Optimal Design of New Magnetorheological Mount for Diesel Engines of Ships

선박용 디젤엔진을 위한 새로운 MR 마운트의 최적설계

Xuan-Phu Do*, Joon-Hee Park*, Jae-Kwan Woo** and Seung-Bok Choi†
도쑤웬푸 · 박 준 희 · 우 제 관 · 최 승 복

(Received September 13, 2012 ; Revised December 11, 2012 ; Accepted December 11, 2012)

Key Words : Magnetorheological Fluid(MR 유체), Engine Mount(엔진 마운트), Optimal Design(최적설계), Diesel Engine Mount(디젤엔진 마운트), Ship(선박)

ABSTRACT

This paper presents an optimal design of a magnetorheological(MR) fluid-based mount(MR mount) that can be used for to vibration control in diesel engines of ships. In this work, a mount that uses mixed-modes(squeeze mode, flow mode, and shear mode) is proposed and designed. To determine the actuating damping force of the MR mount required for efficient vibration control, the excitation force from a diesel engine is analyzed. In this analysis, a model of a V-type engine is considered. The relationship between the velocity and pressure of gas in terms of the torque acting on the piston is derived. Subsequently, by integrating the field-dependent rheological properties of commercially available MR fluid with the excitation force, the appropriate size of the MR mount is designed. In addition, to achieve the maximum actuating force under geometric constraints, design optimization is undertaken using the ANSYS parametric design language software. Through magnetic density analysis, optimal design parameters such as the bottom gap and radius of coil are determined.

요 약

이 논문은 선박용 디젤엔진의 진동제어에 적용할 수 있는 MR 유체기반 마운트(MR 마운트)의 최적설계를 제시한다. 이 연구에서는 압착모드, 유동모드, 전단모드를 포함하는 혼합모드가 제안되었고 설계되었다. 효과적인 진동제어를 위하여 요구되는 MR 마운트의 작동 댐핑력을 결정하기 위하여 디젤엔진의 기진력이 분석되었다. 이 분석에서 V-type엔진이 고려되었으며 피스톤의 토크에서의 속도와 가스압력간의 관계를 유도하였다. 결과적으로 상업적으로 이용 가능한 MR 유체의 장의존적 유동특성과 기진력을 통합함으로써 적절한 MR 마운트의 크기가 설계되었다. 게다가 기하학적 제한조건이 고려된 최대 구동력을 얻기 위해 ANSYS를 이용하여 최적설계가 수행되었다. 자기밀도분석을 통해 바닥간격과 코일의 반지름과 같은 최적설계변수가 결정되었다.

† Corresponding Author ; Member, Department of Mechanical Engineering, Inha University
E-mail : seungbok@inha.ac.kr
Tel : +82-32-860-7319, Fax : +82-32-868-1716
* Member, Department of Mechanical Engineering, Inha University
** Department of Mechanical Engineering, Inha University

A part of this paper was presented and selected as one of best papers at the KSNVE 2012 Annual Autumn Conference
‡ Recommended by Editor Moon Kyu Kwak

1. Introduction

A mount is normally used to support an object and suppress vibration. There are many types of mounts such as passive, active, and semi-active mounts. Semi-active mounts such as magneto-rheological(MR) mounts can be used to isolate vibration at low frequency ranges where diesel engines primarily operate. The advantages of MR mounts include continuous damping control and simple design. It is well-known that MR fluids have three modes of operation - flow mode, shear mode and squeeze mode. These modes give MR fluids some special characteristics such as immediate change in viscosity upon the application of a magnetic field. Hence, MR fluids have become popular in applications such as brakes, mounts and dampers.

The flow mode is a phenomenon in which the fluid flows as a result of the pressure gradient between two stationary plates. The shear mode is a phenomenon in which the fluid flows between two plates that are in relative motion. The squeeze mode refers to the fluid flows between two plates that move perpendicular to their planes.

Recently, there has been an increasing level of interest in determining whether the modes of MR fluids can be combined for the purpose of designing dampers(2,7~14). Some of these researches have noted that squeeze mode dampers can handle higher loads than other modes(15,16), and hence, this mode can be applied in mount systems, while other modes can be used for dampers(15~19). In a research aimed at finding an optimal design mount, Nguyen et al.(1) studied an analytical method based on quasi-static modeling and magnetic circuit analysis. Nguyen et al.(4) also evaluated the zero-field friction heat and geometric dimensions in optimization progress of an automotive brake based on the finite element analysis model. A design of isolator included multi-mode using MR fluids was devised by Brigley(2), but he could not provide an optimal solution to the design. Farjoud et al.(15) and Zhang et al.(16) performed tests to find the characteristics of a squeeze mount that could withstand higher loads. These works are mostly related to MR dampers and small load mounts in automotive systems. The application of MR mounts in engines of ships has hardly been studied yet.

Consequently, the main purpose of this work is to develop a new type of MR mount that can be used for vibration control in the huge diesel engines of ships. This type of MR mount uses mixed modes(flow mode, shear mode, squeeze mode), which are based on the complicated operation of ship engines, involving forces and moments.

To achieve the research goal, the excitation from a diesel engine is analyzed by using a V-type engine. Then, using the field-dependent rheological properties of MR fluids the damping force required to suppress the vibration is calculated. The initial design parameters such as the length of the magnetic pole and the diameter of the piston rod are determined using the equations governing the MR mount. In addition, to maximize the damping force under geometric constraints, an optimization process is undertaken using the commercial software APDL(ANSYS parametric design language). Computer simulation is undertaken to investigate the actuating force with respect to the distance between the piston and the bottom of the housing.

2. Excitation Forces from Diesel Engine

There are two modes in the operation of a diesel engine: the rigid mode and the flexible mode. The rigid mode includes phenomena such as rolling, pitching, bouncing, and yawing. The flexible mode has two sub-modes, which we focus on: the vertical bending mode and the horizontal bending mode. The vertical bending mode is more important in terms of its effects on the capabilities of the operation system.

There are some phenomena in the operation of diesel engines that are undesirable due to their ill effects on human health and the longevity of the engine. Figure 1 shows the external forces and moments in an engine. F_v represents the unbalanced vertical external force, whereas F_H is the unbalanced horizontal external force. Similarly, M_v is the unbalanced vertical external moment, and M_H is the unbalanced horizontal external moment.

To calculate and find out the common features between the forces and moments transmitted to the mount, it is essential to know where a force appears and what the consequence of the moment is. A model of a V-type engine for convenient computation of forces is shown in Fig. 2. From Fig. 2, the inertia forces due to reciprocating parts of piston 1 and piston 2 along the line of stroke are expressed as[5]

$$F_{if1} = m \cdot \omega^2 \cdot R[\cos(\theta - \alpha) + \frac{R}{L_c}\cos 2(\theta - \alpha)] \quad (1)$$

$$F_{if2} = m \cdot \omega^2 \cdot R[\cos(\theta + \alpha) + \frac{R}{L_c}\cos 2(\theta + \alpha)] \quad (2)$$

where m is mass of reciprocating parts per cylinder(kg), ω is angular velocity of crank(rad/s), R is radius of crank(m), θ is inclination of the crank to the vertical(rad), and L_c is length of the connecting rod(m).

The V12-type engine has 12 pistons, with six couple of pistons placed equal distances apart. Engine manufacturers keep the parameters of their engines confidential, but these parameters vary for each company. Therefore, to find the solution, we shall use a mathematical model and then eliminate parameters involved with it.

In the Eq. (1) and Eq. (2) above, the vertical components and horizontal components are defined as follows:

$$\begin{aligned} F_V = 2m \cdot \omega^2 \cdot R(&\cos\theta \cdot \cos^2\alpha \cdot \sum_1^6 \cos\phi_i \\ &- \sin\theta \cdot \cos^2\alpha \cdot \sum_1^6 \sin\phi_i \\ &+ \frac{R}{L_c}\cos 2\theta \cdot \cos 2\alpha \cdot \cos\alpha \cdot \sum_1^6 \cos 2\phi_i \\ &- \frac{R}{L_c}\sin 2\theta \cdot \cos 2\alpha \cdot \cos\alpha \cdot \sum_1^6 \sin 2\phi_i) \end{aligned} \quad (3)$$

$$\begin{aligned} F_H = 2m \cdot \omega^2 \cdot R(&\sin\theta \cdot \sin^2\alpha \cdot \sum_1^6 \cos\phi_i \\ &+ \cos\theta \cdot \sin^2\alpha \cdot \sum_1^6 \sin\phi_i \\ &+ \frac{R}{L_c}\sin 2\theta \cdot \sin 2\alpha \cdot \sin\alpha \cdot \sum_1^6 \cos 2\phi_i \\ &+ \frac{R}{L_c}\cos 2\theta \cdot \sin 2\alpha \cdot \sin\alpha \cdot \sum_1^6 \sin 2\phi_i) \end{aligned} \quad (4)$$

where F_v is vertical force, F_H is horizontal force, and ϕ_i is firing angle of the ith piston.

We can easily find the equilibrium conditions from Eq. (4) as

$$F_V = 0; \; F_H = 0 \quad (5)$$

Equation (5) can be rewritten as follows

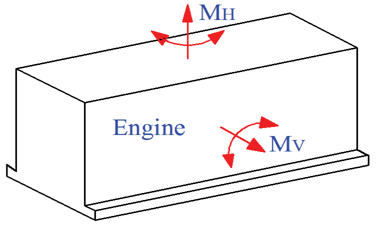

Fig. 1 External forces and external moments in diesel engine

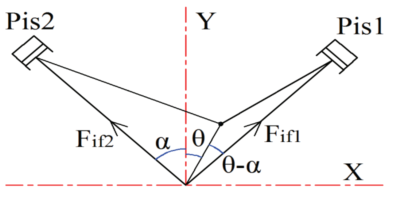

Fig. 2 Model of V-piston of diesel engine

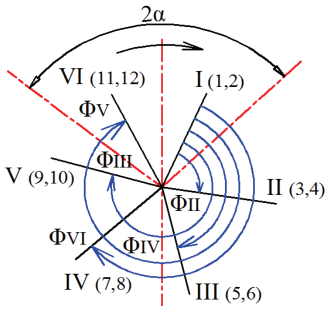

Fig. 3 Operating angle of V12-type engine with V angle 2α=90°

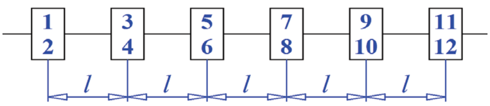

Fig. 4 Configuration of couples of pistons in V12-type engine

$$\sum\cos\phi_i = 0;\ \sum\sin\phi_i = 0; \quad \sum\cos 2\phi_i = 0;\ \sum\sin 2\phi_i = 0 \tag{6}$$

In Fig. 3, the firing angle for a V12-type engine with a V angle of 90°. To use the forces to calculate the moments caused by these forces, we shall use the fact that the distances between each adjacent couple of pistons are equal, as shown in Fig. 4. This figure shows the configuration of the pistons of the V12-type engine.

Let the distance between each adjacent couple of pistons be l. Combining with the firing angle(Fig. 3) and applying the moving force principle to the first couple of pistons, we obtain the vertical unbalanced moment M_V and horizontal unbalanced moment M_H as

$$M_V = m\cdot\omega^2\cdot R\cdot l(-5\cos\theta + 1.72\sin\theta) \tag{7}$$

$$M_H = -m\cdot\omega^2\cdot R\cdot l(5\sin\theta + 1.72\cos\theta + 3.44\sqrt{2}\frac{R}{L_c}\cos 2\theta) \tag{8}$$

Now, to find the vertical force, the relative positions of the mounts with respect to the center of mass of the engine should be noted. Given this information, we can find the exact value of the vertical force that we shall further use to design the mounts.

From Fig. 5, the displacement of piston S is defined as

$$S = (R + \frac{R^2}{4L_c}) - R(\cos\theta + \frac{R}{4L_c}\cos 2\theta) \tag{9}$$

It can be rewritten by using MacLaurin's expansion as[6]

$$S = R(a_0 + \cos\theta + a_2\cos 2\theta + a_4\cos 4\theta + a_6\cos 6\theta + \cdots) \tag{10}$$

where

$$a_0 = \frac{L_c}{R}\left[1 + \frac{1}{4}\left(\frac{R}{L_c}\right)^2 + \frac{3}{64}\left(\frac{R}{L_c}\right)^4 + \frac{5}{256}\left(\frac{R}{L_c}\right)^6\right] \tag{10a}$$

$$a_2 = \frac{L_c}{R}\left[\frac{1}{4}\left(\frac{R}{L_c}\right)^2 + \frac{1}{16}\left(\frac{R}{L_c}\right)^4 + \frac{15}{512}\left(\frac{R}{L_c}\right)^6\right] \tag{10b}$$

$$a_4 = -\frac{L_c}{R}\left[\frac{1}{64}\left(\frac{R}{L_c}\right)^4 + \frac{3}{256}\left(\frac{R}{L_c}\right)^6\right] \tag{10c}$$

$$a_6 = \frac{L_c}{R}\left[\frac{1}{512}\left(\frac{R}{L_c}\right)^6\right] \tag{10d}$$

The relationship between the velocity and pressure of gas in terms of the torque acting on the piston T_{piston}, of piston can be given as

$$T_{piston} = P\cdot A_p\left(\frac{dS}{d\theta}\right) = P\cdot A_p\left(\frac{dS}{dt}\right)\cdot\left(\frac{dt}{d\theta}\right) = P\cdot A_p\frac{\dot{S}}{\dot{\theta}} \tag{11}$$

Using Eq. (10) in this equation, we further have

$$T_{piston} = -P\cdot A_p\cdot R(\sin\theta + 2a_2\sin 2\theta + 4a_4\sin 4\theta) \tag{12}$$

Equation (12) makes it convenient to compute the gas force required to impact the mount. Equation (7)

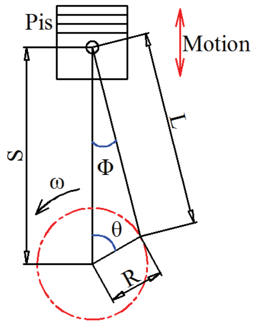

Fig. 5 Model of single piston

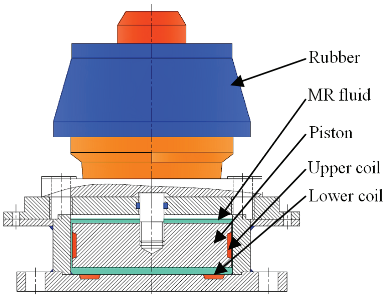

Fig. 6 Schematic configuration of MR mount

and (12) are used to calculate dynamic force and gas force respectively. To find the acceleration, it is assumed that the system vibrations are harmonic. With this assumption, the dynamic force acting on the system can be computed. The total vertical force acting on the mount is the sum of the values of the vertical unbalanced force and the gas force, which are evaluated from Eq. (7) and (12) respectively. To be on the safer in practical applications, this value should be multiplied by a safety factor which in the range of 1.1 to 1.2.

3. Design and Optimization

3.1 Design of MR Mount

In this study, an MR mount has been designed by the addition of an MR piston part to a rubber mount. A schematic configuration of the proposed MR mount is shown in Fig. 6. This design is used to control dynamic vertical force that appears in the operation of the engine. We chose a mount model that uses two coils; one is in the longitudinal body of piston and the other is at the bottom of the housing. The parameters of these two coils are taken to be equal in the calculation of mount. Other parameters of this MR mount are computed using the initial values in the mathematical model of this mount.

Table 1 shows the parameters calculated in this research, while the model for calculating the damping force in an MR mount is shown in Fig. 7. In the figure, L_p is the length of the pole coil, L is the height of the piston, R_{c1} is the radius of the first coil, R_p is the radius of the piston, R_{c2} is the radius of the second coil, R_{tc} is the total radius of the second coil, d_{o1} is the width of the gap between inside the vertical housing and the piston and d_{o2} is bottom gap between the piston and the housing.

Table 1 Calculated parameters of mount

Parameter	Value
Initial mass	4400 kg
Type of MR fluid	MR132DG
Length of magnetic pole(L_p)	10 mm
Radius of piston(R_p)	90 mm
Diameter of piston rod	27 mm
Current(I)	2.5 A
Number of turns(N)	104
Radius of first coil(R_{c1})	85 mm

The type of MR fluid used in this research is MR132DG, which has a medium yield stress given by

$$\tau_y = 52.962B^4 - 176.51B^3 + 158.79B^2 + 13.708B + 0.1442 \quad (13)$$

where τ_y is

the yield stress(kPa) of MR132DG fluid in a magnetic field, with flux density B(T). To evaluate optimization results, it should be noted that

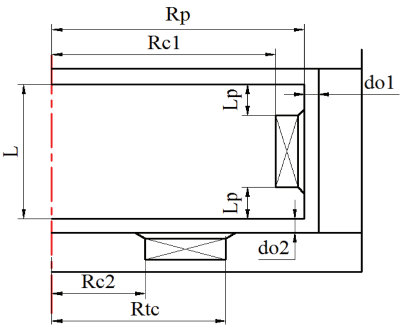

Fig. 7 Geometry for calculating force of MR mount

the saturation of steel is not over 1.8 T, and MR132DG fluid is less than 1.65 T[4].

Damping force F_d of this mount is the sum of four forces, given as[2]

$$F_d = F_{re} + F_{bfm} + F_{vfm} + F_{sm} + F_{sqm} \tag{14}$$

where F_{re} is the force on the rubber mount that can stand static load, F_{bfm} is the force in the flow mode of MR fluid at bottom of the housing, F_{vfm} is the force in the flow mode of MR fluid at longitudinal piston, F_{sm} is the force in the shear mode of MR fluid at longitudinal piston, F_{sqm} is the force in the squeeze mode at the bottom of the housing.

It is observed that the force from the bottom includes two forces. One is the squeeze force in the initial state of the MR fluid whose magnitude is set up in a short time span with less intense vibrations. The other is the flow force that appears when the system has vibrations of large amplitudes, as shown in Eq. (16). The forces may be expressed as[2]

$$F_{sqm} = \frac{3\pi\mu R_p^4}{2d_{o2}^3}\dot{u} + \frac{4\pi\tau_y R_{c2}^3}{3d_{o2}} sign\,(\dot{u}) \tag{15}$$

$$F_{bfm} = \frac{3\pi\mu R_p^4}{2(d_{o2}+u)^3}\dot{u} + \frac{4\pi\tau_y R_{c2}^3}{3(d_{o2}+u)} sign(\dot{u}) \tag{16}$$

$$F_{sm} = \frac{2\pi R_p \mu L}{d_{o1}}\dot{u} + 2\pi R_p L_p \tau_y sign\,(\dot{u}) \tag{17}$$

$$F_{vfm} = \frac{12 A_p \cdot \mu L}{2\pi R_p \cdot d_{o1}^3}\dot{u} + \frac{2 A_p \cdot L_p \cdot \tau_y}{d_{o1}} sign\,(\dot{u}) \tag{18}$$

where μ is the off-state plastic viscosity (Pa.s), $\dot{u}$ is the velocity of pistion (m/s), and u is its displacement(m).

3.2 Optimization of MR Mount

Based on the initial results obtained in the calculation of MR mount, the values are optimized to design a practical mount. The software APDL is used for optimization process. First, the order of the optimization module is used to optimize the design of MR mount. In this module, various steepest descent and conjugate direction searches are enforced for each iteration until convergence is reached. A flow chart for optimization of MR mount is shown in Fig. 8.

Because this research focuses on the vertical forces from the engine, it is important to determine the distance between the bottom of housing and the piston. This area will create a force larger than other areas, that is enough to maintain a high force from MR fluid when current is supplied to the system. As shown in Fig. 9, the total force is inversely proportional to the distance between the piston and the bottom of the housing. Although the recommended values for the distance between the piston and the bottom of the housing are between 1 mm to 2 mm, it is difficult to control its value. Hence, its values is chosen between 1 mm to 5 mm. It is noted that the larger is the distance between the piston and the bottom of the housing, the smaller is the damping force in MR mount.

In this MR design, the squeeze mode occurs at the bottom for a short time span. Figure 10 shows the relationship between the piston-bottom distance and the squeeze mode which does not change. Since the squeeze mode does not remain all through the operation of the system, we use the next mode for the optimization progress. This state is the flow mode, which is the main force for

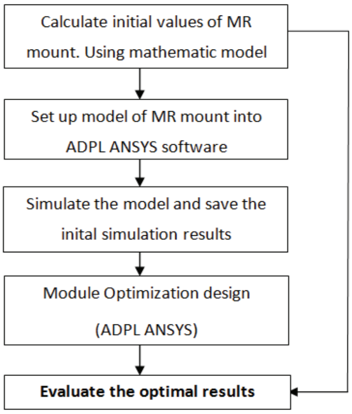

Fig. 8 Flow chart for optimization of MR mount

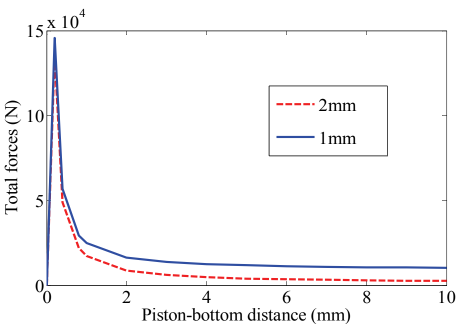

Fig. 9 Relationship between piston-bottom distance and total forces

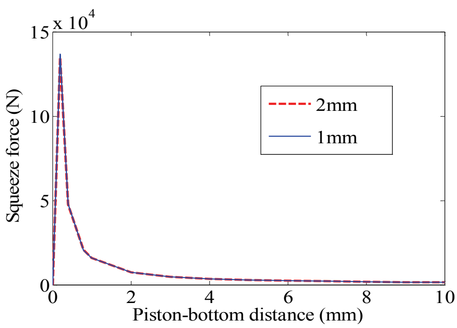

Fig. 10 Relationship between piston-bottom distance and squeeze forces

with standing high vertical loads, and is shown in Eq. (16).

The objective of damping force in the MR mount is 26400 N. Three main parameters that influence the progress of optimization are the bottom gap

Table 2 Optimized parameters

Parameter	Symbol	Min	Max	Result	Unit
Bottom gap between piston and housing	d_{o2}	1	5	2	mm
Length of second coil ($R_{tc}-R_{c2}$)	L_{c2}	20	33	20	mm
Radius of second coil	R_{c2}	47	65	60	mm
Total forces	TTF		26400	26703	N

between the piston and housing d_{o2}, radius of the second coil R_{c2}, total radius of second coil R_{tc}. These parameters directly impact the optimization done to obtain the objective force. The function used to find optimal values and the convergence can be defined by the following equation[(4)]:

$$OBJ=\frac{1}{F_{total}} \tag{19}$$

where F_{total} is objective total force of mount(N), and $F_{total}=F_d$. In Table 2, optimized force is found to be 26703 N, with a tolerance of 1 %.

4. Conclusion

In this work, a new type of MR mount that can be used for vibration control in the diesel engines of ships was proposed and optimally designed. The excitation force of a V-type diesel engine was analyzed, and the appropriate size of an MR mount that can produce an actuating force of over 25000 N was designed. In addition, the actuating force was maximized under geometric constraints using the optimization tool of the APDL software. The optimally designed MR mount will be manufactured and its performance will be evaluated in a future study.

References

(1) Nguyen, Q. H., Choi, S. B., Lee, Y. S. and Han, M. A., 2009, An Analytical Method for Optimal Design of MR Valve Structures, Smart Materials & Structures,

Vol. 18, No. 9, pp. 1~12.

(2) Brigley, M. J., 2006, A Multi-mode Magneto-rheological Axial Isolator, Thesis of Master of Science, University of Maryland, USA.

(3) Frank M. White, Fluid Mechanics, 2006, Fourth Edition, Mc-Graw Hill, New York.

(4) Nguyen, Q. H. and Choi, S. B., 2010, Optimal Design of an Automotive Magnetorheological Brake Considering Reometric Dimensions and Zero-field Friction Heat, Smart Materials and Structures, Vol. 19, N. 11, pp. 1~11.

(5) Khurmi, R. S. and Gupta J. P., 2008, Theory of Machines, Eurasia Publishing House.

(6) Park, J. G., Jeong, W. B. and Seo, Y. S., 2002, Optimal Design of Crank Angles for Reducing the Excitation Forces in a Diesel Engine, Transactions of the Korean Society for Noise and Vibration Engineering, Vol. 12, No. 2, pp. 108~115.

(7) Huyndai Heavy Industries Co. ltd, 2012, Vibration of Marine Diesel Engine.

(8) Yang, G., Spencer, B. F., Jung, H. J. and Carlson, J. D., 2009, Dynamic Modeling of Large-scale Magneto-rheological Damper System for Civil Engineering Applications, Journal of Engineering Mechanics, Vol. 130, No. 9, pp. 1107~1114.

(9) Yang, G., Spencer, B. F., Carlson, J. D. and Sain, M. K., 2002, Large-scale MR Fluid Dampers: Modeling and Dynamic Performance Considerations, Engineering Structures, Vol. 24, pp. 309~323.

(10) Choi, S. B., Hong, S. R., Sung, K. G. and Sohn, J. W., 2008, Optimal Control of Structural Vibrations Using a Mixed-mode Magnetorheological Fluid Mount, International Journal of Mechanical Sciences, Vol. 50, No. 3, pp. 559~568.

(11) Bass, B. J. and Christenson, R. E., 2007, System Identification of a 200 kN Magnetorheological Fluid Damper for Structural Control in Large-scale Smart Structures, Proceedings of the 2007 American Control Conference, pp. 2690~2695.

(12) Hong, S. R., Choi, S. B. and Lee, D. Y., 2006, Comparison of Vibration Control Performance Between Flow and Squeeze Mode ER Mounts: Experimental Work, Journal of Sound and Vibration, Vol. 291, pp. 740~748.

(13) Hong, S. R., Choi, S. B., Choi, Y. T. and Wereley, N. M., 2005, Non-dimensional Analysis and Design of a Magnetorheological Damper, Journal of Sound and Vibration, Vol. 288, pp. 847~863.

(14) Hong, S. R., Wereley, N. M., Choi, Y. T. and Choi, S. B., 2008, Analytical and Experimental Validation of a Nondimensional Bingham Model for Mixed-mode Magneto-rheological Dampers, Journal of Sound and Vibration, Vol. 312, pp. 399~417.

(15) Spencer, B. F., Yang, G., Carlson, J. D. and Sain, M. K., 1998, Smart Dampers for Seismic Protection of Structures:a Full-scale Study, The Second World Conference on Structural Control, Kyoto, Japan.

(16) Farjoud, A., Craft, M., Burke, W. And Ahmadian, M., 2011, Experimental Investigation of MR Squeeze Mounts, Journal of Intelligent Material Systems and Structures, Vol. 22, pp. 1645~1651.

(17) Zhang, X. J., Farjoud, A., Ahmadian, M., Guo, K. H. and Craft, M., 2011, Dynamic Testing and Modeling of an MR Squeeze Mount, Journal of Intelligent Material Systems and Structures, Vol. 22, No. 15, pp. 1717~1728.

(18) Gołdasz, J. and Sapiński, B., 2011, Modeling of Magneto-rheological Mounts in Various Operating Modes, Acta Mechanica et Automatica, Vol. 5, No. 4, pp. 29~39.

(19) Popp, K. M., Zhang, X. Z., Li, W. H. and Kosasih, P. B., 2009, MRE Properties under Shear and Squeeze Modes and Applications, Journal of Intelligent Material Systems and Structures, Vol. 21, No. 15, pp. 1471~1477.

(20) Kuzhir, P., L´opez-L´opez, M. T., Vertelov, G., Pradille, C. and Bossis, G., Oscillatory Squeeze Flow of Suspensions of Magnetic Polymerized Chains, Journal of Physics: Condensed Matter, Vol. 20, No. 20, pp. 1~5.

Seung-Bok Choi received a BS degree in Mechanical Engineering from Inha University in 1979, MS and PhD degrees from the Michigan State University in 1986 and 1990 respectively. He is currently Dean of the School of Mechanical Engineering and fellow professor in Inha

University, Incheon. He is a fellow of NAEK (National Academy Engineering of Korea), KAST (The Korean Academy of Science and Technology). His research interests are robust controller design for linear and nonlinear systems, and design and control of various control systems using smart actuators.

Xuan-Phu Do received his BS and MS degree in Mechanical Engineering from Ho Chi Minh City University of Technology in 2005 and 2008 respectively. He is currently studying PhD course at Inha University. His research interests are robust controller design, system using smart materials, and vibration control.

Joon-Hee Park received the BS degree in Mechanical Engineering from Inha University in 2012. He is currently studying for an MS course at Inha University in Mechanical Engineering. His research interests are robust controller design for linear and nonlinear systems, design and control of mechanical systems using smart materials.