of a Coaster which capsized, having insufficient stability, according to Pescod (Ref. 8). 5.2 is the stability curve, for the same ship, which should be sufficient, as stated by Pescod.



REMARKS. Coaster which developed a heavy list, when sailing in the Baltic Sea, with bad weather and Beaufort 7 wind, due to icing conditions and flooding. It was confirmed that there was not shifting of cargo. (Ref. 9). 6.1 corresponds to the Load arrival condition, whereas 6.2 is the stability in the moment of the accident.

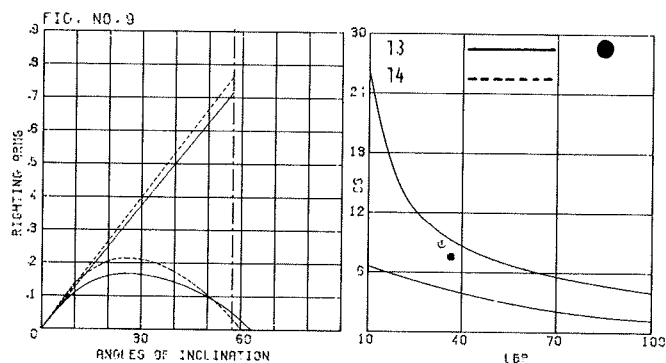


REMARKS. The capsizing of several German trawlers in 1904 prompted an investigation by Germanischer Lloyds and "See-Berufsgenossenschaft", on the stability of German fishing boats. The stability of the ships AUGUST (7), BREMA (8) and EMMY (9) was deemed as insufficient. (Ref. 3 and 10).

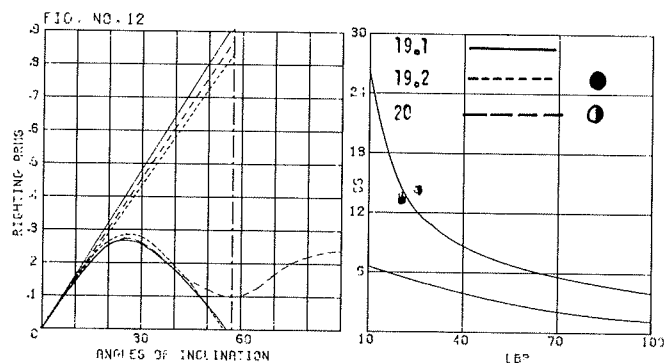


REMARKS. In the same investigation, that corresponds to vessels 7, 8 and 9, the ships ARTHUR FRIEDRICH (10), BRESLAU (11) and BRAUNSCHWEIG (12) were considered to have sufficient stability. (Ref. 3 and 10).

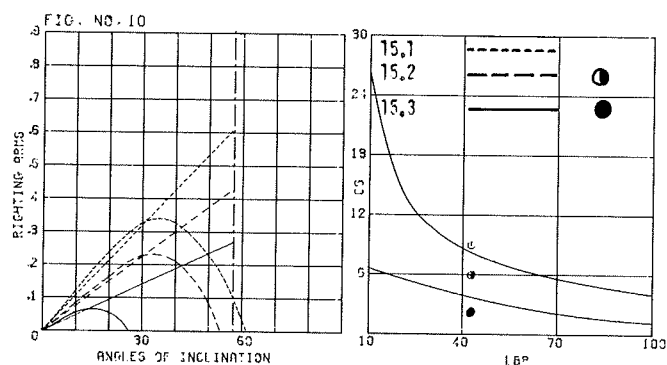




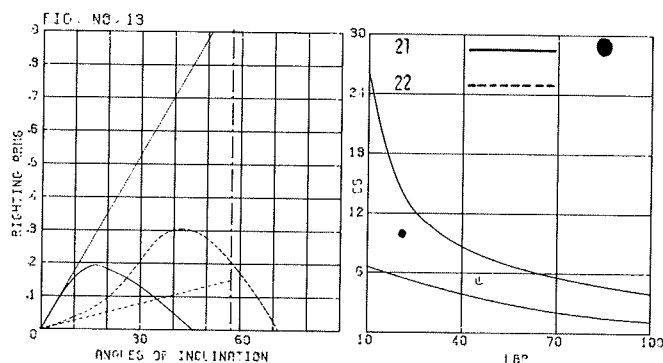
REMARKS. Stability curves of fishing boats, considered insufficient by Rahola (Ref. 3).



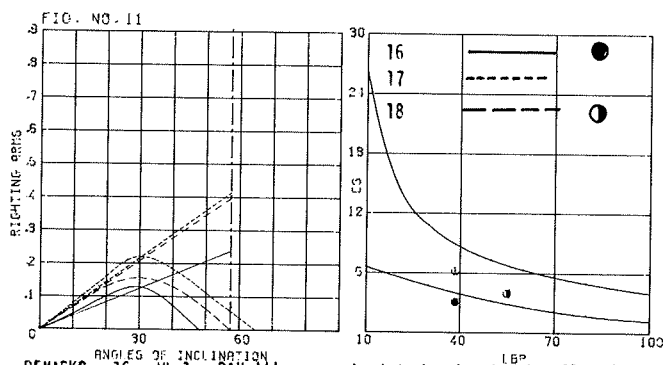
REMARKS. Ship No. 19: Fishing boat, capsized in a storm, with bad weather and wind force 7-8 and following sea. It was assumed that shifting of cargo occurred. (19.1 Load arrival, 19.2 Accident) Ship No. 20: Trawler of sufficient stability, experiencing a very large angle of heel at sea, near 90 degrees, when hoisting the side trawl, with 5 m waves. Once the trawl was released, the ship recovered the upright position.



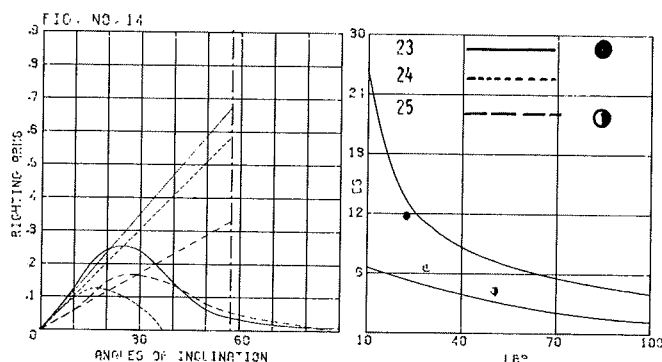
REMARKS. Capsized Coaster, with following sea.  
15.1 Ship stability, according to original design.  
15.2 Ship stability, after conversion, with addition of high weights. This curve is considered insufficient by Rahola (Ref. 3)  
15.3 Ship stability, in the moment of the accident.



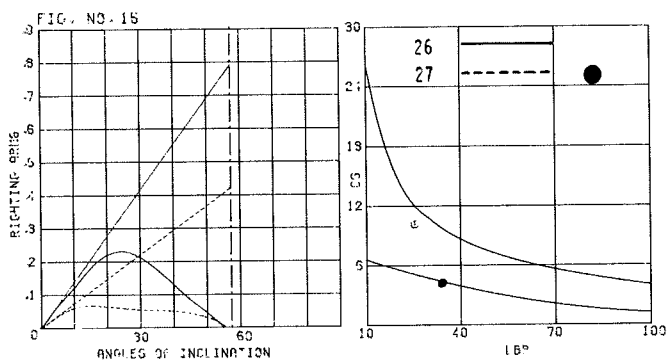
REMARKS. Ship No. 21: Fishing boat, capsized shortly after entering service.  
Ship No. 22: Coaster capsized with following seas.



REMARKS. 16.- Whaler RAU III, capsized during turning in official trials.  
17. Sister ship RAU IV, with better stability conditions.  
18. Coaster GALLEON, lost in a storm. Stability considered insufficient by experts (Ref. 3).



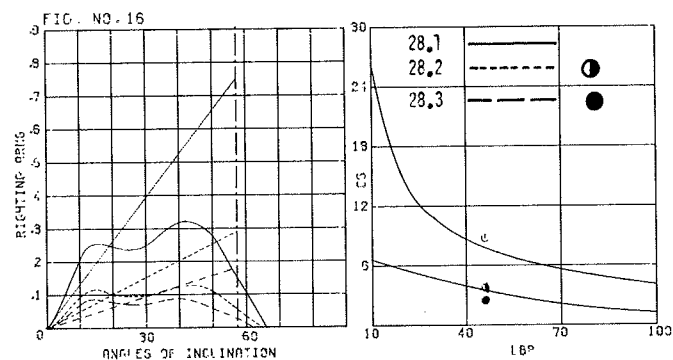
REMARKS. Fishing boats lost at sea.  
Ship No. 23: Capsized, no survivors (Ref. 11).  
Ship No. 24: Ship sunk on 26.03.1950 (Ref. 12).  
Ship No. 25: Ship lost in the North Sea, with following seas and 3 m waves, with length very similar to ship length. This vessel had 35 tons of permanent ballast. (Ref. 13)



#### REMARKS.

Ship No. 26: Ship lost on 25.11.1963, in bad weather, with waves of 6 to 8 m high. Apparently, there was shipping of water on deck, with insufficient freeing ports. (Ref. 14).

Ship No. 27: Fishing boat lost with bad weather and strong wind, of intensity 10-11, Beaufort. (Ref. 15).

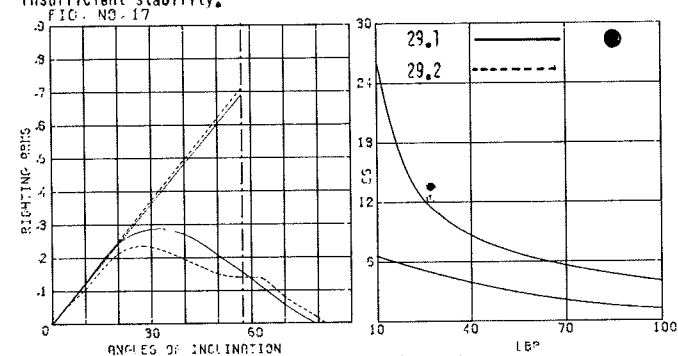


#### REMARKS. Capsized Coaster GROVENORT. (Ref. 16)

28.1 Ship stability, according to original design.

28.2 Ship stability, in the same displacement, but with 85 tons of deck cargo.

28.3 Co, assuming 125 tons of deck cargo and 40 tons void in the hold. The experts considered the original stability, 28.1, to be sufficient. The capsizing was attributed to bad stowage of the cargo, leading to insufficient stability.

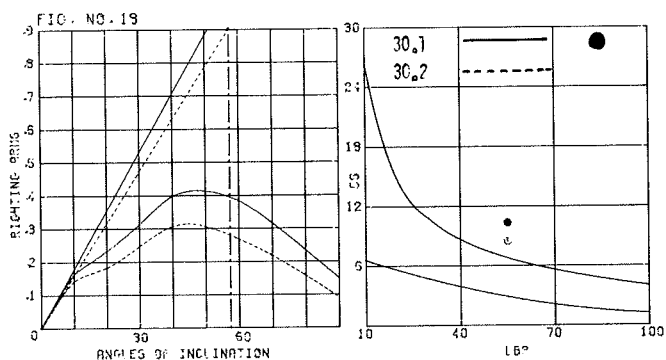


#### REMARKS. Fishing boat sunk, on 30.5.1957. (Ref. 9)

29.1. Ship stability, in the moment of the accident.

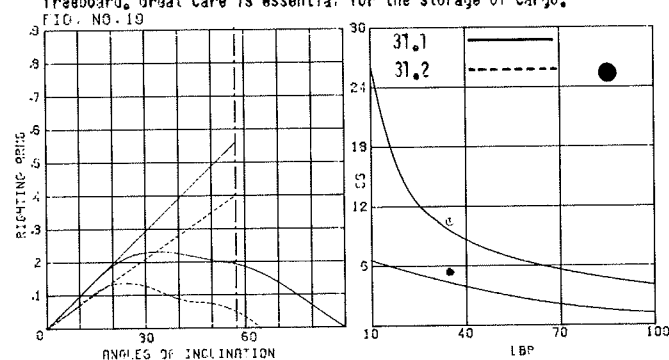
29.2. Load arrival condition.

The ship capsized, in bad weather, with beam waves and wind 10 to 11, Beaufort scale. The Shipmaster had the impression of cargo shifting in the hold.



REMARKS. Cargo boat MARIANNE WEHR, capsized on 14.10.1963 (Ref. 17).  
30.1 Stability at the accident, assuming good stowage and 1.3 t/m.  
30.2 No, assuming a stowage factor of 1.2 t/m<sup>3</sup> and the cargo to have an angle of repose of 40 degrees.

This ship was lost in very bad weather, due to a shifting of the cargo, produced by bad stowage. The experts considered the stability as sufficient, high GM being needed in order to compensate for the small freeboard. Great care is essential for the storage of cargo.

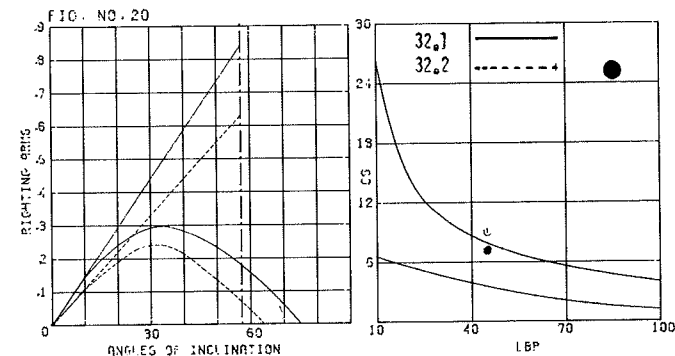


#### REMARKS. Fishing boat lost, with bad weather, in icing conditions.

31.1 Stability, in the moment of the accident, normal conditions.

31.2 Stability, in the moment of the accident, assuming 40 t of ice on decks and rigging.

Ref. Casualty report, IMCO.

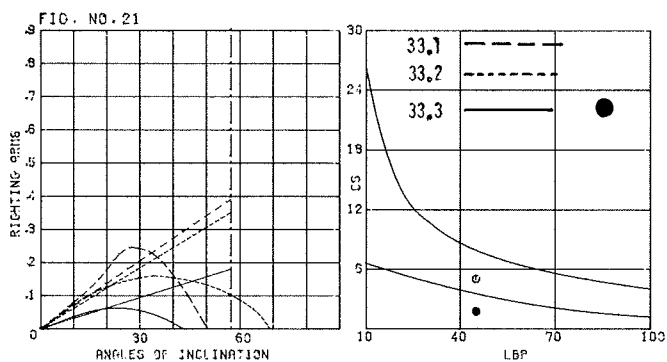


#### REMARKS. Lightship ELBE I, sunk on 27.10.1936 (Ref. 3)

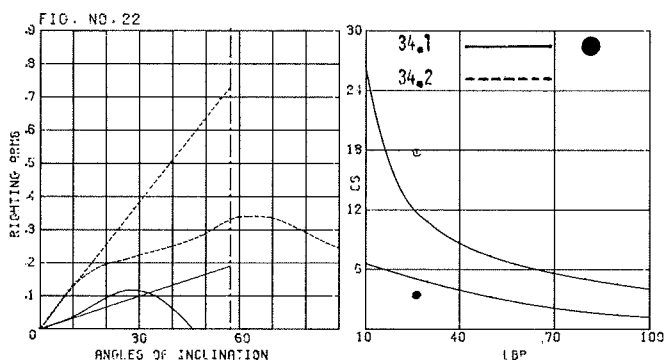
32.1 Stability, full load condition (deemed sufficient by experts).

32.2 Stability, in the moment of the accident.

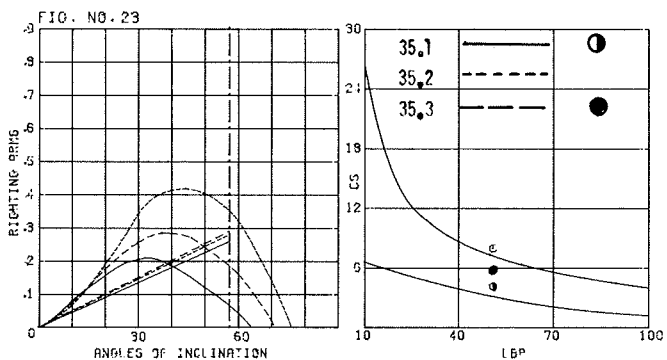
The ship sunk in a heavy storm, with winds 10 to 12, Beaufort scale. The experts considered the stability curve at the moment of the accident, as insufficient.



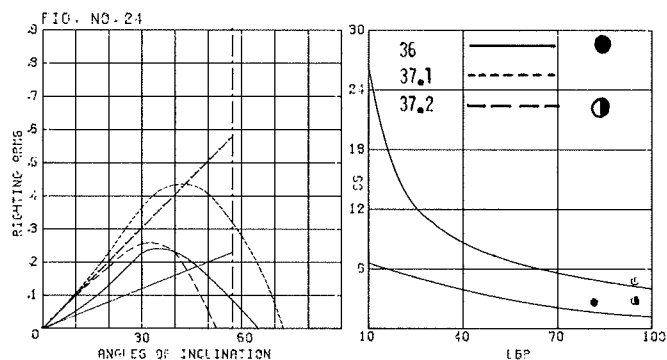
REMARKS. Coaster capsized, shortly after loading, with good weather.  
 33.1 Stability, ballast condition.  
 33.2 Stability, full load condition.  
 33.3 Stability, full load condition, with 70 t deck load (accident).  
 The ship stability did not comply with normal stability criteria.  
 The ship capsized only one month after entering service.



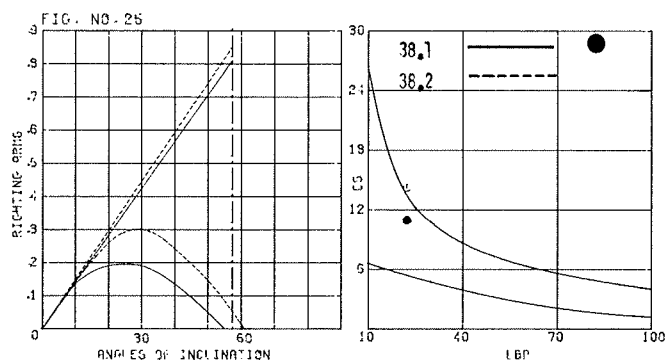
REMARKS. Fishing boat lost on 7.11.1962.  
 34.1 Stability, in the moment of the accident.  
 34.2 Stability, load arrival condition.



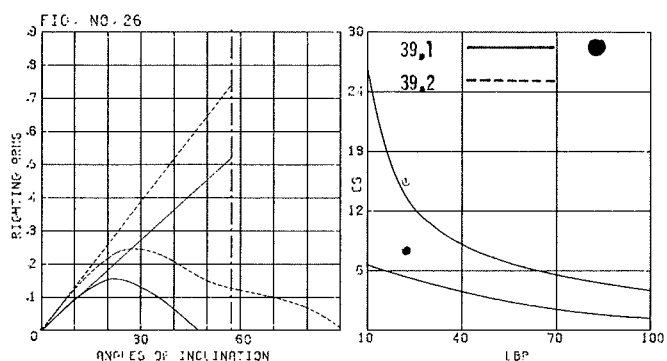
REMARKS. Cargo ship capsized.  
 35.1 Stability, full load, with deck cargo.  
 35.2 Stability, half load condition.  
 35.3 Stability, in the moment of the accident.  
 The ship capsized, in bad weather, with quartering seas. It is worth mentioning that the GZ's comply with Rahola's criterion, but the values of GM, in all loading conditions were very small.



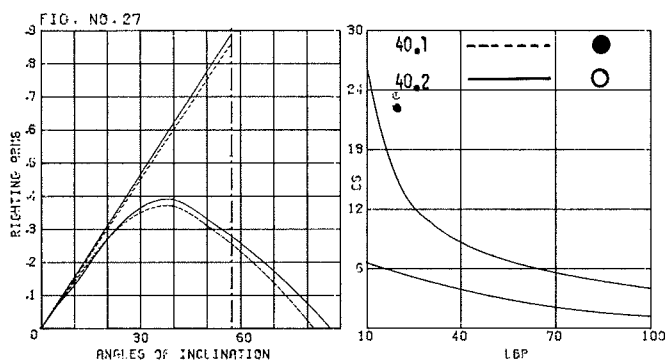
REMARKS. Ship No. 36. Cargo ship IRENE OLDENDORF (Ref 18)  
 Ship lost in bad weather, with quartering seas and no survivors.  
 Stability considered inadequate.  
 Ship 37. Training ship PAMIR (Ref. 19)  
 37.1. Stability, in the moment of the accident (Curve adequate).  
 37.2. Do, but considering non-watertight superstructures (Curve inadequate).



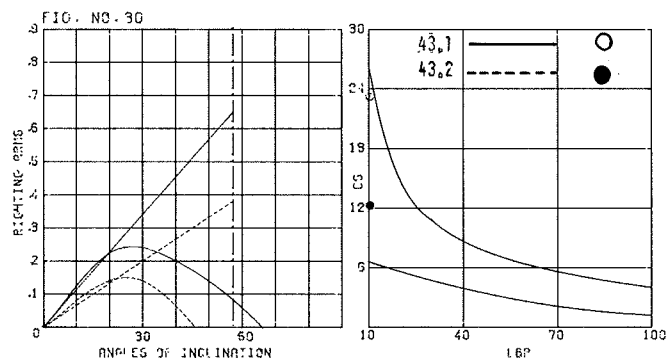
REMARKS. Fishing boat lost on 16.2.1962, without survivors.  
 38.1 Stability, in the moment of the accident.  
 38.2 Stability, Load arrival condition.  
 The ship was lost in the North Sea, with bad weather and quartering seas.  
 Ref. Casualty report, IMCO.



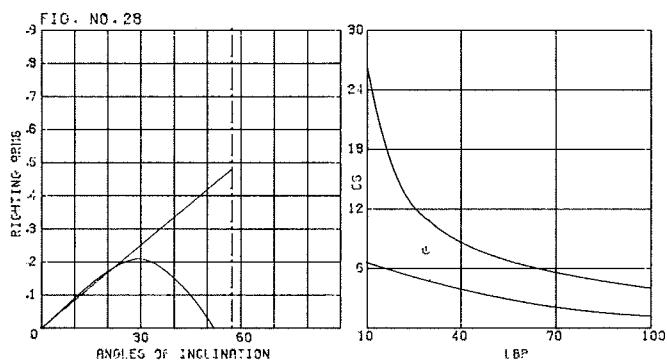
REMARKS. Fishing boat lost on 17.11.1962, without survivors.  
 39.1 Stability, in the moment of the accident.  
 39.2 Stability, Load arrival condition.  
 Ref. Casualty report, IMCO.



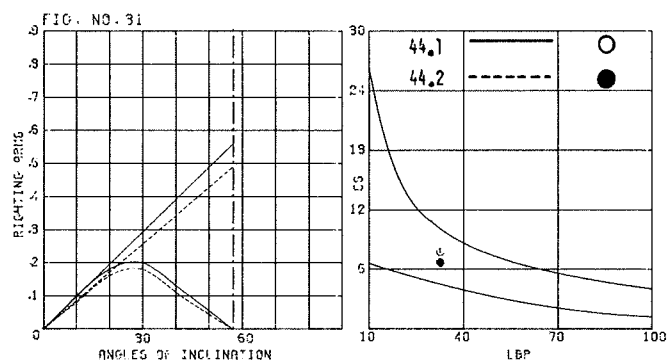
REMARKS. Fishing boat capsized. (Ref. 20)  
 40.1 Ship stability, at the time of loss.  
 40.2 Ship stability, Load arrival condition.  
 The ship capsized suddenly in a restricted sea area, with rough quartering sea and wind Beaufort 8-10. She carried load on deck and there was trapping of water on deck.



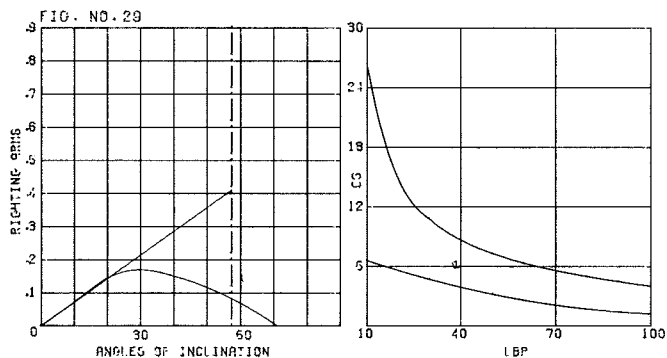
REMARKS. Fishing boat capsized No. 43.  
 43.1 Ship stability, Load arrival condition.  
 43.2 Ship stability, at the time of loss.  
 Sudden capsizing occurred with moderate following sea and Beaufort 8 to 10. (ref. 20)



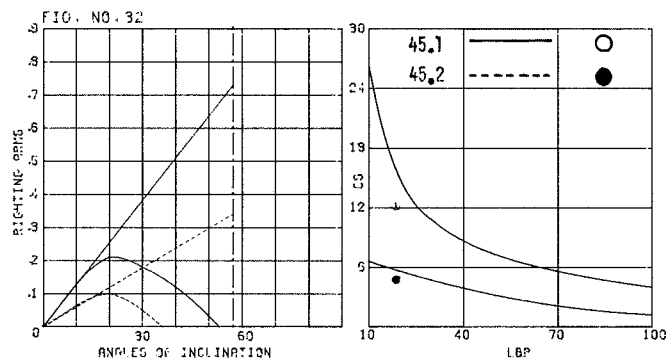
REMARKS. Fishing boat capsized (Ref. 20). No. 41.  
 The ship capsized gradually with a rough following sea and wind Beaufort 8-10.



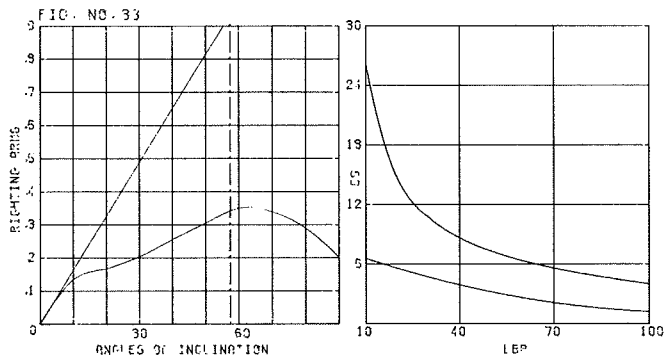
REMARKS. Fishing boat capsized No. 44 (Ref. 20).  
 44.1 Ship stability, Load arrival condition.  
 44.2 Ship stability, at the time of loss.  
 The ship capsized gradually, with rough beam seas and Beaufort 4-7.



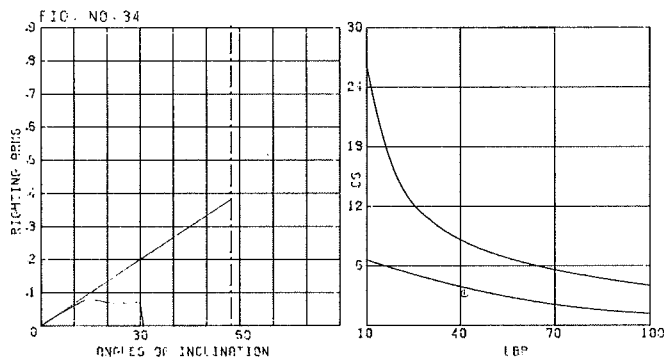
REMARKS. Fishing boat capsized No. 42 (Ref. 20).  
 It was assumed that the ship capsized gradually in a restricted sea area, with rough sea and wind exceeding Beaufort 10. There was icing on decks and rigging.



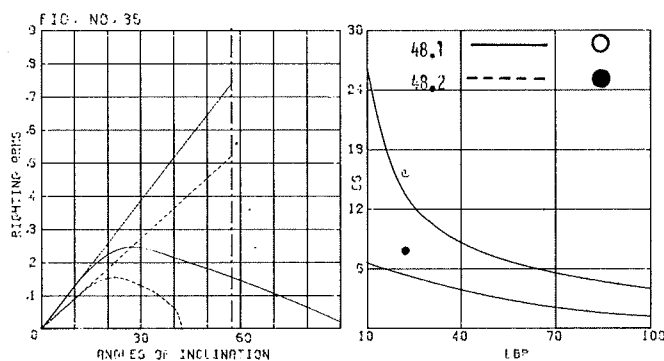
REMARKS. Fishing boat capsized No. 45 (Ref. 20).  
 45.1 Ship stability, Load arrival condition.  
 45.2 Ship stability, at the time of loss.  
 It was assumed that the ship capsized suddenly in open sea. The environmental conditions were of good weather, smooth sea and moderate wind, Beaufort 0 - 3. The ship was hauling in the trawl and carried load on deck. The stability curve was highly insufficient.



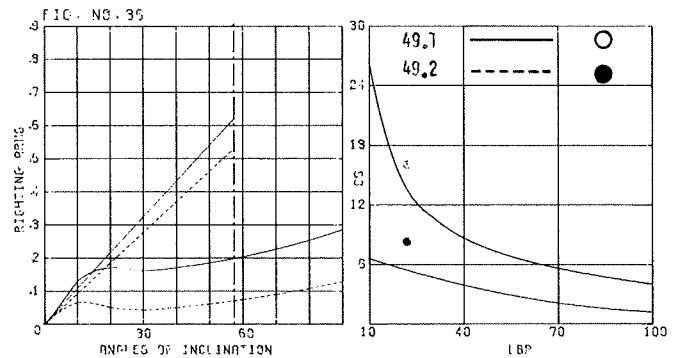
REMARKS. Fishing boat lost No. 46 (Ref. 20)  
The casualty was produced in open rough sea, with wind exceeding Beaufort 10. It was assumed that the ship sailed with following seas. There was water inrush and trapping of water on deck. Ship stability is considered sufficient, assuming watertight integrity of the ship.



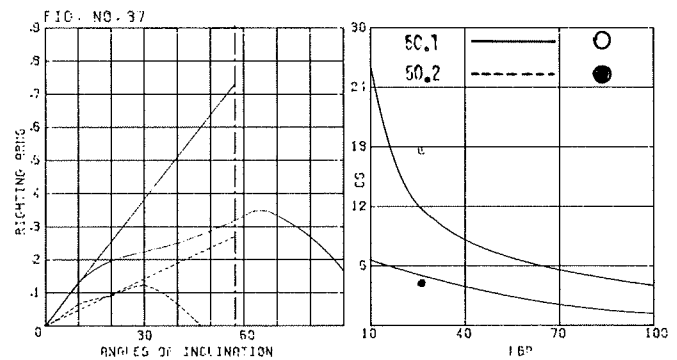
REMARKS. Fishing boat capsized No. 47 (Ref. 20).  
The ship carried load on deck, capsizing with rough sea and Beaufort 8 to 10.



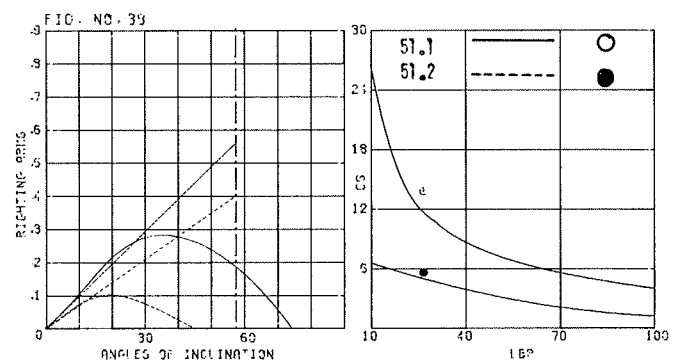
REMARKS. Fishing boat capsized No. 48 (Ref. 20).  
48.1 Ship stability, Load arrival condition.  
48.2 Ship stability, at the time of loss.  
The ship capsized with rough quartering seas, and wind force Beaufort 8 to 10.



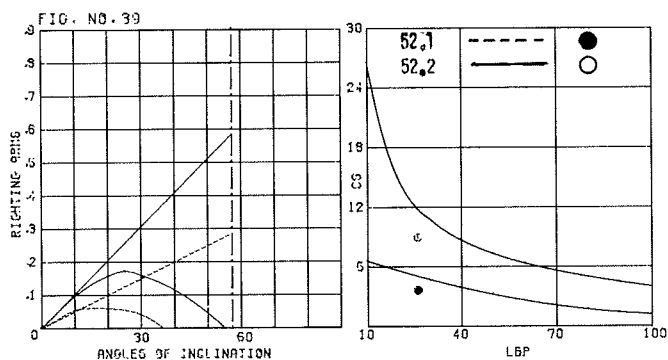
REMARKS. Fishing boat No. 49 (Ref. 20)  
49.1 Ship stability, Load arrival condition.  
49.2 Ship stability, at the time of the accident.  
The stability of this vessel is sufficient in the load arrival condition. The ship experienced a dangerous heel, when sailing with beam seas, Beaufort 4 to 7, being overloaded, trapping water on deck.



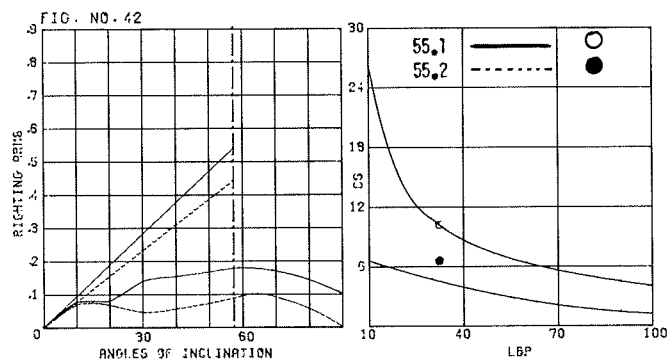
REMARKS. Fishing boat capsized No. 50 (Ref. 20).  
50.1 Ship stability, Load arrival condition.  
50.2 Ship stability, at the time of loss.  
The vessel capsized gradually in rough sea, due to water trapping on deck and free surface effects.



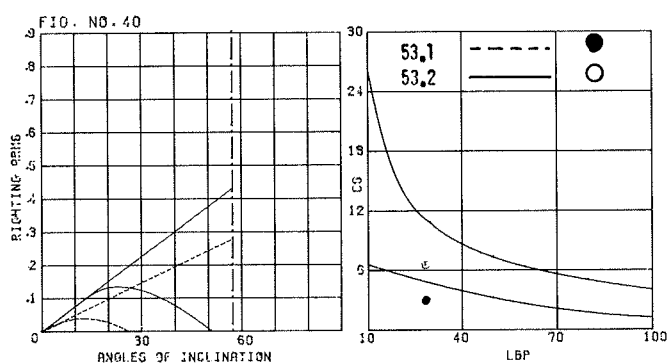
REMARKS. Fishing boat No. 51 (Ref. 20).  
51.1 Ship stability, Load arrival condition.  
51.2 Ship stability, at the time of the accident.  
This vessel experienced a dangerous heel in a beam sea, with moderate wind, Beaufort 4 to 7. There was water inrush and trapping of water on deck.



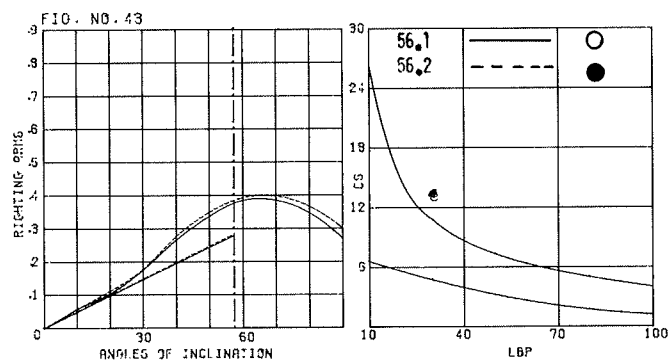
REMARKS. Fishing boat No. 52 (Ref. 20).  
 52.1 Ship stability, at the time of the accident.  
 52.2 Ship stability, Load arrival condition.  
 This vessel experienced a dangerous heeling, when sailing in summer, with rough following sea and moderate wind.



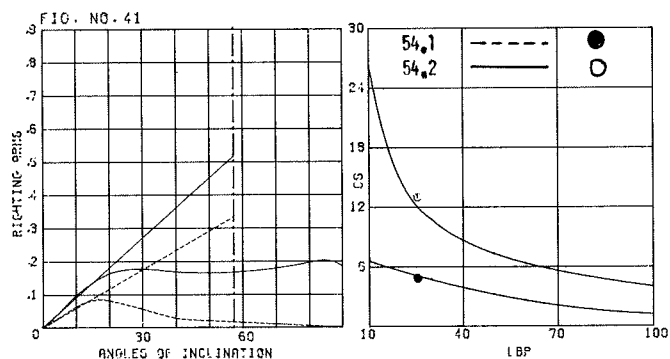
REMARKS. Fishing boat No. 55 (Ref. 20).  
 55.1 Ship stability, Load arrival condition.  
 55.2 Ship stability, at the time of accident.  
 The accident occurred in winter time, with moderate sea and wind force 4 to 7, Beaufort scale.



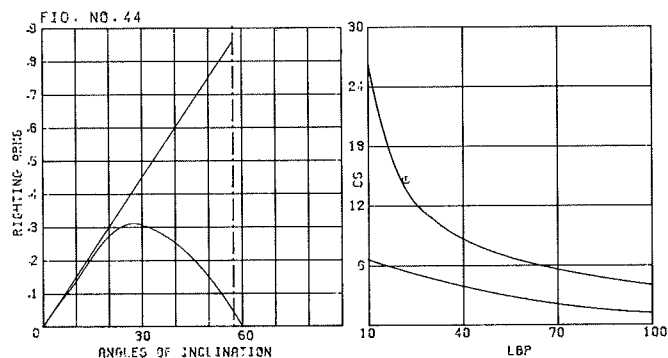
REMARKS. Fishing boat No. 53 (Ref. 20).  
 53.1 Ship stability, at the time of the accident.  
 53.2 Ship stability, Load arrival condition.  
 This ship had a dangerous heeling, due to water trapping on deck, with quartering waves and moderate wind.



REMARKS. Fishing boat capsized No. 56 (Ref. 20).  
 56.1 Ship stability, Load arrival condition.  
 56.2 Ship stability, at the time of loss.  
 Ship capsized in rough beam seas, when carrying a deck load



REMARKS. Fishing boat capsized No. 54 (Ref. 20).  
 54.1 Ship stability, at the time of loss.  
 54.2 Ship stability, Load arrival condition.  
 This vessel was lost by flooding, with beam sea, wind 8 to 10, Beaufort scale, and water trapped on deck.



REMARKS. Fishing boat capsized No. 57 (Ref. 20).  
 This vessel was lost by gradual capsizing, when carrying a deck load, with rough seas, due to shifting of cargo and water trapped on deck. The stability curve corresponds to load arrival. Stability conditions at the time of loss are unknown.

# ON THE CRITERIA FOR EVALUATING INTACT STABILITY OF SHIPS IN WIND AND WAVES

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China

## ABSTRACT

Casualty statistics show that capsizing remains a keen problem to naval architects and ship operators. And ship stability is still of crucial importance as it has been for over a hundred years.

In this paper, the authors confine themselves to the discussion on criteria for evaluating intact stability of ships in wind and waves. Two conditions dangerous to operation of ships are chosen. For the beam sea condition, they ascertain through their experiments that the criterion based on the gust (no heel) situation is more vigorous, and should be adopted instead of that for the steady wind situation. For the following sea condition, a procedure for evaluation is recommended.

## NOMENCLATURE

$A_1^G$  :  $-\int_{-\phi_A^g}^{\phi_m^g} M_F(\phi) d\phi + \int_{-\phi_A^g}^0 \Delta \cdot GZ d\phi$   
 $A_1^S$  :  $-\int_{-\phi_A^s}^{\phi_m^s} M_F(\phi) d\phi + \int_{-\phi_A^s}^0 \Delta \cdot GZ d\phi$   
 $B$  : Breadth moulded  
 $C$  : Wave velocity  
 $D$  : Depth moulded  
 $E_1^G$  :  $\int_0^{\phi_m^g} \Delta \cdot GZ d\phi$   
 $E_1^S$  :  $\int_0^{\phi_m^s} \Delta \cdot GZ d\phi$   
 $GM$  : Metacentric height  
 $\delta GM$  : Amplitued of GM Variation  
 $H_w$  : Wave height  
 $K$  : Damping coefficient  
 $L$  : Length of ship

$M_F$  : Gust heeling moment  
 $T$  : Draught  
 $T_w$  : Wave period  
 $T_\theta$  : Natural period of rolling  
 $\omega$  : Wave frequency  
 $\omega_e$  : Frequency of encounter  
 $\omega_\phi$  : Natural frequency of rolling  
 $\epsilon$  : Phase lag  
 $\phi$  : Tuning factor for rolling  
 $\phi_A^G$  : Rolling amplitude for gust criterion  
 $\phi_m^G$  : Maximum heeling angle for gust criterion  
 $\phi_A^S$  : Rolling amplitude for steady wind criterion  
 $\phi_m^S$  : Maximum heeling angle for steady wind criterion  
 $\lambda$  : Wave Length  
 $\Delta$  : Displacement

## 1. INTRODUCTION

It is not seldom to learn that ships capsized owing to inadequacy in intact stability. Sometimes, even ships in full compliance with the IMCO stability requirements were reported to have capsized[1] and suspicions were consequently aroused about the fitness of stability criteria set forth.

In the People's Republic of China, however, we are more fortunate. Since the enforcement of the rules governing the stability of sea-going vessels in 1960, which was revised in 1974[2], a great number

of ships had been built and none was lost in bad weathers owing to inadequacy in intact stability. Nevertheless, some were reported to have capsized through improper operations. This fact is mainly due to: (1) establishment of explicit criteria for intact stability, and (2) strictness in executing the rules, continual attentions being paid not only to ships as built, but also to environmental conditions in which the ships engage in their whole lifetime. Statistics have shown a low rate of capsizing casualties for Soviet vessels[3], and it is considered to be due to strictness in executing the rules also. Mr. Watson pointed out that continuation of fishing operations in gale conditions instead of hoving to or scurrying to port had led to increase the rate of casualties of trawlers[4]. Dr. Kawashima et al came to the following conclusion through model tests[5]: models in compliance with stability criteria do not capsize, while those below do capsize. All these facts help to demonstrate that stability criteria should be specified in accordance with environmental conditions (wind and waves), and to drive a ship beyond her capability is the primary cause for disaster.

In spite of non-existence of capsizing casualty through inadequacy in intact stability, we do not esteem our stability rules as perfect. On the contrary, to specify more rational intact stability criteria, together with method of evaluation, remains to be one of the most important topics for further investigations in our country.

In this paper, firstly the authors analyse the conditions dangerous to operation of ships which they may encounter, and then propose stability criteria for these dangerous conditions through model tests and analyses.

## 2. CONDITIONS DANGEROUS TO SHIP OPERATION

Ship stability depends on ship form parameters and environmental conditions jointly. To single out only one criterion suited for various ship types under any arbitrary condition will be a difficult task, not to say impossible. The most applicable approach is to ascertain some typical dangerous conditions likely to be encountered according to ships' mission, and to assign a separate criterion for each condition involved. We agree with many investigators [6] [7] [8] [9] that the main factors leading to capsizing are wind, waves and shipping WATER on deck. To them, 'basic criteria' should be assigned with 'additional criteria' for other minor factors. In this paper, however, we confine ourselves only to the discussion on criteria for evaluating stability of ships in wind and waves. Shipping water on deck, which is especially dangerous to small vessels, will be studied by further investigations.

Following conditions are considered

dangerous to ships in wind and waves:

### 2.1 Beam Sea With Wind Broadside On

From casualty statistics[3] [10] [11] over a half of ships capsized were in beam seas. Compared with ships hoving to, ships under way have larger damping for rolling motion, and consequently reduced roll amplitudes. So, ships hoving to with wind broadside on in beam seas are naturally in the most critical condition. Besides, this condition is not unrealistic in operation. So it is natural that most stability rules, including those issued by the Register of Shipping of PRC, adopt this condition as a basic condition to be checked.

### 2.2 Following Or Quartering Sea

Casualty statistics have shown that the ratio of ships capsized in these waves is as high as in beam seas, especially for small vessels. It is well-known to many naval architects that owing to reducing in righting moment it is dangerous for a ship to ride on the waves with wave crest amidships. But all navigators do not take care of this potential menace. In this sense, this condition is even more dangerous than beam sea condition. The existing stability rules of our country do not give any criterion for these conditions, and we consider it inappropriate.

In checking stability in waves, head sea or bow sea conditions can be ignored, since no ships have ever been reported to have capsized in these conditions in the past.

In short, beam sea with wind broadside on and following or quartering sea are the two dangerous conditions which should be checked for intact stability of ships in wind and waves.

## 3. ON EVALUATING INTACT STABILITY IN BEAM SEA AND WIND

In defining the intact stability criteria in the beam sea and wind condition, the IMCO stability requirements adopt a 'statical approach', while most statutory stability rules adopt a 'physical approach', using the well-known weather or windline criterion. In the latter case, however, two different situations are assumed by different countries:

(1) The ship is heeled with a static heeling angle  $\phi_s$  under the action of a steady wind moment,  $M_s$ , and is rolling with a resonant amplitude  $\phi'_s$  under the effect of waves. Then a gust is applied on the ship, with a heeling moment  $M_F$ . The ship will be considered to be stable, if

$$\int_{-\phi'_s + \phi_s}^{\phi'_s} \Delta \cdot GZ d\phi - \int_{-\phi'_s + \phi_s}^{\phi'_s} M_F(\phi) d\phi \geq 0 \quad (1)$$



or,

$$\int_0^{\phi_m^s} \Delta \cdot GZ d\phi - \int_{-\phi_A^s + \phi_s}^{\phi_m^s} M_F(\phi) d\phi + \int_{-\phi_A^s + \phi_s}^0 \Delta \cdot GZ d\phi \geq 0 \quad (2)$$

or,  $\text{Area (CDE)} - \text{Area (OABC)} - \text{Area (OAF)} \geq 0$

Denoting Area (CDE) as  $Ei^S$ , and the sum of Area (OAF) and Area (OABC) as  $Ai^S$ , the intact stability criterion will be:

$$Ei^S - Ai^S \geq 0 \quad (3)$$

This criterion may be termed steady wind criterion.

(2) The ship is rolling with no initial heeling under the effect of waves, the resonant amplitude being  $\phi_A^s$ . When the ship rolls to the windward side until  $\phi_A^s$ , a gust heeling moment  $M_F(\phi)$  is applied on the ship, fig. 2. The ship will be considered to be stable, if

$$\int_{-\phi_A^s}^{\phi_m^s} \Delta \cdot GZ d\phi - \int_{-\phi_A^s}^{\phi_m^s} M_F(\phi) d\phi \geq 0 \quad (4)$$

or,  $\int_0^{\phi_m^s} \Delta \cdot GZ d\phi - \int_{-\phi_A^s}^{\phi_m^s} M_F(\phi) d\phi + \int_{-\phi_A^s}^0 \Delta \cdot GZ d\phi \geq 0$   
or,  $\text{Area (CDE)} - \text{Area (OABC)} - \text{Area (OAF)} \geq 0$

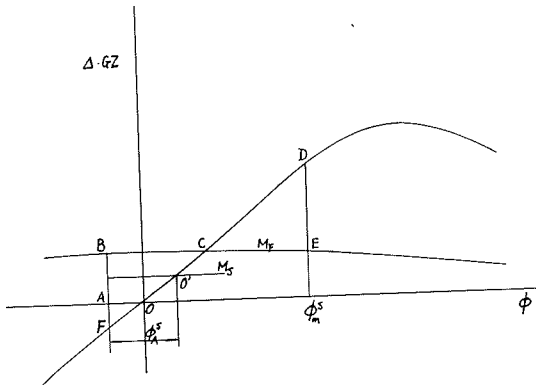


Fig.1 Sketch for Steady Wind Criterion

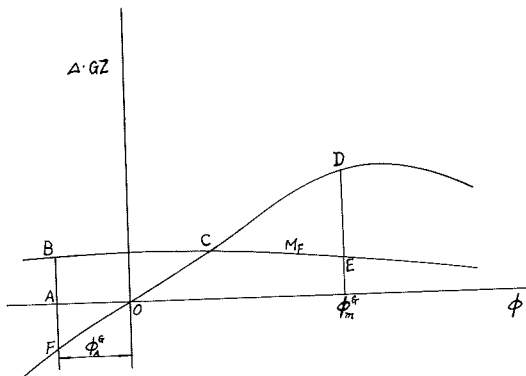


Fig.2 Sketch for Gust Criterion

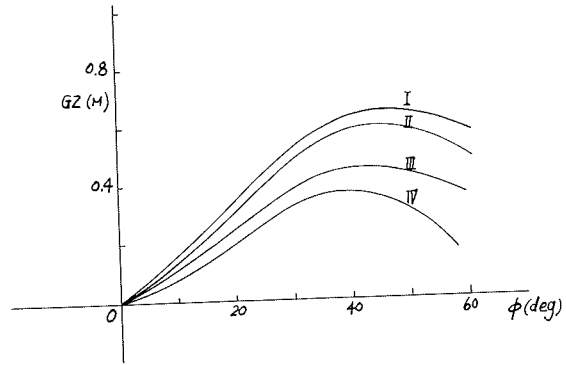


Fig.3 GZ Curves for the Test Variants

Denoting Area (CDE) as  $Ei^G$ , and the sum of Area (OAF) and Area (OABC) as  $Ai^G$ , the intact stability criterion will be:

$$Ei^G - Ai^G \geq 0 \quad (5)$$

This criterion may be termed as gust criterion.

In order to make out which situation is more reasonable, we in MARIC carried out an experiment with a research ship model. Its principal dimensions are given in table 1.

The model was made of wood, its length being 1.75m (scale ratio = 32.6). The above water-portion of the model was also scaled after the actual ship.

In order to investigate the influence of GM on intact stability at the same time, the GM of the model had been varied, with a total of four GZ curves, as shown in fig.3.

The model was tested in a hovering-to condition in regular resonant beam waves ( $T_w = T\phi$ ). The wave conditions remained unchanged both for steady wind situation and for gust situation. The heeling moment of the wind was simulated by a servo-controlled moving weight mechanism which was placed athwart the model. The two situations were simulated by different initial positions of the moving weight. In the steady wind situation, the moving weight was in its midpoint position, and the model would have an initial list  $\phi_s$ . When it rolled in regular waves to windward side until  $\phi_A^s$ , the weight was moved to the leeward side in an instant, and was kept there. In the gust situation, the moving weight was originally in its extreme starboard position, and the model would float upright in calm water. When it rolled in regular waves to windward (starboard) side until  $\phi_A^s$ , the weight was moved to the port side in an instant, and was kept there. Wave elevations, rolling amplitudes, weight moving and the maximum heeling angle caused by the moving weight were recorded simultaneously. Records showed that the weight started moving when

Table 1 The Principal Dimensions of a 800t Research Ship

| Item                          | Symbol   |          |
|-------------------------------|----------|----------|
| Length overall                | $L_{OA}$ | 65.316 m |
| Length on design waterline    | $L_{WL}$ | 60.5 m   |
| Length between perpendiculars | $L_{BP}$ | 57.0 m   |
| Breadth moulded               | B        | 9.2 m    |
| Depth moulded                 | D        | 4.73 m   |
| Draught, forward              | $T_F$    | 2.84 m   |
| Draught, aft                  | $T_A$    | 2.96 m   |
| Mean draught                  | T        | 2.90 m   |
| Block coefficient             | $C_B$    | 0.518    |
| Midship section coefficient   | $C_M$    | 0.890    |
| Displacement                  | $\Delta$ | 820 t    |

the model was rolling windward side to  $\phi_A^*$ , and stopped at the opposite side in a very short duration, complying with the gust requirements.

Test results are given in fig.4 - fig.7, and are summarized in table 2.

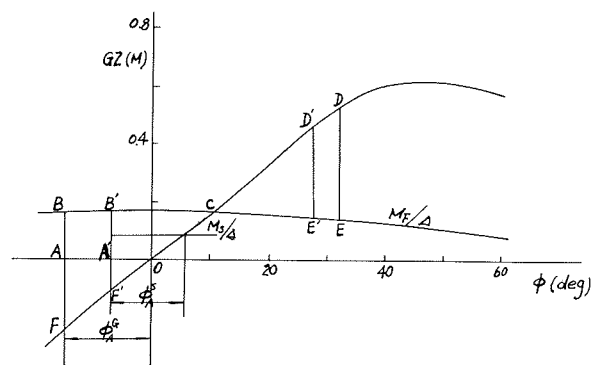


Fig.4 Test Result for Variant I

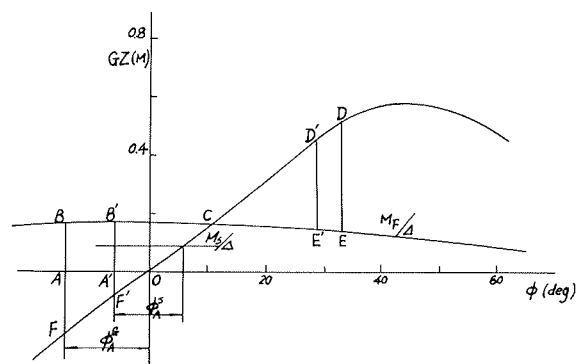


Fig.5 Test Result for Variant II

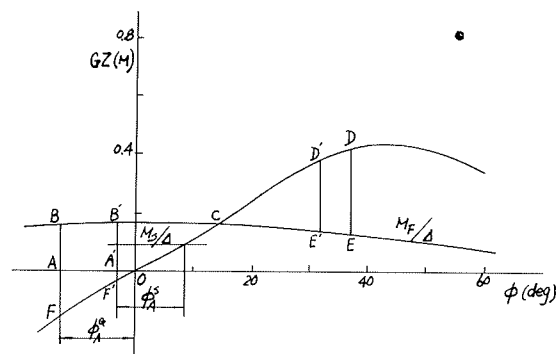


Fig.6 Test Result for Variant III

Table 2. Comprehensive Results of Test Variants

| Item   | Symbol     | Unit | Test Variant |       |       |       |
|--|------------|------|--------------|-------|-------|-------|
|  |            |      | I            | II    | III   | IV    |
| Displacement   | $\Delta$   | t    | 820          | 820   | 820   | 820   |
| Metacentric height                                   | GM         | M    | .893         | .815  | .644  | .479  |
| Wind heeling moment                                  | $M_F$      | t-m  | 138.6        | 138.6 | 138.6 | 138.6 |
| Natural rolling period                               | $T_\phi$   | sec  | 6.63         | 7.09  | 8.29  | 9.2   |
| Wave period  | $T_w$      | sec  | 6.63         | 7.09  | 8.29  | 9.2   |
| Angle of wave slope                                  | $\alpha_0$ | deg  | 3.83         | 3.96  | 3.21  | 2.23  |
| Resonant rolling amplitude for gust situation        | $\phi_A^G$ | deg  | 15.1         | 14.2  | 12.6  | 9.0   |
| Resonant rolling amplitude for steady wind situation | $\phi_A^S$ | deg  | 12.7         | 11.6  | 10.5  | 9.3   |
| Maximum heeling angle for gust situation             | $\phi_m^G$ | deg  | 31.8         | 32.6  | 36.8  | 38.1  |
| Maximum heeling angle for steady wind situation      | $\phi_m^S$ | deg  | 27.1         | 28.4  | 31.6  | 35.1  |
| Area (OABC + OAF) for gust sit.                      | $Ai^G$     | t-m  | 79.24        | 70.14 | 60.86 | 47.86 |
| Area (OABC + OAF) for steady wind sit.               | $Ai^S$     | t-m  | 39.32        | 35.09 | 25.66 | 21.04 |
| Area (CDE) for gust sit.                             | $Ei^G$     | t-m  | 58.91        | 52.75 | 51.52 | 41.38 |
| Area (CDE) for const. wind sit.                      | $Ei^S$     | t-m  | 36.05        | 34.91 | 31.29 | 28.14 |
| $Ei^G/Ai^G$  | -          | -    | 0.743        | 0.752 | 0.847 | 0.867 |
| $Ei^S/Ai^S$  | -          | -    | 0.917        | 0.995 | 1.219 | 1.338 |

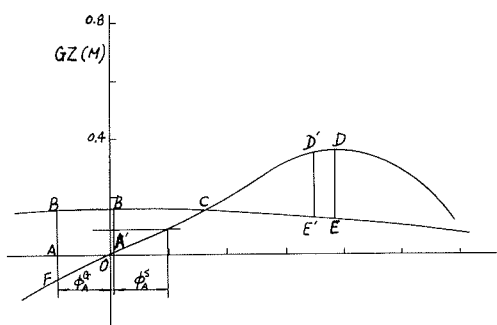


Fig. 7 Test Result of Variant IV

It can be seen from these results that:

(1) Although wave conditions and wind heeling moments were kept unchanged for both gust and steady wind situations, for all the four variants  $Ai^G$  is greater than  $Ai^S$ , and  $Ei^G$  is greater than  $Ei^S$ . It implies that the criterion based on gust situation will be more severe than that based on steady wind situation. According to table 2, there is no difference between  $\phi_A^g$  &  $\phi_A^s$  under similar wave conditions, and the difference between the two ratios ( $Ei^G/Ai^G$  &  $Ei^S/Ai^S$ ) is most likely to have been resulted from the way in which the wind is applied on the ship. For ships in ballast, owing to their increase in areas exposed to the wind, this difference would be even greater. For the sake of safety, in evaluating the intact stability of ships, it is advisable to use criterion based on gust situation.

(2) In table 2, the values of  $Ei^G/Ai^G$  and  $Ei^S/Ai^S$  increase in reverse proportion to GM. This can be accounted for by non-linearity when the heeling becomes great. For all the four variants tested,  $Ei^G/Ai^G$  is always smaller than 1, and  $Ei^S/Ai^S$  is always greater than  $Ei^G/Ai^G$ , and may be greater than 1 in some cases. It follows that taking  $Ei^G/Ai^G > 1$  as a criterion for judging stability in beam sea and wind is on the safe side, while taking  $Ei^S/Ai^S > 1$  as a criterion involves some danger.

(3) From conservation of energy, it is only natural that  $Ei^S/Ai^S$  should not in any case be greater than 1.

We have checked the tests, but the results repeat, and we have to present them as they were. This paradox probably is the result of using 'static' data calculated for upright condition to explain unsymmetrical rolling phenomena. In this respect, it is also favourable to adopt the criterion based on gust (no heel) situation.

#### 4. ON EVALUATING INTACT STABILITY IN FOLLOWING OR QUARTERING SEA

For those engaged in practical work here, little attention is paid to the danger of capsizing in following or quartering

seas since no definite criterion has been specified in our statutory stability rules[2]. Although Jiao Tong University had initiated a research on stability in following seas as early as 1964[12], and a method for assessing GZ curves in astern waves was developed recently together with computer programme[13], they are not employed in practical use. This situation is most regretful.

Owing to periodic variation in GM with a frequency equal to wave encounter frequency, a ship travelling in following waves may suffer heavy rolling under certain conditions[14]. The so-called parametric excitation phenomena can be illustrated by formulating rolling motion in the form of Mathieu equation:

$$\frac{\alpha^2 \phi}{\alpha \tau^2} + [p + q \cos(\omega_e t + \epsilon)] \phi = 0 \quad (6)$$

where

$$p = \frac{1 - k}{\Lambda^2 \phi}$$

$$q = \frac{\delta GM}{GM \cdot \Lambda^2 \phi}$$

where GM is the mean value of the meta-centric height;  $\delta GM$ , the amplitude of GM variation;  $\Lambda \phi = \omega_e / \omega \phi$ , the tuning factor;  $\omega_e$ , the frequency of encounter;  $\omega \phi$ , the natural frequency of rolling; K, the damping coefficient;  $\tau = \omega_e t$ ; t, the time;  $\epsilon$ , the phase lag.

Solutions to this equation are shown in fig. 8. There are regions of stability (shaded) and instability, depending on the values of parameters p and q. Since the linear damping coefficient of roll is small compared with 1, and its square is even smaller, so p is equivalent to  $\omega \phi / \omega_e$ . When  $\omega \phi / \omega_e = \frac{1}{2}, 1, 3/2$ , etc, instability occurs, and the first of these regions is the most dangerous because it corresponds to the widest of the unstable zones and has the most rapid instability growth rate.

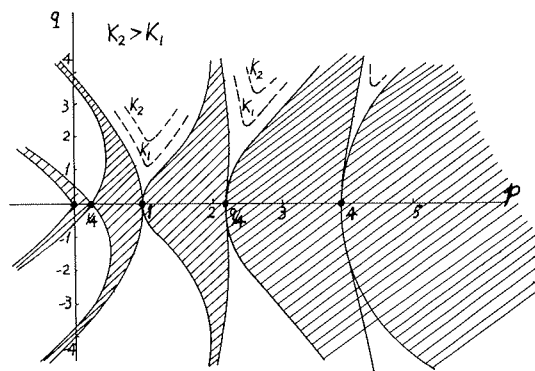


Fig. 8 Solutions for the Mathieu Equation

Parametric resonance may occur in beam seas or quartering seas as well, if only  $\omega_\phi/\omega_e = 0.5$ , and  $\delta GM/GM$  is large enough. One of the authors had confronted with these phenomena in bow, beam and quartering waves in 1965[15], while Dr. Paulling et al found parametric resonance in quartering seas[16]. It is because the model they employed had a  $T\phi = 7.38$  sec. corresponding to 40.4 sec. for the actual ship, and the tests were carried out in natural waves with a modal period of 2 sec.

As to parametric resonance, some points should be noted:

(1) If the ship is stiff, i.e. with large values of GM, parametric resonance is unlikely to occur. The rolling amplitudes may be influenced somewhat by instability as shown in fig.9, but the rolling motion will not build up greatly. Existing test results [1] [5] [15] [16] [17] [18] which are summarized in table 3 indicate that capsizing in following seas occur only for models with small values of GM, most of which are lesser than 5% B. The model tested by one of the authors in 1965 had a GM of 3.9% B, and built up large rolling amplitudes when  $\omega_\phi/\omega_e = 1/2$  [15]. The model he tested recently had a GM greater than 5% B, and no instable rolling was found when  $\omega_\phi/\omega_e = 1/2$ . According to Block [18] (see table 3), his model had a  $\delta GM$  of 0.0157m when GM = 0.042m or 9.1% B, and a  $\delta GM$  of 0.0132m when GM = 0.025m or 5% B. Nevertheless, the model rolled stably with small amplitudes in the former case, whilst it rolled heavily and even capsized in the latter case. All of the models Dr. Paulling tested had small GM, leading to a high ratio of capsizing. Among them, SL-7 model capsized much less than the 'American Challenger' which had a very small GM[19]. Dr. Kawashima gave out similar results. His first test condition had a GM of 0.586m, and the second condition 0.215m. The model capsized many times in the second condition either in regular or irregular waves, but in the first condition it remained uncapsized. Dr. Bovet et al proved through their calculations[20] that the 'American Challenger' would capsize in waves with 250'-400' in length ( $\omega_\phi/\omega_e = 1/2$ ) and 3'-4' in height, if GM = 0.56' or 0.75% B. If GM was raised to 0.86' or 1.15% B, the heights of waves with same lengths required to capsize the ship would be in the order of 15'-20'. It is true that GM is not the unique criterion for judging stability, but it is an important criterion, especially for parametric resonance conditions.

(2) The rolling damping of the ship has a prominent influence on the stable and instable regions. As can be seen from the dotted lines in fig.8, the stable regions widen as rolling damping increases.

(3) When there are other excitations besides regular waves, parametric excitations may still appear, as can be seen from fig.10, which is taken from [15]. The model

was under the excitations of waves and rudder, and the action of the rudder only changed the rolling amplitudes a little. Dr. Paulling carried out his experiments in natural waves, and he got parametric excitations as well [19]. So it seems reasonable to take account of parametric excitations in evaluating stability criterion for following seas.

On the ground of aforesaid points, we propose that the stability in following seas be evaluated as follows:

(1) The GZ curves in waves can be calculated under the following assumptions: the ship is travelling in a longitudinal wave with wave crest at amidships, and the wave has a length equal to the length of the ship and a height equal to  $0.328\lambda$  or 0.63. The computer programme given in [13] may be adopted.

(2) Taking these curves as a basis, a method similar to beam sea condition may be followed. Or, when GM is small, the rolling amplitudes of parametric resonance have to be calculated, taking  $\omega_\phi/\omega_e = 1/2$  and 1, and the action of the gust to be taken into account. Since wind direction may deviate 3 compass points with the prominent wave direction, the heeling moment of the wind may be taken as 0.7 times of that in the beam sea condition. The ship is required to satisfy the inequality (5) under these conditions.

(3) Besides, a minimum value of GM should be specified. In the authors' opinion, the GMs given by the IMCO are too small. In order to take account of other disturbing factors, a minimum GM of 0.5m is recommended. Small vessels should have values of GM higher than 0.5m.

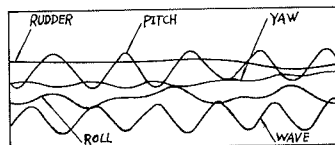


Fig.9 Test Record [15]

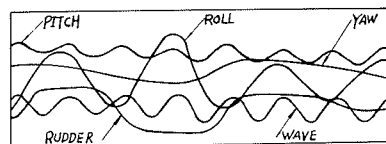


Fig.10 Test Record [15]

Table 3. Model Tests on Capsizing and Parametric Excitations

| Author                                  | Model dimensions<br>L x B x T (m)           | GM<br>( m )        | $\delta GM$<br>(m)     | Wave                                 | Remarks   |
|---|---|--------------------|------------------------|--------------------------------------|---|
| Kure & Bang[1]<br>Kawashima et<br>al[5] | 3.66 x 0.603 x 0.259                        | 0.64               |                        | regular waves in tank                | capsized.   |
|   | 1.771 x 0.514 x 0.154                       | 0.084              |                        | regular & irregular waves<br>in tank | no capsizing occurred, even<br>in case of shift of cargo.                                       |
|   |   | 0.031              |                        | same as above                        | capsized twice in irregular<br>quartering waves, and twice<br>in regular beam waves.            |
| Ji Xi-qi [15]                           | 1.75 x 0.28 x 0.125                         | 0.011              |                        | regular waves in tank                | Capsized in case of shift of<br>cargo.  |
|   |   |                    |                        |                                      | parametric resonances<br>occurred at heading angles<br>of 60°, 90° & 120°, but<br>not capsized. |
| Paulling et<br>al[16]                   | SL-7  | 0.00259            |                        | natural waves in sea bay             | capsized once due to para-<br>metric resonance in<br>quartering waves [19].                     |
|   | 4.99 x 0.588 x 0.178                        | 0.014              | 0.00287                | same as above                        |   |
|   |   | 0.00229            | 0.00231                | same as above                        |   |
|   |   | 0.0115             |                        |                                      |   |
|   | 'American Challenger'<br>5.23 x 0.76 x 0.30 | 0.00259<br>0.00564 |                        |                                      | capsized 22 times out of 132<br>due to parametric resonance<br>in quartering waves.             |
| Roden [17]                              | 2 x 0.32 x 0.163                            | 0.0065             |                        | tested in natural lake               | capsized when GM 0.0065 m.<br>not capsized when GM 0.009 m.                                     |
| Blocki [18]                             | (Cylindrical model)                         | 0.0112             |                        | regular waves in tank                | stable rolling  |
|   | 1.815 x 0.45 x 0.2085                       | 0.026              | 0.0044                 |                                      | do  |
|   |   | 0.042              | 0.157x10 <sup>-1</sup> |                                      | do  |
|   |   | 0.023              | 0.094x10 <sup>-1</sup> |                                      | capsized  |
|   |   | 0.023              | 0.100x10 <sup>-1</sup> |                                      | do  |
|   |   | 0.025              | 0.132x10 <sup>-1</sup> |                                      | do  |
|   |   | 0.023              | 0.181x10 <sup>-1</sup> |                                      |   |

## 5. CONCLUSIONS

(1) In evaluating the stability of ships, two dangerous conditions should be chosen:

- (a) beam seas with wind broadside on;
- (b) following seas under the joint actions of waves and wind.

(2) In beam sea condition, the criterion based on gust (no heel) situation is more rigorous.

(3) More attention should be paid to the stability in following seas, and corresponding criterion has to be augmented to the existing stability rules. A procedure for calculation is recommended.

(4) The importance of GM is emphasized, especially for the following sea condition. A minimum value of 0.5m is recommended.

## ACKNOWLEDGEMENT

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## STABILITY CRITERIA FOR VESSELS OPERATING IN A SEAWAY

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### ABSTRACT

In spite of stability regulations for small vessels enforced in Norway in 1969, several vessels have capsized and disappeared in the years since. Some of the vessels did not comply with the stability regulations, but others did.

The paper summarizes the intact stability research carried out in Norway to improve upon this situation. This research has been, and is still concentrating upon testing the response of vessels in physical situations by use of theory and/or model experiments. The physical situations are selected so as to "extreme or critical", based upon ocean statistics for the area of operation for the vessel.

Primarily, the results from the research are intended to be used to check new designs, to ensure that they will contain ability to sustain dynamic strains imposed by the seaway with a high, accepted probability. Such criteria, taking dynamic behavior into consideration, are lacking today.

The future stability criteria have to take into consideration both the vessel design, the operation and the environmental factors. The probabilistic approach to capsizing of an intact vessel will be the most rational criteria on which to base a stability approval. So far, many uncertain suggestions have to be made.

A more practical approach can be made through demands set to the vessel construction based upon a response analysis and further through operation manuals that describes stability and the vessels' response with routines and warnings.

### 1. INTRODUCTION

Most countries have adopted the IMCO stability recommendations. The rules are reflecting still water situations, but are statisticly also compared and adjusted according to investigations of earlier accidents in rough seas. Thus they are also reflecting safety in waves.

However, several small vessels have been wrecked off the Norwegian coast in recent years. And much effort has been put into finding out the reasons for these accidents. Several of these vessels seems to have capsized in spite of having fulfilled the stability rules. Apparently, the required GZ and initial stability are not a safeguard against all situations which can occur in rough seas.

The stability and safety problem of a vessel in rough seas is very complex and depends upon the weather, the vessel and human factors. Therefore, substantial amounts of work must be done in the future to increase the safety at sea.

The vessel's righting arm curve (GZ-curve) has often a shape as shown in Fig. 1. If  $\phi > \phi_v$ , the vessel will normally capsize, i.e. it will turn over to the stable position of  $\phi = 180^\circ$ .

If  $\phi = \phi_f < \phi_v$ , and the opening in question is allowing large amounts of water to enter, there is no longer a case of intact stability. The further turn of events depends on several factors, but the ultimate result may well be a capsizing if the ingress of water can not be controlled.



In stability regulations, the GZ-curve is regarded as non-existent if  $\theta > \theta_f$  or  $\phi_f$ , implying that these angles are critical to the safety of the vessel.

For several reasons,  $\phi_f$  and  $\phi_v$  may be exceeded due to one or more external moments, by loss of stability, or by combinations of both.

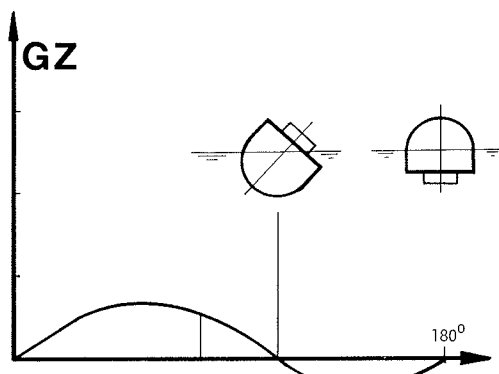


Fig. 1 Stability Curve for a Vessel.  
 $\phi_f$  = angle of flooding  
 $\phi_v$  = angle of vanishing stability

To establish intact stability criteria for reducing the probability of disasters, three main factors must be considered as illustrated in fig. 2, namely:

- the vessel design
- the operation
- the environment

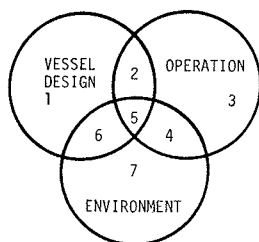


Fig. 2. Factors in safety consideration

As an example the numbers can indicate the following type of accidents:

- (1) Capsize when launched due to faulty design.
- (2) Capsize due to turning at high speed.
- (3) Capsize due to overloading.
- (4) Capsize due to high speed in following waves.

- (5) Capsize due to combination of bad weather, lack of knowledge of stability by operator, low stability.
- (6) Capsize due to excessive amounts of water on deck (too high bulwark), regardless of operational measures (heading).
- (7) Capsize due to heavy icing combined with exceptional seaway that prevent clearing off the ice.

From these examples, it is evident that the probability of capsizing is depending on the factors illustrated in fig. 2.

## 2. STABILITY RESEARCH CARRIED OUT IN NORWAY IN THE RECENT YEARS

### 2.1 "Helland-Hansen"

The research vessel "Helland-Hansen" capsized in 1976 in a breaking wave. The accident happened in deep water in a normal weather condition (light gale). The vessel complied with the IMCO recommendations. During the investigation, model tests were conducted with a model in different breaking waves.

The following main conclusions were arrived at:

- A breaking wave of abt. 5 m height heeled the model to abt. 70°. The model remained in a stable side-position, capsized.
- Increasing stability did not decrease the heel after impact.

In total this investigation gave new insight into extreme response of a vessel due to breaking waves [1].

### 2.2 Ships In Rough Seas (The SIS-project).

A detailed investigation of previous accidents of smaller cargo vessels and fishing boats made it clear that "Helland-Hansen" was not the only vessel that capsized due to waveaction. The weather conditions and the waves are important factors when a capsizing occurs. Present stability requirements are not adequate to prevent capsizings. In the project it was found important to study extreme response and wave situations to try to find a new solution on stability and safety problems. Model experiments and calculations were carried out in order to study capsizing due to breaking waves.

In the initial phase of the project, meteorological and statistical data about the occurrence of breaking waves along the Norwegian coast were collected in order to obtain a picture of how the waves are formed under different weather conditions [2].

Statistics on steepness parameters and group formation for extreme waves and calculations on current and sea bed refraction were elaborated. This work was carried out to be used in a forecasting model for

breaking waves along the Norwegian coastline [2].

Sea areas along the coast with dangerous waves under certain conditions were localized in cooperation between The Pilot Authorities and local fishermen. In such areas the probability for the occurrence of extreme/breaking waves is relatively high [4].

Model tests with a two-dimensional ship section were carried out in simulated deep water breaking waves.

In the model the wave's geometry and velocity distribution acceleration etc. are calculated [6].

The calculations take into account rolling as well as heave and sway motions. The aim was to improve the understanding of which forces and movements occur when a vessel capsized in a breaking wave [6].

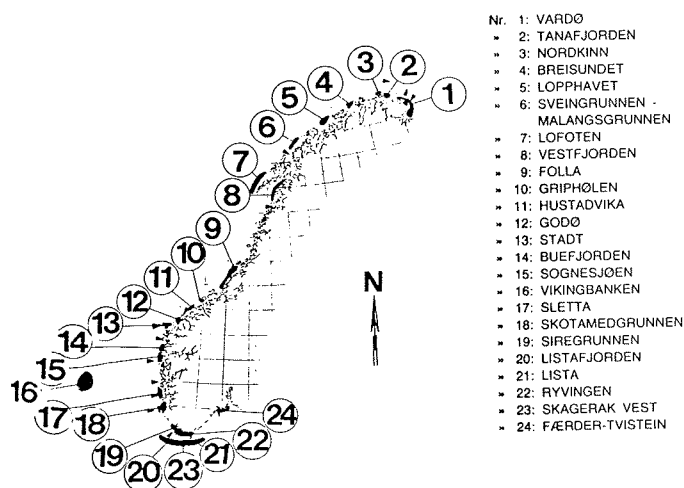


Fig. 3 Exposed areas along the Norwegian Coast. ref [4].

A simulation program that calculates the statistical distribution of rolling maxima for vessels in moderate or irregular waves was developed [7].

The SIS-project also looked into the problems of future stability criteria. It was found important, in principal, that such criteria should be based on physical extreme situations.

### 2.3. Shifting Of Cargo

Shifting of cargo is a problem of great importance when a vessel is capsizing in waves. Last winter three Norwegian cargo vessels of similar type wrecked off in bad weather. Shifting of cargo due to extreme responses can be the direct cause of the capsizing. These problems were looked upon in a research program in 1981/82. It seems to be important to know more about the

accelerations onboard a vessel (in the cargo area). We also have to understand more about the properties of the different cargos. The dynamics of the cargo shifting is an important factor, and is connected both to general cargo and bulk cargoes.

### 2.4 The Research Program "Stability Criteria"

A new research program on stability problems started in late 1981 and will finish in early 1982. The program is called "Stability Criteria" and is following up the ideas on this matter from the SIS-project. The key-word is extreme physical situations.

In 1982 the work is concentrated on model testing. Different types of fishing and cargo vessels are tested in 5-6 extreme or dangerous wave-situations. In principal, the project is built up as a case study of different vessels exposed to extreme external loadings. Theoretical models from the SIS-project are developed further to be used as tools in a parametric study of the various hull forms.

The program intends to give a judgement of a vessels respons-safety in waves. The following aspects are considered:

- The actual extreme wave situations have to be based on wave and wind registrations from the actual area.
- Each deterministic wave situation must be defined through wave heights, periods, geometry and probability of occurring at the actual area.
- Each spesific situation must be defined through the vessel's position, speed direction and load condition.
- The approval of the vessel depends on its respons in waves. Capsizing, critical rolling angles and critical accelerations are parameters that define a extreme or dangerous situation.

### 3. PHYSICAL SITUATIONS - DANGEROUS SITUATIONS

It is known from earlier investigations and from theoretical models of ship respons that the following physical situations might be dangerous:

- Rolling in beam seas, irregular waves.
- Breaking waves in beam seas.
- Loss of stability in following seas on a wave crest.
- Parametric resonans in following seas.
- "Broaching to" in a following, nearly breaking wave.
- Combination situations in three-dimensional waves, beam seas and following, quartering seas.

### 3.1 Rolling in Beam Seas, Irregular Waves

Rolling in beam seas can be critical for some vessels (low stability, wind, and water on deck) and dangerous situations can occur. In general a primary investigation into a vessel's rolling properties would encourage designers to reduce roll motion by improved hull damping and by paying attention to the wave frequency relation of the normal sea condition in the area of operation in relation to the natural frequency of roll. Such an investigation can be carried out through model testing.

However, it can be recommended to carry out a general evaluation of roll motion for a design (angle, velocity and acceleration) in a moderate beam wave train using a computer program where non-linearities of damping and righting moment are included. The moderate wave spectrum appropriate for the area of operation is synthesized into a wave time series, and a roll simulation is carried out, giving a time series of the roll motion. Such a program was developed by the SIS-project and is called ROLLSI, further described in [7].

### 3.2 Breaking waves in beam seas

Most capsizing in rough weather refer to breaking waves in beam seas as the cause of the accident. The roll response of the vessel to breaking waves should therefore be investigated.

In the project "Stability Criteria" model tests have been conducted for several types of vessels to find the level of a dangerous wave. It is however difficult to find a critical wave height to a particular vessel.

The vessels relative size compared to a breaking wave likely to occur, is of course an important factor. But also the vessel's stability and hull design, the vessel's relative position in the breaking zone and the breaking waves appearance in a wave train are important factors.

Capsizing does define a dangerous situation. Great angles of roll and high accelerations occur in such situations and can also be defined as dangerous. It should be noted that if the vessel is situated just after the breaking zone, no critical levels are exceeded.

In future investigative work theoretical computer models should be used to define this situation. Such a program was developed during the SIS-project (described in [7]). Totally, a large amount of the research effort of the SIS-project was concentrated on vessel response to breaking waves.

### 3.3 Loss of stability in following seas, on a wave crest

It is well known that the righting lever arm can be reduced, and sometimes be totally negative with a wave crest amidships. With the vessel in about the same speed as

the wave phase velocity this situation will last for some seconds and the vessel can capsize as in still water due to negative stability.

This situation can be looked upon strictly with hydrostatic calculations. In reality there is some confusing factors as the forward speed of the vessel creates lifting forces on the hull. When the vessel heels, it will set up an additional uprighting moment thus avoiding capsizing. Model tests have however confirmed that the situation is one of the critical ones, especially for vessels with low freeboard and low stability (SIS-project).

### 3.4 Extreme roll motion due to parametric resonans

This effect can for example be caused by the combination of rolling and a wave mid-ships that passes the vessel in a frequency in resonance step with the rolling frequency. Then the vessel can roll out against a wave-crest amidship, which gives a low GZ, thus increasing the motion without any external moment acting. Such resonance rolling can also happen due to great heave-motions in resonance step with the rolling frequency.

The evaluation of this situation can be made by theoretical response models which are active in several countries today, or by model testing. During the last years the situation has especially been looked upon as a problem for wide beam ro-ro vessels.

### 3.5 Broaching in a following, nearly breaking wave

In following seas the vessel is difficult to steer with a wave crest at the stern because the particle speed in the waves is counteracting the water speed past the rudder. Seamen have often experienced this situation as dangerous. But it is when the wave heights increase in the following situations, the wave becoming steeper, nearly breaking, that a completely unexpected situation occurs.

From model tests (stability criteria-project) we have seen that the vessel is taken by the wave thus increasing the speed far above what it can obtain in calm water. Great lifting forces occurs when the vessel runs down the wave. The vessel is being somewhat course unstable and can easily turn over and heel. With low initial stability the vessel is especially exposed in this situation. Small fishing vessels with flat stern ends are more exposed than others.

### 3.6 Combination situation in three-dimensional quartering seas

Several of the above mentioned situations are likely to happen in combination. Survivors from capsizings would sometimes have difficulties in telling which situation caused the accident. In the project "Stability Criteria" experiments with models in shortcrested three-dimensional seas will be carried out in the new Ocean Laboratory in Trondheim.

Wavelengths and average crest height in an irregular seaway are of interest in a dangerous breaking wave situation. The average wave crest length is a most important parameter to consider against the shiplength. The vessel's course to the average dominating wave directions is also to be considered carefully when the vessel is advancing in quartering seas. This parameter will therefore be studied.

### 4. STABILITY PHILOSOPHY

Referring to Figure 2, it is obvious that design regulations for intact stability of smaller vessels must have a closer relation to environmental and operational aspects than has been the case until the present.

The following example can illustrate a probabilistic approach:

A vessel is supposed to carry cargo and be in ballast respectively from A to B as shown in Fig. 3 for its whole lifetime. Even if fuel etc. is consumed when at sea, the loading condition is assumed constant in ballast and loaded, respectively. The weather conditions are assumed constant during the trip, the potential weather coming from 4 sectors as indicated in figure.

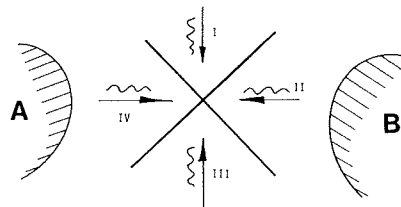


Fig. 3 Vessel route.

Only weather intensity above a certain level is of interest with respect to safety. The intensity will depend on the size and type of vessel, and can be designated to  $h > h_{1/3}$  (critical).

During its lifetime, the vessel will be in one of the situations shown in Table 1.

Table 1. Possible Situation for a Simple Vessel Operation.

| i          | From         | To | Condition | Duration        | No. of times | Weather $h > h_{1/3}$ critical | Sector (Fig.3) | Time                         | Capsize possible |
|------------|--------------|----|-----------|-----------------|--------------|--------------------------------|----------------|------------------------------|------------------|
| 1          | A            | B  | loaded    | $\Delta t$      | $m_1$        | no                             | any            | $m_1 \cdot \Delta t_1$       | no               |
| 2          | B            | A  | ballast   | "               | $m_2$        | no                             | any            | $m_2 \cdot \Delta t_2$       | "                |
| 3          | A            | B  | loaded    | "               | $m_3$        | yes                            | II             | $m_3 \cdot \Delta t_3$       | "                |
| 4          | B            | A  | ballast   | "               | $m_4$        | "                              | IV             | $m_4 \cdot \Delta t_4$       | "                |
| 5          | A            | B  | loaded    | "               | $m_5$        | "                              | I              | $m_5 \cdot \Delta t_5$       | yes              |
| 6          | "            | "  | "         | "               | $m_6$        | "                              | III            | $m_6 \cdot \Delta t_6$       | "                |
| 7          | "            | "  | "         | "               | $m_7$        | "                              | IV             | $m_7 \cdot \Delta t_7$       | "                |
| 8          | B            | A  | ballast   | "               | $m_8$        | "                              | I              | $m_8 \cdot \Delta t_8$       | "                |
| 9          | "            | "  | "         | "               | $m_9$        | "                              | III            | $m_9 \cdot \Delta t_9$       | "                |
| 10         | "            | "  | "         | "               | $m_{10}$     | "                              | II             | $m_{10} \cdot \Delta t_{10}$ | "                |
| 11         | In harbour   |    | "         | $\Delta t_{11}$ | $m_{11}$     | -                              | -              | $m_{11} \cdot \Delta t_{11}$ | no               |
| 12         | On slip/dock |    | "         | $\Delta t_{12}$ | $m_{12}$     | -                              | -              | $m_{12} \cdot \Delta t_{12}$ | "                |
| $\Sigma n$ |              |    |           |                 |              |                                |                | $\Sigma T_6$                 |                  |

The following presentation is given in [8]. Each set of situations are characterized by a matrix  $W_i$ , containing information on position, course, speed and loading condition. In the example above, the weather is supposed to be constant during  $\Delta t$ , although the distance AB can be subdivided. Further, the ship speed is assumed constant, regardless of weather and loading condition.

First, it is noted that the vessel is only endangered for a period of its life time:

$$T_L = \sum_{i=5}^{i=10} m_i \cdot \Delta t_i$$

For the rest of the time,  $T_L - T$  the probability of capsizing ( $\bar{X}$ ) is zero, whereas the probability of non-capsizing ( $X$ ) is 1. Assuming that all the events  $W_i$  are unrelated:

$$P_{T_L}(X) = \prod_{i=5}^{i=10} P(X|W_i)^{m_i} = 1 - P_{T_L}(\bar{X}) \quad (2)$$

$P_{T_L}(X)$  = the probability of not capsizing ( $X$ ) during the time  $T$ .

$W_i$  = situation matrix

$P(X|W_i)$  = the probability of not capsizing, subject to situation  $W_i$ .

$m_i$  = no. of times the situations are (almost) identical

For a fleet of vessels with approximately the same operational pattern, a simplified approach is to regard capsizing events as being a Poisson process, i.e.

$$P_{T_L}(X) = \exp \left[ - \sum_{i=5}^{i=10} (\lambda_{oi} \cdot \Delta t_i \cdot m_i) \right] \quad (3)$$

$\lambda_{oi}$  = capsizing frequency for time  $\Delta t_i \cdot m_i$

Before discussing how the relevant parameters in (2) or (3) can be found, it is obvious that the probability for capsizing for each of the situations  $W_1$  to  $W_5$  should be on a low level. If one situation ( $W_i$ ) dominates, efforts to reduce its probability must be undertaken. First, the factors illustrated in Fig. 2 must be investigated. Clearly, the following main possibilities exist:

- $\lambda_{oi}$  can be reduced by efforts to improve design
- $\Delta t_i \cdot m_i$  can be reduced by operative restrictions (no operation when capsizing is possible (see Table 1) eventually limited to certain areas with high probability of dangerous waves even for  $h < h_{1/3}$  (critical))
- $\lambda_{oi}$  can probably be reduced by better operative training for difficult situations (the essence of the safe handling should be concentrated in an operating manual, which mainly points out the weak sides of the design).

The probability can be found, or rather estimated for some of the situations. There are two approaches:

- Theoretical calculations using a design wave, and calculate the vessel response to different waves with known frequency of occurrence, methods of hydrostatics and hydrodynamics combined with wave statistics.
- Model experiments, exposing the model to design waves as above.

## 5. PRACTICAL APPROACH TO SAFETY IN A SEAWAY

In the future, a probabilistic approach to capsizing of an intact vessel will be the most rational criteria on which to base a stability approval of the design. So far too many assumptions and uncertain suggestions have to be made to make it a practical basis for approval. This situation may change in some years, and this approach should be kept in mind.

However, referring back to Figure 2, it is obvious that design regulations for intact stability of smaller vessels must have a closer relation to environmental and operational aspects than has been the case until the present.

Further, each vessel must be dealt with separately, which requires a more advanced and labour-consuming approbation process.

Also, the control during operation by the operator (skipper) must be improved. It is suggested to produce an operational manual for each vessel. This manual must be understood and followed.

As some smaller vessels may change their operational pattern, and also equipment (fishing vessels), an inclining experiment and operational pattern revision should be undertaken every 4th year, updating the manual mentioned above.

The procedure in the approbation procedure would then be as indicated in Table 2. Referring to Table 1, it should be noted that in some sea-areas, the probability of occurrence of dangerous waves is higher than elsewhere.

| i  | Evaluation procedure                                       | Acceptance level  | Probability of non-capsizing<br>$p_{\Delta ti}(X W_i)$ |
|----|--|---|--|
| 1  | Ship, rolling with beam wind KG varied                     | As 11, 12 below, and<br>$\phi_{max} < \phi_{critical}$  | 1  |
| 2  | "  | "   | 1  |
| 3  | Ship, capsizing with beam wind and water on deck           | -   | 1  |
| 4  | "  |   | 1  |
| 5  | CAPSI with beam wind and water on deck                     | Depending on whether dangerous areas are avoided or not | $\frac{6}{5} \Pi_P(X W_i)^{mi}$                        |
| 6  |  |   |  |
| 7  | Calculation of GZ in wave in conjunction with vessel speed | Depending on operational speed limit is imposed or not  | $\frac{8}{7} \Pi_P(X W_i)^{mi}$                        |
| 8  |  |   |  |
| 9  | As 5, 6  | As 5, 6   | $\frac{6}{5} \Pi_P(X W_i)^{mi}$                        |
| 10 |  |   |  |
| 11 | Present GZ-requirements                                    | Equal or above  | 1  |
| 12 | (IMCO) for cargo ships                                     |   |  |

Table 2. Future evaluation procedure

This would call for a further splitting up of situation 5, 6 and 9, 10. The approval would in some cases depend on operational restrictions for dangerous areas. Possibly, operational restrictions regarding heading of the vessel in relation to waves would have to be given in some cases. Such restrictions, however, are very unsatisfactory for fishing vessels which have to return home in adverse weather to preserve their catch. Further, avoidance of beam seas may involve exposure to quartering and following waves and vice versa.

A further complication is how to deal with icing which may occur quite suddenly, and often in connection with bad weather. The only possibility is to point out the importance of icing forecasts to avoid such situations and further to require acceptable results with icing - of at least IMCO-standards - for areas of operation where icing is known to occur.

The importance of improved weather forecast as regards waves is also pointed out. Such forecasts will enable the skipper to foresee the wave situation in general, and especially in the dangerous areas. The operational manual should therefore contain sufficient information to relate such information to the vessel.

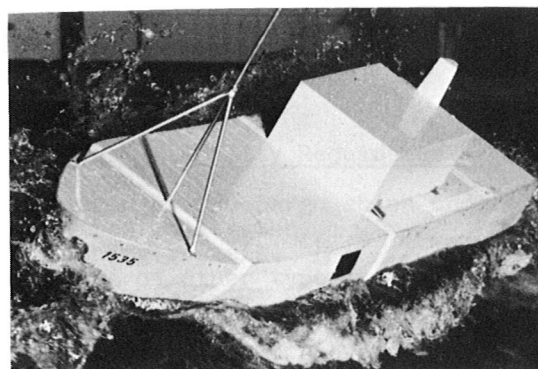


Photo No. 1. Norwegian Fishing Vessel in a breaking wave-situation, model test from the Stability Criteria-project.

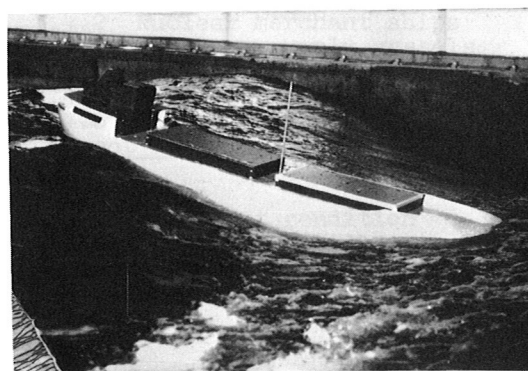


Photo No. 2. Norwegian Coaster in a "broaching to"-situation, model tests from the Stability Criteria-project.

## 6. CONCLUSION

An approbation procedure for intact stability, based on individual judgement of each vessel to ensure an acceptable level of non-capsizing has been outlined. The method is based on wave statistics for the intended areas of operation, and knowledge of the vessel's operations in its lifetime.

If necessary to ensue the acceptable level, operational restrictions are imposed.

All relevant information is to be contained in an operational manual, displayed on a prominent place in the wheel-house.

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## IMO ACTIVITIES IN RESPECT OF INTERNATIONAL REQUIREMENTS FOR THE STABILITY OF SHIPS

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International Maritime Organization

### ABSTRACT

The International Maritime Organization (IMO) known until 22 May of this year as the Intergovernmental Maritime Consultative Organization (IMCO) is the only United Nations specialized agency solely concerned with maritime safety and pollution prevention.

This paper gives a brief account of the work of the Organization in respect of formulating international requirements or recommendations for intact and damage stability of all types of ships. In particular summaries and brief explanations are given in respect of the criteria developed by IMO during the two decades 1962 - 1982.

Following the recommendations by the International Conference on the Safety of Life at Sea, 1960, in principle the Sub-Committee on Subdivision and Stability, later renamed to the Sub-Committee on Subdivision, Stability and Load Lines and the Sub-Committee on Safety of Fishing Vessels, recently amalgamated to the Sub-Committee on Stability, Load Lines and Fishing Vessels Safety, were involved in the development of requirements for intact and damage stability which were included into several IMO instruments - Conventions, Codes and Recommendations - covering a wide range of ship types and sizes, as explained in this paper, which also takes account of current work on intact stability in the amalgamated Sub-Committee.

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# 1 PREAMBLE

The problem of establishing criteria for stability of ships in order to safeguard the safety of ships at sea is an old one; in a practical sense this problem has existed since man first ventured on the seas. It was not until the end of the nineteenth century that attempts were made to evaluate a safe minimum of stability. The lack of basic understanding of the physics of ships capsizing in rough seas and of the influence of different stability parameters on ship's safety prevented the follow-up of proposals such as those put forward by Benjamin[1] or Pierrottet [2] either as statutory requirements or as recommendations.

A statistical analysis with the result of a possible stability criterion was made by Rahola in 1939 [3]. His proposals provided a basis for the criteria adopted or recommended later in several countries. His approach was followed also in IMO's statistical investigations, the results of which conformed roughly with his suggested stability criteria.

Regarding international requirements on intact stability, the 1948 and 1960, as well as the 1974, Conventions for the Safety of Life at Sea (SOLAS) [4] [5] do not include any stability criteria, except the 74 SOLAS for ships carrying grain. The Conventions, however, require that the master shall be provided with information, based upon the results of an inclining experiment, as is necessary to enable him to obtain guidance as to the stability of the ship under varying conditions of service. The intact stability for passenger ships is implicitly regulated by the requirements for positive stability of the ship in the flooded condition. Finally it should be mentioned that the 1966 Load Line Convention [6] also requires the supply of stability data to masters of new ships but without further specifications.

The first important step in the development of international stability requirements, in particular stability criteria, was the recommendation by the 1960 International Conference on the Safety of Life at Sea that:

"The Conference, having considered proposals made by certain Governments to adopt as part of the present Convention regulations for intact stability, concluded that further study should be given to these proposals and to any other relevant material which may be submitted by interested Governments.

The Conference therefore recommends that the Organization should, at a convenient opportunity, initiate studies on the basis of the information referred to above, of: (a) intact stability of passenger ships, (b) intact stability of cargo ships,

(c) intact stability of fishing vessels, and (d) standards of stability information .....

A similar recommendation was earlier adopted by the 1948 SOLAS Conference but intergovernmental exchanges of information did not take place until IMO was established shortly before the 1960 Conference, so that internationally co-ordinated work on stability criteria developed only after the 1960 Conference as a result of the recommendation of the latter conference.

Although losses of passenger ships prompted some national administrations to issue subdivision requirements, international agreement was not attempted until, in 1914, the TITANIC disaster occurred. Delayed due to the First World War, the final agreement on subdivision for passenger ships was reached not earlier than 1929 when a conference concluded the International Convention for the Safety of Life at Sea, 1929. At that time there appeared to be no need for damage stability requirements as passenger ships of those days had relatively narrow beams and deep draughts, which normally provided sufficient residual damage stability. However, design criteria changed over the years when steamers were gradually replaced by motor ships and, in World War Two, many damaged ships capsized before sinking.

Consequent upon this experience, the second SOLAS conference in 1948 included damage stability requirements for passenger vessels in Regulation 7 of Chapter II of SOLAS 1948. These requirements were also included without alteration in the 1960 and 1974 SOLAS Conventions. It was, however, realized that improvements may be desirable as expressed in Recommendation 6 of the 1960 Conference:

"The Conference has considered carefully the question of the watertight subdivision of passenger ships in the light of the results achieved since the International Convention for the Safety of Life at Sea 1948, came into force and has agreed on certain additional requirements calculated to secure greater safety. It recognises, however, that the questions of watertight subdivision and stability deserve further study which the limited time available to the present conference did not permit, and accordingly recommends that the Organization should at the earliest practicable date initiate further studies of watertight subdivision on the basis of proposals which any participating Government may submit, including proposals submitted to the Conference. The objective should be to review the existing criteria of subdivision, stability and damage and to consider the relative merits of these criteria in comparison with other possible criteria from the point of view of safety and practicability."

The ensuing studies were based on the concept of probability of survival [7] and the resulting requirements are given in the Regulations on Subdivision and Stability of Passenger Ships as an Equivalent to Part B of Chapter II of the International Convention for the Safety of Life at Sea, 1960 [8].

## 2 INTACT STABILITY REQUIREMENTS

### 2.1 Introduction

Following the recommendations of the International Conference on Safety of Life at Sea, 1960 [4] the IMO commenced consideration of intact stability in May 1962. At the outset it was recognised that the development of international stability criteria needs to take account of external forces affecting ships in a seaway, and also that this would be a very long-term project. IMO therefore concentrated first on statistical analyses of collated stability data for capsized ships and for ships known to have satisfactory stability. Stability requirements of various nations were also analysed [12]. This work resulted in the adoption in 1968 of the Recommendation on Intact Stability for Passenger and Cargo Ships under 100 metres in length [13] and of a similar recommendation for fishing vessels [14]. The analyses followed, in principle, an earlier approach by Rahola [3] utilizing improved methods, but included a much larger number of ships which had successful stability records and those which had capsized.

Details of the background to the two recommendations were reported by Nadeinski and Jens [15] and by Thomson and Tope [16].

The above two recommendations were subsequently utilized as a basis for recommendations for other types of ships; they also formed a basis for requirements included in the 1977 Torremolinos Convention, although in that Convention an attempt had already been made to include also certain requirements related to the external forces affecting ships in a seaway and during fishing operations. The provisions of resolution A.167 were also later supplemented by other recommendations covering certain additional aspects of intact stability of specific types of vessels.

As always with safety standards, there is no assurance that a ship satisfying the criteria referred to in this paper will, under all circumstances, be safe from capsizing. In the statistical analysis an average line between "safe" and "unsafe" ships was drawn showing that a number of ships having stability parameters lower than recommended had operated successfully over the years and some with higher parameters had capsized. Stability is not the single nor the prime factor which determines safe operation of a ship,

and prudent navigation in a seaway, in particular in severe weather conditions, as well as proper stowage and/or lashing of cargo and proper use of closing appliances, inter alia contribute to a safe voyage. Resolution A.167 therefore includes an important statement:

"Compliance with the stability criteria does not ensure immunity against capsizing regardless of the circumstances or absolve the master from his responsibilities. Masters should therefore exercise prudence and good seamanship having regard to the season of the year, weather forecasts and the navigational zone and should take the appropriate action as to speed and course warranted by the prevailing circumstances."

### 2.2 Passenger and Cargo Ships under 100 Metres in Length

The Recommendation on Intact Stability for Passenger and Cargo Ships under 100 metres in Length as amended to include ships carrying timber deck cargoes [13] constitutes the basic stability requirements for ships of conventional types. It contains all the basic and additional stability criteria for passenger and cargo ships and ships loaded with timber deck cargoes, provisions regarding the inclining test and the stability information, as well as detailed guidelines for a uniform calculation of stability curves, standard conditions of loading and for an approximate stability determination by means of the rolling period.

The principle stability criteria of the recommendation are shown in Figure 1. For ships loaded with timber deck cargoes the basic criteria may be relaxed with respect to metacentric height and dynamic stability, whereas passenger ships have to comply with additional criteria taking into account crowding of passengers to one side of the ship and the angle of heel due to the ship's turning.

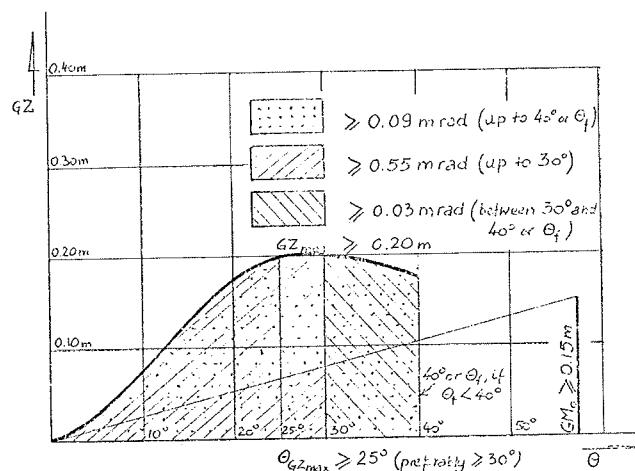


Figure 1.

It was pointed out on many occasions that the criteria in resolution A.167 have a limited applicability to ships larger than 100 metres in length and to very small ships.

An important point to mention is that most of the ships included in the analyses were in loaded condition and those ships which capsized with higher stability parameters than recommended were in lightly loaded or ballast condition.

As only a few cases of ships with small draughts capsized, no attempt was made on a statistical analysis, but it was suggested that the stability of ships with small draughts should be determined by improved criteria, such as weather criteria.

### 2.3 Fishing Vessels

Together with resolution A.167, the IMO Assembly adopted resolution A.168 - Recommendation on Intact Stability of Fishing Vessels [14]. In this recommendation all basic stability criteria are identical to those in resolution A.167 except that a higher metacentric height is required, as the majority of fishing vessels have low freeboards (single deck vessels). Criteria for fishing vessels were obtained by methods of statistical analysis identical with those used for the minimum criteria in resolution A.167 [15][16].

In addition to the above, the resolution A.168 also includes:

1. Recommendation on minimum requirements on icing of fishing vessels.
2. Recommended practice on portable fish hold divisions.
3. Recommended practice for freeing ports on fishing vessels.
4. Recommended practice for exterior hatch coamings and door sills on fishing vessels.
5. Some suggestions to fishermen.

Items 3. and 4. were later further developed and included in the Recommendation on the Construction of Fishing Vessels Affecting the Vessel's Stability and Crew Safety [17].

At the time resolution A.168 was adopted, the point was made that there were many small fishing vessels built without any drawings and therefore no calculations of hydrostatic curves and stability cross curves could be made and the application of the stability criteria of resolution A.168 would not be possible. For that reason a Recommendation for an Interim Simplified Stability Criterion for Decked Fishing Vessels under 30 metres in Length was adopted in 1971 [18].

It was, furthermore, considered that there was a need to make the application of stability criteria more reliable, for

which the Code of Practice concerning the Accuracy of Stability Information for Fishing Vessels was developed [19].

Recognizing that many fishing vessels operate in high latitudes, the Recommendation for Ensuring a Fishing Vessel's Endurance in Conditions of Ice Formation was adopted in 1973 [20].

In 1974 IMO, FAO and ILO agreed on Part B of the Code of Safety for Fishermen and Fishing Vessels [21] which includes the requirements of resolutions A.168 and A.207. As that Code applies to vessels of 24 metres in length and over, in 1979 Voluntary Guidelines for Small Fishing Vessels [22] were developed to cater for the vast majority of such vessels in the world. These guidelines, however, contain only very general stability provisions as at the time of preparation it was felt that more research work was needed to formulate specific criteria for vessels of less than 24 metres in length.

In 1977 an International Conference on Safety of Fishing Vessels [23] was held in Torremolinos, Spain, under the auspices of IMO which adopted the Torremolinos International Convention for the Safety of Fishing Vessels, 1977, applicable to new vessels of 24 metres in length and over, which contains a chapter on stability and associated seaworthiness. The stability criteria of the Convention are basically the same as in resolution A.168 and in the Fishing Vessels Code, except for the required metacentric height which may be reduced to 15 cm for vessels of 70 metres in length and over having complete superstructures.

As an important step forward, the Convention requires consideration of effects such as flooding of fish holds, external forces induced by particular fishing methods, severe wind and rolling, and the presence of water on deck. Guidance is included in the appendices to the Convention on methods of calculation of the effect of severe wind and rolling, of water on deck and ice accretion. The Convention regulates an adequate freeboard for fishing vessels by requiring a minimum bow height and the minimum vertical distance from the deepest operating waterline to the lowest point of the top of the bulwark or to the edge of the working deck. Guidance for calculation of both distances, based on the statistical analysis of motion of fishing vessels on the seaway are also attached to the Convention.

Some recent accidents concerning fishing vessels, e.g. the casualty of MS HELLAND-HANSEN which was thoroughly investigated by Dahle [24] and the results of some model experiments on capsizing phenomena by Morrell [25] led to the plan that the improved stability criteria in the Torremolinos Convention should be complemented by fixing adequate values for the parameters in the various formulae for such criteria.

## 2.4 Ships of Other Types

A number of codes or guidelines developed by IMO for the construction and equipment of types of ships other than passenger and cargo ships and fishing vessels contain additional or modified requirements to those of resolution A.167, as the specific design and operation features of the ships do not always allow direct application of that resolution.

### 2.4.1 Offshore Supply Vessels

The twelfth session of the IMO Assembly in November 1981 adopted Guidelines for the Design and Construction of Offshore Supply Vessels [26]. The basic stability criteria applicable to ships up to 100 metres in length are identical to resolution A.167. However, these vessels have specific design features, such as large deck areas for the carriage of equipment for offshore installations. They also have low freeboards which result in the maximum righting arm occurring at a heeling angle normally lower than 30 degrees. Bearing these features in mind, equivalent stability criteria were adopted (see Figure 2) where relaxation from the requirements of minimum angle of heel at which the maximum righting arm occurs is compensated by increasing the corresponding area under the curve of righting levers up to that angle.

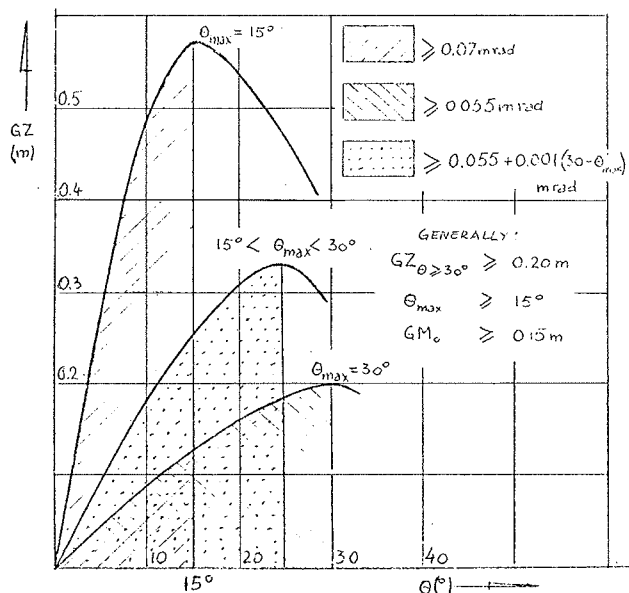


Figure 2.

### 2.4.2 Dynamically Supported Craft

Stability requirements for hydrofoil boats, air-cushion vehicles and similar

craft carrying more than 12 passengers but not more than 450, with all passengers seated and not proceeding more than 100 miles from a port of refuge are dealt with in the Code of Safety for Dynamically Supported Craft [27] for which the provisions of resolution A.167 are obviously not applicable.

This Code makes a distinction between intact stability of the craft in the displacement (hull-borne) mode and in the transient and non-displacement (foil or cushion borne) mode. For the displacement mode specific stability criteria are given, with the inclination of the craft limited to 8 degrees in any direction. For both displacement and non-displacement modes it requires that dynamic stability, roll and pitch stability should be checked for which an outline of the method of stability investigation is given.

The method for displacement mode includes the effect of heeling moments when turning and due to wind pressure and basically comprises equating work of heeling and righting moments and is in principle very similar to the weather criteria of the 1977 Torremolinos Convention and that under consideration at present (see Figure 6). However, the assumed amplitude of roll has to be taken from the upright position of the craft. This weather criteria is given in Figure 3.

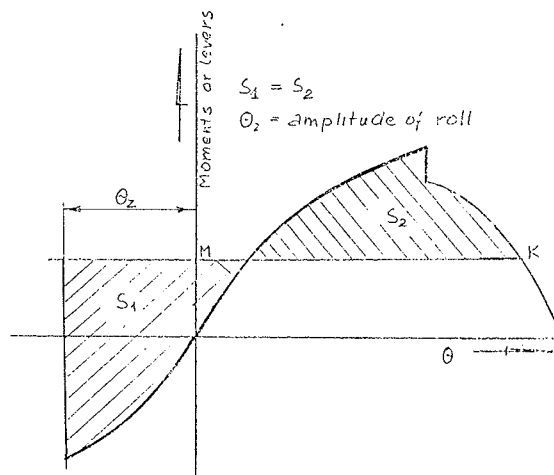


Figure 3.

The method for calculating stability in the transient and foil-borne mode for hydrofoils is not specified and only general recommendations are given. The only exception is a simplified formula for calculating the metacentric height for hydrofoil boats with surface piercing foils which is mentioned as one of the possible methods.

The Code also contains a recommendation for calculating ice accretion similar

to that for fishing vessels.

### 2.4.3 Mobile Offshore Drilling Units

The stability criteria included in the Code for the Construction and Equipment of Mobile Offshore Drilling Units [28] are governed by the entirely different design features of MODUs compared with conventional ships and are therefore different from those in resolution A.167.

The principle for the stability provisions for mobile offshore drilling units is the concept of withstanding of the wind heeling moment (weather criterion). The criterion is given in terms of reserve dynamic stability, i.e. the area between the heeling moment curve and the righting moment curve up to the down flooding angle or to the second intercept (see Figure 4) in percentage of the area under the wind heeling moment curve.

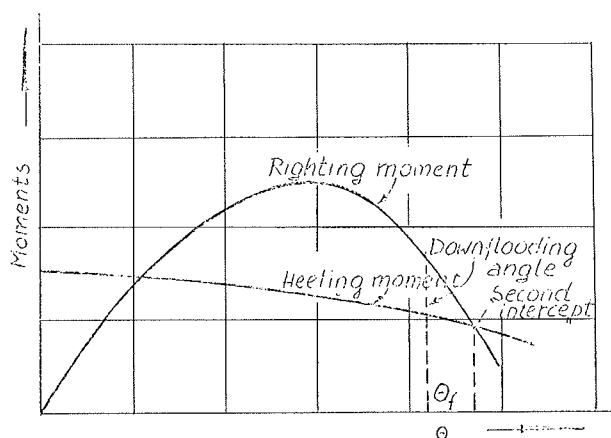


Figure 4.

Details of the calculation of the wind heeling moment which, for the simplest case of rigs of conventional ship type, is a cosine shaped curve, are included in the Code. The assumed wind velocity for normal operational conditions is 70 knots and 100 knots for storm conditions.

### 2.5 Carriage of Grain

Provisions for stability of ships carrying grain in bulk are given in detail in Chapter VI of the 1974 SOLAS Convention [5]. The stability criterion for the assumed shift of grain is illustrated in Figure 5.

Chapter VI also gives the details for the voids to be assumed in the grain holds and the methods for calculating heeling moments due to grain shift.

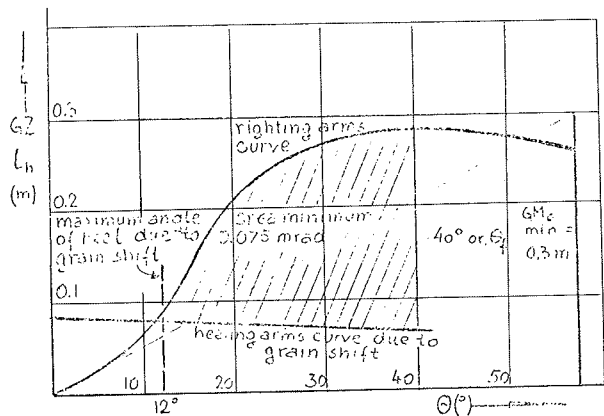


Figure 5.

## 3 DAMAGE STABILITY REQUIREMENTS

### 3.1 Introduction

Damage stability requirements were internationally first introduced in the 1948 SOLAS Convention. However, little and differing experiences had been gained in their application when IMO followed up the Recommendation 6 of the 1960 SOLAS Conference. An extensive study of comparable subdivision and damage stability calculations using the United Kingdom passenger vessel PENDENNIS CASTLE was carried out by numerous delegations which showed a large variety of results, depending on the assumptions taken such as for the service conditions regarding stability, the assumption and distribution of permeabilities for volumes and free surfaces, the tightness of internal divisions and the method of calculation.

There were, in particular, different views concerning the requirement that the damaged ship should be assumed to be in the worst anticipated service condition. On the one hand it was argued that there was a need for verifying the damage stability for one particular intact stability condition only. On the other hand, however, it was pointed out that it would normally not be possible to state exactly which would be the worst service condition concerning intact stability and reserve buoyancy for the numerous damages to be assumed.

In the Equivalent Regulations on Subdivision and Stability [8] the second argument is accepted and it is stipulated that the calculations have to be made for a

number of draughts and the most unfavourable trims in the operating range of draughts as detailed in the Guidance Notes for the Application of Regulations (see [8]).

Subdivision and damage stability requirements for cargo ships were introduced for the first time in 1966 by the International Conference on Load Lines [6]. According to these requirements, tankers are assumed to be loaded to the summer load line only, as the object of the calculations is to verify the assignment of a Type A ship freeboard.

Subdivision and damage stability requirements for tankers were later also included in the International Convention for the Prevention of Pollution from Ships, 1973/78 [9] and the IMO Bulk Chemical and Gas Carrier Codes [10] [11]. These requirements, however, are aimed at keeping the vessels afloat in a stable position, in order to prevent loss of the entire cargo and thus avoid massive pollution of the sea. Details are reflected in the Guidelines for the Uniform Application of the Survival Requirements of the Bulk Chemical Code and the Gas Carrier Code [10] which were adopted in 1980 and appended to the Bulk Chemical Code.

The requirements for compartmentation and damage stability for mobile offshore drilling units and offshore supply vessels are governed by the possibility of contact of the MODUs with these vessels during transfer of goods from the vessels to the units at sea, which may cause restricted damage to both craft.

From the outset in 1962 the Subcommittee on Subdivision and Stability also kept in mind the request of the 1960 SOLAS Conference to consider subdivision and damage stability requirements for cargo ships. This work had been partly postponed pending finalization of the passenger ships' requirements based on the probability of survival concepts and partly carried out for specific types of vessels, as referred to in this section. So far the views are very much divided in respect of formulation of such requirements for general dry cargo ships and the trend to consider specific types of vessels such as roll-on/roll-off ships may continue.

### 3.2 Passenger Ships

The International Conventions for the Safety of Life at Sea, 1948, 1960 and 1974 [4] [5] contain damage stability requirements for passenger ships which were carried forward into the 1981 SOLAS Amendments [29] virtually unchanged. Similarly damage stability requirements are also contained in the Regulations Equivalent to Part B of Chapter II of SOLAS 1960, (resolution A.265) [8].

The damage stability requirements in these SOLAS instruments specify the number of adjacent compartments the flooding of which the ship has to withstand, which are, according to the factor of subdivision, one or two or, in theory, three compartments. They also specify the permeability of spaces and the assumed extent of damage. In resolution A.265 single compartments have to be assumed damaged but when more than 600 persons are carried two compartments have to be considered for a certain part of the ship. These requirements are summarized in Tables 1 and 2. Damage stability criteria for the condition of the ship after damage and, in the case of asymmetrical flooding, after equalization measures have been taken are specified. Concerning the latter, a recommendation on a standard method for establishing compliance with the requirements for cross flooding arrangements [30] was agreed. The damage assumptions and survival requirements for passenger ships are shown in Tables 1 and 2 respectively.

For completeness it should be mentioned that the damage stability requirements of the 1960 SOLAS Convention apply also to ships covered by the Special Trade Passenger Ships Agreement, 1971 [31].

### 3.3 Oil and Chemical Tankers and Gas Carriers

As mentioned above, subdivision and damage stability requirements are given in Regulation 27 of the International Convention on Load Lines, 1966 [6] for Type 'A' ships (tankers). The damage stability criteria have to be satisfied when two adjacent empty compartments excluding the machinery space of a tanker of over 150 metres in length are assumed flooded and if the tanker is over 225 metres in length including the machinery spaces (see Tables 1 and 2). In order to provide for uniform and unambiguous interpretations of the subdivision and damage stability requirements the Organization recommended more detailed requirements as equivalent to Regulation 27 [32].

Damage stability requirements for tankers are also contained in the International Convention for the Prevention of Pollution from Ships, 1973/78 [9]. They differ substantially from those in the 1966 Load Line Convention in that damage to machinery spaces is excluded only for tankers below 150 metres in length and in that range of positive stability minimum GZ value and minimum area under the righting lever curve are specified.

The damage stability requirements for chemical tankers and gas carriers are set out in the Bulk Chemical [10] and Gas Carrier [11] Codes respectively and are similar to those given in the 73/78 MARPOL

Table 1. Summary of damage assumptions

| Type of ship  | Standard of damage   | Assumed extent of damage  |   |   |                               |
|---|--|---|---|---|-------------------------------|
|   |  | Longitudinal  | Transverse  | Vertical  |                               |
| Passenger ships<br>SOLAS 1960<br>SOLAS 1974<br>1981 Amendment<br>SOLAS equivalent       | Floodable length concept.  | 3.05(3.0)m + 0.03L or 10.67(11.0)m whichever is less <sup>a/</sup>  | 0.2B  | From the base line upwards.   |                               |
|   |  | 3.0m + 0.03L or 11.0m whichever is less <sup>b/</sup> .   | 0.2B  | At the level of the subdivision load line   |                               |
| Tankers<br>LL 66  |  | not specified   |   |   |                               |
| LL 66, equivalent<br>MARPOL (amended)<br>Chemical tankers<br>Gas carriers               | Between transverse watertight bulkheads <sup>d/</sup>                    | <u>side damage</u> <sup>c/</sup><br>1/3 <sup>2/3</sup> or 14.5m, whichever is less.<br><u>bottom damage</u><br>1. For 0.3L from the forward perpendicular<br>1/3L <sup>2/3</sup> or 14.5m, whichever is less.<br>2. Any other part of the ship<br>1/3L <sup>2/3</sup> or 5m, whichever is less. | B/5 or 11.5m, whichever is less.<br><br>B/6 or 10.0m, whichever is less.<br><br>B/6 or 5m, whichever is less. | Without limit <sup>f/</sup> .<br><br>B/15 or 6m <sup>e/</sup> , which-ever is less <sup>f/</sup> .<br><br>B/15 or 6m <sup>e/</sup> , which-ever is less <sup>f/</sup> . |                               |
| Nuclear ships   | In all conceivable locations along the length of the ship.               |   |   |   |                               |
| Offshore supply vessels   | Anywhere in the vessel's length between transverse watertight bulkheads. |   | 0.76m <sup>d/</sup>   | From underside of the cargo deck for the full depth.  |                               |
| Dynamically supported craft   |  | <u>side damage</u><br>0.1L or 3m + 0.03L or 11m, whichever is less.<br><u>bottom damage</u><br>As side damage.  | 0.2B or 5m, which-ever is less.<br><br>As side damage.  | Full depth.<br><br>0.02B or 0.5m, whichever is less.  |                               |
| MODUs<br>Surface units<br>Self-elevating units  |  |   | 1.5m  | From base line upwards without limit.   |                               |
| Column stabilized units   | Only columns on the periphery assumed damaged.                           |   |   |   |                               |
| Special purpose ships carrying the following number of special personnel: <sup>i/</sup> | ≤ 50   | Anywhere in the ship's length between transverse watertight bulkhead <sup>d/</sup> , except of machinery space.   | 1/3L <sup>2/3</sup> or 14.5m, whichever is less.  | B/5 or 11.5m, whichever is less.  | Without limit <sup>f/</sup> . |
|   | > 50 ≤ 200   | As above, including machinery space <sup>g/</sup> .   |   |   |                               |
|   | > 200  | as for passenger ships  |   |   |                               |

## Footnotes:

a/ if  $F \leq 0.33$  the longitudinal extent of damage to be increased.

b/ when  $N > 600$ , additional side damage including transverse bulkheads anywhere within a length equal to  $(\frac{N}{600} - 1.0)L_S$  measured from the forward terminal of  $L_S$  (ship's length).

c/ inboard from the ship's side at right angles to the centre line at the level of the summer load line.

d/ spaced at the assumed longitudinal extent of damage.

e/ for gas carriers and nuclear ships this figure is 2m.

f/ taken from the moulded line of the bottom shell plating at centre line upwards.

g/ in any such ships of 100m in length and over the assumed damage should include damage to collision bulkhead.

h/ bottom damage not specified in LL 66 equivalent.



Table 2. Summary of survival requirements

| Type of ship  | At final equilibrium waterline  |   |  | Equalization time |
|---|---|---|--|-------------------|
|   | Waterline   | Maximum angle of heel   | Residual stability   |                   |
| Passenger ships<br>SOLAS 1960<br>SOLAS 1974<br>1981 Amendment<br>SOLAS Equivalent                   | Margin line not submerged.  | 7°; 15° may be allowed in special cases.  | $GM \geq 0.05m$ for symmetrical flooding <sup>a/</sup>   | $\leq 15$ min     |
|   | No part of the bulkhead deck immersed.  | 7° for one compartment flooding; 12° for two or more adjacent compartment flooding; 20° before equalization.  | $GM \geq 0.05m$ ; or<br>$GM = 0.003 \frac{B_2 (N_1 + N_2)}{\Delta F_1}$ ; or<br>$GM = 0.015 \frac{B_2}{F_1}$<br>whichever is greater $a^{**}/b/$                   | $\leq 10$ min     |
| Tankers<br>LL 66  |   | 15°   | $GM \geq 0$  |                   |
| LL 66, equivalent<br>MARPOL, amended<br>Chemical tankers<br>Gas carriers                            | Below the lower edge of any opening through which progressive down-flooding may take place <sup>c/</sup> .                              | 15°; (17° <sup>d/</sup> )<br>25°; (30° <sup>d/</sup> ) (for chemical tankers <sup>a**/c/</sup> )<br>30° <sup>c/</sup>                                       | Range $\geq 20^\circ$ beyond equilibrium position<br>$GZ \geq 0.1m$<br>$e \geq 0.0175m$ rad. $a^{**}/$   |                   |
| Fishing vessels   | Below the lower edge of openings of after end of the top poop superstructure at centreline.   | 20°   | Range $\geq 20^\circ$ beyond equilibrium position<br>$GZ \geq 0.1m$<br>$e \geq 0.0175m$ rad. for upright position.   |                   |
| Nuclear ships   | Below the lower edge of any opening through which progressive down-flooding may take place.   | 15°; (17° <sup>d/</sup> )   | Range $\geq 20^\circ$ beyond equilibrium position<br>$GZ \geq 0.2m$<br>$e \geq 0.035m$ rad. $a^{**}/$  |                   |
| Offshore supply vessels   |   |   | Range $\geq 20^\circ$ beyond equilibrium position<br>$GZ \geq 0.1m$ $a^{**}/$ .  |                   |
| Dynamically supported craft <sup>e/</sup>   | 76mm below the lower edge of openings   | in any dir. 8°; 12° may be permitted <sup>a***/</sup> .   | positive   |                   |
| MODUs   | Below the lower edge of any opening through which progressive flooding may take place.  |   | Sufficient to withstand wind velocity of 25.8m/sec (50 knots) from any direction.  |                   |
| Special purpose ships carrying $\leq 50$ the following $> 50 \leq 200$ number of special personnel: | Below the lower edge of openings through which progressive flooding may take place (before equalization and/or in the process thereof). | 7° in the case of flooding between adjacent watertight bulkheads; 12°, in the case of flooding involving the collision bulkhead; 20° prior to equalization. | Range $\geq 20^\circ$ beyond equilibrium position<br>$GZ \geq 0.1m$<br>$GM \geq 0.05m$ for symmetrical flooding or in upright position for unsymmetrical flooding. |                   |
| $> 200$   | as for passenger ships  |   |  |                   |

## Footnotes:

a/ in the intermediate stages of flooding; <sup>a/</sup> special consideration, <sup>a\*\*/</sup> stability should be sufficient, <sup>a\*\*\*/</sup> heel of 16° may be permitted.

b/  $\Delta$  = displac.(t),  $B_2$  = extr.mould.breadth,  $F_1$  = effect.damage freeboard,  $N_1$  = number of pers.for whom lifeboats are provided,  $N_2$  = number of pers. in excess of  $N_1$ .

c/ also in any stage of flooding.

d/ if no part of the deck is immersed.

e/ should remain afloat for 30 min. or 3 x time of evacuation plus 7 min.



Convention. Although initially damage assumptions and survival requirements in the above three instruments differed, they were harmonized and are at present, in the latest editions of the Codes and in amendments proposed to the MARPOL Convention, virtually identical (see Tables 1 and 2).

### 3.4 Ships of Other Types

#### 3.4.1 Fishing Vessels

Damage stability requirements in the 1977 Torremolinos Convention are given only for vessels of 100 metres in length and over carrying 100 persons or more, which should be capable of remaining afloat with positive stability after any one compartment is assumed damaged. The guidance attached to the Convention specifies damage assumptions and survival requirements, the latter not being very different from those for tankers referred to in 3.3. These include the range of positive stability, minimum GZ value and minimum area under the righting lever curve, as shown in Tables 1 and 2.

#### 3.4.2 Nuclear Merchant Ships

Basic requirements for nuclear powered merchant ships are given in identical form in Chapter VIII of the 1960 and 1974 SOLAS Conventions ([4][5]).

The supplementary requirements concerning subdivision and damage stability in the Nuclear Merchant Ships Code [33] contain damage assumptions which are almost identical to those adopted for oil and chemical tankers and gas carriers, except for the vertical extent of bottom damage (see Table 1). The damage, however, should be applied to all conceivable locations along the length of the ship and at least a two-compartment standard of subdivision should be obtained (see Table 2).

#### 3.4.3 Offshore Supply Vessels

Offshore supply vessels are vulnerable when operating in close proximity to offshore structures. The Guidelines for Offshore Supply Vessels [26] therefore include damage assumptions which in this case were taken as minor damage which might occur during loading and unloading operations at sea (see Table 1). The survival requirements again follow the pattern adopted for other types of ships, but were changed in particular with regard to the admissible heeling angle which is lower, taking into account the low initial freeboard of this type of vessel.

#### 3.4.4 Dynamically Supported Craft

The Code for Dynamically Supported Craft [27] refers to passenger craft and

requires that in the displacement mode, following postulated damage specified in the Code, the range of residual stability after damage is adequate in the judgement of the Administration.

As these craft carry passengers, the admissible angles of heel following damage are considerably lower, as for cargo ships. Details of damage assumptions and damage stability criteria are given in Tables 1 and 2.

#### 3.4.5 Mobile Offshore Drilling Units

The Mobile Offshore Drilling Units Code [28] contains only general requirements in respect of subdivision and damage stability, as indicated in Tables 1 and 2. The reserve stability of the rig in damaged condition is required to be sufficient to withstand the wind heeling moment based on wind velocity of 50 knots (25.8 m/sec). This of course takes into account the specific configuration and specific mode of operation of the units.

The need for amendments to the Code is being considered, having regard to the loss of the hotel platform ALEXANDER L. KIELLAND [34] and also to other accidents of offshore units.

#### 3.4.6 Special Purpose Ships

A Code of Safety for Special Purpose Ships is in the process of finalisation, which includes damage stability requirements. The proposed code covers types of vessels such as survey vessels, training ships, fish and whale processing ships and other types of vessels which the Administration would deem to come under the scope of the Code. The requirements will apply to special purpose ships of 500 gross tonnage and more carrying 12 special personnel and more. The number of passengers carried, which should not exceed 12, must be included in the number of special personnel. The proposed damage stability requirements are also shown in Tables 1 and 2.

## 4 CURRENT WORK ON INTACT STABILITY CRITERIA

### 4.1 Introduction

Following the adoption in 1968 of criteria for the configuration of intact stability curves for ships in resolutions A.167 and A.168, IMO pressed on with consideration of improved criteria including criteria for ships in lightly loaded and ballast conditions and also for very small ships and ships above 100 metres in length also taking into account unusual designs.

Amongst other things the improved stability criteria were to be based on comparison of the heeling moments due to external forces and the ship's righting

moment. At that time no model for this approach had been agreed, i.e. whether to consider the ship in upright or inclined position; in regular beam or irregular seas or in following waves, because it was thought essential to collect first and foremost data on wind and waves.

For this purpose a joint group was established in which experts from ECOR, IAPSO, IMO, ISSC, IUTAM, WMO and UNESCO/IOC\* participated. This group considered for five sessions items such as wind effects, wave height, statistics on wave groups, sea spectra of various sea areas and analyses of stability casualty records. The group concluded that it had provided sufficient information and that "the whole problem of ship stability must be studied with the sea represented in spectral form" and "having regard to statistics of well developed groups of waves". "Quartering and following seas are critical and the study of stability in these conditions must allow response with seven degrees of freedom including rudder." However "capsizing may occur in seas that are not necessarily extreme and poor navigation may place the ship in hazard."

With the information on external forces in hand, in 1974 a long-term work programme on improved stability criteria was agreed, which included a review of theoretical and experimental work concerning the capsizing phenomena, identification of relevant ship parameters, formulation of ship response to wind and sea, performance of comparative calculations and analysis of casualty data.

This long-term programme is of course very ambiguous and for a short term solution a pragmatic approach was used, that is:

1. to improve the statistical stability criteria (resolutions A.167 and A.168) by considering vanishing stability, ships with small draughts and of small size and to extend them to ships of above 100 metres in length;
2. to develop the so-called weather criteria assuming the ship under idealistic conditions, exposed to beam wind and waves, thus constituting an improved comparative assessment of the ship's stability; and
3. to consider stability of ships in following waves.

Not much has been accomplished regarding the first task, doubts not having been dispelled as to whether the criteria of resolution A.167 could be applied to ships greater than 100 metres in length on the one hand and to small ships on the other hand, like fishing boats, which could capsize in a breaking wave even longitudinally. However some clarification was

reached as reported in 4.2 and 4.3. The second task was virtually the main subject on which the work was concentrated, leaving the third item mainly for the future.

#### 4.2 Angle of Vanishing Stability

The possible fixing of the angle of vanishing stability in the IMO intact stability criteria was considered at the time resolutions A.167 and A.168 were under development. However, the statistical analysis of data for ships which operated successfully led to the conclusion that it would not be possible, in the light of the data available, to indicate a value for the angle of vanishing stability  $\theta_v$ .

It was agreed that the first part of the righting arm curve up to  $40^\circ$  or the angle of flooding was of vital importance and was adequately fixed by the values for the area under the curve and GZ co-ordinates at specific angles of heel. The calculated location of the  $\theta_v$  is dependent on trim before and during heeling and the way the buoyancy of the ship's erections etc. are taken into account. As at a large angle of heel a reduction in stability by flooding and/or by a shift of cargo may be expected, the shape and termination of the righting arm curve at such an angle was considered of less significance.

However, in 1976 the Norwegian vessel M/S HELLAND HANSEN capsized off the west coast of Norway. As a result of the investigations, Norway recommended that for small vessels of less than 45 metres in length the righting lever GZ should be at least 0.10 m at all angles of heel up to 80 degrees [24]. With this recommendation the problem of fixing the angle of vanishing stability at least for small ships again came under consideration.

Several proposals for an additional criterion related to the angle of vanishing stability were advanced, but finally, although as large a value of  $\theta_v$  as possible appears to be desirable, it was found to be impossible to agree on a minimum standard for various reasons, such as:

1. for some types of ships this may be impracticable and such standard would either need to be reduced to a relatively small value or alternatively other types of ships for which a large stability range would be desirable would need to be exempted;
2. there is no known theoretical technique that could be employed to determine the reduction of probability of capsizing associated with the increase of  $\theta_v$ ;
3. the choice of an acceptable value of  $\theta_v$  (such as  $50^\circ$  -  $55^\circ$ ) would be arbitrary which could become a design constraint with detrimental effect;

- obtained by applying criteria of resolution A.167 and the weather criterion according to various national methods of calculations to several ship types was undertaken. It was decided to use for this exercise weather criteria used in the Netherlands, the USSR, the United Kingdom and Japan. The main conclusion of the above exercise was, as expected, that generally the weather criterion is more severe at lower displacements than the criteria in resolution A.167; it is also more severe for ships with large windage area.

The results obtained by applying various methods of calculation of the weather criterion differed widely and in particular the estimated roll amplitude and assumptions regarding the flooding angle had paramount effects. Therefore those parameters had to be standardized first of all before a uniform international method of calculation of the effects of wind and rolling can be agreed upon. In Figure 6 the envisaged method, including parameters, is shown.

As mentioned the applicability of the criteria of resolution A.167 to ships in lightly loaded or ballast condition was extensively discussed at the time these criteria were developed and proposals were made to introduce a correction factor to the dynamic stability parameter on this account [2]. In addition to the casualties analysed, also some later casualties e.g. the capsizing of the EDITH TERKOL, led to the conclusion of inadequacy of the application of the criteria derived for the fully laden conditions also to ships in light or ballast condition.

To overcome the problem an additional criterion to resolution A.167 of the type:

$$e_{\theta} = k \frac{\Delta_{\theta}}{\Delta} e_{\theta_0}$$

with:

$$\Delta_o = \text{displacement in the loaded condition}$$
$$\Delta = \text{displacement in the light condition}$$

$k$  = correcting factor  $\neq 1.0$

$e_{\theta_c}$  = righting arm curve area (resolution A.167)

$e_a$  = corrected righting arm curve area

was proposed or the adoption of a weather criterion to cover the stability of ships at all possible draughts.

Eventually the latter proposal was preferred, and it was agreed that the weather criterion should be applied to all normal conditions of loading.

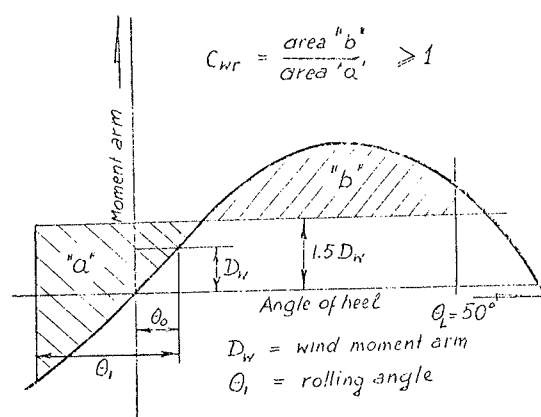


Figure 6.

#### 4.5 Stability of Ships in Following Waves

The main item in the short-term work programme was development of a weather criterion. The work on this subject commenced in 1977 but in fact, at that time, the conference in Torremolinos had already included a regulation on severe wind and rolling with guidance for its evaluation in the Fishing Vessels Convention [23]. The Guidance on a Method of Calculation of the Effect of Severe Wind and Rolling in Associated Sea Conditions, attached to the Torremolinos Convention, 1977, was therefore taken as a basis for further consideration and in 1979 was accepted almost without changes as an addition to resolution A.167.

The guidance of the 1977 Torremolinos Convention does not, however, include specific coefficients for evaluation of the wind and rolling effects, which were left to the judgement of the Administrations. In order to achieve a uniform method of calculation of the effect of wind and rolling a comparison of results

The effect of stability variations in following seas on the safety of a ship and the consequences this may have on the IMO stability criteria was recognized from the very beginning of work on intact stability. However, although numerous papers on theoretical approaches and on model investigations concerning this subject were submitted, no definite conclusions could be reached.

In 1978 stability of ships in following waves was introduced as a separate item as it was felt that evaluation of the loss of stability of a ship on the wave crest

would supplement the other stability criteria. Two basic approaches were considered:

1. an analytical approach where a theory of the non-linear system "ship in following seas" was to be developed and the quantities of the resulting phenomena were to be expressed statistically; and
2. a quasi-static approach where the stability loss of a ship was to be considered by assuming a sine wave with its crest amidships.

Comparative calculations for several sample ships were made but so far no results are available. It has to be noted that in a recent submission it was pointed out that from experiments with a model of a container ship the  $GZ_{max}$  value necessary to proceed safely in following seas was about four times that necessary for compliance with resolution A.167. It is however understood that the full size ship was above 100 metres in length. It also had a high Froude number, i.e. a slim water line and was fitted with bulbous bow. This case points out that, in addition to the verification of the criteria of resolution A.167, for larger ships unusual and novel design criteria need also be given consideration.

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- \* ECOR - Engineering Committee on Oceanic Resources;  
 IAPSO - International Association of the Physical Sciences of the Ocean;  
 IMO - International Maritime Organization;  
 ISSC - International Ship Structure Congress;  
 IUTAM - International Union of Theoretical and Applied Mechanics;  
 WMO - World Meteorological Organization;  
 UNESCO - United Nations Educational Scientific and Cultural Organization;  
 IOC - Intergovernmental Oceanographic Commission.

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#### Weather Criterion

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\*/ Available from the IMO Publications Section in English, French and Spanish.

# 20 YEARS OF EXPERIENCE - STABILITY REGULATIONS OF THE WEST-GERMAN NAVY -

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## ABSTRACT

The newly founded Navy of West Germany started stability supervision in 1956 with Bureau of Ships Regulations of the US Navy. As these proved not to be satisfactory for small vessels in shallow waters with steep waves, an attempt was made with the stability lever balance method, propagated by Prof. Wendel, at that time in the chair for design of ships and ship's theory at the Technical University Hanover. For the first time in naval architecture the effect of regular waves was considered in stability regulations, and there were some doubts whether a dynamic phenomenon could be met by taking into account the results of quasi-static calculations. But in comparison with the preceding efforts - and even with most of the following up to today - the number of unknowns was reduced to a remarkable extent, and therefore the safety margin of the residual righting levers could be determined in a practicable order of magnitude. This paper gives a report about the background and the experiences with this unusual stability regulations, which have remained nearly unchanged throughout the past 20 years, in spite of the results of further investigations and model tests. Clear and direct presentation of the essential effects, unmistakable demands concerning ship design and easy control of results have proved to be the outstanding advantages of the method employed.

## NOMENCLATURE

$h_G$  = Righting lever in smooth water in m  
 $h_B$  = Righting lever in wave crest condition in m

$h_T$  = Righting lever in wave trough condition in m  
 $h_S$  = Righting lever mean value in waves in m  
 $h_{Res}$  = Residual righting lever of a stability balance in m  
 $H$  = Wave height from crest to trough in m  
 $L$  = Wave length from crest to crest in m ( LAMBDA)  
 $L_{KWL}$  = Waterline length at design draught in m  
 $K_F$  = Heeling lever due to free surfaces in m  
 $K_D$  = Heeling lever due to centrifugal force in a turning circle in mm  
 $K_W$  = Heeling lever due to beam wind pressure in m  
 $K_{W70}$  = Heeling lever due to pressure of a 70 Kn beam wind, in m  
 $K_Q$  = Heeling lever due to transverse pull by repl. gear in m  
 $K_{Tr}$  = Heeling lever due to transverse pull of towlines in m  
 $K_P$  = Heeling lever due to people crowded on one ship's side in m  
 $P_W$  = Wind pressure in kN/m<sup>2</sup>

## 1. HISTORICAL BACKGROUND

When starting with the reconstruction of the new Navy of the Federal Republic of Germany about 25 years ago, very soon stability problems became evident. During the 10 years gap since world war II, there were essential changes in shipbuilding technology, machinery, armament, communication equipment and even in sea warfare, some of them having started already during the war. Secondly, since the Federal Navy had no central design department, with the first series of newbuildings there was a need for stability regulations in order to overcome the individual stability calculations of the different shipyards.

A first attempt was made with the Bureau of Ships Regulations for the US-Navy (1). But a closer look led to the conclusion that these were more suitable for large warships from cruiser size upward, than for frigates, minesweepers and patrol boats of which the West-German Navy would mainly consist. Further it seemed to be disadvantageous that all criteria were based upon smooth water righting levers only, whereas the North Sea as well as the Baltic Sea - the main operational areas of the West German Navy - are known for heavy weather and steep waves.

At the same time, Prof. Wendel, being in the chair for ship design and ship's theory at the Technical University of Hanover, had investigated some stability accidents of cargo vessels (2) by means of the stability balance and taking into account the influence of waves on the righting levers of a vessel. For this reason he was entrusted with the elaboration of stability regulations for the surface vessels of the West German Navy. These came into force as Specification 103, having a preliminary character first, only a few months later, in summer 1961.

This short period may be astonishing in comparison with the attempts of finding stability criteria for merchant ships, but one should keep in mind that in this case the following special circumstances had to be taken into consideration:

- The post-war interruption favoured a new beginning.
- There was a demand for progress, as most of the German escort vessels in World War II had proved poor seaworthiness in bad weather.
- New findings on stability were available.
- There were no economic interests of the vessels which could be affected.
- Owners and controlling authorities were not just the same people but the same institution, therefore alterations or modifications could be performed easily, if necessary.
- A quick flow-back of experience and observations was assured.

So it was possible to issue definite stability regulations in 1969 as Specification No. 1033, which have been in force

since, with only slight modifications. There were no great differences between the preliminary and the definite version, just a more precise wording in some paragraphs, restriction of some calculations, which obviously were unequal to the accuracy of the results, and incorporation of some new findings about external forces. But the characteristic features remained unchanged, as there are:

- Comparison of righting and heeling levers by means of balances, which cause the naval architect to consider the strains to be encountered with the vessel in question in the design stage already. They also supply the command of the vessel with clear information concerning the different effects on safety against capsizing and sinking. Further, this method seems to be appropriate to take into account the fact that righting and heeling forces may differ widely among the several types of vessels and their operational areas, with special emphasize to future designs and military demands.
- Computation of the righting lever arms in some phases of given waves and taking into account dynamic effects implicitly by including wave crest as well as wave trough lever arms into the stability criteria. This seems to be a more realistic way than the comparison of areas among the righting lever arm curve valid for smooth water and the heeling lever arm curve of transverse wind, in spite of the lack of exact knowledge of the ship-wave interaction.
- Watertight subdivision to such extent that the probability of withstanding damages is as high as practicable without affecting other important features of the vessel in a way that would restrict its operability.

## 2. THE MAIN DETAILS

The goal envisaged when elaborating these stability regulations was to find a method for easy and reliable evaluation of the stability of ships and to have a better base for comparison of stability calculations and evaluations. This was achieved by standardizing figures, algorithms and methods to a wide extent.

### 2.1 Operational Groups of vessels

According to operational requirements, ships are classified in the following groups:

- Group A: Ships for locally unlimited operating areas.
- Group B: Ships for operating areas in North Atlantic, North Sea, and Baltic Sea being capable of evading wind velocities exceeding 70 Kn.
- Group C: Ships for operation in coastal areas being capable of duly calling at a harbour in case of storm warning.
- Group D: Ships for operation as harbour craft and estuarial craft.
- Group E: Motor boats, personnel boats, etc. for operation in harbour and coastal areas.

## 2.2 Loading Conditions

The standardized loading conditions are summarized in Table I. The conditions are numbered from 0 to 6, conditions 1 and 2 with variations, due to different operational groups or ice accumulation. Loading conditions without number are theoretical ones and are serving statistical purposes only. The figures in the table indicate the percentage of the different weights to be included.

## 2.3 Righting levers

The righting levers are to be derived from cross curves of stability in smooth water ( $h_G$ ) and in a wave ( $h_s$ ) of given height/length ratio, where the wave length is to be equal to the length of the design waterline (LKWL):

$$H \text{ (Wave)} = \text{LKWL} : (10 + 0.05 \text{ LKWL}) \quad (1)$$

( $H$  and  $L$  in m). The ship is assumed to be lying stationary in the crest ( $h_B$ ) or trough ( $h_T$ ) of a trochoidal wave without trimming, or - with trimming - in 10 different phase stages.

For both, the smooth water and the wave condition, the same amount of watertight body, erections and appendages have to be chosen, and the calculation is to be performed for the same angles of heel.

## 2.4 Heeling Levers

For the following heeling forces formulae, loads, factors etc. are given (further details published in (3)):

- Free surfaces in tanks and cells ( $K_F$ ). For tanks resulting in more than 3 cm heeling lever arm at 30 deg. heel, cross curves are to be computed instead of using the free surface formula.
- Centrifugal force in a turning circle ( $K_D$ ). From a collection of test data (4) the most unfavourable combination of steady state velocity and turning circle diameter has been selected.
- Beam wind pressure ( $K_W$ ). The lever of the heeling moment is to be measured from the centre of pressure of the wind lateral area to 0.5 draught amidships. The wind pressure per area unit, which is calculated with a resistance coefficient of  $c_w = 1.2$  is subject to the operational group of the ship:

$P_W$  = wind pressure in  $\text{kN/m}^2$  for ships of

|          |                      |                       |
|----------|----------------------|-----------------------|
| Group A: | 1.50 $\text{kN/m}^2$ | (wind velocity 90 kn) |
| Group B: | 1.00 $\text{kN/m}^2$ | (70 kn)               |
| Group C: | 0.50 $\text{kN/m}^2$ | (50 kn)               |
| Group D: | 0.30 $\text{kN/m}^2$ | (40 kn)               |
| Group E: | 0.10 $\text{kN/m}^2$ | (20 kn)               |

A wind pressure  $P_W = 0.30 \text{ kN/m}^2$  is to

be assumed for ships of all groups for the following conditions of loading:

- 0 = unloaded displacement
- 0V = warping displacement
- 4 = special full load displacement
- 5 = special limit displacement

- Transverse pull by replenishment - at-sea gear is only to be considered if it exceeds 5 cm in upright condition ( $k_Q$ ).
- Transverse pull of towlines for screw tugs ( $k_{Tr}$ ).
- People crowded at one side of the ship ( $k_P$ ).

Accumulation of ice on deck and superstructures is not looked upon as heeling force, but as a modification of certain loading conditions (see Table I), as it is assumed to occur symmetrically, and therefore gives no heeling moment but an increase in displacement and CoG above the base line.

As ice accumulations may be unlimited in certain environmental conditions, a layer of  $0.5 \text{ kN/m}^2$  on exposed decks and on front bulkheads and of  $1 \text{ kN/m}^2$  on the front areas of weapons, equipment and other obstacles up to 10 m above the waterline has to be assumed. The loss of stability caused by this amount of ice accumulation shall give the command an indication, whether countermeasures are to be taken or not.

## 2.5 Stability balances

For every vessel and loading condition, normally two stability lever arm balances have to be calculated, one in smooth water, taking into account the righting lever in smooth water, the heeling levers due to free surfaces  $K_F$ , moderate transverse wind  $K_W$  ( $0.3 \text{ kN/m}^2$ ) and centrifugal force in a turning circle  $K_D$ , and the second one for unfavourable sea and wind conditions. The combinations of righting and heeling levers to be assumed are shown in Table II.

## 2.6 Stability Criteria for intact ship

Safety against capsizing is evaluated due to the static angles of heel and the residual righting lever arm at given angles of reference (see Fig. 1) The limit values are:

- 15 deg., if at 35 deg. the resid. lever exceeds 10 cm,
- 20 deg., if at 45 deg. the resid. lever exceeds 15 cm,
- 25 deg., if at 55 deg. the resid. lever exceeds 20 cm,

As modern naval vessels with extended erections and small displacement may suffer large angles of list under beam winds, the angle of static heel may exceed even 25 deg., if there is a sufficient amount of residual righting lever arms. Originally, the maximum limit value of 25 deg. was chosen with



Table I

This table is a summary only; in case of doubt, the text of rule 3 prevails.

|  | Unloaded Ship, Ready for Service | Unloaded Displacement | Warping Displacement | Limit Displacement with Ballast water | Limit Displacement with Accumulation of Ice | Limit Displacement for 90, 70, 50 and/or 40 kn Wind | Full Load Displacement | Full Load Displacement with Accumulation of Ice | Medium Displacement | Special Full Load Displacement | Special Limit Displacement | Standard Displacement | Designed Displacement | Training Cruise Displacement |
|--|----------------------------------|-----------------------|----------------------|---------------------------------------|---|---|------------------------|---|---------------------|--------------------------------|----------------------------|-----------------------|-----------------------|------------------------------|
| Condition of loading   | -                                | 0                     | 0V                   | 1                                     | 1E  | 1A; B; C  | 2                      | 2E  | 3                   | 4                              | 5                          | -                     | -                     | 6                            |
| Ship including operating media in machinery, piping, armament, and equipment, possibly including permanent ballast | 100                              | 100                   | 100                  | 100                                   | 100   | 100   | 100                    | 100   | 100                 | 100                            | 100                        | 100                   | 100                   | 100                          |
| Crew and effects   | -                                | 100                   | 100                  | 100                                   | 100   | 100   | 100                    | 100   | 100                 | 100                            | 100                        | 100                   | 100                   | 100                          |
| Consumables and provisions   | -                                | -                     | -                    | 10                                    | 10  | 10  | 100                    | 100   | 50                  | 100                            | 10                         | 100                   | 100                   | 10                           |
| Freshwater   | -                                | -                     | 1)                   | 10-50                                 | 10-50                                       | 10-50   | 100                    | 100   | 50                  | 100                            | 10-50                      | 100                   | 100                   | 10-50                        |
| Boiler feedwater   | -                                | -                     | 1)                   | 10-50                                 | 10-50                                       | 10-50   | 100                    | 100   | 50                  | 100                            | 10                         | -                     | 100                   | 10-50                        |
| Operating media (fuel)<br>(lubricating oil)  | -                                | -                     | 1)                   | 10                                    | 10  | 10*)  | 100                    | 100   | 50                  | 100                            | 10                         | -                     | -                     | 10                           |
|  | -                                | -                     | 1)                   | 50                                    | 50  | 50  | 100                    | 100   | 70                  | 100                            | 50                         | -                     | -                     | 50                           |
| Ammunition   | -                                | -                     | -                    | 33                                    | 33  | 33  | 100                    | 100   | 50                  | 100                            | 33                         | 100                   | 100                   | 100                          |
| Airplanes  | -                                | -                     | -                    | 100                                   | 100   | 100   | 100                    | 100   | 100                 | 100                            | 100                        | 100                   | 100                   | 100                          |
| Special load (mines etc.)  | -                                | -                     | -                    | -                                     | -   | -   | -                      | -   | -                   | 100                            | 33                         | -                     | -                     | -                            |
| Landing troops and equipment   | -                                | -                     | -                    | -                                     | -   | -   | -                      | -   | -                   | 3)                             | 50                         | -                     | -                     | -                            |
| Cargo  | -                                | -                     | -                    | 33-100                                | 33-100                                      | 33-100  | 100                    | 100   | 50                  | 3)                             | -                          | 100                   | 100                   | -                            |
| Ballast water  | -                                | -                     | 1)                   | 1)                                    | 2)  | -   | -                      | -   | 1)                  | -                              | 1)                         | -                     | -                     | 1)                           |

1) To the extent required to provide satisfactory stability and trim

2) As in corresponding condition of load without accumulation of ice

3) Only if required

\*) at least

Table II

Lever arm balances for unfavorable sea and wind conditions

| Groups  | Lever arms to be compared |                            | Conditions of Loading |    |    |   |   |
|---------|---------------------------|----------------------------|-----------------------|----|----|---|---|
|         | Righting Arms             | Heeling Arms               |                       |    |    |   |   |
| Group A | $h_s$                     | $k_F + k_{W90}$            | 1                     | 1A | 2  | 3 | 6 |
|         | or                        | $k_F + k_{W50}$            | 1B                    | 1E | 2E |   |   |
|         | $h_G$                     | $k_F + k_{W40}$            | 1C                    | 4  | 5  |   |   |
|         | $k_G$                     | $k_F + k_{W40}$            | 0                     | 0V |    |   |   |
| Group B | $h_s$                     | $k_F + k_{W70}$            | 1                     | 1A | 2  | 3 | 6 |
|         | or                        | $k_F + k_{W50}$            | 1B                    | 1E | 2E |   |   |
|         | $h_G$                     | $k_F + k_{W40}$            | 1C                    | 4  | 5  |   |   |
|         | $h_G$                     | $k_F + k_{W40}$            | 0                     | 0V |    |   |   |
| Group C | $h_s$                     | $k_F + k_{W50}$            | 1                     | 2  | 3  |   |   |
|         | or $h_G$                  | $k_F + k_{W40}$            | 1C                    | 4  | 5  |   |   |
|         | $h_G$                     | $k_F + k_{W40}$            | 0                     | 0V |    |   |   |
| Group D | $h_G$                     | $k_F + k_{W40}$            | 0                     | 0V | 1  | 2 |   |
|         |                           | $k_F + k_{Tr} + k_{W40}$   | 1                     | 2  |    |   |   |
| Group E | $h_G$                     | $k_F + k_{W20} + k_{Pers}$ | 3                     |    |    |   |   |

respect to safe operation of machinery and penetration of water through openings of the hull. From the point of view of stability a 25 deg. list may not be a limiting value.

A second criterion is the righting lever arm on a wave crest (or wave trough, whichever may be smaller), which should be positive over a 10 degree-range between 0 and 45 degrees of heeling with a maximum value of at least 5 cm.

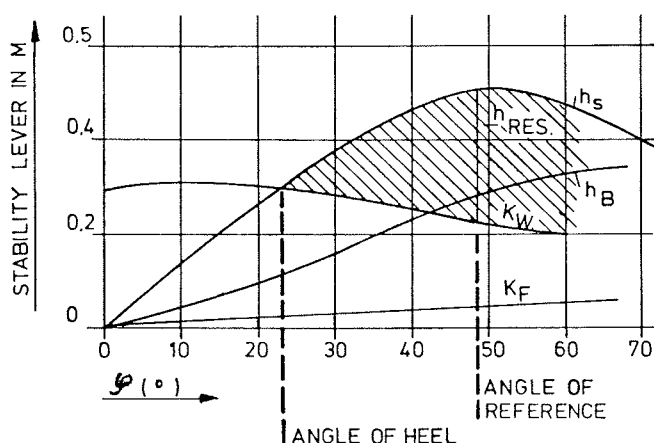


Fig. 1 Stability lever balance for unfavourable sea and wind conditions

## 2.7 Stability and Buoyancy of Damaged Ship

The stability regulations contain some subdivision guidelines which have proved advantageous for safety against sinking. The minimum length of a compartment is 1.8 m. Shorter spaces have to be added to the adjacent compartments.

For ships less than 30 m in length, only one compartment has to be assumed flooded. For other ships the length of a leak is 0.18 of the design waterline length - 3.6 m, but not exceeding 18 m. The leak may be at any position of the ship, thus resulting at least in two compartment flooding, if only one transverse bulkhead is penetrated, or a threecompartment flooding, if two adjacent compartment are shorter than the leak length.

In transverse direction the leak stretches up to the centre line, but centre line bulkheads being undamaged. Side bulkheads may be assumed undamaged, if the occurring list might be larger than with this bulkhead damaged.

In vertical direction the leak is to be assumed from keel to bulkhead deck.

For stability evaluation purposes the righting lever arm of the residual undamaged part of the ship is to be compared with the heeling levers due to free surfaces and a moderate beam wind of 0.3 kN/m<sup>2</sup>.

The safety is considered sufficient, if

- without beam wind pressure and symmetrically flooded the ship is floating upright,

with nonsymmetrical flooding the bulkhead deck may not

be submerged due to heel and trim in final condition, in intermediate conditions 25 deg. of list are not exceeded and the residual righting lever arm is larger than 5 cm,

- with beam wind pressure openings of intact compartments are not submerged, 25 deg. of list are not exceeded - even with nonsymmetrical flooding - and the residual righting lever arm is larger than 5 cm.

If these conditions cannot be met totally, a probability factor of safety against sinking has to be calculated in order to show the difference to normal safety standard. This difference then will serve as a help for decision making concerning the required exemption.

## 2.8 Further standardized values

In order to achieve compatibility as far as possible, the stability regulation contains fixed values for the density of sea water, Diesel oil, fuel oil, lubricating oil etc., as well as permeability factors for damage stability calculations. Furthermore, the layout of stability sheets for intact and damage conditions is shown on example sheets, and also the minimum contents of stability booklets are listed, which are the source of information for the command.

## 3. THE PHILOSOPHY BEHIND

The balance of all forces which may act simultaneously on a vessel shows the relevant physical relationship, such giving an information to the design engineer as well as the command, which improvements can be made by the means available, e.g. ballasting the vessel, change course, reduce speed, avoid ice accumulation etc.

The idea of this method was first published by Reed in 1868 (5), taking into account the heeling due to wind pressure on sails only. But still in the early fifties of this century, the main attention was fixed upon one side of the balance only - at least among German naval architects, which discussed with emphasize the problem how to increase the accuracy in calculating the cross curves of stability for small angles of heel in still water. On the other side of the balance the tolerances were at least an order of magnitude greater, but in spite of that this method was helpful already in elucidating some stability accidents. In one of these cases the influence of regular waves on the righting lever arms was taken into consideration (2), as course and speed of the vessel and the data of the seaway made allowance for a quasistatic treatment of the problem.

This first application of hand-calculated righting levers in a seaway was followed by further investigations at the Chair of Prof. Wendel in Hanover, mainly concer-

ning the order of magnitude of the righting arm alterations, the influence of ship's length versus wave dimensions ratio, the effect of changing directions of encounter, and the amount of calculations necessary to get results with sufficient accuracy. First model tests were performed in order to get confirmation of the calculations and to gain some insight in the influence of the ship-borne wave system and the motion dynamics of the ship in waves (6).

At this state of affairs, the Ministry of Defence asked for the elaboration of stability regulations, and it was decided that the influence of waves on the righting lever arms could not be neglected as it was of the same order of magnitude as wind pressure or centrifugal force in turning circles, in spite of the uncertainties mentioned before. But these uncertainties too were of roughly the same order of magnitude as those of wind pressure, centrifugal force in turning circles, and even the heeling lever due to free floating liquids in tanks could be of similar inaccuracy if all bottom tanks of a vessel had to be partially filled in order to encounter the effect of underwater detonations in mine endangered areas.

The next question then was, how to make allowance for the effect of waves on the stability lever balance. Three effects had been derived from the calculations and model experiments (7):

- With high frequencies of encounter in a longitudinal or near to longitudinal seaway the vessel has an integrating effect due to its mass inertia. Therefore the mean value of the righting levers through all phases of encounter can be looked upon as effective. (In order to reduce the amount of computations, only the mean value of crest and trough levers were introduced, as intermediate phases are of minor importance.)
- With frequencies of encounter in the magnitude of the ships own roll frequency or the 0.5-, 1.5-, 2.0- and 2.5-fold of this frequency parametric excitation may occur (8) (with higher manifolds the effect is diminishing). The excited roll motion may become unstable, i.e. the vessel capsizes, if the residual righting levers are below a certain limit.
- With frequencies of encounter approaching to zero (following seas), the condition in which the smallest righting lever arms are effective (normally the crest-amidships situation, but under extreme conditions the trough-amidships phase may be the most unfavourable) may last as long that there will be sufficient time for the vessel to capsize during just one period of encounter.

Consequently the righting lever arms in wave conditions were introduced in two ways: The mean value of crest and trough situation  $h_s$  and the minimum righting lever are to be

taken into account. These values represent in a direct way the resistance against capsizing in high and very low frequencies of encounter, whereas at the same time the difference between crest and trough righting levers and therefore the exciting force in resonance frequencies of encounter is restricted.

Experiments with free running models in natural seaways (9) or in regular waves in model tanks (10) showed, that the range of uncertainty between safety against capsizing and danger of capsizing is of moderate magnitude only. So the residual righting levers mentioned in para. 2.6 were derived from the experiments as being sufficient.

For beam seas, no special stability criterion was established, as no capsizing in beam seas was observed even with models, which capsized in following seas. Therefore, if the stability criteria of the Specification 1033 are met, the probability of safety against capsizing seems to be very close to 1 (11).

It may be surprising, that in the stability balance for unfavourable sea and wind conditions, righting lever arms in longitudinal waves are combined with beam wind. This combination is accounting for the fact that even strong winds may change their direction within a short time only, whereas the waves are proceeding in the direction in which they were excited. Waves and wind from different directions can be observed especially near storm centres, where winds are circulating due to the eddy character of low-pressure disturbances.

Contrary to most other stability criteria, wind pressure is considered as a static force. This pays attention to the fact that gusts with quick changes of wind speed occur only over small areas, and therefore have a slight dynamic influence only upon the heeling moment of ships. In spite of lack of simultaneous measurements of wind velocities at several points of an area of say 500 m<sup>2</sup>, at least the assumption that gusts may cause heeling angles twice as large as a steady state wind of the same velocity has not been confirmed by observations during full scale trial trips.

#### 4. EXPERIENCES

During the past 20 years the Specification 1033 has fulfilled the expected demands. Its effects:

- Consideration of the essential forces in combinations as realistic as possible, thus the probability of surprising effects is reduced to a very low level.
- Definite stability criteria and therefore easy communication between parties involved. Discussions about square millimetres of areas will not occur.
- Informative presentation of the results. Every effect is presented due to its importance. Improvements of stability can easily be derived.
- Compatibility of results and experiences with different ships. Special features with

- respect to stability are shown clearly.
- Easy performance and control of stability calculations. Nobody has to perform investigations of his own in order to find the right coefficient or formula - except for outstanding effects. As control instrument a small ruler is sufficient.
  - Flexibility with respect to new types of ships. Particular features of a design can be taken into account.

At the beginning some comparisons were made with the stability regulation of the US-Navy, which in the opinion of some specialists makes better allowance for dynamic effects. Fig. 2 shows the results of some stability comparisons of German destroyers and a mine-sweeper of World War II. Here the evaluation is similar in all cases, in spite of different methods, but the performance of evaluation is easier with the Specification 1033, as no areas are to be integrated. This is an effect which is highly appreciated by the administrative bodies.

As critical conditions have been proved the Limit Displacement (see Table II) for vessels with large wind lateral areas and the Full Load Displacement for vessels with low freeboard. Fig. 3 shows the development of the wind lateral areas of German destroyers and frigates between 1936 and 1975. It is obvious that heeling due to beam wind plays an important role in stability control. Some of the Navy vessels had to be ballasted with fixed ballast in the course of modernization measures, due to additional equipment on deck and a small range of righting lever arms.

The behaviour of the ships in service was in good conformity with the results of the stability balances, even in some damage conditions which were caused by collisions. This experience lead to considerations as to how to decrease heeling levers or to increase righting levers by means of new developments in ship design. One of these efforts is shown in Fig. 3. The frigate down below has an extended freeboard due to a superstructure deck stretching from bow nearly to the stern, and masts and weapons concentrated on lower levels. The wind lateral area has increased in comparison with the lateral plan, but the wind plane centre is lowered and the righting levers at large angles of heel-especially in waves-are increased remarkably.

Another interesting development is shown in Fig. 4. This design of lines (12) has nearly identical righting levers in smooth water as well as in the wave crest and in the wave trough condition, thus minimizing the excitation of rolling in a longitudinal seaway and at the same time danger of capsizing in following seas will disappear.

The fact that the Royal Dutch Navy has taken over the Specification 1033 almost literally already some years ago, and nearly all naval vessels which have been built in West German shipyards for foreign navies are

supplied with stability documents coinciding with this specification, may be looked upon as a confirmation of the ideas involved in the stability regulations of the West German Navy. A secondary effect of this extension to other navies is, that further experiences are gained in areas far away from the origin of the Specification 1033.

## 5. FURTHER DEVELOPMENT

In order to confirm these stability regulations or to improve them in accordance with the progress in naval architecture in general and in the range of stability research in particular, the activities in this field are observed thoroughly and investigations are promoted by the West German Naval Authorities.

In 1977 model tests were performed with the model of a frigate in the Hamburg towing tank (HSVA), with special emphasize on the wave-induced alterations of the righting lever arms (10). The main findings were:

- The model did not capsize, as long as it met the criteria of Specification 1033, based upon the mean value of crest and trough righting levers
- The maximum heeling angles were in good conformity with those gained with quasi-static calculations due to Specification 1033
- The model did not capsize in some conditions, where the heeling levers exceeded the righting levers in the wave crest amidships - position (Fig. 5.1). On the other hand capsizing occurred with righting levers in the crest amidships-position larger than the heeling levers (Fig. 5.7).

Main emphasize therefore is to be laid on the mean value criterion by which the residual amount of righting energy is determined due to the size and position of the minimum residual righting lever. This special criterion will be altered in such way that the limitation of statical heel of 25 deg. will be given up, in order to gain more flexibility. As already pointed out, the limit angle of 25 deg. was determined from the viewpoint of operation reasons rather than from stability. The criterion will then read as follows:

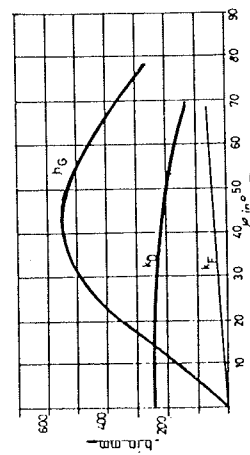
$$\begin{aligned} \text{Angle of reference} &= \text{angle of static heel} + 5 \text{ deg.} & (2) \\ \text{residual righting lever } h &= (\text{angle of stat. heel} - 5 \text{ deg.}) \cdot 0.01 \text{ in cm} & (3) \end{aligned}$$

These formulae do not mean a loss of safety against capsizing, compared with the Specification 1033 at the time being, but give the design engineer more freedom in development of new types of naval vessels.

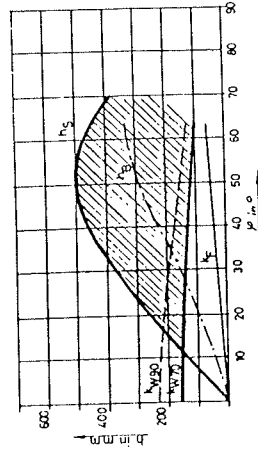
With respect to the dynamic behaviour in a seaway, studies are under way at the Institute of Shipbuilding at the Hamburg University, with emphasize on the analysis of the coefficients which are determining motion dynamics of ships.

Summing up the experiences with the stability regulations of the West German

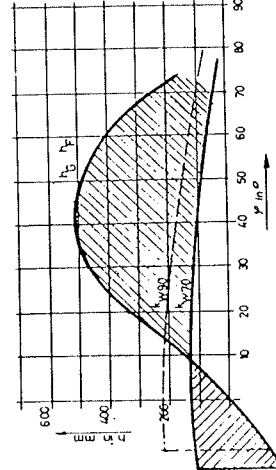
S 1033



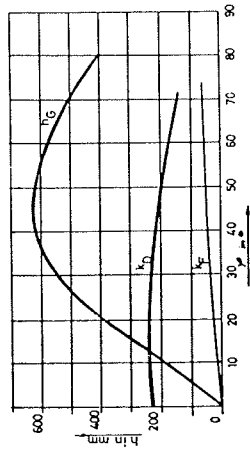
S 1033



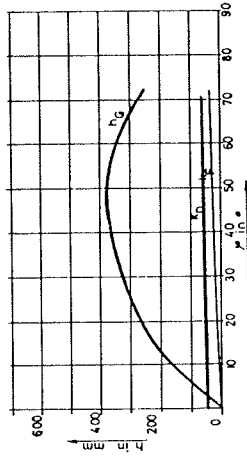
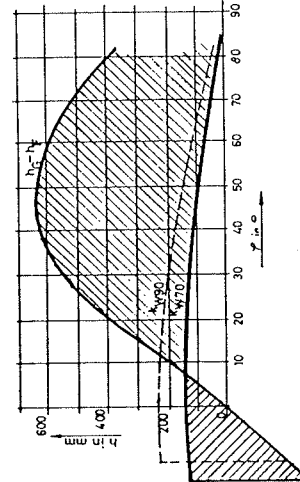
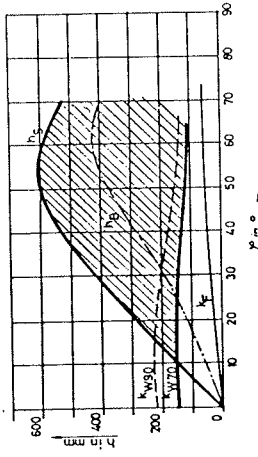
Bureau  
of Ships



Destroyer 36A



Destroyer 36B



Mine-Sweeper 40

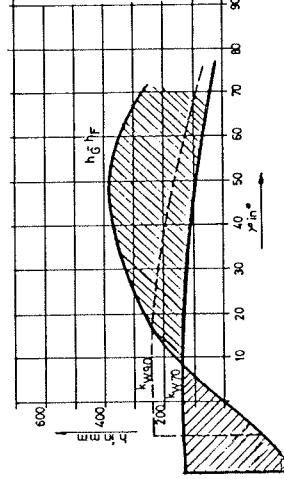
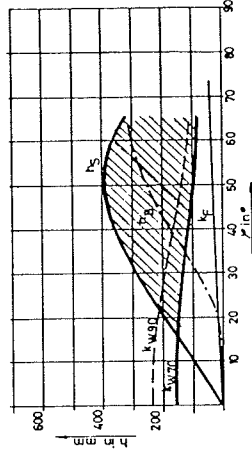


Fig. 2 Comparison of stability lever balances with area method due to Bureau of Ships

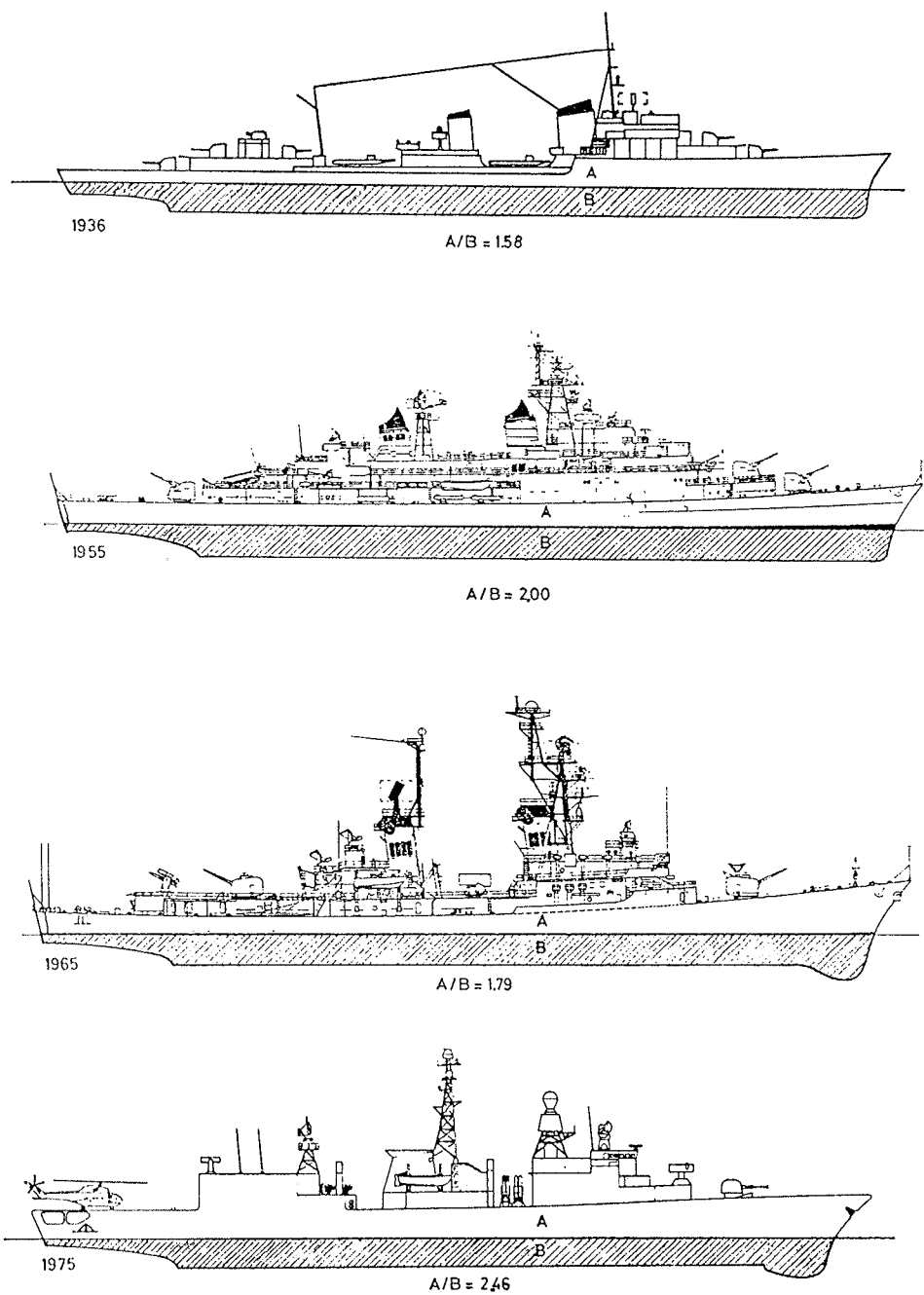


Fig. 3 Development of wind lateral areas of German Naval vessels

Navy during the past 20 years, and taking into account the different types of special purpose ships, both of naval and merchant vessels, there seems to be no way back to a simple overall stability criterion, which would cover with darkness the relevant physical relationships and neglect the peculiarities of the individual ship.

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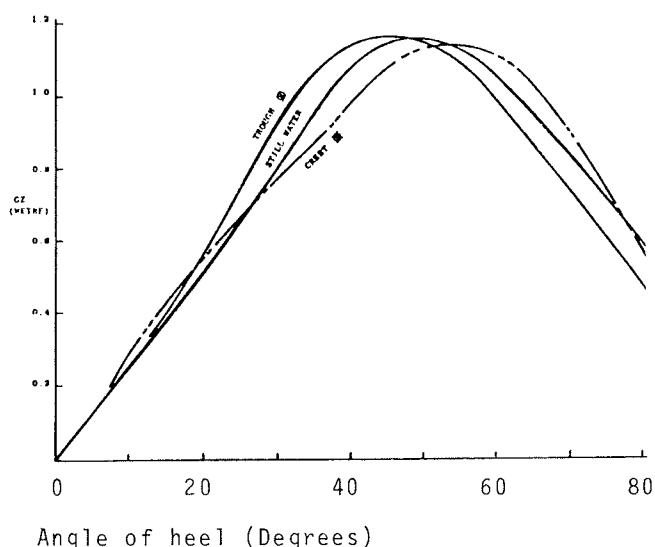
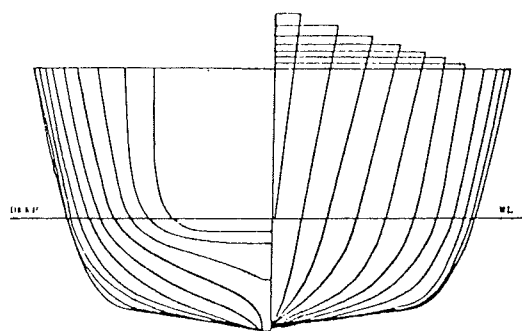


Fig. 4 Lines with only slight changes of righting levers in waves (12)

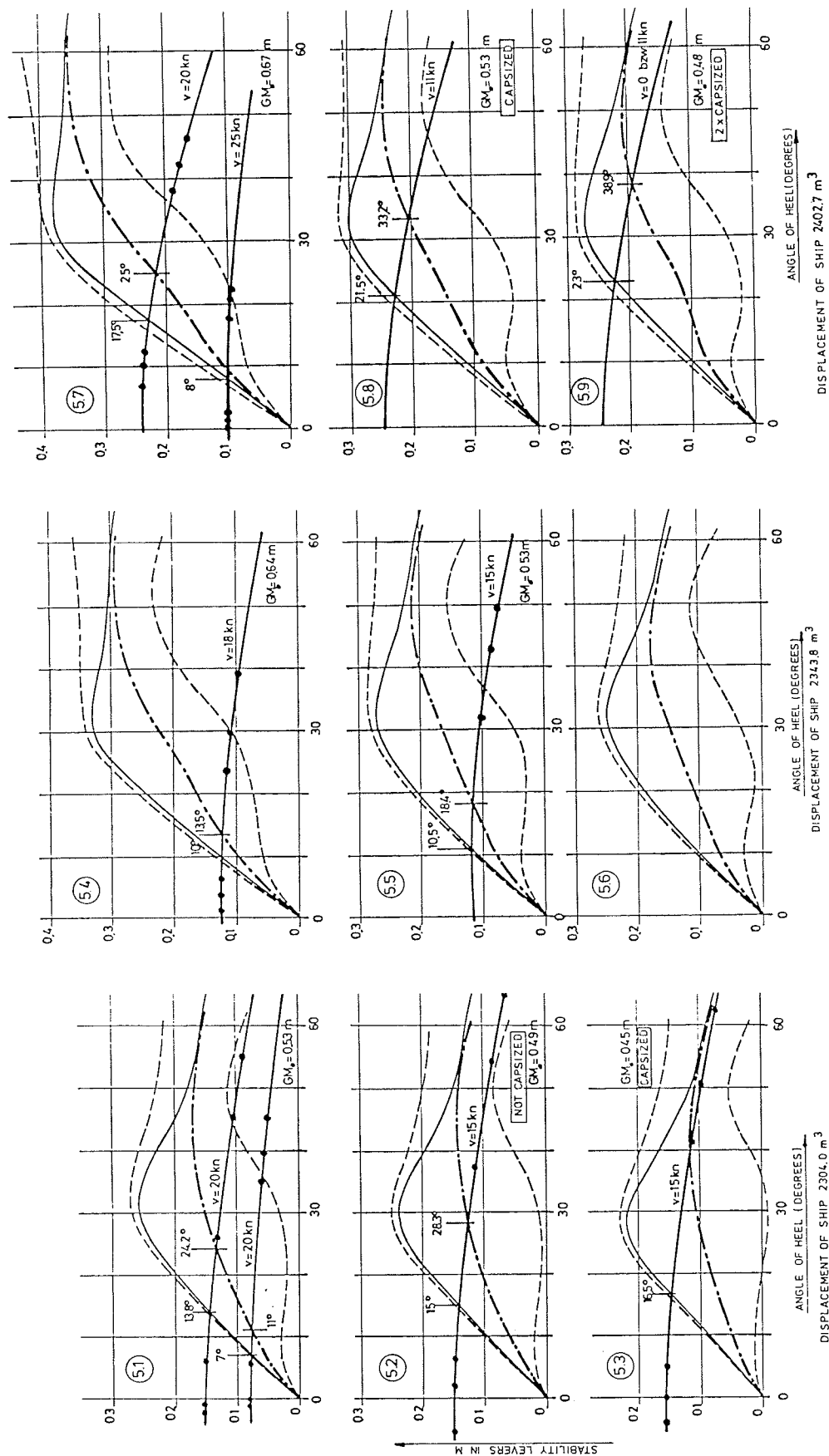


Fig. 5 Some results of towing tank tests with a model of a frigate of the West-German Navy.



## *Closing Session*

*Chairman*

Prof. Seizo Matora  
Nagasaki Institute of Applied Science  
Japan

## SUMMARY OF STABILITY '82

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### INTRODUCTION

It is difficult - if not impossible - to summarise all that has happened during the last five days and at the same time do justice to everyone who has contributed to the success of this conference. We have, after all, listened to forty-three papers, the views of twelve panellists, and over 120 contributions to discussion, without mentioning the countless informal debates and exchanges of idea which have taken place over coffee, lunch and tea, and during the technical visit.

However, we cannot allow this conference to end without making some attempt to assess the progress made and the lessons learnt, to draw some conclusions, and to consider what future activities we should try to encourage. I should like, therefore, discuss some key points under the four headings:

1. From the First to the Second Conference
2. Highlights of STABILITY 82
3. Future Activity
4. The Message of the Conference.

Finally I should like to propose a vote of thanks to those who made this conference possible.

### FROM THE FIRST TO THE SECOND CONFERENCE

I should like to start by considering the question of aims. As Professor Motora stated on Monday at the Opening Session, the aims of this conference were precisely the same as those of the first conference at the University of Strathclyde Glasgow, in March 1975. That is, we wanted to provide a forum at the international level for the discussion of research findings on stability, and to facilitate the use of these findings as guidelines for future research efforts and the practical application of the results. It would therefore be both helpful and informative to use these aims as a means for assessing the progress made during the past seven years.

#### a) Increase in Scope of Interest

At the 1975 conference we had around a hundred and thirty delegates. This time our numbers are in the region of two hundred and fifty.

At the first conference twenty-four papers were presented, on topics in three broad areas: General Reviews, Theoretical Studies, and Experimental Investigations; and a Quiz Session gave participants the opportunity to ask a panel of experts questions related to ship

stability. This conference has heard almost double the 1975 number of papers, and the International Technical Committee was unable to accommodate all the offers of papers actually received.

To allow us to examine in greater detail key aspects of stability, the programme for this conference has also included two Panel Sessions, one on "Philosophy and Research", and the other on "Criteria and Regulations".

In view of the increased activity in the North Sea and other offshore locations throughout the world the scope of the conference now embraces the stability of ocean structures; and fresh topic-areas covered include damage stability, stabilisation devices, stability of semi-submersibles, shipping of water, and broaching.

It is clear, therefore, that interest and technical activity in this area have both virtually doubled since 1975.

b) Cases of Capsizing

If the importance of our subject was not fully appreciated in 1975 we shall certainly not find many in 1982 who are complacent about the issues of stability and capsizing.

It is sad, but true, that our progress in recent years has been aided by several tragic accidents. The capsizing and loss of the fishing vessels "Gaul" and "Helland Hansen" and of the semi-submersibles "Alexander Kielland" and "Ocean Ranger" are known of by almost everyone connected with naval architecture and marine technology. These events have focused attention on the need to test the stability of the vehicle at every stage of the design process, and investigative studies have yielded much valuable

information on this subject. Examples are Morrall's on the loss of the "Gaul" and Dahle's on the "Helland Hansen" which were published after their studies had been completed, while the case of the "Alexander Kielland" was considered at the very first technical session of this conference.

These studies are important, and the papers make a valuable contribution to the subject, but we must not forget that we have still to solve the problem of the continuing regular loss of small fishing vessels through capsizing. It is our hope that systematic and determined efforts will continue to be made to solve this problem, even though the major losses are now fading from the public's memory.

c) Session Chairmen

By dividing the papers up into several sessions and having the chairing duties of each session shared by two people I believe that the organisers of this conference have developed a most effective new form of international co-operation. This arrangement not only meant a sharing of the workload but also made possible the renewal of old friendships and the development of new ones, and I suspect that in certain cases it gave an opportunity for continuing long-standing debates!

d) Panel Discussions

Shortage of time for discussion after presentations is a problem at most international conferences and STABILITY 75 and STABILITY 82 were no exception. This time, however, two three-hour panel discussions allowed us to examine questions of "Philosophy and Research", and "Criteria and Regulations", at a depth which would not have been possible otherwise.

The beneficial features of panel discussions are that critical debate can take place on specific problems, expert opinion can be challenged, and every delegate can be involved in the discussion. This means that a much better understanding of the problem can be acquired by all concerned.

In view of the foregoing, we can state categorically that considerable progress has taken place been STABILITY 75 and STABILITY 82.

#### HIGHLIGHTS OF STABILITY 82

To my mind the conference has provided us with eight highlights:

##### a) Help for Users

One of the most gratifying features of this conference has been the genuine attempt by a number of authors to produce tools and techniques which can be readily adopted by designers and operators. These tools and techniques do not in themselves provide a complete answer to given stability problems but they do offer interim solutions. Such efforts should be encouraged, even if they are not fully satisfying to those who are looking for complete answers. We must realise that the solving of stability problems and the establishing of fully effective stability criteria are step-by-step processes.

##### b) Development of Criteria

While official bodies are considering ways of improving the criteria presently operated by the International Maritime Organisation, or IMO - known until recently as IMCO - to take into consideration the effects of wind, our priority should be to look at how the whole question of criterial development should be approached.

One suggestion during the second

Panel Session was that a Venn diagram incorporating "Design", "Operation" and "Environment" should be taken as a starting point. This, in my view, identifies the main features which must be taken into consideration in the development of criteria, but this development can only be done in stages, and has to be related to the amount of information and knowledge available at any given point in time. This approach is called the "levels of stability" approach. It would involve having a range of levels of stability based on criteria of increasing degrees of sophistication for use at different stages of the design and operation procedure. The criteria at each level would be determined by the quantity and quality of information available at that point. For example, if the only data available at a given stage are the volume of the vessel, its draught, and its dimensions, the stability criteria adopted at that stage would have to be somewhat approximate.

There was some discussion on whether there should be special operational criteria alongside those that exist for design purposes but it seems to me that this might complicate the situation without making any real contribution to final solutions.

We must always bear in mind that any criteria that are developed must be presented in forms that can be readily applied.

##### c) The Exchange of Ideas

A main aim of any conference is the encouragement of the exchange of ideas and this conference has provided numerous opportunities for this, and also shown how useful it would be if more effective exchanges could take place. Various theoretical papers presented here will

benefit from published experimental results. In one case the authors stated that they now plan to set aside work described in their paper and to embark on an alternative approach on the basis of suggestions made at a technical gathering. We also noted during the session on "Stabilisation Devices" that the conclusions drawn in one paper would certainly have been different had knowledge been available earlier of work described in another paper. It is certainly true that wide scope exists for the a fuller exchange of ideas on semi-submersibles, theoretical investigations, and the use of model experiment results to back up theory.

d) Theory and Practice

The conference has considered and discussed two extremes of interest and activity. On the one hand, some highly theoretical studies were presented in the "Fundamental" Section, and the importance of these was highlighted by the point made in the first Panel Discussion during which it was stated that we do not yet have sufficient knowledge to be able to develop reliable methods for calculating the forces experienced by vessels in seaways. On the other hand, several presenters have commented on the "ignorance of operators" about the stability of ships. The validity of this comment was borne out by a visual presentation in which a vessel for catching king crabs had deck loads to a height double the depth of the vessel!

All research work must take cognisance of these two extremes, otherwise the present gap between them may become so wide that it will never be bridged. One discussor, in fact, suggested that "all research should be used with caution". I believe, however, that, provided we direct all our studies primarily at the safety of ships, we shall be able to achieve

a closer link between theory and practice.

e) Technological Advances

In recent years there have been a number of major technological advances: in the development, for example, of microcomputers, sensing devices and transducers. The outstanding features common to all these cases are a reduction in size and hardware cost, and an increase in reliability. We must be careful to avoid putting blind faith in "improved" hardware, but nevertheless the availability of this equipment today does open up for us possibilities which did not exist in the past.

The paper which described the use of microcomputers and sensing devices on small fishing vessels provided an example of ways in which development is likely to take place in the future. Such devices can provide general guidance to small fishing vessel operators and every encouragement should be given to this type of work.

In the second Panel Discussions it was suggested that a black box, similar to those used in aircraft, should be employed to make continuous measurements of vessel motion. This, however, is not a new idea. It was considered at STABILITY 75 but has so far not been adopted for several reasons:

- i) The necessary hardware is bulky, and not yet reliable enough
- ii) The employment of the equipment calls for considerable human effort.
- iii) It has not yet been established that there is a sufficient market for such devices.

Recent technological advances can certainly deal with the first point, but I think the market demand for

this equipment is still very small and much effort is still needed to overcome various problems in operating the device under normal working conditions.

f) Impact of Computers

In the papers presented here the role played by the computer has been quite outstanding and there is no doubt that we are going to be relying on this facility more and more in the future. However, I should like to sound some warnings:

i) The cost of computer hardware has come down, but the cost of preparing suitable software has escalated.

ii) While we are now able to do very sophisticated calculations using numerical techniques, considerable thought still needs to be given to how our information will be used by designers and operators of ships and semi-submersibles.

iii) Users are going to have to be educated so that the full benefit can be gained from the use of the computer in the field of stability.

g) Regulations

The formulation of suitable regulations for practical use is the most obvious way of applying research results, and it was suggested by the Panel on Criteria and Regulations that there are three basic steps in this process:

- i) Criteria development
- ii) Criteria determination
- iii) The development of regulations

At step (i) criteria may be developed for a vessel in still water, or for the same vessel in waves, or the vessel's complete spectrum of motions may be taken into consideration in the criteria

proposed.

At Step (ii) the method on which the proposed criteria are based can be physical, deterministic or probabilistic.

As regards Step (iii), it is clear from the discussion that the development of regulations is a slow process and their implementation can take even longer.

It must be realised that regulations in themselves do not guarantee safety, and it is essential that the outcome of the conference should be used to increase interaction between users and those formulating regulations.

h) Advances in Design

In the past people generally had a very good idea what a "ship" looked like but today the term "ship" is loosely used to cover a whole range of floating structures, bearing, in some cases - little resemblance to the traditional seagoing vessel. One example of this is the semisubmersible now being so-much used in hydro-carbon exploration. Continuing advances in ship design make it very important that a flexible approach should be adopted towards stability. Changes need to be quickly taken into consideration and appropriate methods have to be rapidly developed to deal with the problems that arise. Stability regulations must keep pace with this situation.

To sum up what I have said in this section: it can be confidently stated that the organisers of STABILITY 82 have achieved their aim. However, we must not rest on our laurels: the momentum generated here must be maintained so that still further advances can be made in the safety of ships and ocean vehicles.

FUTURE ACTIVITIES

It would be useful to consider which acti-

vities we should concentrate our energies on in the future, and seven areas appear specially significant:

a) Correlation of Theoretical and Experimental Work

Theoretical studies fall into two broad groupings. "Fundamental studies", are aimed at testing out ideas and at establishing whether a basic principle is valid and can be developed further. "Comprehensive studies", have the object of developing full solutions that may one day become suitable practical techniques. These two approaches require different strategies and produce different kinds of results. Experimental studies can also be considered under two very similar headings: "simple experiments to establish fundamental principles" and "comprehensive programmes of experiments to validate theoretical solutions". Examples of both types of work have been presented to us during the conference. We should be actively encouraging further theoretical studies and experiments to check basic concepts and ideas. However, for the experimentalists and theoreticians who are deeply involved with comprehensive programmes, we would urge much closer co-operation between them because these studies are both costly and time consuming, and because advances can only be achieved by good correlation of results.

Such co-operation calls for a wider dissemination of the ideas that are being studied at the theoretical level and could provide material for experimental work, and for a more general announcement of intended work in the experimental field so that workers on theory will be alerted to look out for and obtain results as they become available. This conference has made a valuable contribution in this direction, but more effort is needed.

b) Improved Communication about Research

Good communication already exists in the field of stability between researchers and those wishing to use their results. For example, conference papers are published and informal discussion takes place whenever researchers and operators meet together; but there is scope for considerable expansion both in communication between the different types of research as well as between research workers and those who can benefit from their results. This could be done in several ways:

i) It would be useful to have an international databank of the research being performed worldwide on stability which could be consulted by those wishing to know what progress has been made, or planning to work, in a specific area.

ii) In addition to international conferences such as the present one there should be workshops in which specialists could debate a specific subject.

iii) Serious thought must be given to the actual presentation of research findings to potential users. As regards reports prepared for general circulation I would like to advocate the

philosophy, "Simplicity is beautiful", even at the expense of not being able to provide a detailed picture of the problem. We must ensure that such reports actually focus on the issues under consideration and that potential readers are not put off by a mass of complicated equations or mathematical symbols.

c) Feedback on Performance

All our work is dependent on the feedback of information, especially information regarding the operation of ships and ocean vehicles, and we

need some systematic method of gathering and recording such information so that researchers can have ready access to it in their own work.

It is important that all of us should try to identify the problems requiring solution and we would ask those who make use of our information to "market" their problems to researchers, so that highly trained minds can be brought to work on them.

A particular weakness at present is lack of information on "near accident" and "near capsized" situations. We must encourage operators to provide us with a wider range of information about their experiences with ships and ocean vehicles in seaways if we are to make progress.

d) Closing the Gap between Theory and Practice

The need for bridging the gap between user and researcher is appreciated by all concerned, but we have not yet found the best way of doing this. It is, however, clear from what has been said during the conference, that progress can be made if we take "the application of stability" as the basic objective of our research. To this end the following might be a useful procedure: identify a given problem; collect information on it; establish how a solution would be used in practice; devise a method of approach; determine a solution; apply the findings.

It should be borne in mind that a "complete solution" will not be possible, but it is still necessary to provide useable criteria at every stage.

e) Harnessing General Interests

This heading may not be quite clear to everyone so I shall try to explain its significance.

A comparison of the list of authors presenting papers at this conference with a similar list for the 1975 conference reveals that about 40% of the papers we have heard here were presented by new authors, and the majority of these belong to the younger generation. The interest and enthusiasm of these young researchers has been one of the key features of this conference for me and they must be encouraged in every way to continue their work, and be guided towards tackling the major real problems in this field.

Several of the speakers on the first day discussed demand and capability probability curves and indicated that their aim was to ensure a minimum overlap between these two curves. In the present context, however, we really want to have a good deal of overlap, so that there is a pool of able workers available to tackle continuing and emerging problems, and this pool of workers can only be found among these young enthusiasts.

f) Special Research Workshops

During the past few days various opinions have been expressed regarding the date of the next Stability conference. Some have mentioned 1985, and others have simply said that a seven-year gap is too long. I myself do not think that a date should be fixed at this stage as there is no guarantee that sufficient progress will have been made by that date to justify a conference. There are too many examples of three-yearly international conferences where the same ideas are repeated time after time, with very little new material being presented on each occasion. I would agree, all the same, that something between three and five years would probably be an



appropriate interval before the third Stability conference.

However, this present programme has covered a very broad spectrum of subjects and many of them call for much more detailed discussion and debate. In order to facilitate this, and to provide opportunities for keeping in touch I would propose that during the next two years or so a number of specialist international workshops should be arranged on specific topics such Water on Deck, Following and Quarters Seas, Experimental Methods, or Computation of Restoring. These workshops would be attended by small groups of interested workers and the programme could be conducted mainly on the panel discussion principle. Findings could be made generally available thereafter in published form. This would allow us to achieve the best of both worlds and provide the basis for the next conference

g) Levels of Stability

We are unlikely ever to arrive at a perfect set of criteria so what are the alternatives? One possible approach is to take one small step forward and to test its applicability before going on to the next step. However, this method of progress is so slow as to be totally unacceptable. Another possibility is to provide various "levels" of criteria, as we have already considered, and more effort is needed in this direction.

THE MESSAGE OF THE CONFERENCE

I believe that the participants in this conference will not allow the momentum it has generated to be lost, but it will take some time for us to see exactly how we are going to move forward for the benefit of everyone concerned.

Having spent five days here I am sure we should all like to have some particular message to remind us of the conference and I should like to propose the

following for you to take away:

"To research for the sake of research, we say, 'No'.

To research for the sake of administration, we say, 'No'.

To research for the sake of safety and application, we say a very big 'Yes!'"

VOTES OF THANKS

On behalf of the participants in the conference I should like to convey sincere thanks to a number of people who have made this event possible. First of all we are indebted to the following sponsors:

- |                |  |
|----------------|--|
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| Mr. Sato -     | President of Nippon Kaiji Kyokai                           |
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| Mr. Kondo -    | President of the Japanese Shipowners' Association          |

and members of the International Technical Committee for their contribution in connection with the technical programme.

We must also thank the Japanese Organising Committee for the splendid organisation of this conference, making particular reference to:

- |                  |   |
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| Mr. Mizushima -  | Chairman of the Committee                       |
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Dr. Sugai - of the Ship Research  
Institute, who has  
also acted as  
Conference Secretary

We are grateful to the conference secretariat for all their work, in particular: Professor Nakato; Professor Fujino; Professor Maeda; Professor Kinoshita; Dr. Kudo; Dr. Fukasawa; and especially Miss Ikeda, Miss Tsuruoka and their colleagues at the Conference Desk whose pleasant ways have created such a friendly atmosphere.

Finally we come to Professor Matora. It was his initiative that paved the way for holding this conference in Tokyo at this time. As you know, he is Chairman of the International Technical Committee and Vice-chairman of the Japanese Organising Committee, and he has worked tirelessly for the success of the conference.

Professor Matora has been a distinguished Professor of Naval Architecture at the top university in Japan for over thirty years and is President of the Society of Naval Architects of Japan, and his name is familiar to all who have studied ship motions, manoeuvring and stability. But for all of us who have come to know him it is his personal qualities, rather than his achievements, that make the deepest impression. What we shall remember from this conference are his sincerity, his generosity and his friendliness. I should like to ask you all to join me in saluting everyone who has made this conference possible, but in particular, Seizo Matora.

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