浮体式波力発電装置に関する動揺制御による エネルギー吸収効率向上の検討

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動力学的に継がっている振点系は，動揺制御装置として浮体式波力発電装置のエネルギー吸収効率向上に応用できる。そこで，この振点系に関する時間領域における制御を用いてエネルギー吸収効率向上を検討し，制御に使われるエネルギー量を算出した。また，2種類の模型を用いた水槽実験を行った結果，振点系による制御の有効性とこれを用いた装置の設計ポイントが明らかになった。

キーワード：浮体式波力発電装置，浮体運動，動揺修正装置，動力学，制御，エネルギー吸収効率

Use of an On-board Motion Compensated Block to Enhance Wave-energy Conversion by Floating Devices

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This paper describes an application of a system that uses favorable dynamic interaction of coupled, controlled oscillators to provide motion compensation on a floating offshore structure. Such motion compensation can be used to absorb energy from the floating-hull motion on the JAMSTEC "Mighty Whale" device.

The energy requirements for such a system are examined. Also discussed are model experiments utilizing a passive version of the proposed apparatus. Results from tank tests on two types of floating devices (including the Mighty Whale) indicate that the system would be beneficial, though careful design would be essential for both small and large-scale experiments using active control.

Key Words: Wave energy, floating devices, motion compensation, dynamics and control, conversion efficiency

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1 Introduction

The present work on wave-energy conversion at JAMSTEC follows a series of major developments, which began over two decades ago with the world's first large-scale floating wave-energy device "Kaimei". Development of the "Mighty Whale" device has been in progress since 1987 [Hotta *et al.* (1995)[1]]. The prototype for sea trials is shaped like a whale, and supports three oscillating water columns (OWC) facing the predominant wave direction. The prototype is 50 m in length, 30 m in breadth, 12 m in depth and is designed to be ballasted to float even keel at a draft of 8 m. The operating water depth is about 40 m. Wave-induced oscillation of the water-level inside the three chambers forces an alternating air-flow over three tandem-type Wells air turbines. The turbines power three out of four induction generators available on board. The overall rated capacity is 110kW. The prototype is shortly to be moored at an open-sea location just outside Gokasho Bay off Mie Prefecture.

The advantages of floating devices such as the Mighty Whale are threefold : The first is the significant economy that results from the ability of floating hulls to experience reduced impact loads in extreme wave conditions. The second advantage is that available energy in the waves generally increases with increasing water depth (exceptions are shallow-water regions where refractions due to favorable bottom topography can lead to focusing of energy). Finally, the floating-body motion of the hull can increase energy absorption in certain wave conditions, due to increased relative motion between the OWCs and the hull.

The work in this report concerns the third effect. For axisymmetric, primarily heaving buoys supporting OWCs [e.g. McCormick (1976)[2], Masuda (1985)[3]], the buoy heave motion is generally found to increase energy absorption. However, theory and laboratory experiments on a Kaimei-type floating OWC device (Maeda, *et al.* 1985)[4] have shown the floating-body motion of the hull to reduce energy absorption at most wave frequencies of interest. Similar observations have also been made with Mighty Whale laboratory models. For this reason, a system is being developed in this work, whereby the floating-hull motion of the Mighty Whale can be utilized to supplement the energy already being generated by the OWCs. The design constraints are : to achieve this without the use of actuated moorings or tethers, to retain the original advantages of a floating hull, and to restrict any energy required to operate this system to a minimum.

Figure 1 shows an example of such a system, which can also be set up to work in the surge or sway directions of hull motions. Note that the figure is illustrative in that it does not reflect the exact configuration of the prototype or of the proposed system.

2 Theory

The system under study is schematically shown in Figure 2. Mass M_s represents the floating hull, mass M_c the block to be motion-compensated. Mass M_m is undamped as far as practical. For multiple OWC chambers, a motion compensated block M_c may be provided for each chamber. Throughout this paper, however, only one motion compensated block is used. The equations of motion can be written as,

$$M_m\ddot{x}_m = -k_m(x_m - x_c) + f_m \quad (1)$$

$$M_c\ddot{x}_c = -f_m + f_c - C_c(\dot{x}_c - \dot{x}_s) - k_c(x_c - x_s) - T_L + k_m(x_m - x_c) \quad (2)$$

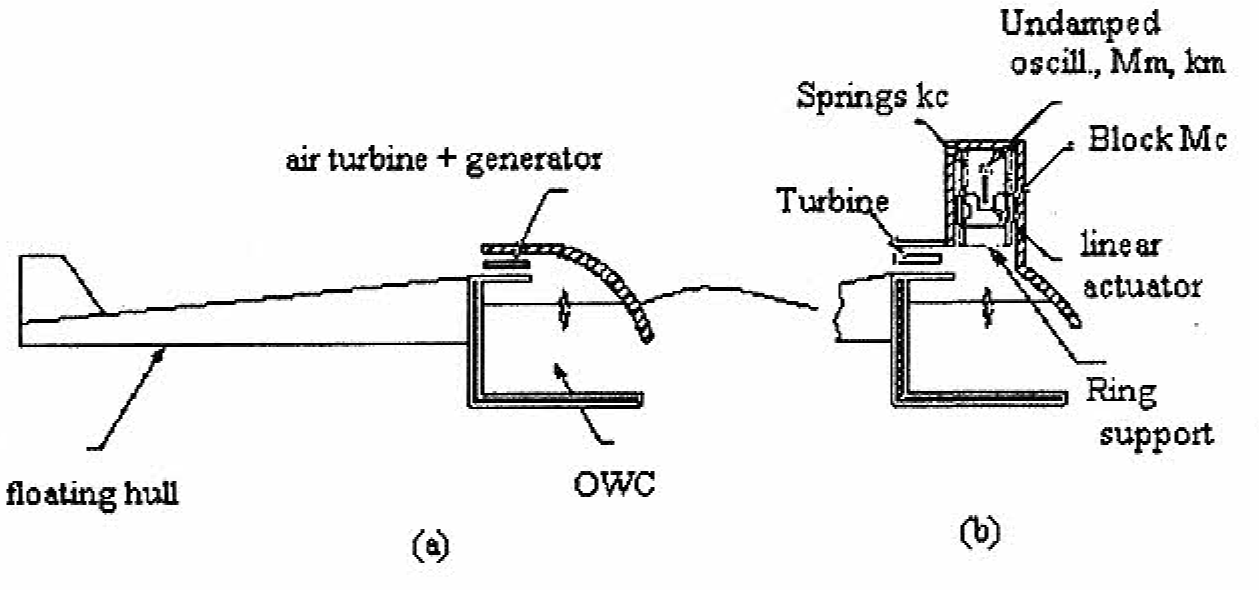

Fig. 1 Illustration of a floating device fashioned after the Mighty Whale prototype and possible modification. See Washio, *et al.* 1998[5] for actual prototype design; see Figure 3 for implementation in present experiments.

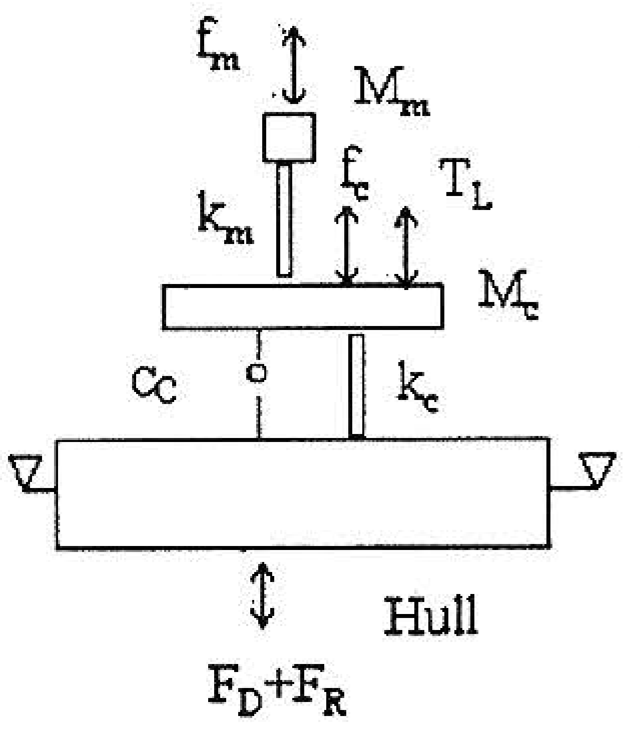

Fig. 2 Schematic model of the motion compensator

$$M_s\ddot{x}_s = F_D - f_c - F_R - k_s x_s + C_c(\dot{x}_c - \dot{x}_s) + k_c(x_c - x_s) \tag{3}$$

where

$$F_R = M_\infty \ddot{x}_s + \int_0^\infty h(\tau)\dot{x}_s(t-\tau)d\gamma \tag{4}$$

with $h(\tau)$ denoting the causal impulse response function, and M_∞ the infinite-frequency added mass in heave for the hull. x_s, x_c, and x_m denote instantaneous vertical displacements. T_L denotes the force acting on M_c due to alternate compression and expansion of air in the OWC chamber. Note that T_L is a function of chamber pressures, which in turn depend on incoming wave conditions, device geometry, and turbine-generator settings. Therefore, it is necessary to solve a system of equations where Eqns. (1-3) are supplemented with three additional equations for three chamber pressures. The aim of the present discussion is to illustrate the dynamics of the proposed system. T_L can be left as a variable for that purpose, as evident from Eqns. (10) and (11) below.

f_m and f_c are control forces applied to masses M_m and M_c respectively. Substituting Eqn. (4) in Eqn. (3) and applying the Fourier transform to Eqns. (1-3),

$$-\omega^2 M_m \hat{X}_m = -k_m(\hat{X}_m - \hat{X}_c) + \hat{f}_m \tag{5}$$

$$-\omega^2 M_c \hat{X}_c = -\hat{f}_m + \hat{f}_c - i\omega C_c(\hat{X}_c - \hat{X}_s) - k_c(\hat{X}_c - \hat{X}_s) - \hat{T}_L + k_m(\hat{X}_m - \hat{X}_c) \tag{6}$$

$$-\omega^2 M_s \hat{X}_s = \hat{F}_D - \hat{f}_c + \omega^2(M_\infty + M_\omega)\hat{X}_s - i\omega\mu_\omega \hat{X}_s - k_s\hat{X}_s + i\omega C_c(\hat{X}_c - \hat{X}_s) + k_c(\hat{X}_c - \hat{X}_s) \tag{7}$$

where M_ω and μ_ω are the frequency-dependent added mass and radiation damping coefficients for the ship in heave, and are defined by

$$h(\tau) = \frac{2}{\pi}\int_0^\infty \mu_\omega(\omega)\cos\omega\tau d\tau = -\frac{2}{\pi}\int_0^\infty \omega M_\omega(\omega)\sin\omega\tau d\omega \tag{8}$$

The symbol^ denotes Fourier transforms.

For a passive system without active control we set $f_m = 0$ and $f_c = 0$. Eqns. (5-7) can be rewritten in matrix form $A\hat{X} = \hat{F}$ with,

$$A = \begin{bmatrix} -\omega^2 M_m + k_m & -k_m & 0 \\ -k_m & -\omega^2 M_c + k_c + k_m + i\omega C_c & -i\omega C_c - k_c \\ 0 & -i\omega C_c - k_c & -\omega^2(M_s + M_\infty + M\omega) + i\omega(\mu_\omega + C_c) + k_c + k_s \end{bmatrix} \tag{9a}$$

$$\hat{X} = \begin{bmatrix} \hat{X}_m \\ \hat{X}_c \\ \hat{X}_s \end{bmatrix} \tag{9b}$$

$$\hat{F} = \begin{bmatrix} 0 \\ -\hat{T}_L \\ \hat{F}_D \end{bmatrix} \tag{9c}$$

The variations μ_ω, $M_\infty + M_\omega$, and $\hat{F}_D$ versus ω can be determined numerically [e.g. Pizer, 1997)[6)]].

Denoting the determinant of the matrix A as $|A|$, we can express the complex amplitudes $\hat{X}_m$, $\hat{X}_c$, and $\hat{X}_s$ as,

$$\hat{X}_m = \frac{1}{|A|}(-D_{12}\hat{T}_L + D_{13}\hat{F}_D) \tag{10a}$$

$$\hat{X}_c = \frac{1}{|A|}(-D_{22}\hat{T}_L + D_{23}\hat{F}_D) \tag{10b}$$

$$\hat{X}_s = \frac{1}{|A|}(-D_{32}\hat{T}_L + D_{33}\hat{F}_D) \tag{10c}$$

where the elements D_{ij} of the adjoint of A are given by

$$D_{12} = k_m[-\omega^2(M_s + M_\infty + M_\omega) + i\omega(\mu_\omega + C_c) + k_c + k_s] \tag{11a}$$

$$D_{13} = k_m(i\omega C_c + k_c) \tag{11b}$$

$$D_{22} = (-\omega^2 M_m + k_m)[-\omega^2(M_s + M_\infty + M_\omega) + i\omega(\mu_\omega + C_c) + k_c + k_s] \tag{11c}$$

$$D_{23} = D_{32} = (-\omega^2 M_m + k_m)(i\omega C_c + k_c) \tag{11d}$$

$$D_{33} = (-\omega^2 M_m + k_m)(-\omega^2 M_c + k_c + k_m + i\omega C_c) - k_m^2 \quad (11e)$$

and

$$|A| = (-\omega^2 M_m + k_m)(D_{11}) + k_m(-D_{12}) \quad (11f)$$

where

$$D_{11} = (-\omega^2 M_c + k_c + k_m + i\omega C_c)[-\omega^2(M_s + M_\infty + M_\omega) + i\omega(\mu_\omega + C_c) + k_c + k_s] - (i\omega C_c + k_c)^2$$

Note from Eqns. (10b), (11c) and (11d) that $\hat{X}_c = 0$ at a frequency at which $(-\omega^2 M_m + k_m) = 0$; which is the uncoupled natural frequency of oscillator (M_m, k_m). Consequently, for a passive system described by Eqn. (9) block M_c would be perfectly heave-compensated if the excitation force $\hat{F}_D$ were a perfect sinusoid with frequency $\omega_m = \sqrt{\frac{k_m}{M_m}}$.

For an active system we measure vertical accelerations $\ddot{x}_m$, $\ddot{x}_c$, and $\ddot{x}_s$ and use an analog circuit comprising op-amp-based integrators and amplifiers to compute input voltages to actuators driving M_m and M_c. The voltages are proportional to forces f_m and f_c given by,

$$f_m = M_m(\ddot{x}_m - \ddot{x}_c) + k_m(x_m - x_c) \quad (12)$$

$$f_c = k_l(x_c - x_s) \quad (13)$$

Substituting Eqns. (12) and (13) in Eqns. (1-3) we find,

$$M_m \ddot{x}_c = 0 \quad (14)$$

$$(M_c - M_m)\ddot{x}_c + M_m \ddot{x}_m + C_c \ddot{x}_c - C_c \dot{x}_c - - C_c \dot{x}_s + (k_c - k_l)x_c + (k_l - k_c)x_s = -T_L \quad (15)$$

$$M_s \ddot{x}_s - C_c \dot{x}_c + C_c \dot{x}_s + (k_s + k_c - k_l)x_s + (k_l - k_c)x_c + \int_0^\infty h(\tau)\dot{x}_s(t-\tau)d\tau = F_D \quad (16)$$

Fourier transformation leads to the following equations:

$$-\omega^2 M_m \hat{X}_c = 0 \quad (17)$$

$$-\omega^2 M_m \hat{X}_m + -\omega^2(M_c - M_m)\hat{X}_c + i\omega C_c \hat{X}_c - i\omega C_c \hat{X}_s + (k_c - k_l)\hat{X}_c + (k_l - k_c)\hat{X}_s = -\hat{T}_L \quad (18)$$

$$-\omega^2(M_s + M_\infty + M_\omega)\hat{X}_s - i\omega C_c \hat{X}_c + i\omega(C_c + \mu_\omega)\hat{X}_s + (k_s + k_c - k_l)\hat{X}_s + (k_l - k_c)\hat{X}_c = \hat{F}_D \quad (19)$$

Rewriting in the form $A\hat{X} = \hat{F}$,

$$A = \begin{bmatrix} 0 & -\omega^2 M_m & 0 \\ -\omega^2 M_m & -\omega^2(M_c - M_m) + k_c - k_l + i\omega C_c & -i\omega C_c + k_l - k_c \\ 0 & -i\omega C_c + k_l - k_c & -\omega^2(M_s + M_\infty + M_\omega) + i\omega(\mu_\omega + C_c) + k_s + k_c - k_l \end{bmatrix} \quad (20a)$$

$$\hat{X} = \begin{bmatrix} \hat{X}_m \\ \hat{X}_c \\ \hat{X}_s \end{bmatrix} \quad (20b)$$

$$\hat{F} = \begin{bmatrix} 0 \\ -\hat{T}_L \\ \hat{F}_D \end{bmatrix} \quad (20c)$$

In this case, D_{ij}, the elements of $adj(A)$, are

$$D_{12} = \omega^2 M_m[-\omega^2(M_s + M_\infty M_\omega) + i\omega(C_c + \mu_\omega) + k_s + k_c - k_l] \quad (21a)$$

$$D_{13} = \omega^2 M_m[-i\omega^2 C_c + k_l - k_c] \quad (21b)$$

$$D_{22} = 0 \quad (21c)$$

$$D_{23} = D_{32} = 0 \quad (21d)$$

$$D_{33} = -\omega^4 M_m^2 \quad (21e)$$

and

$$|A| = -\omega^4 M_m^2[-\omega^2(M_s + M_\infty + M_\omega) + i\omega(C_c + \mu_\omega) + k_s + k_c - k_l] \quad (21f)$$

$|A|$ is nonzero for $\omega > 0$. Eqns. (10 a-c), and (21 a-f) lead to the conclusion that provided f_m and f_c are chosen according to Eqns. (12) and (13),

$$\hat{X}_m = \frac{\hat{T}_L}{\omega^2 M_m} + \frac{[-i\omega C_c + (k_l - k_c)]\hat{F}_D}{\omega^2 M_m[-\omega^2(M_s + M_\infty + M_\omega) + i\omega(C_c + \mu_\omega) + k_s + k_c - k_l]} \quad (22a)$$

$$\hat{X}_c = 0 \quad (22b)$$

$$\hat{X}_s = \frac{\hat{F}_D}{-\omega^2(M_s + M_\infty + M_\omega) + i\omega(C_c + \mu_\omega) + k_s + k_c - k_l} \quad (22c)$$

Eqns. (22) show that the effect of the chosen control is to lock the motion $\hat{X}_c$ of block M_c to zero for $\omega > 0$. Force f_m brings about the actual cancellation of $\hat{X}_c$, while force f_c restricts the motion of M_m in the presence of f_m.

To summarize, for a passive implementation of the system there exists one frequency at which the oscillators M_m and M_c interact such that M_c remains stationary regardless of the magnitude of the vertical exciting forces F_D and T_L. Active control can extend this behavior to a range of frequencies as shown by Eqns. (22). An active system is expected to be tested on a 1/62.5th scale model of the Mighty Whale. It is expected to be used in both vertical and horizontal directions so as to utilize hull heave and surge respectively. The aim of those tests will be to study the extent of energy absorption improvement enabled by an active system. Prior to both active and passive experiments, however, it is necessary to consider an important question — how much energy is required to operate the above compensator system? Clearly, the gain in wave energy absorption must exceed the energy supplied to the compensator for the project to be of any value. This issue is addressed below.

3 Energy Requirement

At each frequency, the average power required to drive the masses M_m and M_c is given by,

$$P_{av} = P_{avm} + P_{avM} \tag{23}$$

where (*denotes complex conjugate)

$$P_{avm} = \frac{1}{2}\Re\{[-\omega^2 M_m + k_m] (\hat{X}_m - \hat{X}_c)^* i\omega(\hat{X}_m - \hat{X}_c)\} \tag{24}$$

and

$$P_{avM} = \frac{1}{2}\Re[k_l(\hat{X}_c - \hat{X}_s)^* i\omega(\hat{X}_c - \hat{X}_s)] \tag{25}$$

At $\omega \neq 0$, $\hat{X}_c = 0$ so that

$$P_{avm} = \frac{1}{2}\Re[i\omega(-\omega^2 M_m + k_m)\hat{X}_m^* \hat{X}_m] = 0 \tag{26}$$

$$P_{avM} = \frac{1}{2}\Re[i\omega k_l(\hat{X}_s^* \hat{X}_s)] = 0 \tag{27}$$

Frictional/heat losses in hardware (motors/hydraulic rams, couplings, etc.) are assumed to be small in the equations above. As shown, $P_{av} = 0$, $\omega > 0$, or control requires no net energy input, except what may be consumed as losses in hardware. However, if the zero-frequency instability apparent in Eqn. (21f) is prevented by allowing small oscillations at very low frequencies, then it would become necessary to supply some net average power to the system. It may be noted that the damping C_c causes some dissipation of the energy in hull oscillations. This is found to be, $P_{dis} = \frac{1}{2}\omega^2 C_c |\hat{X}_s|^2$ for $\hat{X}_c = 0$. We conclude that from an energy-economics point of view, the system being studied is suitable for further development. For completeness we add that the force requirement is high at low frequencies [Korde (1998)[7]] and good hardware design would be necessary at full scale.

4 Experiments

The aim of the present experiments is to test the passive concept (i.e. without control) on laboratory models in a 2-dimensional, small-scale tank. Figure 3 shows a schematic for the system without control. The passive system provides motion compensation of M_c at a single frequency, which can be calculated for chosen designs. The interest is in observing the possible increase in absorbed power, and in studying likely problems in the design of such systems. It is expected that this exercise will clarify problems that must be resolved before the tests with active control. Experiments in regular waves were carried out in the JAMSTEC 2-dimensional wave tank of dimensions 20m×0.5m×0.6m (water depth), with a single flap-type absorbing wavemaker. The tank allows tests on a 1/62.5 th scale model of the Mighty Whale prototype. Prior to these experiments, the concept was tested on a model of the axisymmetric floating OWC buoy mentioned above [Masuda, (1985)[3]]. The goal was to gain experience with the measurement system in the tank, and to gain quantitative understanding of the effect of the compensator on device dynamics. It was expected that this would help with the interpretation of results from the Mighty Whale tests.

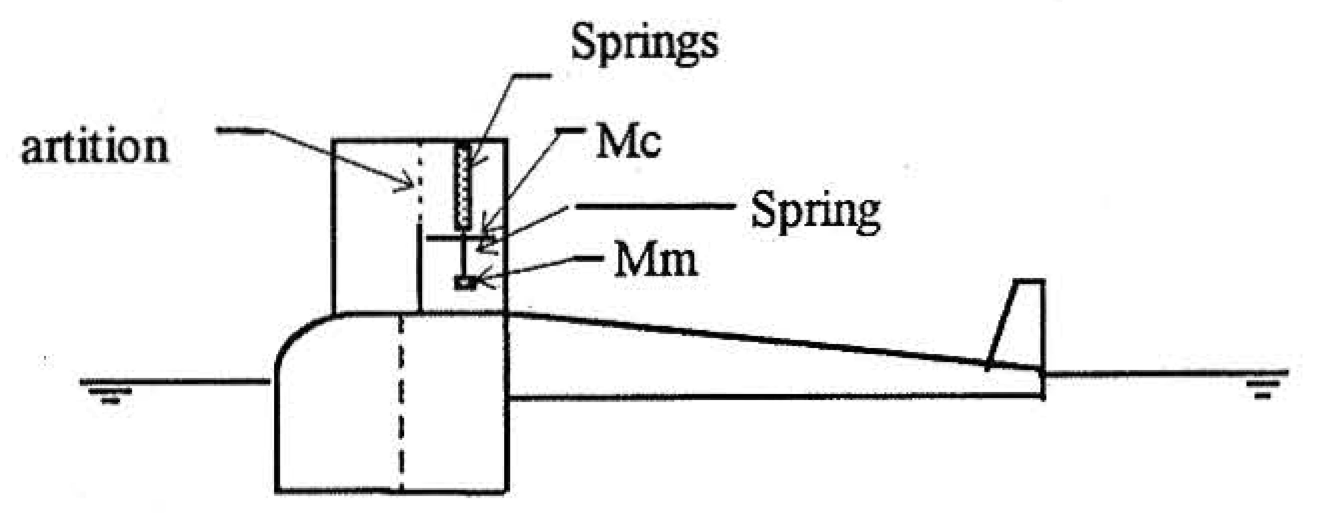

Fig. 3 Passive implementation of compensator on a 1/62.5 th scale model

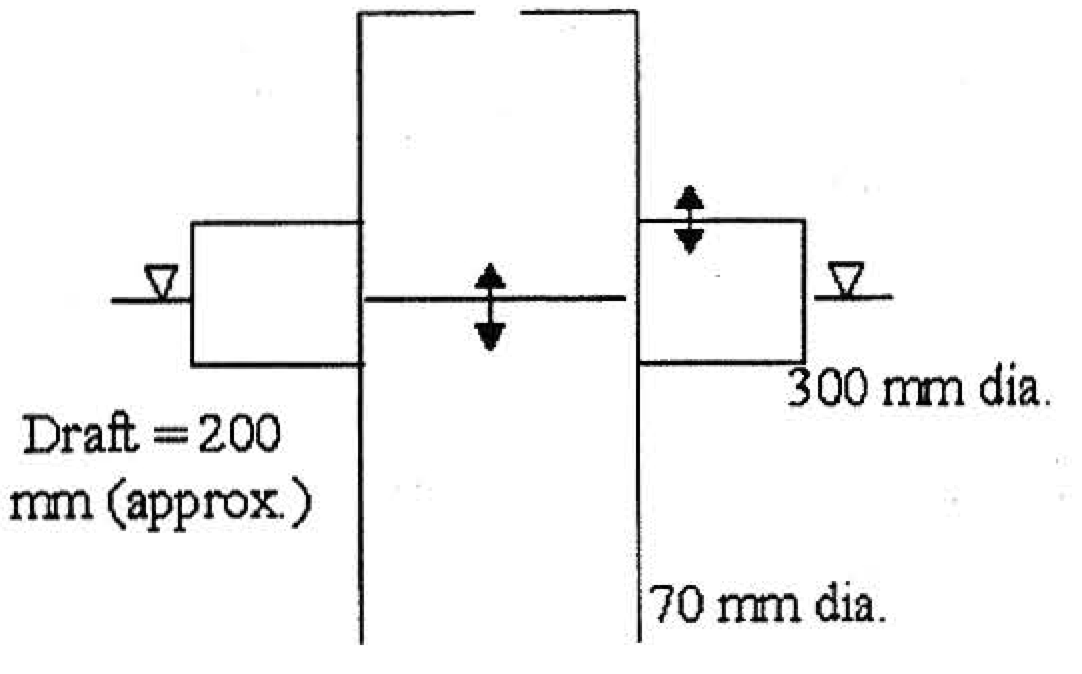

Fig. 4 Axisymmetric buoy used in these tests

4.1 Tests on axisymmetric buoy

Figure 4 shows a schematic of the buoy tested. Experiments were carried out over the wave-period range 0.48 s to 1.0 s at a wave height of 2cm. Mass M_c was suspended from the pipe cap by a spring, and mass M_m was suspended from M_c by another, weaker spring. Orifices were drilled into the cap and mass M_c to allow damping of air flow. The quantities measured were (i) wave profile in front of the model, (ii) wave profile behind the model, (iii) vertical acceleration of mass M_c, (iii) vertical acceleration of buoy, and (iv) pressure in the air chamber. Measurements were recorded digitally in a data logger, and analyzed on a Personal Computer. Both wave gages were moved through a distance of about 1 wavelength along the tank to resolve the incident, reflected, and transmitted wave components. Reflections from the beach back to the model were also computed. The device efficiency was inferred using energy balance (i.e. energy received by the device − energy rejected by the device = energy absorbed). Note that it is more common in wave energy literature to compute efficiency (capture factor) relative to energy incident on the width of the device, while the efficiency here is defined as the fraction absorbed from the total incident energy over the tank width. Vertical oscillations of M_c and the device were obtained by two successive numerical integrations of the corresponding acceleration records. The ratio of the two oscillation amplitudes at each frequency indicated how well the expected phenomenon worked.

Tests at each frequency were also carried out without the compensator system. In this case, the wave profiles (incident, reflected, transmitted, and reflected back from the beach) and chamber pressure were measured. Efficiency of the device in this situation was computed using the same method as above, and values in the two situations were compared at each frequency.

4.2 Mighty Whale model tests

The tests on the buoy provided familiarity with the tank testing procedures and the measurement system at JAMSTEC, and tests on the Mighty Whale model could be carried out with relative ease. The central OWC in the 1/62.5 th scale Mighty Whale model was modified (Figure 3) to allow the passive compensator system to be set up. Air-flow nozzles (orifices) on the port and starboard air chambers were sealed off, and only the central air chamber was used in these tests.

The mass M_c was suspended on 4 parallel springs within a superstructure (Figure 3) connecting the central air chamber with the buoyancy chamber behind it. Mass M_m was suspended on a single spring from mass M_c. The model was constrained from above to permit heave motion only. Orifices were drilled in appropriate locations on the superstructure to dissipate energy in the air flow. Note that the air-flow now results from both the water column oscillations and the relative oscillation between the hull and mass M_c. Experiments were carried out at 6 regular wave periods in the range 0.5-1.1 s at wave heights = 1cm and 2cm. Two situations were studied with the compensator: with the partition between OWC and mass M_c partially open, and (ii) with the partition fully closed. Two or more values of mass M_m were tried in each case. The quantities measured were: (i) wave profile in front of the model, (ii) wave profile behind the model, (iii) vertical oscillations of mass M_c, and (iv) vertical oscillations of the hull. The procedure for measurement, recording and analysis was identical to that for the buoy. The wave profile measurements were also carried out without the compensator in place, for comparison with the results with the compensator. Efficiency was defined as in the axisymmetric buoy case, and calculated in a similar manner.

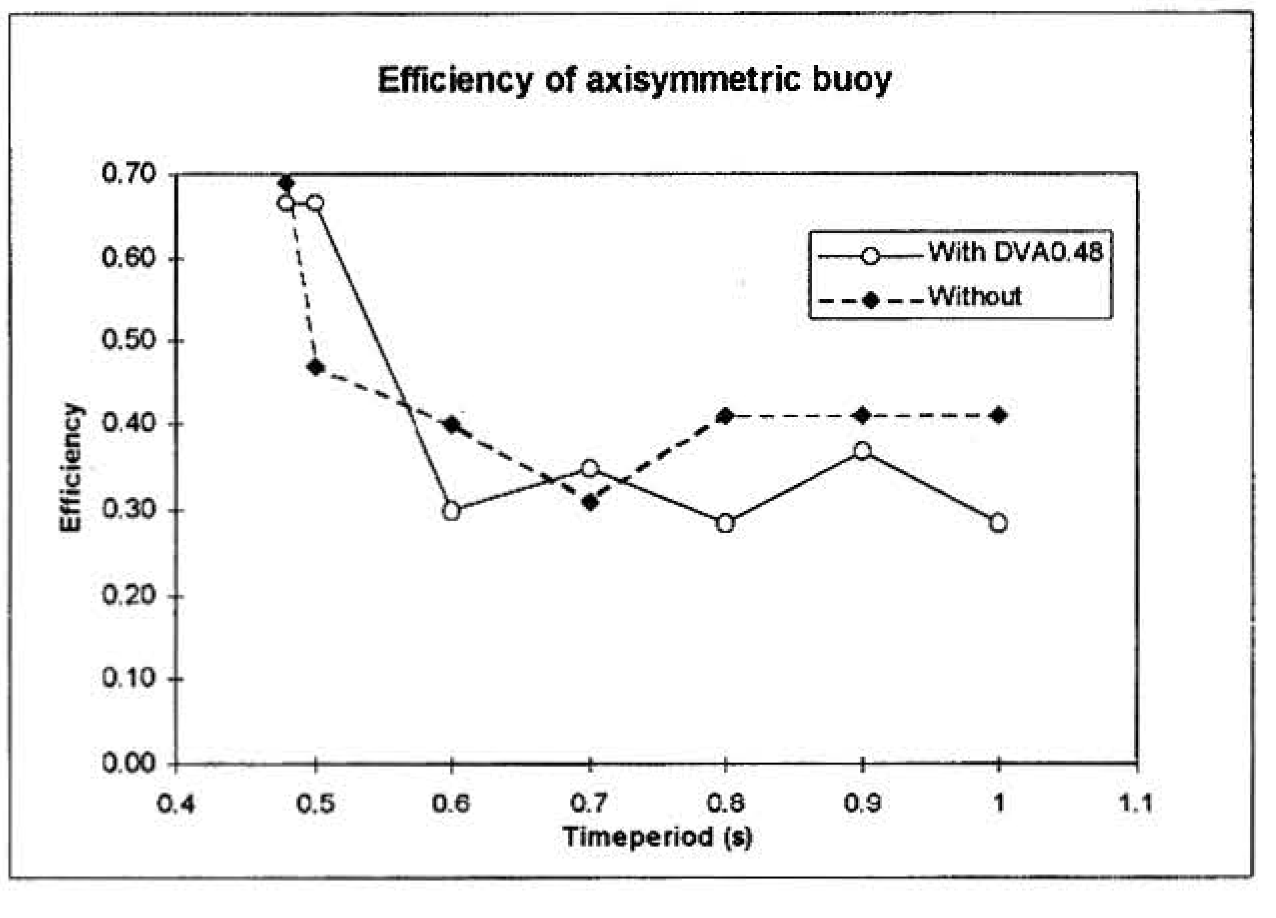

Fig. 5 Efficiency results for buoy (DVA 0.48 : Mass M_m tuned to 0.48 s)

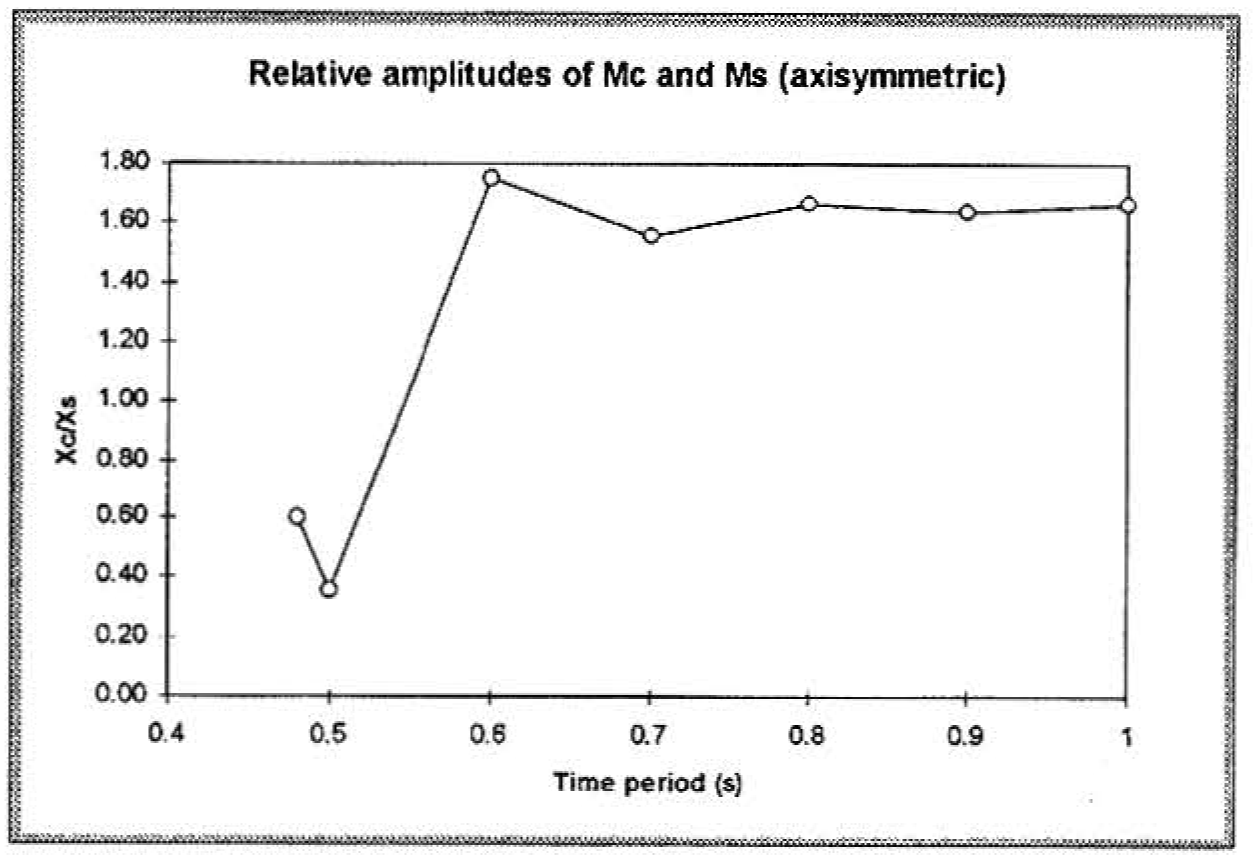

Fig. 6 Displacement of M_c for buoy (DVA 0.48 : Mass M_m tuned to ≈ 0.48 s)

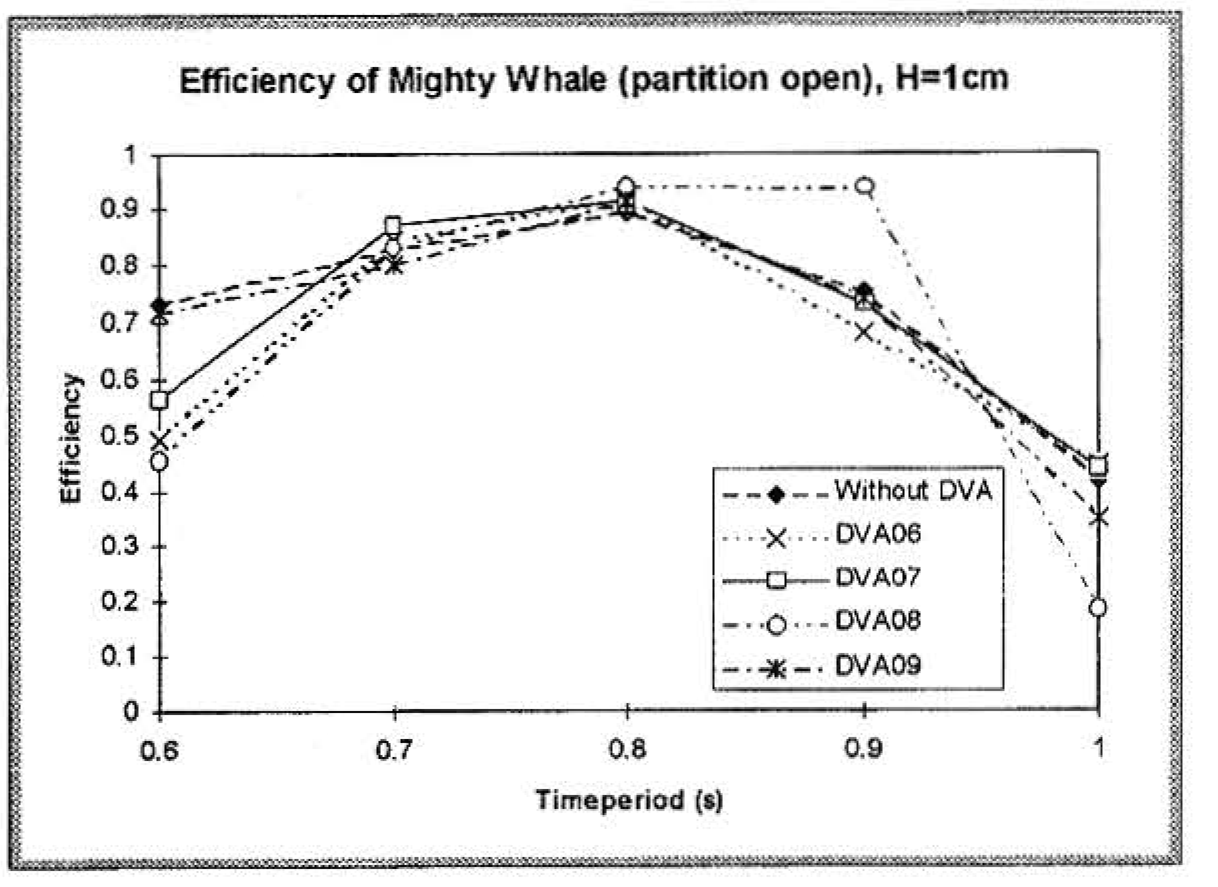

Fig. 7 Efficiency comparisons for Mighty Whale model; Wave height (H)= 1 cm (DVA NN : Mass M_m tuned to N. N s)

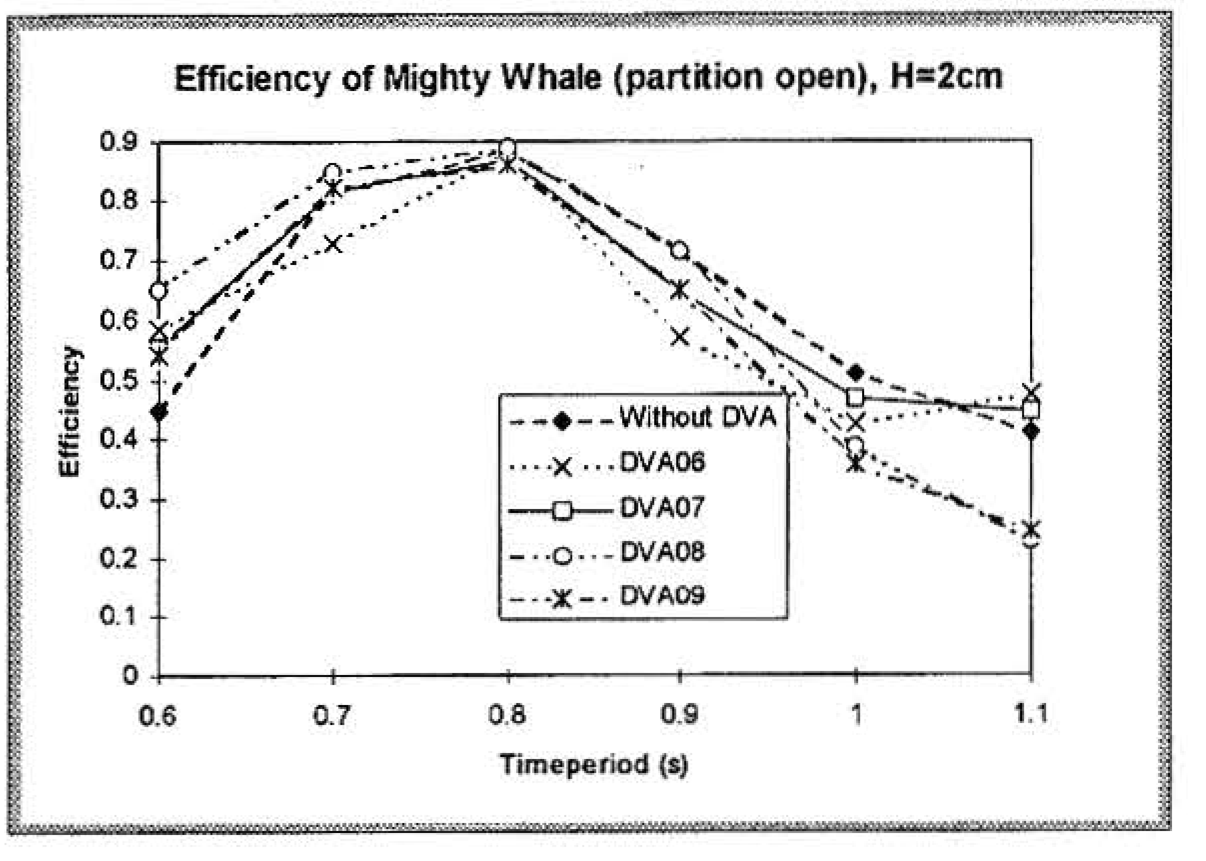

Fig. 8 Efficiency comparisons for Mighty Whale model; Wave height (H)= 2 cm (DVA NN : Mass M_m tuned to N. N s)

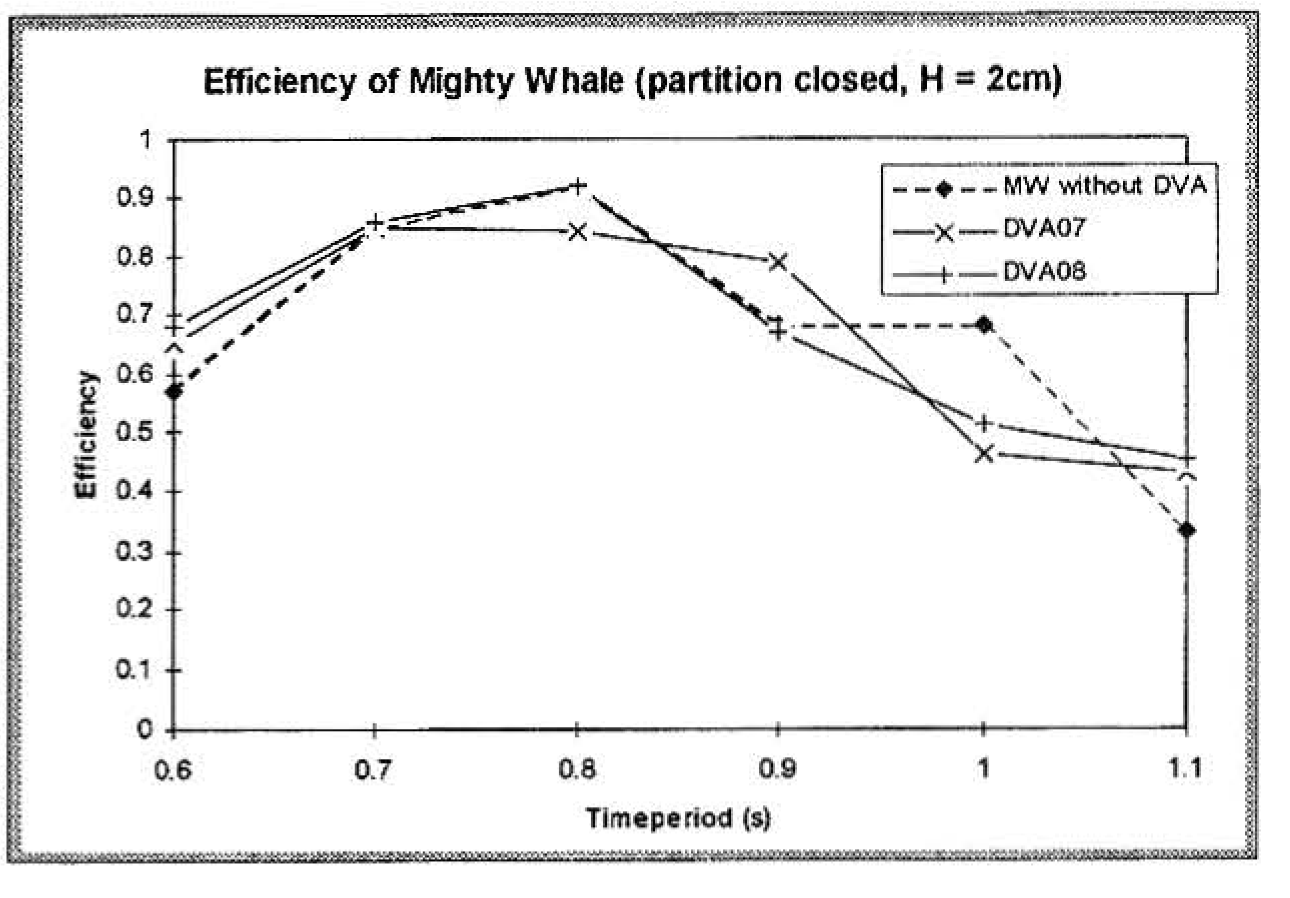

Fig. 9 Efficiency comparisons for Mighty Whale model with partition closed (DVA NN : Mass M_m tuned to N. N s)

5 Discussion of Experimental Results

5.1 Buoy Tests

Measurements above showed that the oscillations of M_c were negligible at a frequency very close to that calculated from theory. Efficiency results (Figure 5) revealed that the device absorbed substantially more energy at this frequency (2.0 Hz, corresponding to wave period = 0.5 s) than in the absence of the compensator. M_m was tuned to 0.48 s but greatest efficiency improvement was seen at 0.5 s. Further, the greatest reduction in X_c was observed at 0.5 s, and X_c was never actually driven to zero. This effect was likely a result of some nonlinearity in the spring k_m at large displacement of M_m. As anticipated, the oscillations of M_c were considerable at other frequencies, and this led to decreased energy absorption (Figure 6).

Increasing M_m and reducing k_m would have made the compensator frequency less than 2.0 Hz. However, this was not possible in the present tests, because it would have caused excessive initial stretching of spring k_m, for which there was no room in the present model. The tests were carried out several times to ensure repeatability of results presented here.

5.2 Mighty Whale Model

One disadvantage of the use of incident, reflected and transmitted wave profile measurements to compute efficiencies was the inability to account suitably for dissipation along tank walls, model edges, and gaps between model and tank walls. By default, these effects are seen in the graphs below as increased energy absorption than previously measured [Washio, *et al.* (1998)[5]] in all situations tested here. Nevertheless, the comparisons below are fair in that the same measurement procedure was used for every trial with and without the compensator. Again, repeated runs of the experiments established the repeatability of the results. The frequencies at which tests were carried out included the values calculated for all tested values of M_m. Efficiency improvement was observed with the compensator working, though this was not as high as expected. At the frequencies where M_c was expected to be stationary, it was found unavoidably to undergo small oscillations. This indicated that the calculated frequency of stationarity of M_c differed somewhat from the actual value.

It was observed further that the hull heave was reduced somewhat in the presence of the tested system. This effect combined with the small oscillations of M_c could have resulted in the efficiency improvement being no more than 10-15%. It was concluded that air flow across the considerable gap between M_c and the superstructure walls caused increased damping of the hull oscillations.

6 Conclusion

The purpose of this study was to investigate an actively controlled compensation system based on favorable interaction of coupled oscillators. At least in theory, the smaller oscillator can be controlled such that motion of a spring-supported block (platform) is effectively locked to zero over the frequency band covered by irregular ocean waves. Because the body whose motion is being controlled is not in contact with the water surface, the required control method is theoretically straightforward, and knowledge of current oscillations only is necessary.

It is evident from the experimental results on the Mighty Whale model that more careful design is necessary in order to reduce the air gap along the periphery of mass M_c. It was also observed that at frequencies where favorable interaction between M_m and M_c did not take place, the device efficiency frequently decreased. Since the device is to operate in irregular waves in practice, M_c should be made stationary over a wide frequency range. Active control appears necessary to achieve this, and must be implemented in practice. A system based on these findings is currently under construction, and is expected to be tested in the JAMSTEC two-dimensional wave tank.

While Eqns. (10) and (11) indicate isolation of block M_c from floating-body heave motion over a wide frequency range under the proposed control, these are based on assumption of linearity. It is possible that the large oscillations of M_m in the passive system led to spring nonlinearity, and the expected behavior was could not be produced exactly.

A potential application of the actively controlled motion compensator exists on drill ships, where it is particularly important to isolate the drill string/riser from the possibly large heave motions of ship hull.

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