Vibration and Noise Control of Structural Systems Using Squeeze Mode ER Mounts

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This paper presents vibration and noise control of flexible structures using squeeze mode electro-rheological mounts. After verifying that the damping force of the ER mount can be controlled by the intensity of the electric fild, two different types of ER squeeze mounts have been devised. Firstly, a small size ER mount to support 3 kg is manufactured and applied to the frame structure to control the vibration. An optimal controller which consists of the velocity and the transmitted force feedback signals is designed and implemented to attenuate both the vibration and the transmitted forces. Secondly, a large size of ER mount to support 200 kg is devised and applied to the shell structure to reduce the radiated noise. Dynamic modeling and controller design are undertaken in order to evaluate noise control performance as well as isolation performance of the transmitted force. The radiated noise from the cylindrical shell is calculated by SYSNOISE using forces which are transmitted to the cylindrical shell through two-stage mounting system.

Key Words : ER Mount (electrorheological mount), Seueeze Mode, Radiated Noise, Vibration Control, Noise Control, Two-stage Mounting System

1. Introduction

As modern naval ships are getting more and more silent, it is necessary to reduce vibration and acoustic pressure radiated from ships. One of the attractive approach to attenuate unwanted vibration and noise is to utilize mounting system, especially two-stage mounting system. In two-stage mounting system, there is an intermediate structure between upper and lower mounts to achieve required vibration and noise reduction goal. A two-stage mounting system has the functions of diminishing the vibration transmitted from machinery installed to ship's hull, and reducing shock wave propagated from underwater explosion. The essential element in the two-stage mounting system is mount itself.

Mounts serve two principal functions ; isolation and support. To reduce transmitted vibration and noise from machinery, softer mounts become necessary. However mounts must also limit or control machinery dynamic behavior. To provide control, it is important that mounts be stiff and heavily damped.

To meet the conflicting requirements of isolation and control, many researchers have studied electro-rheological (ER) mounts whose fluid un-

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dergoes instantaneous and reversible changes in damping, and the vibration control has been initiated in automotive engineering applications (Morishita et al., 1992; Williams et al., 1993; Petek et al., 1995; Choi et al., 1996; Choi et al., 1999; Choi et al., 2000). This idea can be easily exploited in structural vibration control (Wang et al., 1994; Choi et al., 1996). This method can be applied for the vibration control of relatively large-sized structures or complicated structures such as foundation structure in a ship.

When ER mount is used for vibration control, the operating mode of the ER mount can be classified by three different types: flow mode (Petek et al., 1995), shear mode (Choi et al., 1999) and squeeze mode (Williams et al., 1993). In general, steady-state excitation amplitude of equipments is approximately lower than 0.1-0.2 mm in the frequency range of above 20 Hz. Unlike the former two modes, in the squeeze mode the electrode gap is varied and the ER fluid is squeezed out by a normal force. For this reason, the squeeze mode ER mount can be effectively used for vibration control with small excitation amplitude. The authors proposed squeeze mode ER mount, which supports 3 kg static load, and applied to the vibration control of beam structure (Jung et al., 2002).

Intermediate structure in the two-stage mounting system is generally massive and consists of flexible structure members such as beam and plate. Flexible structure has its own natural modes and this is the reason for large vibration and sound radiation in certain frequencies called resonance. It is necessary to control the resonance mode of flexible structure, especially 1st and 2nd mode which are offsetting the benefits associated with mounting system. Usually, resonance frequencies for 1st and 2nd bending mode are about 30-40 Hz in most two-stage mounting system.

The main contribution of this paper is to effectively suppress the 1st and 2nd bending mode of frame and two-stage mounting system using squeeze mode ER mounts which supports 3 kg and 200 kg loads, respectively. To achieve this goal, after establishing control model in the space representation, an optimal controller which consists of two feedback signals: the velocity and the force transmitted from the exciting point to the mount position-is formulated. The simulated and experimental control responses such as acceleration are compared together for the frame structure. Radiated noise of a cylindrical shell is also calculated by using controlled transmitted forces and LMS-SYSNOISE S/W. It is noted that none deals with the squeeze mode ER mount subjected to 200 kg of static load for vibration control of a two-stage mounting system.

2. Vibration Control of Frame Structure

2.1 Squeeze-mode ER mount to support 3 kg

The schematic configuration of the 3 kg ER mount used in this paper is shown in Fig. 1(a). The lower electrode is fixed to the base, while the upper electrode is to be moved up and down. Thus, the squeeze-mode motion of the ER fluid occurs in the housing. The coil spring mount is attached to provide a static load capability of 3 kg. The total force of the squeeze mode ER mount can be obtained by

$$F(t)=F_s(t)+F_v(t)+F_{er}(t) \tag{1}$$

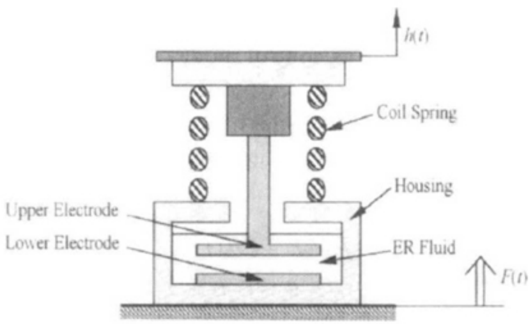
$$\begin{aligned} &F_s(t)=kh(t), \\ &F_v(t)=c_f\dot{h}(t)=\frac{3\pi\eta R^4}{2[h_0+h(t)]^3}\dot{h}(t), \\ &F_{er}(t)=\frac{4}{3}\frac{\pi\eta R^3\tau_y(E)}{[h_0+h(t)]}\operatorname{sgn}(\dot{h}(t)) \end{aligned} \tag{2}$$

In the above, $F_s(t)$, $F_v(t)$ and $F_{er}(t)$ are spring force, viscous damping force and controllable damping force, respectively. k is the stiffness constant of coil spring, η is the viscosity of the ER fluid, $c_f(t)$ is the damping coefficient of the ER mount in the absence of the electric field, $h(t)$ is the exciting displacement, h_0 is the initial gap between lower and upper electrodes, and R is the radius of the circular electrode. $\tau_y(E)$ is the field-dependent yield stress which is given by αE^β, where α and β are 437, 1.2, respectively. The value of η is 0.08. By considering the mass to be supported, an appropriate size of the ER mount is manufactured as shown in Fig. 1(b). The geometry data for squeeze mode ER mount to support 3 kg are ;

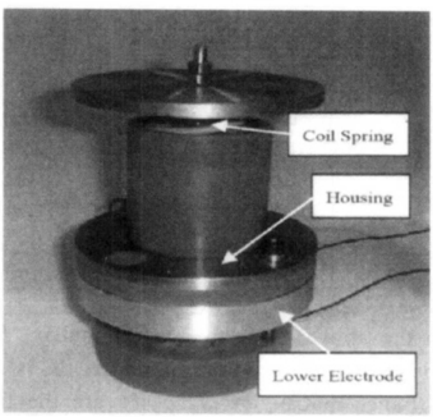
-radius of circular electrode : 200(mm), initial gap between electrodes : 3(mm)
-stiffness of coil spring : 4000(N/m), outer size : $\phi 105 \times H160$(mm)

Figure 2(a) shows the measured force-displacement relation of squeeze mode ER mount for the excitation frequency of 75 Hz. The area of the closed elliptic circle is increased as the voltage increases.

The area of the closed elliptic circle means the energy dissipation capability of ER mount. As the voltage increases, the damping energy dissipation capability also increases.



(a) Schematic diagram

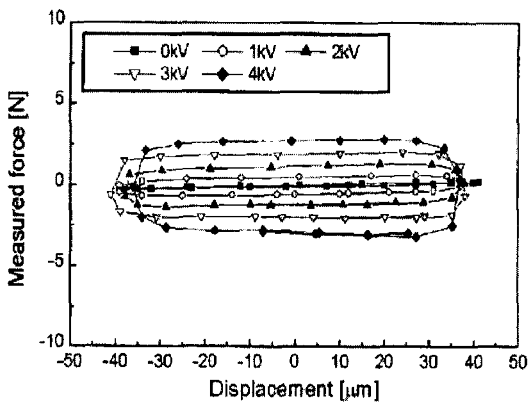


(b) Photograph

Fig. 1 Schematic diagram and photograph of 3 kg squeeze mode ER mount

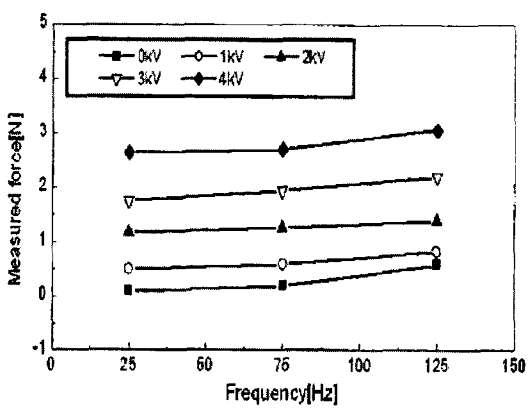
Figue 2(b) presents the measured force-frequency relation of squeeze mode ER mount for the excitation of $\pm 40(\mu m)\sin(2\pi f)t$, $f=25$ Hz, 75 Hz, 125 Hz. The measured damping force increases as the applied voltage and exciting frequency are increased.

In practical application of a squeeze-flow as well as steady-state response, it is necessary to consider the transient behavior when the ER fluid is energised and de-energised. Time constant is usually used for the response measure of output quantity to a step change in applied voltage. Time constant for the ER fluid employed here is approximately within 6.5(msec) in average (Hong et al., 2002).



excitation frequency : 75 Hz

(a) Force-displacement diagram



excitation amplitude : $\pm 40(\mu m)$

(b) Force-frequency diagram

Fig. 2 Measured damping force of squeeze mode ER mount to support 3 kg

2.2 Dynamic modeling of frame structure

The schematic configuration of the frame structure supported by four spring mounts and two ER mounts shown in Fig. 3. The frame structure consisting of L-shaped cross-sectional steel components (30 mm × 30 mm × 3 mm) has the length of 1200 mm and the width of 800 mm. Four spring mounts are fixed at four corners (p_1, p_3, p_4, p_6), and two ER mounts are attached at p_2 and p_5 positions. The position p_7 is to be excited by a shaker. Accelerometers are attached on the frame structure at the position of p_2 and p_5, and velocities at the corresponding positions are obtained by the integrator circuits.

Force transducers are installed at the bottom of spring mounts (p_3 and p_4) to measure the force transmitted to the base through spring mounts. The equation of motion for frame structure with control force is obtained by

$$\begin{aligned}&[M_s]\{\ddot{y}(x,t)\}+[C_s]\{\dot{y}(x,t)\}+[K_s]\{y(x,t)\}\\&=[B_s]\{u(t)\}+[D_s]\{f(t)\}\end{aligned} \tag{3}$$

where $[M_s]$, $[C_s]$, $[K_s]$ are $n\times n$ dimensional mass, damping and stiffness matrix, $u(t)$ is $m_1\times 1$ dimensional control force vector, $f(t)$ is $m_2\times 1$ dimensional external force vector. $[B_s]$ is $n\times m_1$ matrix, $[D_s]$ is $n\times m_2$ matrix, respectively. From the modal analysis using mass normalized modal matrix $[\phi]$, generalized coordinate $\{q(t)\}$, transverse deflection $y(x,t)=[\phi]\{q(t)\}$ where x is the position of frame structure, the decoupled matrix equation is derived by

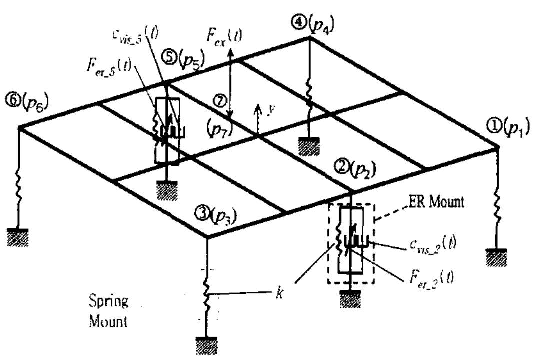


Fig. 3 Frame structure with squeeze mode ER mounts

$$\begin{aligned}&[I]\{\ddot{q}(t)\}+diag[2\zeta\omega_n]\{\dot{q}(t)\}+diag[\omega_n^2]\{q(t)\}\\&=[\phi]^T[B_s]\{u(t)\}+[\phi]^T[D_s]\{f(t)\}\end{aligned} \tag{4}$$

In the above, x is the position of frame structure. From Eq. (4) and Fig. 2, the decoupled ordinary differential equation for i^{th} mode is derived by

$$\begin{aligned}&\ddot{q}_i(t)+2\zeta_i\omega_{ni}\dot{q}_i(t)+\omega^2_{ni}q_i(t)=Q_i(t)+Q_{exi}(t)\\&Q_i(t)=-\phi_i(p_2)[f_{vis\_2}(t)+F_{er\_2}(t)]\\&\qquad-\phi_i(p_5)[f_{vis\_5}(t)+F_{er\_5}(t)]\\&Q_{exi}(t)=\phi_i(p_7)F_{ex}(t)\end{aligned} \tag{5}$$

$$f_{vis\_j}(t)=C_{vis\_j}\dot{y}(p_j,t)=\frac{3}{2}\frac{\pi\eta R^4}{(h_0+y(p_j,t))^3}\dot{y}(p_j,t),\ j=2,5$$

$$F_{er\_j}(t)=\frac{4}{3}\frac{\pi\eta R^3}{[h_0+y(p_j,t)]}\tau_y(E)\,\mathrm{sgn}(\dot{y}(p_j,t)),\ j=2,5$$

where $f_{vis\_j}(t)$, $F_{er\_j}(t)$ are viscous damping force, controllable damping force with the electric field, respectively. $F_{ex}(t)$ is the external force.

2.3 Controller design

In this paper, we considered the first two bending modes as control modes since these two modes are dominant for the transverse vibration. Thus, the dynamic model of the structural system can be expressed in a state-space form as follows from Eq. (5)

$$\begin{aligned}&\dot{x}(t)=Ax(t)+Bu(t)+\Gamma d(t)\\&x(t)=\{q_1(t)\ \ \dot{q}_1(t)\ \ q_2(t)\ \ \dot{q}_2(t)\}^T\\&u(t)=[F_{er\_2}(t)\ \ F_{er\_5}(t)]^T,\ d(t)=[F_{ex}(t)]^T\end{aligned} \tag{6}$$

$$A=\begin{bmatrix}0 & 1 & 0 & 0\\ -\omega_1^2 & -2\zeta_1\omega_1 & 0 & 0\\ 0 & 0 & 0 & 1\\ 0 & 0 & -\omega_2^2 & -2\zeta_2\omega_2\end{bmatrix}$$

$$B=\begin{bmatrix}0 & 0\\ \phi_1(p_2) & \phi_1(p_5)\\ 0 & 0\\ \phi_2(p_2) & \phi_2(p_5)\end{bmatrix},\ \Gamma=\begin{bmatrix}0\\ \phi_1(p_7)\\ 0\\ \phi_2(p_7)\end{bmatrix}$$

In the above, A is the system matrix, B is the control input distribution matrix, Γ is the disturbance matrix, $q_1(t)$, $q_2(t)$ are the first and second modal coordinate, $\phi_1(p_i)$, $\phi_2(p_i)$, $i=2, 5, 7$ are the first and second bending mode shape at p_i position.

Among many controller candidates, an optimal controller which is known to be very effective for structural vibration control is adopted. As a first step to formulate the optimal controller, we choose two important variables; velocity at p_2, p_5 of frame structure and transmitted forces at p_3, p_4. Thus, by defining these variables as new state variables, we can rewrite the control model expressed by Eq. (6) as follows.

$$\begin{aligned} \dot{z}(t) &= A^* z(t) + B^* u(t) + \Gamma^* d(t) \\ z(t) &= [\dot{y}_{p_2}(t), F_{p_3}(t), F_{p_4}(t), \dot{y}_{p_5}(t)]^T = Px(t) \end{aligned} \quad (7)$$

$$A^* = PAP^{-1},\ B^* = PB,\ \Gamma^* = P\Gamma$$

$$P = \begin{bmatrix} 0 & \phi_1(p_2) & 0 & \phi_2(p_2) \\ k\phi_1(p_3) & 0 & k\phi_2(p_3) & 0 \\ k\phi_1(p_4) & 0 & k\phi_2(p_4) & 0 \\ 0 & \phi_1(p_5) & 0 & \phi_2(p_5) \end{bmatrix}$$

In the above, P is the state transformation matrix. The control purpose is to regulate unwanted vibrations of the structural system and transmitted forces to the base with appropriate control input voltage. Thus, the performance index to be minimized is chosen by

$$J = \min \int_0^\infty \{ z^T(t) Q z(t) + u^T(t) R u(t) \} dt \quad (8)$$

where Q is the state weighting semi-positive matrix, R is the input weighting positive matrix. Since the system (A^*, B^*) in Eq. (7) is controllable, we can obtain the following state feedback controller.

$$u(t) = -R^{-1} B^{*T} L z(t) = K z(t) \quad (9)$$

In the above, K is the state feedback gain matrix, and L is the solution of the algebraic Riccati equation:

$$A^{*T} L + L A^* - L B^* R^{-1} B^{*T} L + Q = 0 \quad (10)$$

Now, the control damping force of the ER mount at position p_j can be represented as

$$u_j(t) = k_{j1}\dot{y}_2(t) + k_{j2}F_3(t) + k_{j3}F_4(t) + k_{j4}\dot{y}_5(t), \quad j=2, 5 \quad (11)$$

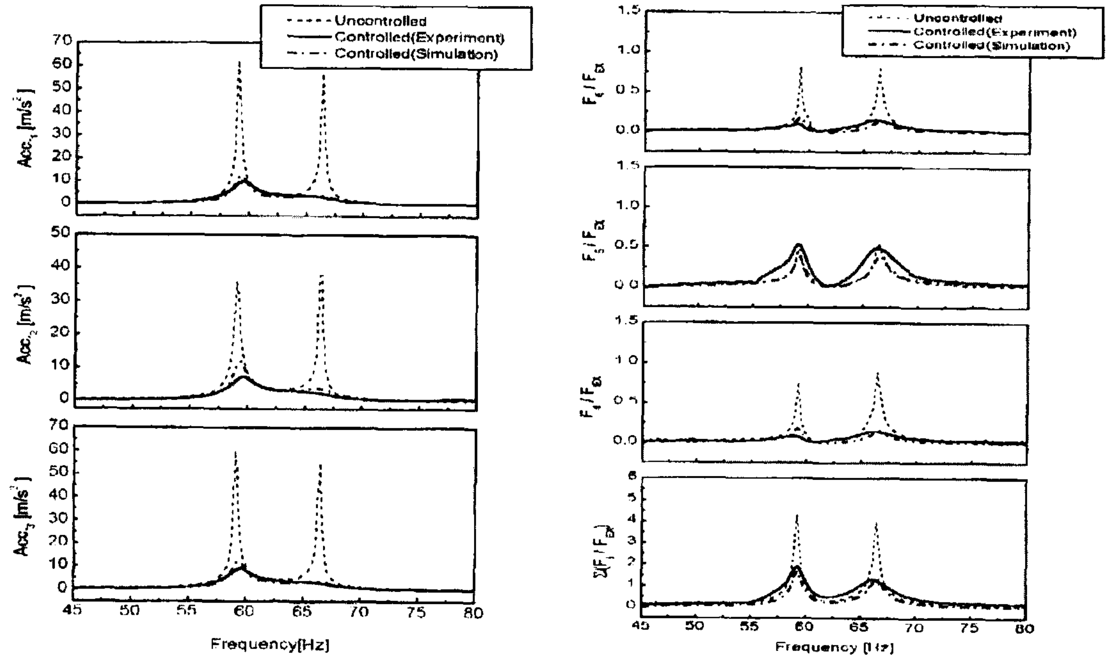
It is noted that forces $F_3(t)$ and $F_4(t)$ are directly measured using force transducers, and the velocity signals $\dot{y}_2(t)$ and $\dot{y}_5(t)$ are obtained from the integrator circuits with the measurements of the corresponding accelerations. Thus, state estimator is not need for the controller implementation. The squeeze mode ER mount is semi-active. Therefore, control signal needs to be applied according to the following actuating condition

$$u_j(t) = \begin{bmatrix} u_j(t) \text{ for } u_j(t)\dot{y}(p_j, t) > 0 \\ 0 \text{ for } u_j(t)\dot{y}(p_j, t) \le 0 \end{bmatrix}, \ j=2, 5 \quad (12)$$

For the computation of the optimal controller gains, the weighting matrix Q and R need to be predetermined. If the elements in matrices Q and R are determined, we can get the state feedback matrix K in Eq. (9) by obtaining the matrix L which is the solution of algebraic Riccati equation in Eq. (10). Therefore, the choice of matrices Q and R has an important role in computing the optimal controller gains. In this paper, the weighting matrices have been chosen by the trial and error using optimal controller with semi-active condition given by Eqs. (11) ~ (12).

Table 1 Parameters of state space model for frame structure

Parameters	1st bending mode		2st bending mode	
Modal frequency (Hz)	f_1	59.253(Hz)	f_2	66.522(Hz)
Modal damping	ζ_1	0.0015	ζ_2	0.0001
Mode shape at p_1	$\Phi_1(p_1)$	−3.4821E-1	$\Phi_2(p_1)$	4.0263E-1
Mode shape at p_2	$\Phi_1(p_2)$	3.2816E-1	$\Phi_2(p_2)$	−3.9110E-1
Mode shape at p_3	$\Phi_1(p_3)$	−3.4821E-1	$\Phi_2(p_3)$	4.0263E-1
Mode shape at p_4	$\Phi_1(p_4)$	−2.8293E-1	$\Phi_2(p_4)$	−4.5616E-1
Mode shape at p_5	$\Phi_1(p_5)$	2.6291E-1	$\Phi_2(p_5)$	4.4114E-1
Mode shape at p_6	$\Phi_1(p_6)$	−2.8293E-1	$\Phi_2(p_6)$	−4.5616E-1



(a) Acceleration (b) Force

Fig. 4 Vibration control of frame structure with force-velocity feedback optimal control

2.4 Results and discussion

Since the first elastic mode is twisting mode and the first bending mode occurs from the second elastic mode of the frame structure as shown in Fig. 3, the second and the third elastic modes are considered as control modes. Parameters of state-space model and control gain values for frame structure are listed in Table 1.

Control gains for frame structure used in this paper are ;

$$k_{21}=10.9,\ k_{22}=-1.1,\ k_{23}=-0.8,\ k_{24}=-0.4,$$
$$k_{51}=-1.0,\ k_{52}=1.0,\ k_{53}=0.7,\ k_{54}=10.5$$

The measured and simulated acceleration levels at the position p_1, p_2, p_3 are presented in Fig. 4(a) for the external disturbance when excited by $2\sin(2\pi ft)$ varied from 25 Hz to 80 Hz. Here, Acc_1, Acc_2, Acc_3 means acceleration levels at p_1, p_2, p_3 in Fig. 3.

Figure 4(b) displays the measured and simulated transmitted forces at the position of p_4, p_5, p_6 for the external disturbance with $2\sin(2\pi ft)N$ varied from 25 Hz to 80 Hz. Here, F_4, F_5, F_6 are the transmitted force levels at F_4, F_5, F_6 in Fig. 3. The vibration of frame and transmitted force through mounts are effectively attenuated. It is also observed that the good agreements between simulated and experimental results exist from Fig. 4.

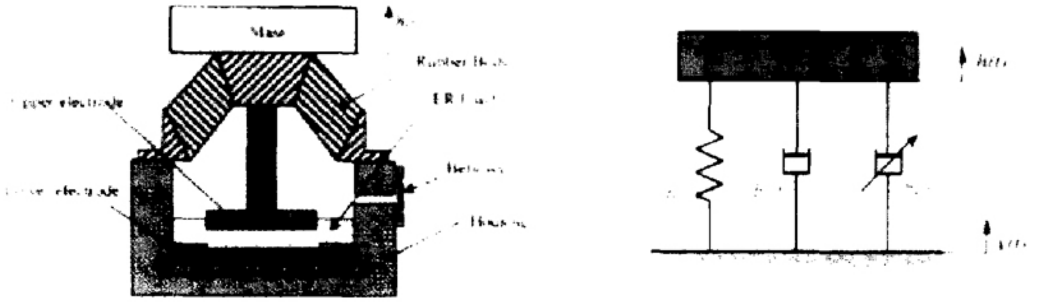
3. Noise Control of Cylindrical Shell

3.1 Squeeze-mode ER mount to support 200 kg

The schematic configuration and mathematical model of the 200 kg squeeze mode ER mount used in this paper is shown in Fig. 5.

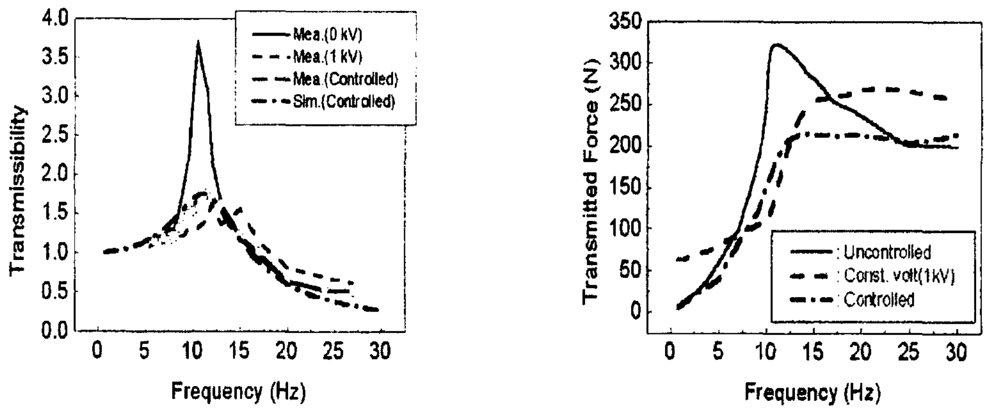
The lower electrode is fixed to the base, while the upper electrode is to be moved up and down. Thus, the squeeze-mode motion of the ER fluid occurs in the housing. The rubber mount is attached to provide a static load capability of 200 kg. The geometry data for 200 kg ER mount are as follows

-radius of circular electrode : 50 mm,
initial gap between electrodes : 3 mm
-stiffness of rubber mount : 1.3×10^6(N/m),



(a) Schematic diagram (b) Mathematical model

Fig. 5 Schematic diagram and mathematical model of 200kg squeeze mode ER mount



(a) Acceleration response (b) Transmitted force response

Fig. 6 Transmissibility of 200 kg squeeze mode ER mount

outer size: $W220 \times L220 \times H250$

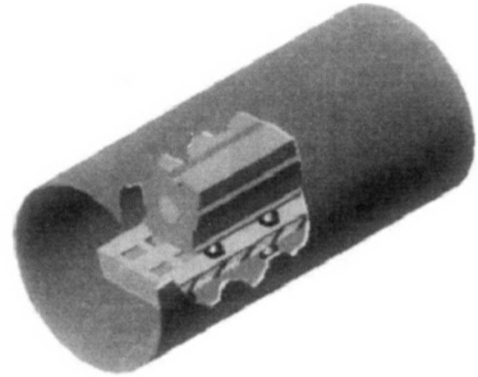
The total force of the squeeze mode ER mount can be obtained using Eqs. (1)-(2) and mathematical model in Fig. 5(b). The vibration control effectiveness of 200 kg squeeze mode ER mount is shown numerically and experimentally with constant voltage and skyhook control algorithm in Fig. 6. The skyhook gain is set to be 6000.

3.2 Dynamic modeling of cylindrical shell

The radiated noise plays a very important role in the acoustic stealth of the naval ships since the acoustic wave in the sea can be transmitted to longer range than any other signature. Therefore, the prediction and reduction of radiated noise must be conducted from the early design stage. The calculation process of radiated noise in a submerged cylindrical shell is shown in Fig. 7, where F_{hull} is the force transmitted to the shell and P_{rad} is the radiated noise by the transmitted force (F_{hull}). Therefore, we have to first analyze the transmitted force. If there is a mounting system within a shell, there must be a supporting structure with some finite impedance below the mount. Transmitted force through the mount can be affected by the supporting structure below it.

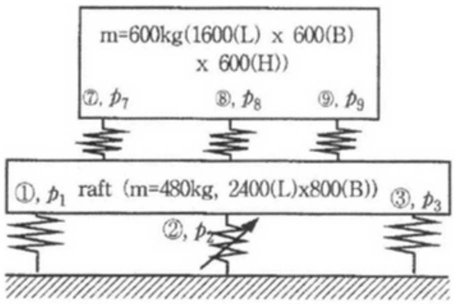
However, if there is a large impedance mismatch, about 10-20(dB) between mount and supporting structure, the influence of supporting structure is known to be ignored (Verheij 1986). The schematic configuration of the cylindrical shell with two-stage mounting system is shown in Fig. 7(b) The vibrating machine is resiliently mounted to a heavy intermediate structure (raft) which is resiliently mounted to the hull. Vibrations must be transmitted through the raft before they reach the hull. Resilient mounts are

(a) Noise generation mechanism

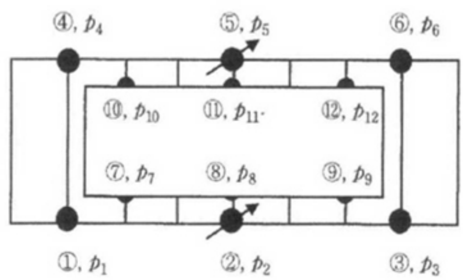


(b) Cylindrical shell with two-stage mounting system

Fig. 7 Schematic diagram of radiated noise generation mechanism and cylindrical shell



(a) Side view



(b) Top view

Fig. 8 Side view (a) and top view (b) of two-stage mounting system

installed between the vibrating machine and raft, and between the raft and hull in order to obtain the full benefits of two-stage mounting system as shown in Fig. 7(b).

The upper mass represents engine, and lower mass is the intermediate structure (raft). The weight of engine is assumed to be 600 kg, and raft to be 480 kg. The upper six rubber mounts between engine and raft can support 100 kg of static load, and the lower four ones can provide 200 kg of static load. The vertical stiffness of upper and lower mounts is $k_z=2.47\times10^5$(N/m), and $k_z=1.36\times10^6$(N/m), respectively. The transverse and longitudinal stiffness of upper and lower mounts are assumed to be half to the vertical stiffness. Two ER mounts, shown in Fig. 4, are installed at the positions of p_2 and p_5 below raft. The intermediate structure has the frame structure form. The finite element model of two-stage mounting system is shown in Fig. 9.

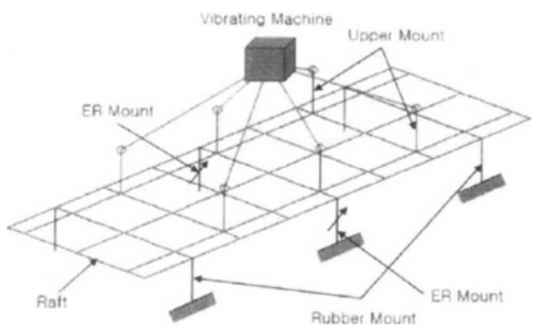


Fig. 9 Finite Element Model of two-stage mounting system

It is difficult to experimentally investigate the control effectiveness of two-stage mounting system with 200 kg ER mounts for many reasons. Thus, in this paper, we use numerical simulation for the transmitted forces in two-stage mounting system. In order to identify modal parameters such as natural frequency and mode shape of

Table 2 Parameters of state space model for two-stage mounting system

Parameters	1st bending mode		2st bending mode	
Modal frequency (Hz)	f_1	70.691(Hz)	f_2	92.441(Hz)
Modal damping	ζ_1	0.002	ζ_2	0.0015
Mode shape at p_1	$\Phi_1(p_1)$	−3.1389E-2	$\Phi_2(p_1)$	4.8128E-2
Mode shape at p_2	$\Phi_1(p_2)$	5.5174E-2	$\Phi_2(p_2)$	−7.7863E-2
Mode shape at p_3	$\Phi_1(p_3)$	−3.1389E-2	$\Phi_2(p_3)$	4.8128E-2
Mode shape at p_4	$\Phi_1(p_4)$	−3.1389E-2	$\Phi_2(p_4)$	−4.8128E-2
Mode shape at p_5	$\Phi_1(p_5)$	5.5174E-2	$\Phi_2(p_5)$	7.7863E-2
Mode shape at p_6	$\Phi_1(p_6)$	−3.1389E-2	$\Phi_2(p_6)$	−4.8128E-2

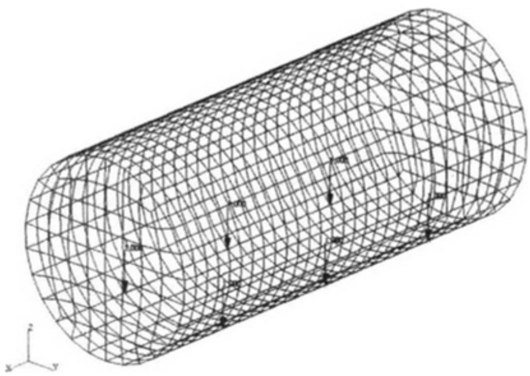


Fig. 10 Transmitted forces acting on the cylindrical shell through mounts

structures, we use a commercial finite element code (NASTRAN).

Radiated noise of a cylindrical shell is also calculated by using a commercial code (SYSNOISE). SYSNOISE model and acting positions of transmitted forces are shown in Fig. 10. The length, radius and thickness of shell are 3 m, 0.8 m and 0.03 m, respectively.

3.3 Controller design

Since the first two bending modes mainly contribute to the underwater radiated noise, we considered them as control modes like beam and frame structure. Two signals are fed back to the controller: velocity signals at the position of p_2 and p_5, and force signals at the position of p_1 and p_6. The state variables of the two-stage mounting system can be expressed by

$$\begin{aligned} z(t) &= [F_{p_1}(t),\ \dot{y}_{p_2}(t),\ \dot{y}_{p_5}(t),\ F_{p_6}(t)]^T \\ &= Px(t) \end{aligned} \tag{13}$$

$$P=\begin{bmatrix} k\Phi_1(p_1) & 0 & k\Phi_2(p_1) & 0 \\ 0 & \Phi_1(p_2) & 0 & \Phi_2(p_2) \\ 0 & \Phi_1(p_5) & 0 & \Phi_2(p_5) \\ k\Phi_1(p_6) & 0 & k\Phi_2(p_6) & 0 \end{bmatrix}$$

The remaining processes are similar to the case of frame structure.

3.4 Results and discussion

In this paper, the forces (F_{hull}) transmitted to the shell through mounts in two-stage mounting system are derived from the mathematical model with rigid boundary condition at the underneath lower mounts. Since the first elastic mode is twisting mode and the first bending mode occurs from the second elastic mode, the second and the third elastic modes are considered as control modes like frame structure with mounts.

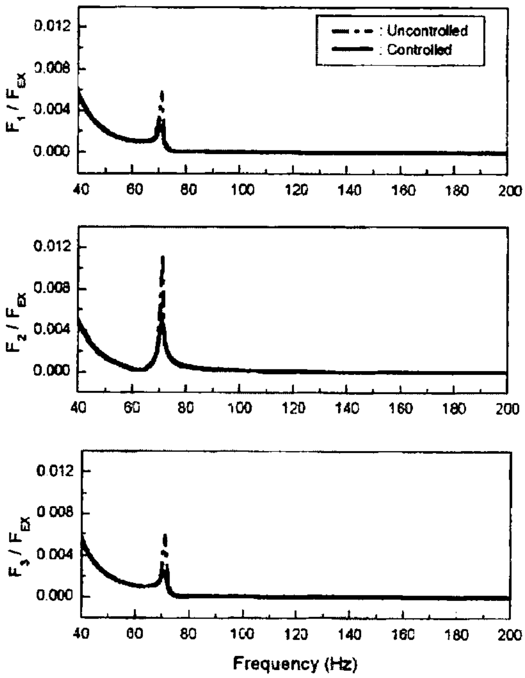
Parameters of state-space model and control gain values for two-stage mounting system are listed in Table 2.

Optimal controller and gains for cylindrical shell used in this paper are;

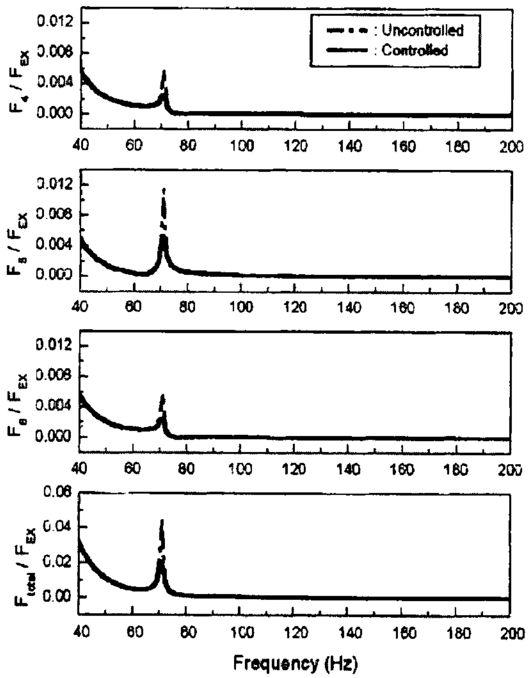
$$u_j(t)=k_{j1}F_1(t)+k_{j2}\dot{y}_2(t)+k_{j3}\dot{y}_5(t)+k_{j4}F_6(t), \quad j=2,\ 5 \tag{14}$$

$k_{21}=-0.2694$, $k_{22}=1200.0$, $k_{23}=2$, $k_{24}=0.0313$, $k_{51}=0.0313$, $k_{52}=2$, $k_{53}=1200$, $k_{54}=-0.2694$

Fig. 11(a), (b) represent the simulated force transmissibility. It is clearly observed that force transmissibility are reduced by activating the controller. The force transmissibility at the rubber mounts (p_1, p_3, p_4, p_6) is attenuated less than that of the ER mounts (p_2, p_5) positions. This is due



(a) Force transmissibility at p_1, p_2, p_3



(b) Force transmissibility at p_4, p_5, p_6

Fig. 11 Force transmissibility of two-stage mounting system

to the symmetric system arrangement and the first bending mode shape, in which large deflection occurs at the center. However, the total force transmissibility which is the summation of the forces at every mount position was effectively reduced at first resonance frequency as shown in the bottom of Fig. 11(b).

The equation of motion for the submerged shell with internal forces (F_{hull}) exerted to shell in Fig. 7(a) is obtained by

$$[M_s+M_A]\{\ddot{w}(x,t)\}+[C_s+C_A]\{\dot{w}(x,t)\} + [K_s]\{w(x,t)\}=\{F_{hull}\} \tag{15}$$

where $[M_s]$, $[C_s]$, $[K_s]$ are $n\times n$ dimensional structural mass, damping and stiffness matrix, $[M_A]$, $[C_A]$ are $n\times n$ dimensional added mass, damping matrix caused by structure-fluid interaction, $\{F_{hull}\}$ is $n\times 1$ dimensional exciting force vector, $\{w(x,t)\}$ is $n\times 1$ dimensional deflection response vector, x is the position at the shell surface. Eq. (15) can be solved using LMS-SYSNOISE S/W which utilise Finite Element/Boundary Element Method. The acoustic pressure P_{rms} at the distance R from the source point becomes from the Helmholtz integral equation (Junger et al., 1986)

$$\frac{P_{rms}}{2}=\int_s\left[p(R_o)\frac{\partial g}{\partial n}+j\omega v_n g\right]dS(R_o) \tag{16}$$

where R_o is the radius of radiating surface, g is the Green's function, v_n is the velocity components perpendicular to the shell deflection w.

The radiated noise in this paper is calculated with the acoustic pressure P_{rms} at the distance, 200 m, and then is converted to acoustic pressure source level of 1 m as follows;

$$\begin{aligned} &SPL_{1m}(dB)=SPL_{200m}(dB)+20\times\log_{10}(200)\\ &SPL_{200m}(dB)=10\times\log_{10}\left(\frac{P_{rms}}{P_r}\right),\\ &P_r=1\times10^{-6}(\mathrm{N/m^2}) \end{aligned} \tag{17}$$

where, we use the forces (F_{hull}) transmitted to the shell presented in Fig. 11 for the uncontrol and control case by activating two ER mounts.

Fig. 12 presents the uncontrolled and controlled response of the radiated noise. From this, we

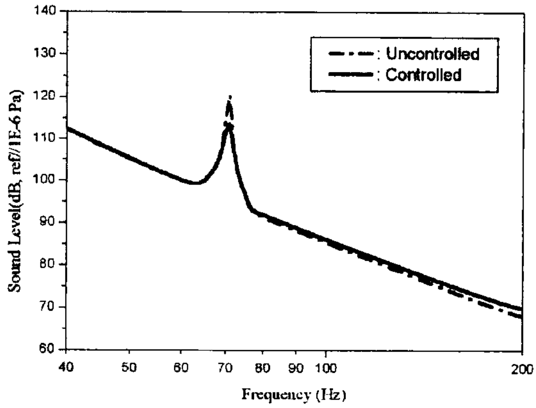


Fig. 12 Sound pressure at 1m from the center by the transmitted forces

see that radiated noise can be reduced about 6.9 (dB) at the resonance frequency by activating ER mounts.

4. Conclusion

Vibration control of frame structure and noise control of cylindrical shell with two-stage mounting system were investigated by applying the 3 kg and 200 kg squeeze-mode ER mounts. After verifying simulation method using frame structure supported by spring mounts and ER mounts, cylindrical shell with two-stage mounting system has been modeled. It has been demonstrated through experiment and simulation that vibration control such as acceleration and transmitted force agree well for frame structure. It is also shown that transmitted forces and radiated noise of cylindrical shell can be substantially reduced at the resonance frequencies by activating the ER mounts. The control results presented in this paper are quite self-explanatory justifying that the squeeze-mode ER mount can be effectively employed to flexible structures in order to suppress unwanted vibration and radiation.

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