Towing Test and Motion Analysis of a Motion-Controlled Ship- Based on an Application of Skyhook Theory

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ABSTRACT

A novel–concept catamaran called the Motion-Controlled Ship (MCS) Type-6 is proposed. It consists of a cabin, two hulls, four suspension setting units as well as four relative independence control system units. The cabin and hulls are separated by suspensions. By implementing adaptive control algorithms, the motion modulation of the cabin is realized. A hull-excited bench test is conducted to validate the function of the control systems, following which a model ship towing test is performed in regular wave conditions. The motion responses of the MCS in terms of heave, pitch and roll are analysed under five control algorithms at two different towing speeds. Compared to a rigid body catamaran (in which suspension systems are invalid), the pitch is eliminated by a maximum of 93% and an average of 74.8% under certain test conditions.

Key Words: *Catamaran, Motion Control, Stability, Skyhook, Suspension System*

1. INTRODUCTION

Ride comfort plays an important role in ground vehicle evaluation. Numerous researchers have dedicated themselves to the investigation and improvement of devices for shock absorption or vibration elimination. Currently, suspension setting, which comprises springs and dampers, is commonly applied in such motion control systems.

Comparing road profiles to the ocean surface, one finds that the latter is much rougher and can easily lead to violent shaking; however, suspension settings have seldom been used in ocean vehicles to improve ride comfort and stability.

The development of a Motion-Controlled Ship (MCS) has been ongoing since 2008. The MCS Type-1, shaped similar to a tricycle, had three small hulls and one big submerged float (Figure 1). One suspension setting, which consisted of a spring and an oil damper, was equipped between the cabin and one of the hulls. It was found that strong dampers had a relatively high efficiency in reducing the motion of the small hulls but had less effect on the cabin (Lu. 2010).

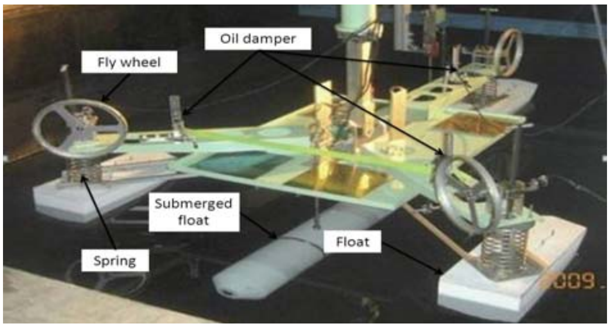

Figure 1 Structure of the MCS Type-1

The MSC Type-2 was a catamaran. It had two suspension setting units on each hull, located at the front and rear (Figure 2). The test results indicated that the reduction of the heave and pitch motions of the cabin was improved along with the increase of the damping coefficient when the towing speed was $1.5m/s$. The results also suggested that the relative displacement between the cabin and hulls could produce sufficient kinetic energy to be reused (Tsukamoto. 2012).

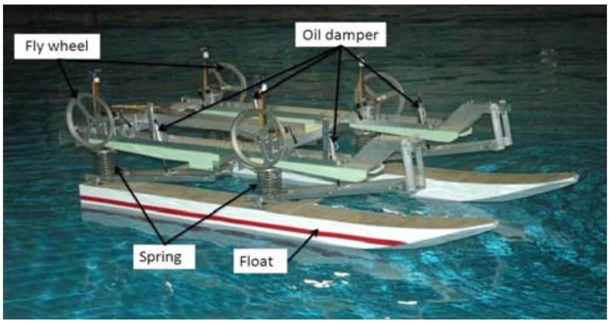

Figure 2 Structure of the MCS Type-2

Instead of oil dampers, the so-called electronic damper was formed and applied to the MCS Type-3 (Figure 3). A stepping motor was connected to a load resistor in series to construct an electrical circuit; by tuning the value of the resistance, the current in the circuit was made to vary and therefore the rotations of the motor shafts were adjusted. This affected the angular velocity of a pinion that meshed with a rack, leading to a change in the relative velocity between the cabin and hulls. This can be seen as an equivalent result of that obtained by damper tuning. A towing test was performed; it was shown that the ability of motion elimination increased along with the reduction of load resistance, which meant an increase in the damper coefficient. It also implied that a strategy of simultaneously enhancing motion control and energy harvesting is possible. A compromise between those aims is necessary and should be made according to the use of the ship (Han. 2013a).

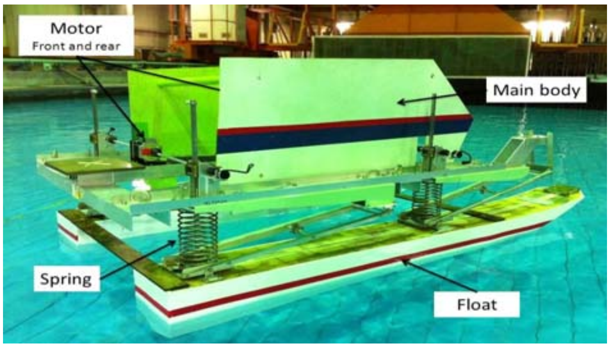

Figure 3 Structure of the MCS Type-3

A semi-active motion control system was developed for the MCS Type-4. The ship structure was similar to the Type-3, except the number of motors in one control system was increased from one to two. The control system analysed the feedback signals of the acceleration of the cabin as well as the relative velocity between the cabin and hulls, then determined whether or not to trigger the motion-control system. Through the inductive force generated by the motors, the heave and pitch of the cabin could be reduced. This was proved by a towing test. In the wave energy harvesting phase, the motors acted as generators, and a wave energy harvesting potential (defined as the ratio of the harvested energy to the wave energy contained by the crest over the width of the hulls) of 110% was achieved (Han. 2013b).

Between Type-2 and Type-4, the MCS can be seen as a high speed ship with hulls that planed on the sea surface. Type-5 adopted displacement type hulls.

In this paper, the MCS Type-6, which is equipped with a pair of displacement hulls, is introduced. The suspension settings are improved so as to yield roll motion. Based on an application of skyhook control theory, the model ship is tested and evaluated for its potential to eliminate heave, pitch and roll motion.

2. CHALLENGES OF THE MODEL

There are three challenges facing the design of the proposed novel catamaran. The first is motion separation between the cabin and hulls. A traditional ship has a rigid body, which generates motion in six degrees of freedom: translational motion—surge, yaw and heave; rotational motion—roll, pitch and sway. In our design, the two hulls are connected rigidly. By means of springs which are mounted between the cabin and hulls, the motion of the ship increases by up to 12 degrees of freedom. Considering the stability of the ship, the suspension supports are designed to refrain from generating phase differences in surge, sway and yaw between the cabin and hulls. Hence, a 9 degree-of-freedom model is proposed, the heave, pitch and roll of the cabin as well as those motion of the hulls; surge, sway and yaw of the whole ship. A blueprint for this design is given in Figure 4. It shows how the relative forward and lateral motions between the cabin and hulls are restricted.

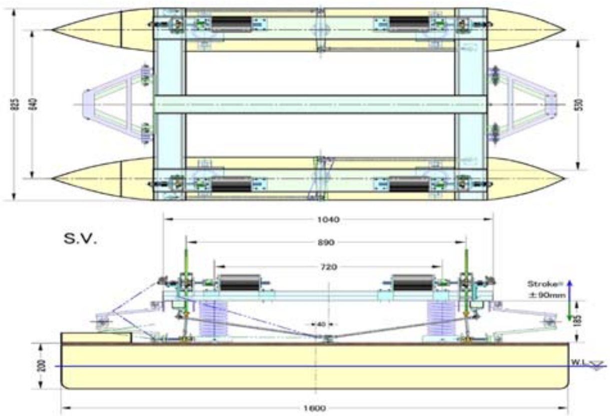

Figure 4 Blueprint of the MCS Type-6

The second challenge is effective power transfer between mechanical and electrical forms. To solve this, a crank and connecting rod (con-rod)-type mechanism is at first considered. A con-rod is connected to one of the hulls, while a crank rotates the motor shaft. Through testing, we find that the transmission efficiency from electrical to mechanical energy is unexpectedly low. Thus, we return to the former rack-pinion mechanism and attempt to make a modification. The final proposal, which adds an adding ball joint to the bottom of the rack to offer another degree of freedom for the roll motion of the cabin, is given in Figure 5.

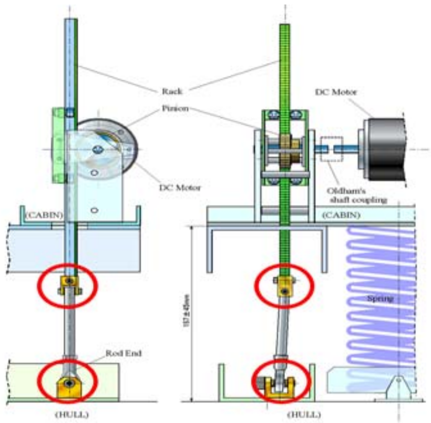

Figure 5 Configuration of the revised rack

The third challenge is high feasibility of motion-control system to achieve a certain level of stability of the cabin. In the model ship, there are four control spots, which work independently along with the input signal of the acceleration at each spot. We assume that the cabin could simultaneously obtain its pre-concerted motion state, if the four control sports achieve their. More specific details will be introduced in the next section.

3. CONTROL ALGORITHMS

3.1. Skyhook Control Theory

For an ideal skyhook control, we consider a design consisting of a damper connected to a suspended mass and an inertial reference which is fixed in the sky. When the base reference is excited, the damper will provide a force to eliminate the motion of the mass. Although this is a purely imaginary configuration, it serves as an inspiration for the design concept of the proposed motion control system.

In the model ship, the hulls are excited by waves. The wave force can be illustrated as a combination of a spring force (K_W) and a damping force (C_W). Meanwhile, the suspension system, set between the cabin and hulls, provides another spring force (K), while the motor fit on the cabin produces a reaction force meant to counteract the force that acts on the cabin. This skyhook-like dynamic configuration is shown in Figure 6.

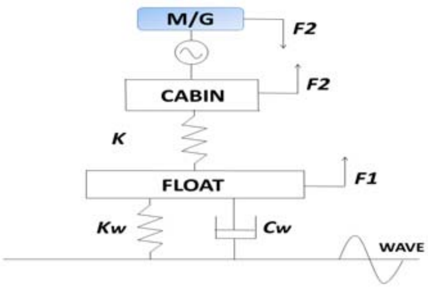

Figure 6 Configuration of the model ship dynamic

Specifically, when a spring starts to expand or contract from its neutral length, the motor applies power to restrain it (motor mode); when the spring expands or contracts to the normal length, the generator absorbs the spring power and converts it into electricity (generator mode); hence, the external force acting on the cabin is supposed to be zero.

3.2. DC Motor and Sensor

Considering the affordability of the control system, a brushed DC motor Maxon-353300, made by Maxon Japan Co. Ltd, is selected and tested. The stall torque of the motor is $1.41 Nm$, the terminal resistance is 1.06Ω . The sensitivity of the G-sensor is $1.2V$ per gravitational acceleration.

3.3. PI Control System

Our control contains both proportional and integral elements and is therefore known as PI control. In P control the system acts in such a manner that the control effort is proportional to the error, while which of I control is proportional to the integral of the error.

In the current study, only P control is activated. The acceleration of the cabin is detected by a G-sensor and transferred to an integral operator to calculate the cabin's absolute velocity. The difference between the reference velocity (set to zero) and the cabin absolute velocity is analysed. By tuning the gain of the P control, the instruction signal measured in voltage (V_{out}) is varied. This signal is fed to the motor and determines the value of the torque force it generates. Such torque force acts on the hull through a rack-pinion unit that eventually restrains the motion of the cabin. Therefore a new acceleration is generated, and causes the control circle to repeat until the current velocity of the cabin reaches 0m/s, this procedure is shown in Figure 7.

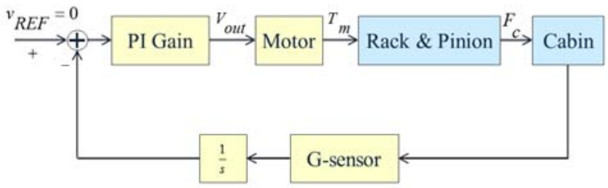

Figure 7 PI control procedure

4. HULL-EXCITED BENCH TEST AND SIMULATION

A hull-excited bench test is performed to validate the proposed skyhook control system, simplified to one degree of freedom.

4.1 Experiment Setting

An oscillation machine is settled on a heavy steel framework (Figure 8). The oscillation operator is connected to a metal support on which the model ship rides. The hulls are tightly tied to the bottom of the metal support, when the oscillation machine exerts a force at the centre of gravity of the frame-ship structure, the hulls move along with the metal support, therefore yielding heave motion in the cabin.

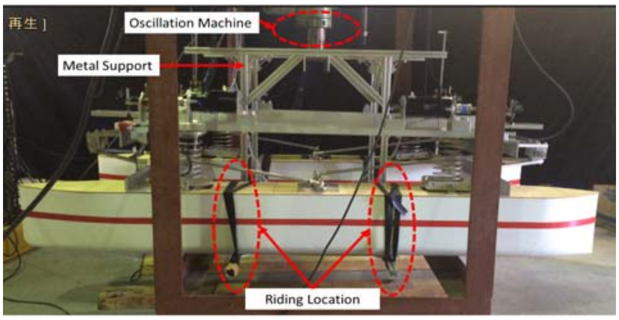

Figure 8 Experimental set-up of the bench test

4.2 Experiment Conditions

A simulation model with one degree of freedom is built in LTspice®, which is an open source analogue electronic circuit simulator produced by the semiconductor manufacturer Linear Technology (LTC).

The value of the mechanical friction is estimated in two steps. First, the skyhook control system is eliminated; therefore a simple mass-spring-mass structure is constructed. Under this condition a bench test is implemented and the heave response of the cabin is recorded. Second, tuning the value of the friction in the simulation program until the similar motion response of the cabin is obtained. The value of friction is determined at this point.

The P-gain (G_P) based on the design of our PI control system is expressed as a multiplication of several components, shown in Equation (1):

$$G_P = \frac{1}{s} \times G_G \times G_{PC} \times G_{PA} \times G_A \quad (1)$$

where G_G is the gain of the G-sensor, equal to $0.122V/(m/s^2)$; G_{PC} is the gain of the proportional circuit, expressed as 25; s is the Laplace Operator; G_{PA} is the gain of a power amplifier, equal to 4; G_A is the gain of an adjusting unit, which is a ratio of a reference resistance ($47k\Omega$) to R_{sky}. Note that R_{sky} is an adjustable resistance, so G_P therefore can be described as

$$G_P = 573.4 / R_{sky} \quad (2)$$

Here, R_{sky} can be seen as a medium for modulating the magnitude of the PI control system. According to the test results, we find that if R_{sky} is smaller than $4k\Omega$, an unstable motion is observed for the cabin. Therefore, we consider that $4k\Omega$ is the threshold value of R_{sky}. In order to make a comparison between the performance of several skyhook control conditions, another two R_{sky} are decided: $10k\Omega$ and $20k\Omega$. The equivalent P-gains of these cases are 143.35, 57.34 and 28.67, respectively.

The stroke amplitude of the hulls is set as $3cm$, while frequency is $0.8Hz$, $1.0Hz$, $1.2Hz$ and $1.4Hz$.

4.3 Experimental Results

The results of the bench test and the simulation for the dimensionless heave and power consumption are shown in Figure 9 to Figure 12. The x-axis represents stroke frequency, while y-axis is either dimensionless heave or power consumption.

In Figure 9, it is found that among the three resistances, $R_{sky} = 4k\Omega$ shows the strongest motion-elimination ability, reducing the heave by more than 50%. When R_{sky} gets bigger, such elimination ability gets weaker. Moreover, in the same control algorithm, a higher stroke frequency achieves better motion elimination. The heave motion is mitigated to 20.5% when $R_{sky} = 4k\Omega$ and $f = 1.4Hz$.

In Figure 10, the simulation results show the same trend, along with the variety of R_{sky} and f. However, compared to Figure 9, a significant deviation of the magnitude of heave is observed, which might be caused by the rough estimation of the friction or other unconsidered factors.

In Figures 11 and 12, it is easy to determine that when $R_{sky} = 4k\Omega$, the power consumption maintains its highest level, which decays if R_{sky} grows bigger. This proves that in order to achieve better motion control, more power consumption is necessary. The power can be seen as partially used to maintain the stability of the cabin and partially devoted to overcoming friction. Note that the stability in the bench test is worse than that in the simulation, which implies that the power consumption should be smaller. However, the overall power consumption in the bench test is higher than that in the simulation, suggesting that more power is consumed in overcoming friction during the test than the simulation. This conclusion agrees with the inference in the above motion analysis.

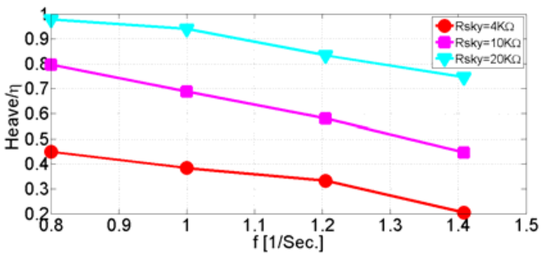

Figure 9 Dimensionless heave in bench test

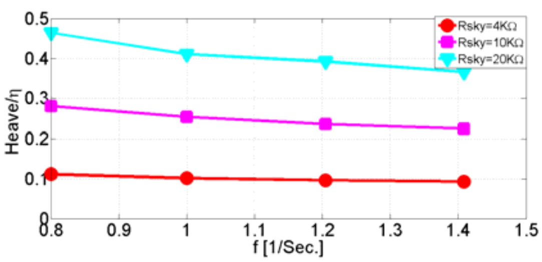

Figure 10 Dimensionless heave in simulation

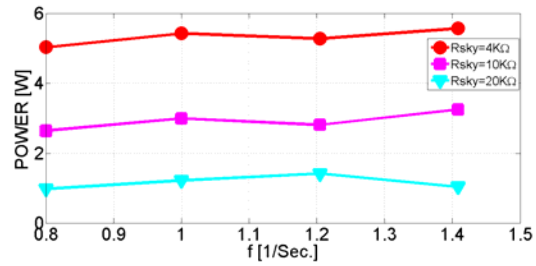

Figure 11 Power consumption of bench test

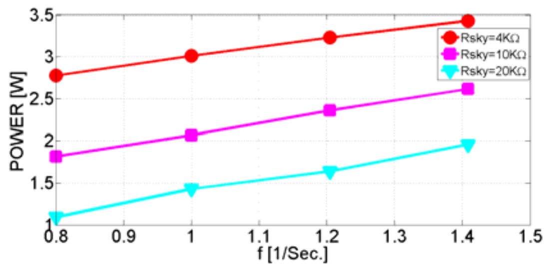

Figure 12 Power consumption of simulation

4.4 Conclusions

Through the comparison between the simulation and the bench test, the one-degree of-freedom control system was evaluated. Conclusions can be summarized as:

- The control circuit and control panel are well-designed and the blushed DC motor is well functioned. Four one-degree-of-freedom control system sets perform reasonably.

- Higher P-gain produces better heave motion reduction. $R_{sky} = 4k\Omega$ is currently the optimal skyhook control condition and could be adopted as a test condition in the towing tank test.

- A better motion control strategy consumes more power.

- The friction of the ship structure should be discussed further so as to improve the accuracy of the simulation program.

5. TOWING TEST AND RESULTS

5.1 Model Ship Specifications

The components and the structure of MCS Type-6 are given in Figure 13. The model ship is 1.6m in length and 0.83m in width. The weight of the cabin session is $34.71kg$, which of the hulls session is $13.14kg$. The mass of the suspension parts is equally distributed into those two sessions. The spring constant is $615N/m$ and water surface is $0.498m^2$. The locations of the four control spots are arranged symmetrically 0.445m from the centre of gravity of the hulls from bow to stern.

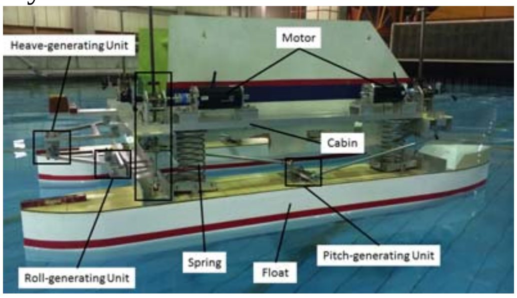

Figure 13 Structure of the MCS Type-6

5.2 Experiment Descriptions

A towing test was performed in December 2014, at Ocean Engineering Basin in the Chiba Campus of the University of Tokyo.

In order to evaluate the efficiency of the skyhook control system, a rigid body catamaran is used as a reference model. By connecting the cabin and the hulls with four metal plates, the suspension system was invalid, therefore an equivalent model of rigid body catamaran is formed, called Rigid Body mode.

Control OFF mode, is a test condition when the skyhook control system is turned off, by only letting springs be functioned.

Skyhook control algorithms are $R_{sky} = 4k\Omega$, $10k\Omega$ and $20k\Omega$, which are the same as the bench test.

The towing test is performed in regular wave conditions which are listed in Table 1 and Table 2. The direction of wave propagation β is 180° and 90°.. Due to the limitation of the towing tank, the towing speed is chosen as 0.0m/s and 1.5m/s.

Table 1 Regular wave conditions $\beta = 180°$

Wave Period [sec]	Frequency [Hz]	Wave Amplitude [cm]	Wave slope
0.67	1.5	1.12	0.100
0.77	1.3	1.47	
0.83	1.2	1.71	
0.91	1.1	2.06	
1.00	1.0	2.48	
1.11	0.9	3.06	
1.25	0.8	1.94	0.050
1.43	0.7	2.54	
2.00	0.5	4.97	

Table 2 Regular wave conditions $\beta = 90°$

Wave Period [sec]	Frequency [Hz]	Wave Amplitude [cm]	Wave slope
0.67	1.5	1.12	0.100
0.77	1.3	1.47	
0.83	1.2	1.71	
0.91	1.1	2.06	
1.00	1.0	2.48	
1.11	0.9	3.06	
1.25	0.8	3.88	
1.43	0.7	5.08	
2.00	0.5	4.97	0.050

5.3 Motion Responses

The results of the heave, pitch and roll of the cabin are given from Figure 14 to Figure 19. The x-axis represents the encounter wave frequency, while the y-axis represents dimensionless value.

When $\beta = 180°$, Control OFF mode shows the biggest motion response and significant resonances are observed. Compared to the rigid body mode, control modes show several levels of reduction in heave and pitch motion, the potential of which increases along with the decrease of R_{sky}. In other words, $R_{sky} = 4k\Omega$ shows the best motion elimination. It agrees with the conclusion in the bench test.

When $\beta = 90°$, only the control mode in $R_{sky} = 4k\Omega$ is tested. Comparing the control mode to the rigid body mode, heave motion is greatly reduced, especially around the resonance frequency. However, in roll motion such reduction only appears around the resonance frequency. In other wave frequencies, rigid body shows relatively smaller motion response. In general, Control OFF mode generates the largest motion magnitude.

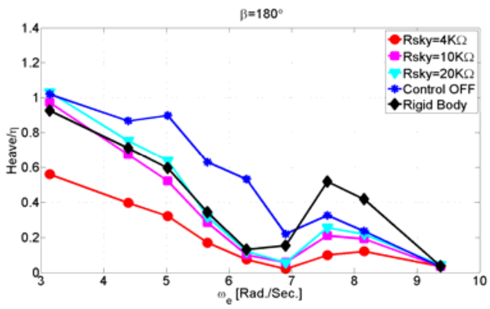

Figure 14 Dimensionless heave when $V = 0.0$m/s at $\beta = 180°$

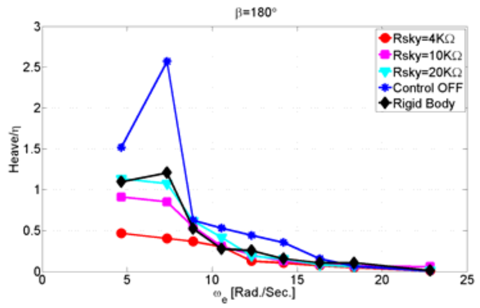

Figure 15 Dimensionless heave when $V = 1.5$m/s at $\beta = 180°$

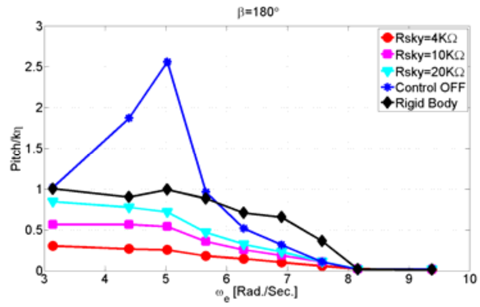

Figure 16 Dimensionless pitch when $V = 0.0$m/s at $\beta = 180°$

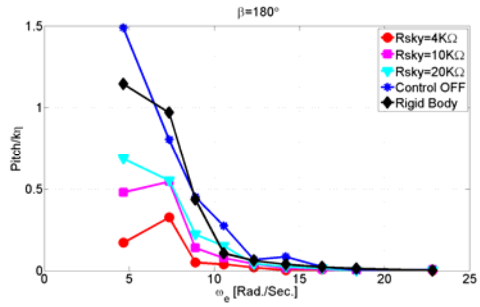

Figure 17 Dimensionless pitch when $V = 1.5$m/s at $\beta = 180°$

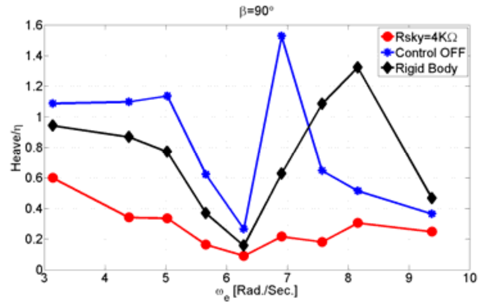

Figure 18 Dimensionless heave when $V = 0.0$m/s at $\beta = 90°$

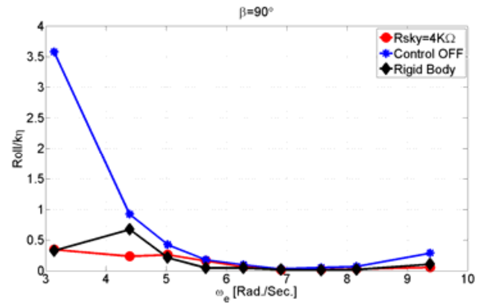

Figure 19 Dimensionless roll when $V = 0.0$m/s at $\beta = 90°$

5.4 Stability Evaluation of the Cabin

A ratio of motion response in Control Mode against Rigid Body mode is applied as an index of stability evaluation, which is expressed as

$$Ratio = {ControlMode}\Big/{RigidBodyMode} \times 100\% \qquad (3)$$

The lower the ratio is, the better the stability represents. Assuming the benefit of a control mode is expressed in Equation (4). When the value is positive, a certain benefit is gained, otherwise a certain loss is obtained.

$$B = 100\% - Ratio \qquad (4)$$

The results of the ratio are shown from Figure 20 to Figure 24. When $\beta = 180°$, in most of cases benefits are obtained. The highest benefit reaches up to 93% in pitch when $R_{sky} = 4k\Omega$, $V = 1.5\text{m/s}$, $\omega_e = 14.19\text{Rad/Sec.}$

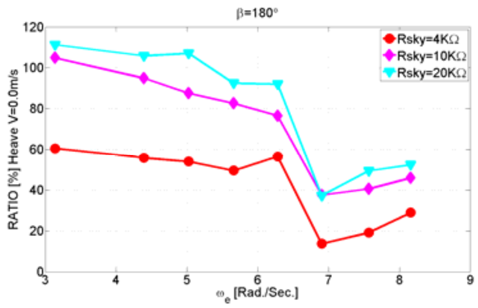

Figure 20 Stability of heave when $V = 0.0\text{m/s}$ at $\beta = 180°$

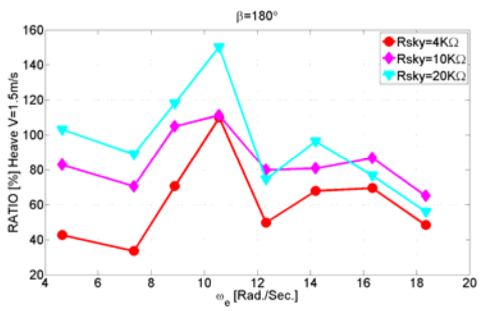

Figure 21 Stability of heave when $V = 1.5\text{m/s}$ at $\beta = 180°$

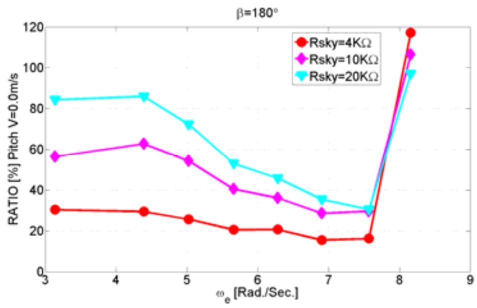

Figure 22 Stability of pitch when $V = 0.0\text{m/s}$ at $\beta = 180°$

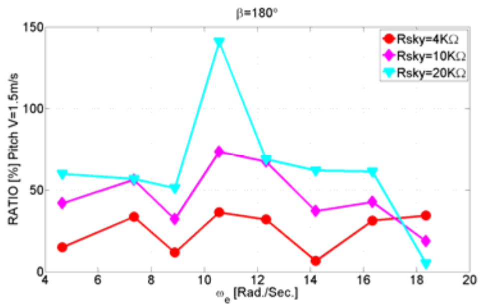

Figure 23 Stability of pitch when $V = 1.5\text{m/s}$ at $\beta = 180°$

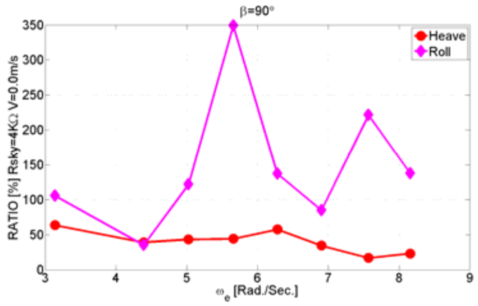

Figure 24 Stability of the cabin when $V = 0.0\text{m/s}$ at $\beta = 90°$

If the average benefit is given by

$$B_{avg} = {1}\!\Big/\!{8} \sum_{n=1}^{8} B_n \qquad (5)$$

Then the B_{avg} at $R_{sky} = 4k\Omega$ can be summarized and shown in Table 3. An average benefit in pitch reaches up to 74.8%, when the towing speed $V = 1.5\text{m/s}$ and $\beta = 180°$. However, an average loss of 49.35% in roll is obtained when $V = 0.0\text{m/s}$ and $\beta = 90°$.

Table 3 Average benefit level at $R_{sky} = 4k\Omega$

$\beta = 180°$			$\beta = 90°$	
	0.0m/s	1.5m/s		0.0m/s
heave	57.7	38.5	heave	59.7
pitch	65.5	74.8	roll	-49.35

6. CONCLUSIONS

In the current study, a $1/5$-scale model ship that contains suspension systems and brushed DC motors was tested and evaluated. A hull-excited bench test and a tank towing test were performed. The motion reduction of the heave, pitch and roll of the cabin under several control modes were validated.

The bench test showed that a reasonable heave motion reduction was obtained and the highest level reached up to 79.5% of reduction. A higher P gain of the control algorithm generated stronger motion elimination ability. However, because of the friction of the structure, extra power consumption was unavoidable. Further work will be required to understand friction control.

In the towing test, the optimal control algorithm agreed with the bench test, with an average benefit level of 74.8% in pitch when towing velocity is $1.5m/s$. The peak reduction of pitch reached to 93% at a certain wave and control condition. However, this reduction was not reproduced in roll motion.

In next step, the research might focus on improving the motion control system by developing an absolute position control model, and evaluating the motion responses in irregular wave conditions.

ACKNOWLEDGEMENT

I am sincerely grateful, to Mr. Yukitsugu Hirota who is a responsible, sophisticated and enthusiastic electro-dynamics expert and taught me a lot of principles about system controls and motors and to Mr. Hiroshi Itakura, an excellent technical staff member, who gave me sufficient support about parts manufacture and ship building.

The major part of this paper is based on results obtained from a project commissioned by the New Energy and Industrial Technology Development Organization (NEDO). A part of the paper was supported by Japan Society for the Promotion of Science (JSPS), Grant-in-Aid for Challenging Exploratory Research, 25630399.

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