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Semiactive vibration control for ship structures (I)

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ABSTRACT

The reduction of main hull vibration of ships by a semiactive dynamic absorber is investigated. The dynamic absorber system includes a moving mass, support springs, dynamic dampers and a control system. Only small electrical power supply is needed for the valve control of the damper and the operation of the control system. In this investigation, the operation theory of the dynamic absorber is first described. Then, a suboptimal control law for the absorber is derived based on the optimal theory. The numerical simulation results show that the dynamic absorber achieves better efficiency in hull vibration reduction than the passive type absorber during critical periodical excitation from the propeller. The vibration caused by multi-frequency excitation can also be suppressed by the dynamic absorber.

1. INTRODUCTION

Since the high screw propeller and long stroke diesel engine have become more widely used, widespread problems relating to main hull vibration deserve greater attention. Its presence can affect the comfort of passengers and crew, damage the structure and impair the fighting efficiency of warships. The passive type of vibration absorber was first introduced for the vibration reduction of ship hulls in the nineteenth century [1]. Until now, many types of the passive absorber have been installed on the ships [2-4]. However, the performance of the passive absorber was low for the hull with multiple vibration frequency, which was generally due to propeller and engine excitation.

Recently, an active control system for the reduction of multi-mode vibration due to the engine and propeller excitation was introduced [6,7]. Since high power supply and high performance of actuator devices and controller are required, high initial and maintenance cost is the primary disadvantage of the active absorber system.

A semiactive type dynamic absorber is proposed for the hull vibration reduction of ships in this paper. The semiactive type absorber system represents a compromise between the passive and the active type absorber [8,9]. In this system, only a small electrical power supply is needed for the valve control of the dynamic damper and the operation of the control system. In this investigation, the operation theory of the semiactive absorber and the dynamic equations of the hull with the absorber is first described. Then, a suboptimal control law for the dynamic absorber will be derived basing on the optimal theory. Finally, an oil tanker is investigated in the numerical example to understand the performance of this scheme.

2 THEORY

A free-free beam model was considered for the vibration analysis of the ship hull girder. The coordinate system used for the analysis is shown in Fig. 1. In this paper, the hull is first idealized by an appropriate number of uniform cross-section segments. The vertical and shear deflection of the hull segment are approximately expressed in terms of these generalized coordinates by means of an appropriate set of assumed shape functions. By the principle of virtual work, the equations of motion of each hull segment i can be expressed as a matrix form [12,13]

$$
\begin{aligned}
&M_h^{(i)} \frac{d^2 W_h^{(i)}(t)}{dt} + C_h^{(i)} \frac{dW_h^{(i)}(t)}{dt} + K_h^{(i)} W_h^{(i)}(t) \\
&= F_{ext}^{(i)}(t) - ¢_k^{(i)} \cdot \left(\frac{1}{l_k} (a - \sum_{j=1}^{k-1} l_j) \right) f_a(t)
\end{aligned}
\tag{1}
$$

where $M_h^{(i)}, C_h^{(i)}$ and $K_h^{(i)}$ are the mass, damping and stiffness matrices of the hull element i; $W_h^{(i)}(t)$ is the vertical translation and shear slope of the nodes; $F_{ext}^{(i)}(t)$ is the generalized load vector; $\cdot$ is the shape function vector of the segment; l_k is the length of the segment k, where the absorber is located; $¢_k^{(i)}$ is the Kronecker delta function .

The model of a dynamic absorber system under consideration for hull vibration reduction is shown in Fig. 2 where m_a is the absorber mass attached to the main hull by a linear spring with constant stiffness coefficient k_a and a dynamic damper. The absorber mass is driven by the spring and the damper given by

$$m_a \frac{d^2 \overline{w}_a(t)}{dt^2} + k_a \tilde{w}_d(t) = f_{ad}(t) \qquad (2)$$

where $\overline{w}_a(t)$ is the absolute upward displacement of the moving mass with respect to the initial coordinate system. $\tilde{w}_d(t)$ is the relative upward displacement of the absorber mass to the hull. $f_{ad}(t)$ is the acting force of the damper, called damper force. Simultaneously, an equivalent reaction force induced by the spring and damper, called absorber force, is acting on the main hull. Basing on the equations of (4) and (5), the dynamics of the hull and absorber can be combined as a set of second order differential equations

$$\begin{aligned} & M \frac{d^2 W(t)}{dt} + C \frac{dW(t)}{dt} + KW(t) \\ & = \begin{Bmatrix} k_a \overline{w}(a,t) \\ F_{ext}(t) - k_a \overline{w}(a,t)\Delta_k \end{Bmatrix} - B_a f_{ad}(t) \end{aligned} \qquad (3)$$

where a_k is the distribution function,

$$W = \begin{Bmatrix} \tilde{w}_a \\ W_h \end{Bmatrix}, \quad M = \begin{bmatrix} m_a & 0 \\ 0 & M_h \end{bmatrix}, \quad C = \begin{bmatrix} 0 & 0 \\ 0 & C_h \end{bmatrix}$$

$$K = \begin{bmatrix} k_a & -k_a \Delta_k^T \\ -k_a \Delta_k & K_h + k_a \Delta_k \Delta_k^T \end{bmatrix}, \quad B_a = \begin{bmatrix} 1 & 0 \\ 0 & -\Delta_k \end{bmatrix}$$

$\overline{w}(a,t)$ is the absolute upward displacement of the hull at the location of the absorber with respect to the initial coordinate system.

In this investigation, we consider the dynamic absorber, whose damping coefficient is designed to be adjustable. Then, the damper force is function of the velocity difference between the main hull and absorber mass as well as the controlled damping coefficient. So, the desired force can be obtained if the damping coefficient is controlled by an appropriate input. However, since external power is not offered, only the control force generated by the absorber is feasible. So the damper force should be constrained under the following conditions

$$f_{ad}(t) \frac{d\tilde{w}_d(t)}{dt} \le 0 \qquad (4)$$

The determination of $f_{ad}(t)$ is the kernel of this design. In this study, the turning of the damper is determined using the linear quadratic control scheme. In order to obtain the control law by the control scheme, the governing equations are rewritten as first order matrix differential equations:

$$\frac{dZ(t)}{dt} = A_o Z(t) - B_o f_{ad}(t) + \begin{Bmatrix} M^{-1} F_e(t) \\ 0 \end{Bmatrix} \qquad (5)$$

where

$$Z(t) = \begin{Bmatrix} \dfrac{dW(t)}{dt} \\ W(t) \end{Bmatrix}, \quad A_o = \begin{bmatrix} -M^{-1}C & -M^{-1}K \\ I & 0 \end{bmatrix}$$

$$B_o = \begin{bmatrix} M^{-1} B_a \\ 0 \end{bmatrix}, \quad F_e = \begin{Bmatrix} k_a \overline{w}(a,t) \\ F_{ext}(t) - k_a \overline{w}(a,t)\Delta_k \end{Bmatrix}$$

I is the identity matrix.

A performance index J is defined to describe the total performance of the vibration reduction and power requirement such as

$$J = E\left(\int_0^\infty \left(Z^T(t) Q Z(t) + r f_{ad}^2(t) \right) dt \right) \qquad (6)$$

where Q is a real symmetric positive semidefinite matrix, and r is a positive value. After these weights are given, we can choose an optimal control signal so that the value of the cost function is minimized. We assume that the external excitation load is a white, Gaussian, zero mean. In addition, the initial states of the hull and actuator motions are assumed to be random variables, which are Gaussian and independent of the loading. Based on the method of calculus of variation and the stochastic theory, this problem can be solved by analytical operation [14], in which the optimal damping force can be obtained and given by

$$f_{ad}(t) = r^{-1} B_o^T P Z(t) \qquad (7)$$

where P satisfies the matrix Riccati equation,

$$A_o^T P + P A_o - r^{-1} P B_o B_o^T P + Q = 0 \qquad (8)$$

Since the length of the hull is very long and the number of the nodes is enormous, the measurement system would be very complex and unreliable. To overcome this problem, the Kalman filter can be used to estimate these states based on the motion measured from the actuator of the absorber and the hull [8]. In addition, This filter may also have the property of noise rejection for measured signal. The flow chart of the dynamic absorber system with the estimator is shown in Fig. 3. If a viscous damper is used in this design, the damping coefficient of this design can be obtained

$$c_d(t) = \frac{r^{-1} B_o{}^T P\ Z(t)\left(1 - sign\left(r^{-1} B_o{}^T P\ Z(t) \frac{d\tilde{w}_d(t)}{dt}\right)\right)}{2\frac{d\tilde{w}_d(t)}{dt}} \quad (9)$$

where

$$sign(u) = \begin{cases} 1 & \text{for} \quad u > 0 \\ -1 & \text{for} \quad u \le 0 \end{cases} \quad (10)$$

3 NUMERICAL EXAMPLES

An oil tanker having the principle parameter list in Table 1 is considered for numerical analysis. The natural frequencies with respect to the vertical two nodes, three nodes and four nodes vibration mode of the hull girder are 63.5cpm, 136.5cpm, and 766.3cpm. The relative damping ratio to these modes is 0.34%, 0.56% and 1.1%. Three types of configurations, the original type (the ship hull without any absorber), the dynamic type (the ship hull with the dynamic absorber) and the passive type (the ship hull with the passive absorber), are considered for comparing the performance of the dynamic absorber to others. The moving mass and spring of the passive are the same size used in the dynamic absorber. The moving mass, spring constant and the damping ratio of the support are 200 ton, 187.9 ton/m and 5%.

In the first case, a sinusoidal excitation with an amplitude of 10 tons and frequency equal to the natural frequency of two node mode, excitation I, are considered. Fig. 4 represents the stern displacement response of the hull with the dynamic or the passive type absorber. The fluctuation of the stern diminishes from 88mm to 3mm pp (peak to peak) within 30sec in the semiactive type. The stern acceleration is also reduced from 81gal to 8 gal pp in the same period shown in Fig. 5. The decay rate of the oscillation of the stern in the semiactive type is higher than that of the passive type. Similar effect also appear in the midsection response, see Fig. 6. Fig. 7 shows that the efficiency of both absorbers in this excitation condition is not significantly different. The movement of the moving mass of both absorbers is very close. From Fig. 8, we find that the damper force generated by the dynamic absorber is very small compared to the passive one in most time.

In the second simulation case, the frequency of the excitation force equal to the natural frequency of three node mode, excitation II, is considered. In this case, the excitation frequency is different from the natural frequency of the absorber. Figs. 9 and 10 show the acceleration response of the stern and the quarter length before the stern of those four types of configuration. The hull vibration can not be reduced by the passive absorber but can be suppressed effectively by the dynamic one. Fig. 11 show that the damper force generated by the dynamic absorber is much larger than the damper force generated by the passive absorber during the first 5 seconds. The displacement of the actuator in the dynamic absorber is smaller than that of the displacement in the passive one described in Fig. 12.

4 CONCLUSIONS

A semiactive type absorber for vertical vibration reduction of the hull girder has been developed in this paper. By this system, only a small electrical power source is used for the valve control of the damper. Huge power supply like that used in the active type absorber is not needed. Based on the dynamic characteristics of the dynamic absorber, a semi-optimal control law is then derived. From the numerical analysis, we found the dynamic absorber has the same performance as the passive one when the excitation frequency matches the natural frequency of the absorber. If the excitation frequency is different from the natural frequency of the absorber, the efficiency of the passive absorber will drop. However, the dynamic absorber also has good performance for the vibration reduction when the excitation frequency is variant.

ACKNOWLEDGMENTS

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TABLES AND FIGURES

Table 1 Principle parameters of the sample ship

Length overall	245.0 m
Breadth	32.0 m
Draft	14.8 m
Displacement	87,000 ton

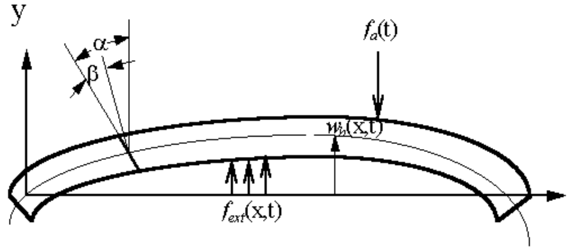

Fig. 1 Dynamic structure model of the ship hull girder

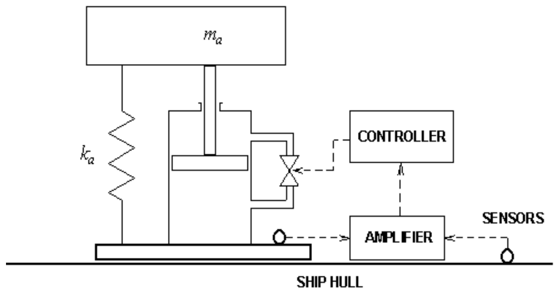

Fig. 2 Configulation of the dynamic absorber system

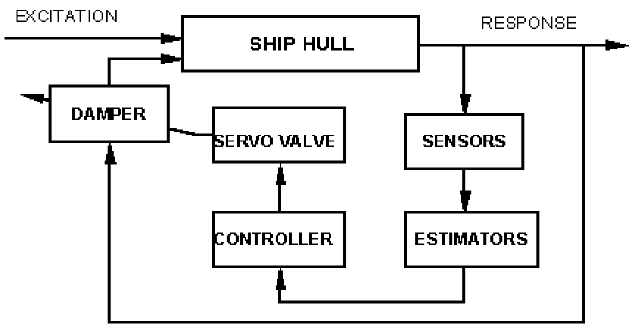

Fig. 3 Flow chart of the dynamic absorber system

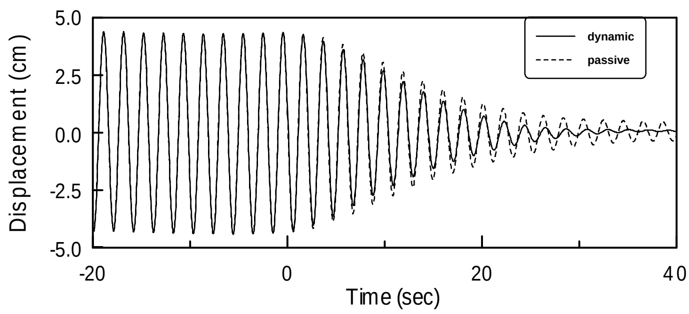

Fig. 4 Displacement response at bow for excitation I

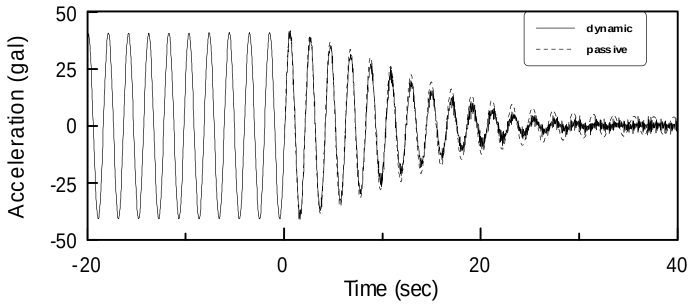

Fig. 5 Acceleration response at bow for excitation I

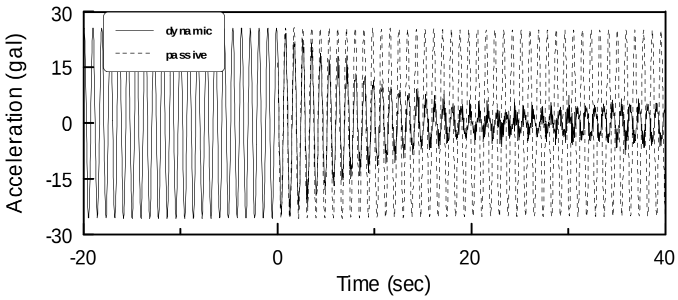

Fig. 9 Acceleration response at bow for excitation II

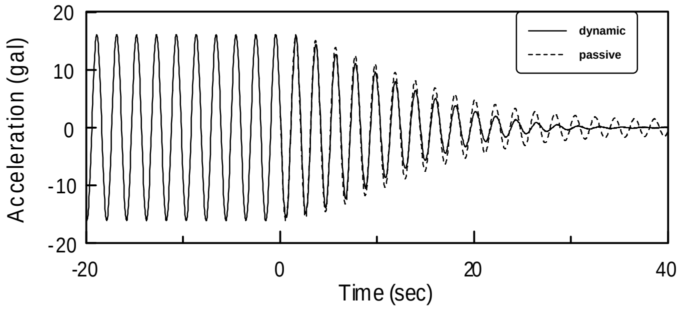

Fig. 6 Acceleration response at midship for excitation I

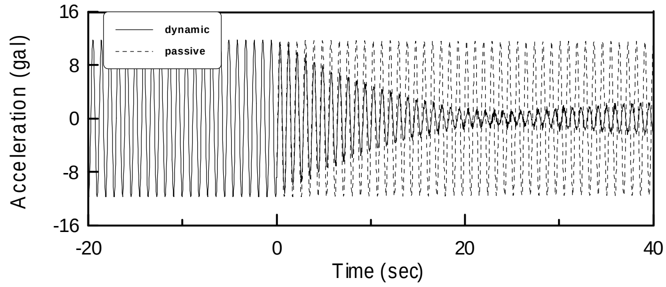

Fig. 10 Acceleration response at quarter length before midship for excitation II

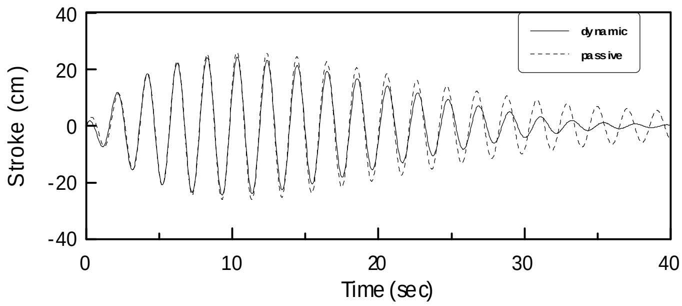

Fig. 7 Displacenment of the actuator for excitation I

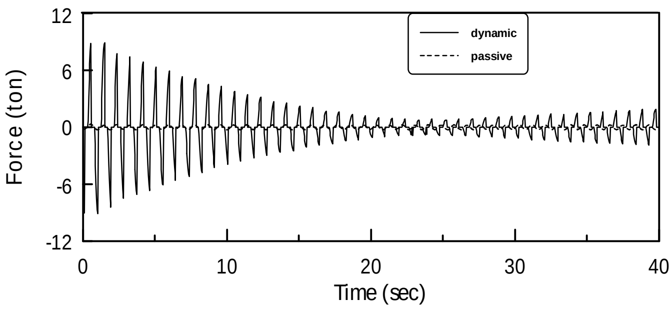

Fig. 11 Displacenment of the actuator for excitation II

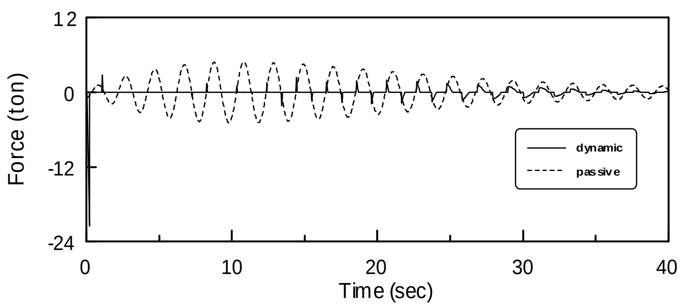

Fig. 8 Force generated by the absorber for excitation I

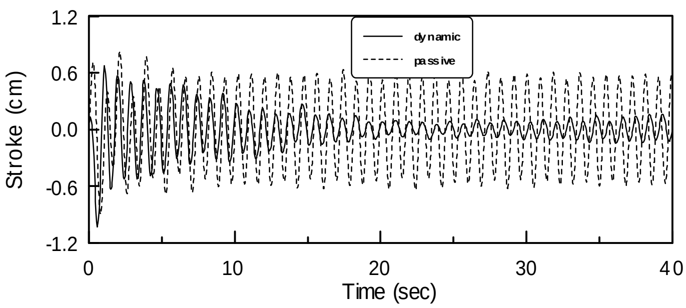

Fig. 12 Force generated by the absorber for excitation II