Design of Bow Absorber for Vibration Reduction of Hull Girders

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Summary

An active bow absorber for vibration reduction of hull girders is presented. In this investigation, the chain locker of ships is designed as the tuned mass of the absorber, which is driven by a hydraulic system. Since the chain locker is very heavy, the absorber will have a high efficiency for vibration reduction. Including the dynamics of the hydraulic system, the control law of the absorber is derived basing on the optimal control and optimum estimation theory. Finally, frequency and time response are considered for a bulk carrier to understand the feasibility of this design. From the simulations, we find the bow absorber is highly effective for vibration reduction of hull girder due to main engine excitation, especially for the lower frequency mode, such as v-2, v-3 and v-4 modes. The required hydraulic force and the displacement of the actuator for reducing the resonance exciting are reasonable. The results also show the effect of vibration suppression by the bow absorber is similar to that by the stern absorber even for the excitation in stern.

1. Introduction

Since hull vibration affects the comfort of passengers and crew and the failure of structures, it has been an interesting subject to shipbuilders and marine engineers for a long time. In 1884, Schlick first discovered that hull vibration will become excessive when the frequency of engine's revolution meets the resonance frequency of the the hull[1]. He also found the severe hull vibration could be avoided if the frequency of any exciting sources did not close the resonance frequency of the hull. So, it becomes very important to predict the resonance frequency of the hull in the design. Until now, several empirical formulas and calculation methods have been developed[2),3),4)] but the problem of hull vibration has still not been solved completely. The dominant reason is that to accurately predict the resonance frequency is difficult because of the complexity of ship structure. In addition, the hull resonance frequency is variant due to the different load conditions. Therefore some passive absorbers, such as tuned mass type[5),6)] and tuned liquid type [7),8),9)] and some active absorbers [10)] have been developed to reduce the peak level of response at dominant resonance frequency.

Recently, a wideband absorber was designed for reducing multi-mode vibration of hull girders by the authors. The results show this scheme does offer a high efficiency of vibration reduction due to the main engine excitation[11)]. In the system, the smaller moving mass the absorber has, the larger capacity the hydraulic system should have. So, we can decrease the size of the hydraulic power system for the absorber by increasing the weight of the moving mass. However, this additional weight will become a significant drawback if the moving mass of the absorber is very heavy. In this paper, the chain locker is designed as the moving mass of the absorber sysem. Since the chain locker is a requisite but also very heavy equipment on ships, the capacity of the hydraulic power can be reduced without additional moving mass. If the absorber is shifted to bow, the performance of the vibration reduction by this design will be different from that of the absorber in the stern, in which most of the excitation sources are located. Finally, a bulk carrier is considered as a numerical example to understand the performance of this design.

2. Methods

2.1 Dynamics of the Bow Absorber

An active absorber with chain locker as the moving mass is designed for vibration reduction of the hull girder. The chain locker is separated from the main structure of the ship and supported by some springs shown in Fig. 1. A hydraulic actuator is fixed on the main structure under the chain locker to generate forces underneath the chain locker. When the chain locker is

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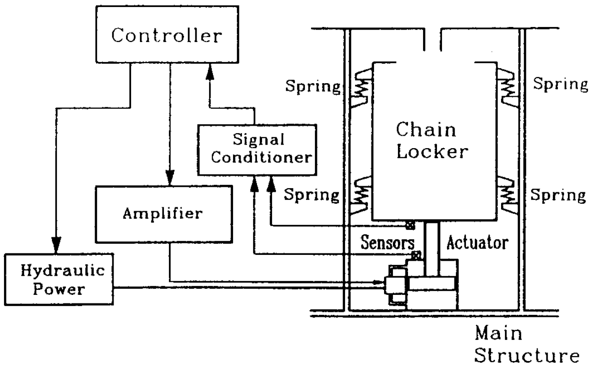

Fig. 1 The Configuration of the Bow Absorber System

moved by the actuator, the active force f_a will act on the hull girder given by

$$f_a = m_1 \frac{d^2 y_p}{dt^2} \tag{1}$$

where m_1 is the mass of the chain locker, y_p is the displacement of the chain loker. Due to the constrain of the springs, the relationship between the active force and the output force from the actuator, f_h, is

$$f_a = f_h - k_1 y_d \tag{2}$$

where k_1 is the total stiffness of those springs, y_d is the displacement of the chain locker relative to the cylinder, which is fixed on the hull.

If the active force is adjusted to be out of phase in respect to the motion of the hull girder, the vibration of the hull girder may be reduced by this force. The movement of the actuator is controlled by a servovalve, which is operated by a suitable mommand calculated by the controller according to the motion of the hull.

By the dynamic equation of the valve orifices[12], the load flow in the vicinity of the operating point depends on the displacement of the spool and the load pressure difference, expressed as

$$Q_d = k_q x_v - k_c P_d \tag{3}$$

where Q_d is the mean value of the forward and return flows, x_v is the displacement of the spool, P_d is the load pressure difference, k_q and k_c are the flow gain and the flow-pressure coefficients of the valve. Although both coefficients are variable and dependent on the displacement of the spool and the load pressure difference, a linearization approach at the operation point is usually applied for design.

The movement of the spool is driven by a coil and an amplifier according to the control signal. It assumes that the displacement of the spool is proportional to the control signal described as

$$x_v = k_s v_c \tag{4}$$

where k_s is the gain of the servovalve.

If the compressibility of the fluid and the leakage between the cylinder and the piston are considered, the relationship between the flow rate and the hydraulic pressure of the forward and the return chambers can be obtained by a continuity equation[13], such as

$$Q_d - c_k P_d = A_p \frac{dy_d}{dt} + \frac{V_t}{4\beta_e}\frac{dP_d}{dt} \tag{5}$$

where c_k is the cross-port leakage coefficient, A_p is the area of the piston, V_t is the total volume of the forward and the return chambers, and β_e is the effective bulk modulus of the oil.

If the resistant force between the piston and the cylinder is considered as a linear viscous friction, the force output from the piston rod leads to

$$f_h = P_d A_p - c_d \frac{dy_d}{dt} \tag{6}$$

where c_d is the viscous friction coefficient.

From Eq(1) to Eq(6), the dynamic equation for the active absorber can be represented by 1st order differential equation sets

$$\frac{dZ_1}{dt} = A_{11} Z_1 + B_{11} v_c + B_{12} \frac{dy_c}{dt} \tag{7}$$

where

$$Z_1 = \begin{Bmatrix} \dfrac{dy_p}{dt} \\ y_d \\ P_d \end{Bmatrix} \tag{8}$$

$$A_{11} = \begin{bmatrix} -\dfrac{c_d}{m_1} & -\dfrac{k_1}{m_1} & \dfrac{A_d}{m_1} \\ 1 & 0 & 0 \\ -\dfrac{4\beta_e A_d}{V_t} & 0 & -\dfrac{4\beta_e (k_c + c_k)}{V_t} \end{bmatrix} \tag{9}$$

$$B_{11} = \begin{bmatrix} 0 & 0 & \dfrac{4\beta_e k_q k_s}{V_t} \end{bmatrix}^T \tag{10}$$

$$B_{12} = \begin{bmatrix} \dfrac{c_d}{m_1} & -1 & \dfrac{4\beta_e A_d}{V_t} \end{bmatrix}^T \tag{11}$$

2.2 Dynamic Equation of the Ship Hull

The ship hull girder is idealized by a non-uniform simple beam. For simplification, the buoyancy and hydrodynamic viscous force are considered as distributed spring and damping; the structural damping is assumed to be distributed Rayleigh damping. Then, the hull girder can be modeled as a non-uniform cross-section Timoshenko beam. The schematic diagram is shown in Fig. 2. The equation of motion of the hull is then expressed as[14),15)]

$$m \frac{\partial^2 w(x,t)}{\partial t^2} + c_e \frac{\partial w(x,t)}{\partial t} + k_e w(x,t)$$

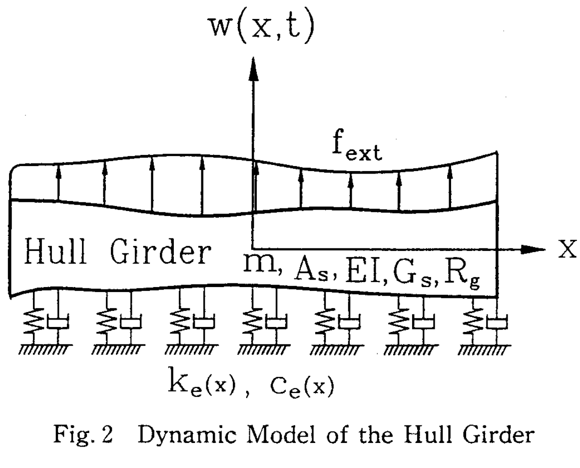

Fig. 2 Dynamic Model of the Hull Girder

$$+EI\left(c_s\frac{\partial^5 w(x,t)}{\partial x^4\partial t}+\frac{\partial^4 w(x,t)}{\partial x^4}\right)$$
$$+\frac{m^2R_g^2}{A_sG_s}\frac{\partial^2 w(x,t)}{\partial t^2}-\left(mR_g^2+\frac{EIm}{A_sG_s}\right)\frac{\partial^4 w(x,t)}{\partial x^2\partial t^2}$$
$$=f_{ext}(x,t) \tag{12}$$

in which m is the summation of the light ship weight, the cargo weight, and the added mass of the ship; c_e is the distributed hydrodynamic damping; k_e is the distributed hydrodynamic spring; c_s is the structural damping coefficient; EI is the flexural rigidity; R_g is the square root of gyration; A_s is the effective shear area; G_s is the shear modulus; w is the displacement of the hull; f_{ext} is the summation all of exciting forces, which include the active force load f_{act} and exciting loads f_e.

According to the concept of the finite element method, the motion distribution of each sub-beam is approximately expressed by the motion of the end points and some shape functions. If vertical translation and slope are chosen as the degree of freedom for each node, there will be $2n$ dofs in the discrete hull model when n-1 sections are taken. By the principle of virtual work[16], the governing equation becomes

$$M\frac{d^2W}{dt^2}+B\frac{dW}{dt}+KW=F_e+F_a \tag{13}$$

where M is the mass matrix; W is the vertical translation and slope of every node; B is the camping matrix; K is the stiffness matrix; F_e is the generalized exciting load vector; and F_a is the generalized active force vector.

2.3 Dynamics of the Total System

Since the absorber is fixed on the ship hull, the cylinder's motion follows the hull vibration. For the location of the absorber $x=\alpha$, the relationship between the active force load of the hull girder and the active force of the absorber is given by

$$f_{act}(x,t)=-f_a(t)\delta(x-\alpha) \tag{14}$$

where δ is the Dirac delta function. If the active absorber is chosen as located at the node k of a discretized model of the hull girder, then

$$F_a=\bar{I}_{2k}f_a \tag{15}$$

where $\bar{I}_{2k}$ is an $2n\times 1$ matrix with a zero value except when element $2k$ is 1.

From Eq(14) and Eq(15), the dynamic equations of the motion of the absorber and the the hull girder can be combined as

$$\frac{dZ}{dt}=A_0Z+B_{01}v_c+B_{02}F_e \tag{16}$$

where

$$Z=\begin{bmatrix} Z_1 \\ W \\ \dfrac{dW}{dt}\end{bmatrix} \tag{17}$$

$$A_0=\begin{bmatrix} A_{11} & O_{3\times 2n} & B_{12}\bar{I}_{2k}^T \\ O_{2n\times 3} & O_{2n\times 2n} & \hat{I}_{2n} \\ -M^{-1}\bar{I}_{2k}B_{21} & -M^{-1}K & -M^{-1}C-c_dM^{-1}\bar{I}_{2k}\bar{I}_{2k}^T \end{bmatrix} \tag{18}$$

$$B_{01}=\begin{bmatrix} B_{11} \\ O_{4n\times 1}\end{bmatrix} \tag{19}$$

$$B_{02}=\begin{bmatrix} O_{4\times 2n} \\ M^{-1}\end{bmatrix} \tag{20}$$

$$B_{21}=[-c_d \quad -k_1 \quad A_d] \tag{21}$$

$O_{i\times j}$ is $i\times j$ zero matrix and $\hat{I}_i$ is an $i\times i$ identity matrix.

2.4 Controller Design

In order to suppress the vibration of the hull effectively, the value v_c must be cautiously adjusted to be dependent on the vibration. Therefore, the objective of this design is to find a suitable turning law for the controller, so that the vibration of the hull can be reduced to the desired level and the operation of the actuator remains feasible. A higher efficiency of vibration reduction is needed, the greater power from the actuator is required. Then we can define a performance index to express the total performance of the vibration reduction and power requirement such as

$$J=E\left(\int_0^\infty\left(\sum_{i=1}^{4n+3}(q_iz_i^2)+(rv_c^2)\right)dt\right) \tag{22}$$

where J is the performance index, z_i is the i th element of Z. In general, q_i is selected as a nonnegative value, and r as 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[17], in which the optimal input command $v_{c_{opt}}(t)$ can be obtained and given by

$$v_{c_{opt}}(t)=-\frac{1}{r}B_{01}^TPZ(t) \tag{23}$$

where P satisfies the matrix Riccati equation,

$$PA_0+A_0^TP-\frac{1}{r}PB_{01}B_{01}^TP+Q=0 \tag{24}$$

and Q is a diagonal matrix with the diagonal element q_i.

From the control law, we know the input command of the controller is a function of all of the states, which include the displacement of the actuator, the hydraulic pressure, and the displacement and velocity of the nodes of the hull girder. In general, 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, an estimator is introduced to estimate these states based on the motion measured from the actuator and bow. In practice, some noise will be included in the measured signal, thus

$$Y(t)=CZ(t)+v(t) \tag{25}$$

where Y is the measured signal, v is the measurement noise. Suppose that the Kalman filter is chosen as the estimator. The structure of the estimator is an analogy to that of the total system, such as

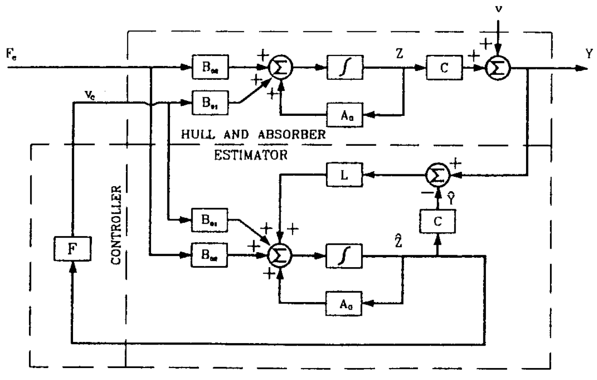

Fig. 3 The Block Diagram of the Total System with the Estimator

$$\frac{d\hat{Z}}{dt}=A_0\hat{Z}+B_{01}v_c+B_{02}F_e+L(Y-\hat{Y}) \qquad (26)$$

where $\hat{Z}$ is the estimated state of the state Z, L is the observer gain. The last item of Eq(26) is the correction item, which is used to compare the measured signal Y and output of estimator $\hat{Y}$ for the estimator modification shown in Fig. 3. The main object of the estimator design is to find an observer gain, for which the difference between the estimated value and actual value is minimum. Thus we consider the criterion

$$J_e=E\{\tilde{Z}(t)U(t)\tilde{Z}(t)\} \qquad (27)$$

where

$$\tilde{Z}(t)=Z(t)-\hat{Z}(t) \qquad (28)$$

is the observation error, U is a symmetric positive matrix. Assume that both F_e and v are stationary white, Gaussian, uncorrelated, and have zero mean. The initial state $Z(t_0)$ is also a Gaussian random variable and uncorrelated with F_e and $v(t)$. According to the stochastic estimation theory[17], the optimal observer gain can be obtained for the minimization of the criterion J_e.

$$L_{opt}=-SC^TW_e^{-1} \qquad (29)$$

Where W_e is the intensity of F_e, S is the variance matrix of $\tilde{Z}(t)$ satisfying the Riccati equation

$$A_0S+SA_0^T-SC^TW_e^{-1}CS+V=O \qquad (30)$$

where V is the intensity of v.

Upon including this estimator, the optimal control law may be rewritten as

$$u_{c_{opt}}(t)=-\frac{1}{r}B_{01}^TP\hat{Z}(t) \qquad (31)$$

3. Results

A bulk carrier having the principle parameter list in Table 1 is considered. The weight of the chain locker with anchor chain in this ship is 70 *ton*. The parameter of the hydraulic system used for the absorber is presented in Table 2. In the expression of the performance index for control law, the weight of each state is chosen by: (1) the weight of each vertical velocity is 10^{12}; (2) the weight of each slope velocity is 10^{15}; (3) the weight of y_d is 10^{10}; (4) the weight of input signal is 1; and (5) other weights are zero.

Table 1 Principle Parameter of the Sample Ship

L_{pp}	245.0 m
B	29.6 m
d	10.8 m
DWT	38,700 ton

Table 2 Parameter of the Hydraulic Servo System

A_d	10^{-2}	m^2
c_d	10^{2}	N–sec/m
c_k	10^{-13}	
k_c	10^{-11}	m^5/sec–N
k_s	1	m/Volt
k_q	10^{3}	m^2/sec
V_d	5×10^{-3}	m^3

To understand the effect of vibration reduction by the bow absorber the frequency and time responses are considered on the excitation from the vertical unbalance force of the main engine, which is located at $0.25L$ aft midship. An active absorber located at the stern with the same specifications is also calculated for comparing the performance of the absorber at the bow and stern. Fig. 4 shows the frequency response of the stern for the cases of the hull without any absorbers, the hull with the bow absorber, and the hull with the stern absorber. In this figure, 6.83*cpm* and 7.76*cpm* are the resonance frequencies of the rigid body motion, which is due to the pitch motion of the ship. The resonance frequencies 63.5*cpm*, 136.5*cpm*, 201.9*cpm*, 538.6*cpm*, and 766.3*cpm* are with respect to v-2, v-3, v-4, v-5, and v-6 mode vibration. The highest peak of the response will be reduced from 4 *gal/ton* to 0.35 *gal/ton* by the bow absorber or stern absorber. Other resonance peaks are also suppressed efficiently by the bow absorber and stern absorber. The performances of the bow absorber and the stern absorber are very similar except in very high frequency ranges. The highest resonance peak of the response at the midship, shown in Fig. 5, is 3.1 *gal/ton* in v-4 mode, which can be decreased 92% by the bow absorber and 90% by the stern absorber. The resonance peak of the bow is also diminished about 90% by the bow or the stern absorber shown in Fig. 6.

In the second case, a time history simulation based on

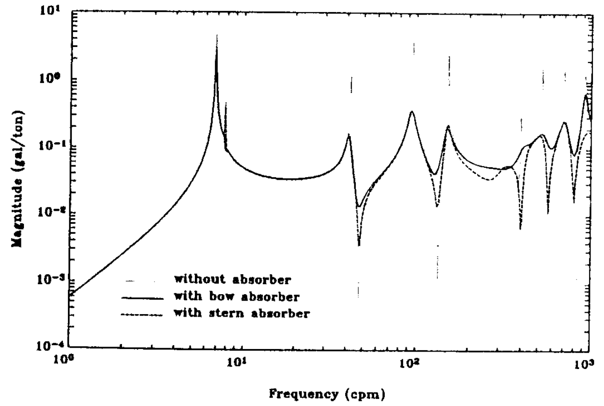

Fig. 4 Frequency Transfer Function of the Stern with respect to the Engine Excitation

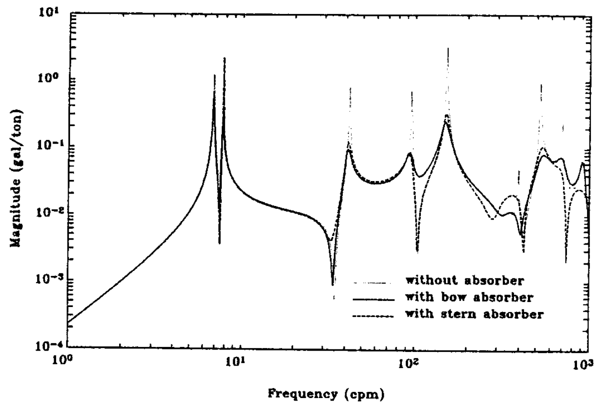

Fig. 5 Frequency Transfer Function of the Midship with respect to the Engine Excitation

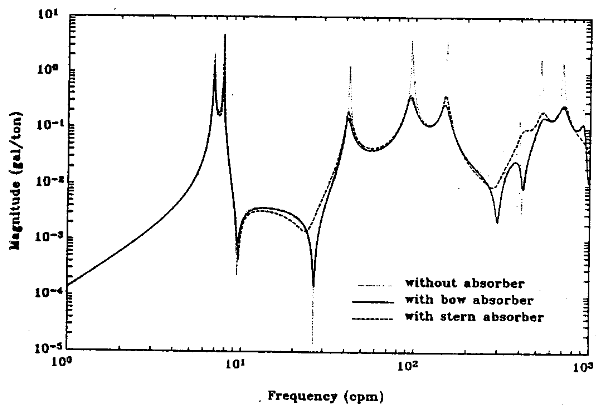

Fig. 6 Frequency Transfer Function of the Bow with respect to the Engine Excitation

an sinusoidal exciting with an amplitude of 10 tons and combined frequencies of v-2, v-3, and v-4 modes from the main engine, $f_e=10(\sin 4.38\,t+\sin 10.00\,t+\sin 15.90\,t)$ *ton*, is considered. The displacement of the hull girder at the stern, as plotted in Fig. 7, increases gradually over time in the case of the hull without absorber, but this fluctuation can be decreased significantly except in low frequency ranges, which is due to rigid body motion. From Fig. 8, we find the maximum value of the acceleration response at the stern is suppressed under the range of 6 gal by the bow or the stern absorber. The vibration at the midship and bow, as shown in Fig. 9 and Fig. 10, is also reduced effectively by the bow absorber. In Fig. 11 and Fig. 12, we find the maximum displacement of the actuator is 3.5cm and the maximum value of the force output by the actuator is less than 25 tons.

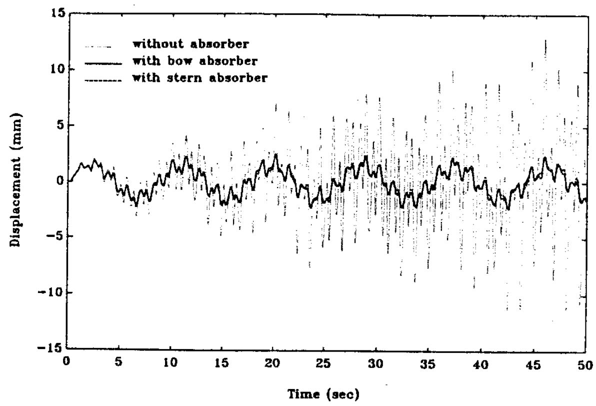

Fig. 7 Displacement Response of the Stern

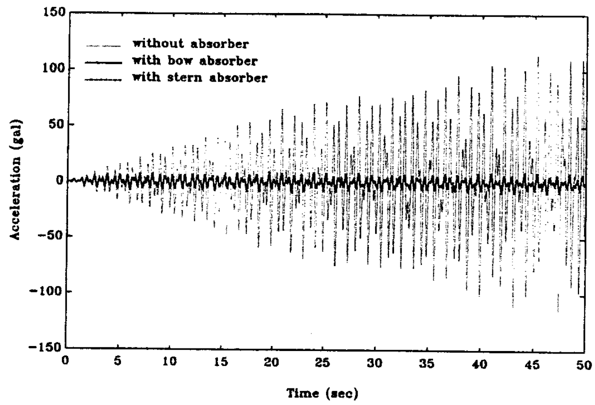

Fig. 8 Acceleration Response of the Stern

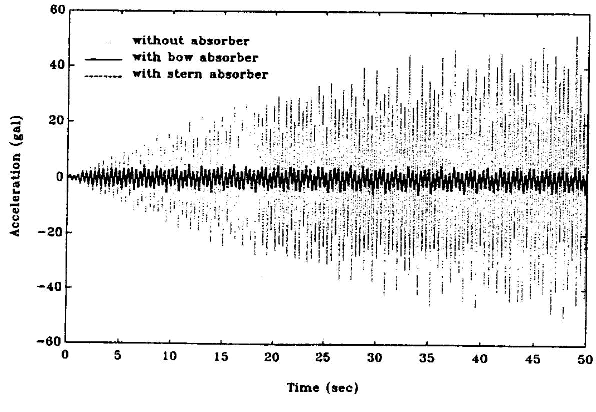

Fig. 9 Acceleration Response of the Midship

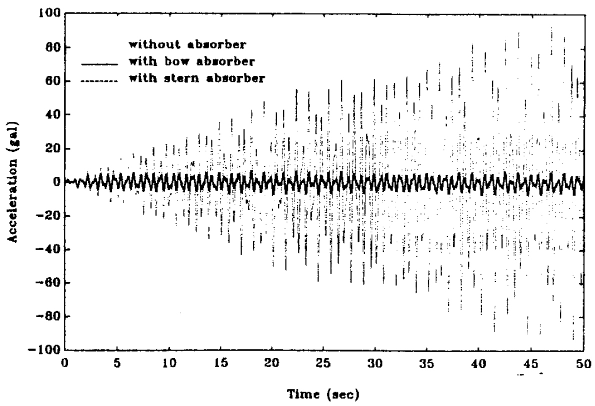

Fig. 10 Acceleration Response of the Bow

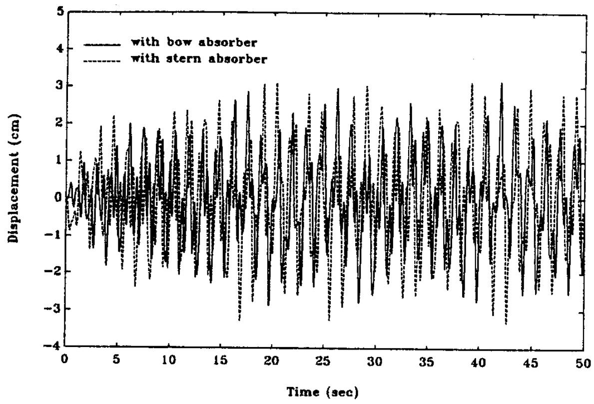

Fig. 11 Displacement of the Actuator

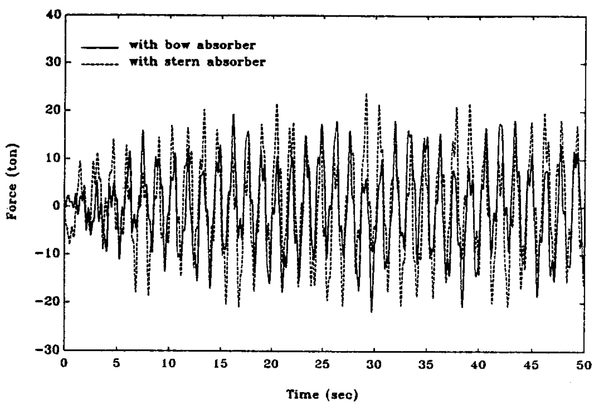

Fig. 12 Force Output of the Actuator

4. Conclusions

A design for a hydraulic bow absorber for vibration control of the hull girder has been studied in this paper. The dynamic characteristics of the hydraulic system was considered in the analysis and design to simulate real operating conditions. Based on the stochastic optimal control theory, an optimal performance for the absorber was derived for considering the efficiency of vibration and the power required for the actuator. In addition, an optimum estimator was used to estimate the distributed vibration states of the hull. In the numerical analysis of a 38,700ton bulk carrier, we found the hull vibration is reduced very efficiently with regard to the main engine excitation. Moreover, the energy consumption of the absorber is limited even in critical excitation.

Although the feasibility of this scheme has been shown in this paper, more advanced study should be conducted along with further experiments to certify the reliability.

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References

1) Schlick O., "On the Vibration of Steam Vessels", *Institution of Naval Architects,* 1884.
2) Todd, F. H., *Ship Hull Vibration,* Arnold, 1961.
3) Kawai, K. H., "The application of Finite Element Methods to Ship Structures", *Computer and Structures,* Vol. 3, 1973.
4) Daidola, J. C., "Natural Vibration of Beams in a Fluid with Applications to Ships and Other Marine Structures", *SNAME Transactions,* Vol. 92, 1984.
5) McGoldrick R. T., "Experiments with Vibration Neutralizers", *Taylor Model Basin,* No. 449, 1938.
6) Powell, M. J., Long, G., and Farrell, A., "Reduction of Hull Vibration in a Landing Craft by Means of a Vibration Absorber, *"RINA,* 1980.
7) Hydrodynamic Vibration Damper, *Shipping World and Ship Builder,* May 1971.
8) Matsuura, Y., Matsumoto, K., Mizuuchi, M., Arima, K., and Jouichi, H., "A study of Hydrodynamic Dampers to Reduce Ship Vibration", *Journal of the Kansai Society of Naval Architects, Japan,* No. 199, Dec. 1985.
9) Kagawa, K., Koukawa, H., Fujita, K., Zensho, Y., and Matsuo, M., "Development of Tuned Liquid Damper for Ship Vibration", *Transactions of the West-Japan Society of Naval Architects,* No. 78, 1989.
10) Kulik, M., "Absorber with Auxiliary Force (AAF)", *Ship Technology Research,* Vol. 37, No. 2, June 1990.
11) Hsueh, W. J. and Lee, Y. J., "A Hydraulic Absorber for Wideband Vibration Reduction in Ship Hulls", *Journal of Offshore Mechanics and Arctic Engineering, ASME,* Vol. 116, February, 1994.
12) Merritt, H. E., *Hydraulic Control Systems,* John Wiley & Sons, 1967.
13) Martin, D. J. and Burrows C. R., "The Dynamic Characteristics of an Electro-Hydraulic Servovalve", *Journal of Dynamic Systems, Measurement, and Control,* Dec. 1976.

14) Clough, R. W. and Penzien, J., *Dynamics of Structures,* McGraw-Hill, 1975.
15) Hsueh, W. J. and Lee, Y. J., "Active Vibration Control on the Mast of Warships", *International Shipbuilding Progress,* Vol. 39, No. 417, 1992.
16) Washizu, K., *Variational Methods in Elasticity and Plasticity, 2nd ed.,* Pergamon Press Ltd, New York, 1975.
17) Lewis, F. L., *Applied Optimal Control and Estimation,* Prentice-Hall, 1992.