Design and Evaluation a Multi-coil Magneto-rheological Damper for Control Vibration of Washing Machine

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Key Words : Magnetorheological (MR) damper, washing machine, modified sliding mode control, design optimization

ABSTRACT

This paper presents a design of magnetorheological (MR) damper for control vibration of washing machine. This design is based on the requirements such as small dimensions with high damping force, and minimal consumed energy. The MR damper is designed using the shear mode of MR fluid, and Bingham plastic model is used for optimization process. In this design, a multi-coil design is adopted for damper to enhance damping force and reduce optimally structural parts. In optimization process, ADPL (Ansys Parametric Design Language) program is applied. Base on the optimal parameters, MR damper is manufactured and tested. In evaluation of MR damper, a modified sliding mode control is formulated and applied in both simulation and experiment. Results of experiment show that the MR damper satisfy the requirement of damping force for vibration control of washing machine.

요 약

이 논문은 세탁기의 진동제어를 위한 MR 댐퍼의 설계과정을 제시한다. 이 연구에서는 MR 댐퍼의 작은 크기와 함께 높은 댐핑력과 저전력을 소비하도록 설계되었다. MR 댐퍼는 전단모드를 사용하도록 제안되었고 빙햄모델을 이용하여 최적설계를 진행하였다. 이 과정에서 멀티코일이 댐핑력을 높이기 위해 적용되었고 최적화된 구조를 찾아내기 위하여 APDL 을 이용하여 최적설계가 수행되었다. 이 후 최적의 수치 값들을 통해 제작하여 실험을 수행하였다. 이 때 슬라이딩모드 제어기를 적용하여 시뮬레이션과 제어실험을 모두 수행하였다. 실험의 결과로부터 MR 댐퍼가 세탁기의 진동제어를 위한 요구 댐핑력을 만족시켜줌을 확인하였다.

1. Introduction

Magnetorheological (MR) fluid is a well-known smart material which can be applied in various devices such as damper, clutch, mount and other actuators. The property of MR fluid is its ability in change the initial state from fluid to semi-solid when the system subjected to external magnetic field, inversely. This characteristic affects to MR fluid to absorb vibration, and increase the viscous property in a small time (milliseconds). In study of MR damper, there are numerous researches in both theory and application. Design and experiment of MR damper were presented for automobile suspension [1]. This research studied a simple structure of damper including piston, single coil, and housing. In related model of MR damper, a review of models for calculation damping force is also presented in [2], which concentrated in parametric dynamic modeling.

MR valve structure is applied in design MR damper, which is presented in [3, 4 ,7]. The MR valve structure in these studies was annular and radial fluid flow resistance channels. An application for tremor suppression of MR damper is presented in [5], in which MR damper is designed in small scale. The structure in this design was simple based on the piston-housing structure. Strategy for design MR smart structure is studied in [6], which analyzed piston-housing structure. It presented such as choosing a MR fluid, calculating response time, saturation analysis, power ratio, optimal design. Saturation problem of MR damper also is solved in [8], where piston-housing structure is analyzed. The saturation is evaluated to guarantee the objective of damping force. The non-dimensional analytical method for optimizing MR valve is also presented in [9], which concentrated on integral and bypass configuration. A review of all structure in design MR damper is presented in [10] which shows structures such as piston-housing, valve, and modified of them.

In [11,13], structures of multi-coil MR damper were presented in which coils were wound outside of core, and rod of damper as a piston.

From the above views, the structures of MR damper are almost piston-housing type with single coil/multi-coil which is integrated at outside piston, and valve structures such as annular, radial, integral, bypass configurations. Hence this study presents a new design of MR damper using multi-coil which is designed on a core with minimal volume MR fluid which is applicable to vibration control of washing machine. The paper is organized as follows. In section 2, we present configuration and optimization of the proposed MR damper. Section 3 describes analysis system, proposed control, simulation system, and experimental results. Finally, our conclusion is drawn in section 4.

2. Configuration and Optimization

The proposed damper will be used for front loaded washing machine. In addition, the requirements for this design are small volume of MR fluid, and its structure is not large for assembling inside of washing machine. As was reviewed in the previous section, the proposed design of MR damper is a new and shown in Figure 1. In Figure 1(a), damper has four coils which are separated and theirs function as excited sources for establishing magnetic field. The damping force of MR damper is changed depending upon the intensity of applied current. Axis of damper plays a role as a piston in this design, and MR fluid is inserted between piston and core in which multi-coils are fixed. Hence the components of core damper include six parts as shown in Figure 1(b). It is noteworthy that the proposed damper is operated based on the shear mode of MR fluid.

Model of the proposed damper for optimization is built in a half section with two coils. These coils are arranged in 45^0 in one quarter of section. This arrangement is reasonable for multi-coils in this proposed damper and prevents the disturbance of magnetic field, i.e magnetic lines. The initial parameters for optimization are presented in Table 1. It has four parameters which influence to the model such as r_1, r_2, d_0, l. These parameters are shown in Figure 2. The parameter l is the height of coil which is not shown in Figure 2.

Materials for the proposed damper are adopted as follows:
+ Housing, piston, core: Steel 1018 with maximal saturation magnetic 2 Tesla.
+ MR fluid: MR132DG [Lord Company] with maximal saturation magnetic 1.65 Tesla.

Yield stress of MR132DG is expressed as follows

[11]:

$$\tau_y = 0.15 + 0.30858H + 2.83544e^{-4}H^2$$
$$- 5.34429e^{-6}H^3 + 9.20846e^{-9}H^4 \quad (kPa) \tag{1}$$

where τ_y is the yield stress (kPa), H is the magnetic field intensity (kA/m). In the proposed damper, damping force is calculated based on the relation of geometric parameters, yield stress by viscosity from MR fluid between piston and core, and Coulomb friction of seals. The damping force F_d is expressed as follows:

$$F_d = 2\pi r_3 l \tau_y + 2f_s \tag{2}$$

where r_3 is the radius of piston (m), l is the height of core damper (m), f_s is the friction force of seals (N).

The friction of seals in this damper is estimated as follows [11]:

$$f_s = 87.5l \tag{3}$$

On the other hand, the damping force of damper for washing machine can be determined as follows [11]:

$$|F_d| = \frac{k\pi\xi m_u r^3 R}{2m} \sqrt{\frac{1}{(1-r^2)^2 + (2\xi r)^2}} \tag{4}$$

where k is the spring stiffness (N/m), m is the mass of the suspended stub assembly (kg), R is the radius of tub (m), ξ is the damping ratio, r is the equivalent required damping force, $r = \sqrt{1-\xi^2}$, and m_u is the unbalanced mass (kg). From the equation (4), the objective damping force for the proposed damper is nearly 120N. The objective function is setup as follows:

$$OBJ = \frac{1}{F_d} \tag{5}$$

Results of optimization are shown from Figure 3 to Figure 6. In Figure 3, meshing model and magnetic lines after optimizing are shown. Figure 3(a) show that the meshing model is clear and no confused areas which cannot show meshed areas are appeared. The meshing step affects to the results of optimization process.
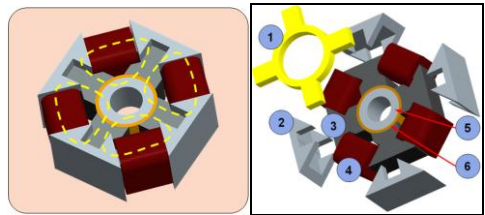
Fig. 1 Configuration of multi-coil MR damper: (a) Distribution of coils; (b) Components of core: (1) Upper cover, (2) Bracket, (3) Bobbin, (4) Coil, (5) Piston, (6) MR fluid.
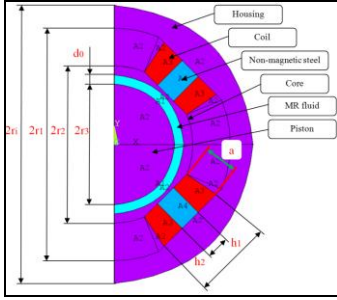
Fig.2 Parameters of optimization model.

Table 1 Initial parameters and optimized parameters.

Parameter (mm)	Initial value	Setup for opt.	After optimization
Housing diameter r_i	18.5	Fixed	18.5
Core diameter r_1	16.5	**15~17**	**15.4**
Bottom coil dia. r_2	12	**10~13**	**10.4**
Piston diameter r_3	8.0	Fixed	8.0
Damping gap d_0	1.5	**1.2~1.5**	**1.2**
Width coil a	4.65	Fixed	4.65
Length coil h_1	9.79	Fixed	9.79
Length coil l	70	**65~80**	**78**

If the meshing is not smooth, optimized parameters cannot satisfy the requirements or the progress is stop undesirably. It is remarked that the model for optimization does not require contribution fully. The section of damper is symmetric, so a half section is established. In Figure 3(b), distribution of magnetic lines after optimizing is regular, and this will guarantee that the damping force of the system can achieve the desired value.

In Figure 4(a), values of magnetic flux density are shown. The values in MR fluid area are less than 0.5 Tesla and less than the standard of saturation 1.65 Tesla. In the piston area, the magnetic flux densities are also not over 0.5 Tesla.


(a) (b)
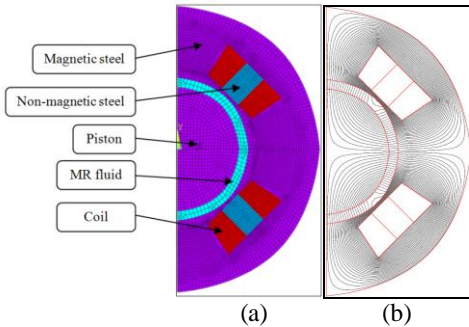
Fig. 3 Conditions of optimization process: (a) Meshing; (b) Magnetic lines.

(a) (b)
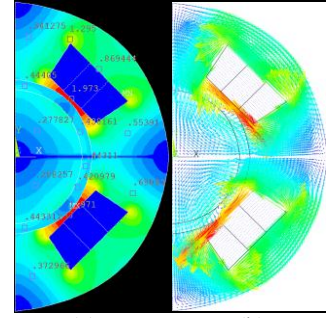
Fig. 4 Results of optimization process: (a) Magnetic flux density (Tesla); (b) Magnetic flux density vector.

In core area, maximal value is 1.976 Tesla which is less than the standard of steel's saturation 2 Tesla. In housing area, the value is not over 1.6 Tesla. In Figure 4(b), vector of magnetic flux density is shown. From this figure, the magnetic vectors are distributed regularly in gap of MR fluid. So the results of optimization are corresponding to the requirements.

The result of iteration is depicted in Figure 5. It includes 11 iterations for finding optimized four parameters such as r_1, r_2, d_0, l.
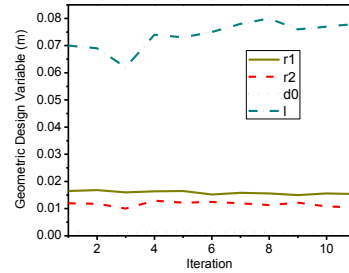
Fig. 5 Iteration of optimization process.

3. Dynamic Modeling of Washing Machine

3.1. Analysis System

The governing equation of the washing machine shown in Figure 6 can be represented as follows [11]:

$$m\ddot{u} + c\dot{u}\left[sin^2(\varphi + \beta_2) + sin^2(\varphi - \beta_1)\right] + ku\left[sin^2(\varphi + \alpha_1) + sin^2(\varphi - \alpha_2)\right] = F_u \quad (6)$$

where m is the mass of the suspended tub assembly including the drum, laundry, shaft, counter weight, rotor and stator (kg), c is the damping coefficient of each damper (N/s), k is the stiffness of each spring (N/m), φ is the angle of an arbitrary direction in which the vibration is considered, u is the displacement of the tub center in the u-direction (m). The excitation force due to unbalanced mass in the u-direction, F_u, is determined as follows:

$$F_u = m_u \omega^2 R_u \cos \omega t \quad (7)$$

where m_u and R_u are the mass and radius from the rotation axis of the unbalanced mass. By choosing $\alpha_1 + \alpha_2 = 90^0$ and $\beta_1 + \beta_2 = 90^0$, the equation (6) can be rewritten as follows:

$$m\ddot{u} + c\dot{u} + ku = F_u \tag{8}$$

(a)   (b)
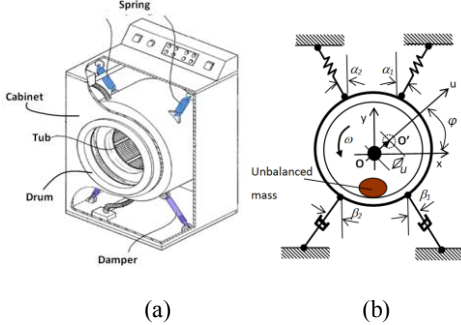
Fig. 6 Washing machine model: (a) Configuration of the drum-type washing machine, (b) Mechanical modeling of washing machine.

3.2. Controller Design for Vibration Control

Base on the equation (8), the system can be symbolized as in Figure 7. For controlling vibration of the system, a modified sliding mode control is suggested. The sliding surface is defined as follows [12]:

$$s = \theta u + \dot{u} \tag{9}$$

where θ is a positive constant. Relation of the equation (8) and (9) is depicted as:

$$\dot{s} = \left(\theta - \frac{c}{m} + \frac{k}{\theta m}\right)\dot{u} + \frac{F_u}{m} \tag{10}$$

From the equation (7) and (10), the unbalanced force can be rewritten as:

$$F_u = -F_{max}sat(s) \tag{11}$$

where F_{max} is a positive constant, $sat(s)$ is a saturation function which is used to limit the chattering in the system.

The damping force of MR damper can be written as

$$F_{MR} = c\dot{u} + Fsat(\dot{u}) \tag{12}$$

where F is force related the yield stress of MR fluid, c is the viscosity of MR fluid. It is noteworthy that the force F has two values such as 0 or F_{max}. From Figure 7, the equation of the system can be defined as:

$$m\ddot{u} + F_{MR} + ku = 0 \tag{13}$$

Substituting the equation (12) into (13):

$$m\ddot{u} + c\dot{u} + ku = -Fsat(\dot{u}) \tag{14}$$

Base on the equation (14), the equation (9) is rewritten as follows:

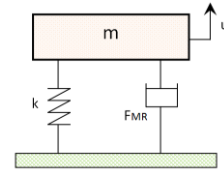$$\dot{s} = \left(\theta - \frac{c}{m} + \frac{k}{\theta m}\right)\dot{u} - \frac{Fsat(\dot{u})}{m} \tag{15}$$

When $s > 0$, from equation (9), the following inequality is determined as:

$$\dot{u} > -\theta u \tag{16}$$

From (11), it has two subcases such as $\dot{u} > 0$, $\dot{u} < 0$. With the subcase $\dot{u} > 0$, the equation (15) is expressed as:

$$\dot{s} = \left(\theta - \frac{c}{m} + \frac{k}{\theta m}\right)\dot{u} - \frac{F}{\beta_1 m} \tag{17}$$

where β_1 is a small positive constant. The condition for sliding mode control is given by $s\dot{s} < 0$.

Fig. 7 Simple model of washing machine.

Hence, value $\dot{s}$ must be less than zero, and this can be written as follows:

$$F > \beta_1 m \left|\theta - \frac{c}{m} + \frac{k}{\theta m}\right|\dot{u} \tag{18}$$

With the subcase $\dot{u} < 0$, the equation (15) is rewritten as follows:

$$\dot{s} = \left(\theta - \frac{c}{m} + \frac{k}{\theta m}\right)\dot{u} \tag{19}$$

In equation (19), the value F must be zero which satisfies the condition of $s\dot{s} < 0$.

When $s < 0$, from equation (4), the following inequality is determined as:

$$\dot{u} < -\theta u \tag{20}$$

It also has two subcases such as $\dot{u} > 0$, $\dot{u} < 0$.

With the subcase $\dot{u} > 0$, the equation (15) is rewritten as $\dot{s} = \left(\theta - \frac{c}{m} + \frac{k}{\theta m}\right)\dot{u}$ which value F is zero for satisfying $s\dot{s} < 0$. The another subcase, $\dot{u} < 0$, the equation (15) is $\dot{s} = \left(\theta - \frac{c}{m} + \frac{k}{\theta m}\right)\dot{u} + \frac{F_{max}}{\beta_1 m}$ in which value F is maximal for satisfying $s\dot{s} < 0$.

As a result, the force F can be summarized as follows:

$$F = \begin{cases} F_{max} & if \ (s > 0, \dot{x} > 0) or \ (s < 0, \dot{x} < 0) \\ 0 & if \ (s \geq 0, \dot{x} \leq 0) or \ (s \leq 0, \dot{x} \geq 0) \end{cases} \tag{21}$$
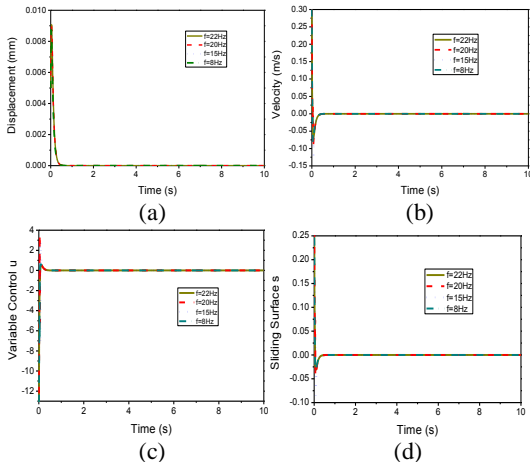
3.3. Simulation and Experiment

3.3.1. Simulation Results

Parameters for simulation the system are listed in Table 2. Using these parameters and Simulink model of Matlab, responses of the system are determined such as displacement, viscosity, variable control and sliding surface.

Table 2. Parameters for simulation system.

| Parameter	Unit	Value
Mass of washing machine m	kg	40
Unbalanced mass m_{u1}	kg	15
Excited frequency f	Hz	8,15,20,22
Unbalanced radius R_u	m	0.13
Constant θ	[]	6
Damping coefficient c	N/s	1920
Stiffness spring k	N/m	10000

Results of simulation are shown in Figure 8. These simulations are carried out without disturbances in the system. Exciting frequencies are used as follows; 8 Hz, 15 Hz, 20 Hz, 22 Hz. From Figure 8, the modified control shows that the vibration of damper is reduced in a small time. In Figure 8(a), with the initial values of displacement 0.005 m, vibration of displacement decays to zero after less 1 second. For different frequencies, the responses of displacement are similar. In Figure 8(b), velocities of different frequencies are also similar showing the stability in a small time. Variable control u in Figure 8(c) and sliding surface **s** in Figure 8(d) are also converge to zero after less 1 second. These results show that the proposed control is efficient in control the vibration of MR damper.

Fig. 8 Results of simulation: (a) Displacement versus Time; (b) Velocity versus Time; (c) Variable control u versus Time; (d) Sliding surface s versus Time.

3.3.2. Experimental Results

The experiment setup is shown in Figure 9. Results of experiment are depicted in Figure 10. In Figure 10(a), (b), (c) damping force of the system varies according to the excited frequency. The excited frequencies are used as follows ; 8, 25, 20, 22 Hz. The damping force at 0, 8, 15, 20, 22 Hz is 17.13 N, 79.55 N, 93.75 N, 100 N, 101.48 N, respectively. At excited frequency 22 Hz, the maximal controlling current is 1.754 A.
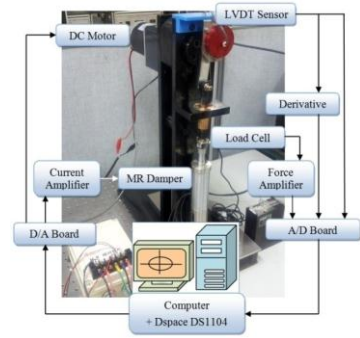
Fig. 9 Experimental setup.

Response time of the system is expressed in Figure 10(d) which is 0.035 s at on-state, and 0.0635 s at off-state. It is noteworthy that the response time belongs to the property of MR fluid and damper configuration. It is proven that from the simulation and experiment that the proposed MR damper is very effective for vibration control of washing machine subjected to various exciting frequencies.
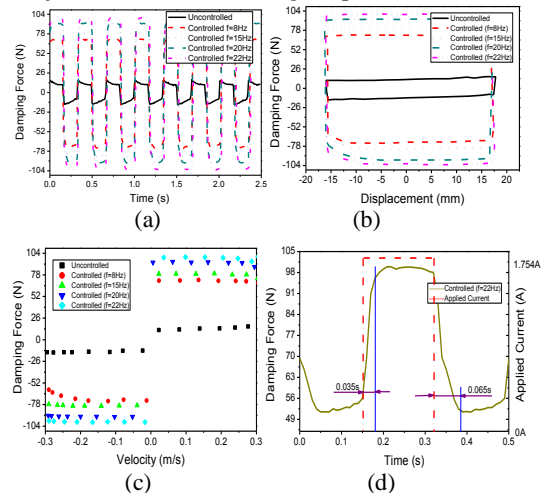
Fig. 10 Results of experiments: (a) Damping force versus Time; (b) Damping force versus Displacement; (c) Damping force versus Velocity; (d) Response time of system.

4. Conclusion

The configuration of multi-coil MR damper is

devised and manufactured through the optimization method for vibration control of washing machine. This proposed configuration is develop a new direction toward design a small damper with high damping force, minimal volume of MR fluid and minimal consumed energy. From the optimization results, the structure of MR damper is designed to satisfy the requirements. The optimized magnetic flux density of damper is not over the standard of materials saturation, and the magnetic lines are optimized in full area of piston which guarantees efficiency of damping force. Simulations of modified sliding mode control are also show that the system is robust and stable. Results of experiment have also proved that the proposed system is very good performance and compatible with the simulation. Therefore, the proposed design of MR damper is well-designed and can be effectively applied washing machine for vibration control.

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