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Design of a self-power magneto-rheological damper in shear mode for front-loaded washing machine

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Abstract
In this research work, a self-power shear-mode magneto-rheological (MR) damper that can replace conventional passive damper of a front-loaded washing machine is designed and experimentally evaluated. The self-power damper consists of two main components: the damping component and the self-power one. The damping component is composed of an inner housing with a thin wall, on which the coils are wound directly, and an outer housing to cover the coil and to create a closed magnetic circuit of the damper. The gap between the inner housing and the shaft is filled with MR fluid to create damping force. The self-power component consists of magnetized permanent magnets fastened together on the moving shaft and a slotted stator core covered by an outer housing. The coils are wound directly on the slots of the stator core and connected to the coils of the damping component to generate the required current when the shaft moves reciprocally. After an introduction to suspension system for front-loaded washing machine, the configuration of the self-power MR damper for front-loaded washing machine is proposed. Optimization of the proposed self-power MR damper is then performed considering required damping force, off-state friction force, size, power consumption and low cost of the damper. From the optimal results, simulated performance of the optimized MR damper is obtained and a detailed design of the MR damper is then conducted and presented with discussion.

Keywords: Front-loaded washing machine, Self-power MR damper, Suspension system, Optimal design.

1. Introduction

It is well-known that a washing machine is one of helpful equipment in human life as it releases people from hard washings for more free time. Recently, in order to satisfy the upgrading demand of customers, the laundry capacity and the spin speed are increased, while the machine weight and cost are reduced. With these, the vibration of washing machine becomes a more challenging issue that should be under consideration. The vibration of washing machine is transferred from the drum to the frame and next to the floor causing acoustic noises, uncomfortable feeling for human and gradual decrease of the machine life-span.

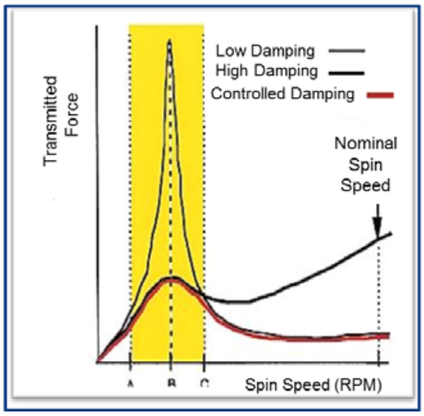

Figure 1. Control of washing machine vibration featuring semi-active suspension system

The vibration of washing machine is mostly due to the unbalanced mass of laundry disposed in the drum. Particularly, in a front-loaded washing machine, the impact of gravity makes the unbalance more serious. Various suspension methods have been developed to control the vibration of washing machine. In this study, the vibration of front-loaded washing machine is suppressed based on the damping control of suspension system. It is found that during the spinning process, the first resonance usually appears at quite low frequency, around 100-250 rpm while another one occurs at high speed, usually above 1000 rpm. In a conventional suspension system, since passive dampers (constant damping coefficient) are applied, there is a tradeoff between the probability of vibration suppression at low frequency and its increased transmissibility at high frequency. Consequently, in order to effectively attenuate the vibration of washing machine at low resonance frequency whereas the one at high excitation frequency is well isolated, a semi-active suspension system with controllable damping coefficient should be employed, as shown in Figure 1. For that, in this paper, a shear-mode magneto-rheological (MR) damper is designed.

MR fluid is a type of smart material consisting of spherical or ellipsoid magnetic particles suspended within carrier oil. When an external magnetic field is applied, these particles form chains along the lines of magnetic flux resulting in solidification of MR fluid. Applications of MR fluid can be found in various industry fields such as brake, clutch, valve, damper… due to its possibilities of rapid response, easy control and reverse. There are some research works on semi-active suspension system for front-loaded washing machines featuring MR fluid. Michael and Carlson [1] have developed a low-cost MR fluid sponge damper for washing machines. Although the sponge MR damper can well eliminate the vibration of washing machine at low frequency [2], wearing and durability of the sponge are still significant challenges. Aydar et al. [3] have researched on design and application of a flow-mode MR damper to control vibration of washing machine. However, the optimal design of the MR damper has not been considered, and the zero-field friction force (the damping force when no current is applied to the coils of the damper), also called the off-state force, is still very high, which may cause a grave vibration of washing machine at high frequency. Furthermore, because a large amount of MR fluid is required for flow-mode, the cost of the damper is also high. Recently, Nguyen et al. [4] have developed a shear-mode MR damper for front-loaded washing machines. It is potential that the MR damper can provide a damping force up to 120 N while the zero-field friction force can be considerably small. Yet, this MR damper has not been verified on a front-loaded washing machine, where sealing and assembly of the damper are significant problems that should be completely taken into account. To overcome this drawback, an optimal design of shear-mode MR damper considering the sealing ability and the assembly space in the washing machine has been proposed by Bui et al. [5], and also the hysteresis behavior of MR damper has been investigated [6]. However, the application of a controller based on this nonliner hysteresis model will meet a lot of difficulties in algorithm and make the operating system more complex.

The main contribution of this research is to design a self-power MR damper in shear-mode which can operate by itself in front-loaded washing machine without external power supply, making the control of suspension system easier. Firstly, a suspension system for front-loaded washing machine featuring MR dampers is introduced. The configuration of the self-power MR damper for front-loaded washing machine is proposed and the design optimization is carried out considering required damping force, sealing ability, power consumption, available space and cost of the system. From the optimal results, simulated performance of the optimized MR damper is obtained and a detailed design of the MR damper is then conducted and presented with discussion.

2. Dynamic modeling of washing machine

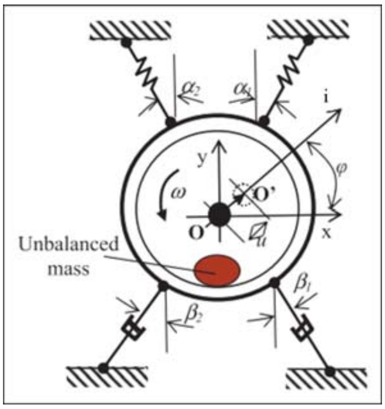

Figure 2. 2D simplified schematic of the prototype washing machine

In this work, the front-loaded washing machine object is a prototype based on the Samsung WF8690NGW washing machine manufactured by Samsung Electronics. A 2D simplified schematic of the washing machine is shown in Figure 2. According to [4, 7], the drum and the tub were modeled as rigid bodies with 3 degrees of freedom in consideration of the rotational motion in x-direction and the translational motions in y and z-directions. From the figure, the governing equation of the washing machine can be expressed

$$m\ddot{u} + c\dot{u}\left[\sin^2(\varphi+\beta_2)+\sin^2(\varphi-\beta_1)\right] + ku\left[\sin^2(\varphi+\alpha_1)+\sin^2(\varphi-\alpha_2)\right] = F_u(t) \qquad (1)$$

where m is the mass of the suspended tub assembly including the drum, laundry, shaft, counter weight, rotor and stator; c is the damping coefficient of each damper; k is the stiffness of each spring; φ is the angle of an arbitrary direction, i, in which the vibration is considered; u is the displacement of the tube center in the i-direction; and F_u is the excitation force due to an unbalanced mass in the i-direction, defined by $F_u = F_0 cos(\omega t) = m_u\omega^2 R_u cos(\omega t)$, where m_u and R_u are the mass and radius from the rotation axis of the unbalanced mass, respectively. From equation (1), the damped frequency of the suspended tub assembly is calculated by

$$\omega_d = \omega_n\sqrt{1-\xi^2} \qquad (2)$$

in which the natural frequency is

$$\omega_n = \sqrt{\frac{k\left[\sin^2(\varphi+\alpha_1)+\sin^2(\varphi-\alpha_2)\right]}{m}} \tag{3}$$

and the damping ratio is

$$\xi = \frac{c\left[\sin^2(\varphi+\beta_2)+\sin^2(\varphi-\beta_1)\right]}{2\sqrt{mk\left[\sin^2(\varphi+\alpha_1)+\sin^2(\varphi-\alpha_2)\right]}} \tag{4}$$

It can be observed that the damped frequency, natural frequency and damping ratio of the tub assembly in the *i*-direction are functions of the angle φ. Therefore, the tub assembly exhibits different resonant frequencies in different directions of vibration. This causes the vibration to become more severe and hard to control. In the design of the suspension system for the tub assembly, the frequency resonance in all directions should be restricted as much as possible. From this, it is easy to realize that, by choosing $\alpha_1 = \alpha_2 = \beta_1 = \beta_2 = 45°$ considering manufacturing and available space, equation (1) can be simplified as follows

$$m\ddot{u} + c\dot{u} + ku = F_u(t) \tag{5}$$

Then, the damped frequency, natural frequency and damping ratio of the tub assembly do not depend on the *i*-direction of the vibration and we have

$$\omega_n = \sqrt{\frac{k}{m}}\ ; \qquad \xi = \frac{c}{2\sqrt{mk}} \tag{6}$$

3. Configuration of self-power MR damper in shear-mode

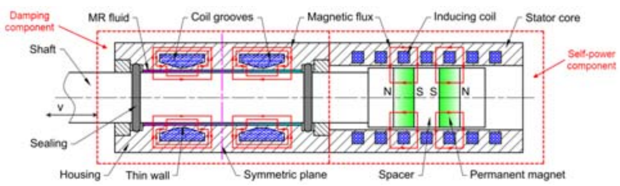

Figure 3. Schematic configuration of proposed self-power shear-mode MR damper

Figure 3 shows the configuration of proposed self-power shear-mode MR damper. As shown in the figure, the shaft of damper moves reciprocally due to the vibration of washing machine, developing damping force via direct shearing of the MR fluid. Among three operational modes of MR damper (shear-mode, flow-mode and their combination, mixed-mode), this shear-mode is proposed for our study due to its simple design, small off-state friction force and low cost. Compared with traditional MR damper using external current applied to the coils, the self-power MR damper operates based on energy harvested from its operating environment via an energy-harvesting device. In general, the self-power MR damper consists of two main components: the damping component and the self-power one (energy-harvesting device).

The damping component is composed of an inner housing with a thin wall, on which the coils are wound directly, and an outer housing to cover the coil and to create a closed magnetic circuit of the damper. The gap between the inner housing and the shaft is filled with MR fluid to create damping force. The section of the thin wall is designed to have a small area so that the magnetic flux going through it promptly reaches to saturation and hence is drived across the MR gap. In response to the magnetic field, the MR fluid solidifies and resists the relative movement between the shaft and the housing, producing the damping force. In the literature, a configuration with more than two coils can be used in order to increase the magnetic flux density across the MR gap. However, this increases the length of the damper and so the assembly space will not be satisfied. On the other hand, more coils means more applied powers required, which results in high cost of operation and the rising of heat emission. Taking all above issues into account, the configuration with two coils is recommended for the damping component.

The self-power component consists of magnetized permanent magnets and spacers alternately fastened together on the moving shaft and a slotted stator core covered by an outer housing. Each one magnet and one adjacent spacer are grouped into one pole pair. The magnets are placed in cross-pole positions in order to force the flux going through the spacers and across the air gap. The coils are wound directly on the slots of the stator core and connected to the coils of the damping component. Under external excitation caused by the vibration of the washing machine, the relative movement between the shaft end carrying magnets and the stator core appears, the coils experience a change in flux linkage and electrical power is generated for the damping component.

4. Optimal design of MR damper

In this work, optimal design of the proposed shear-mode MR damper is performed based on the quasi-static model of the MR damper and the dynamic equation of the washing machine presented in "Dynamic modeling of washing machine". From the Figure 3, by assuming a linear profile of velocity of the MR fluid in the duct between the shaft and the housing, the damping force F_{sd} and the zero-field friction force F_{s0} can be respectively determined by

$$F_{sd} = 2\pi R_s L_e(\tau_y + \eta\frac{v}{d}) + 2F_{or} + F_{mf} \tag{7}$$

$$F_{s0} = 2\pi R_s L_d (\tau_{y0} + \eta_0 \frac{v}{d}) + 2F_{or} + F_{mf} \quad (8)$$

where R_s is the shaft radius, d is the gap size of the MR fluid duct, v is the relative velocity between the shaft and the housing, η and τ_y are, respectively, the field-dependent post-yield viscosity and yield stress of the active MR fluid in the duct, L_d is the length of the MR fluid duct, and L_e is the effective length of the active MR fluid in the duct. For the proposed MR damper, L_e can be approximated by $L_e \cong L_d$. F_{mf} is the friction force caused by the magnetism of the permanent magnets and is found to be around 19 N in this study. F_{or} is the coulomb friction force between the shaft and the o-ring which can be approximately calculated by [8]

$$F_{or} = f_c L_r + f_h A_r \quad (9)$$

in which L_r is the length of seal rubbing surface, f_c is friction per unit length due to o-ring compression, A_r is the projected area of seal, and f_h is friction force due to fluid pressure on a unit projected area of seal. It is noteworthy that for the shear-mode MR damper, the pressure on the o-rings is very small and thus can be neglected, $f_h \cong 0$. Moreover, the compression of o-rings should be set at a moderate ratio so that the off-state force is not too high while the sealing of the MR damper is well ensured during the operation of washing machine. Therefore, in this paper, 70-durometer rubber o-rings are used and the compression of o-rings is set by 10%. With these, it can be found that f_c = 116.75 N/m.

The MR fluid 132-DG made by Lord Corporation is employed for our proposed MR damper. Based on Bingham model, the rheological properties of MR fluid depend on applied magnetic field and can be estimated by the following equation [9]

$$Y = Y_\infty + (Y_0 - Y_\infty)(2e^{-B\alpha_{SY}} - e^{-2B\alpha_{SY}}) \quad (10)$$

where Y represents one of the rheological parameters of MR fluid such as yield stress and post-yield viscosity. The value of parameter Y tends from the zero applied field value Y_0 to the saturation value Y_∞. α_{SY} is the saturation moment index of the Y parameter. B is the applied magnetic density. The values of Y_0, Y_∞, α_{SY} are determined from experimental results using curve-fitting method and the results are presented in Table 1.

Table 1. Rheological properties of MR fluid 132-DG

Bingham model parameters		
μ_0 = 0.1 pa•s	τ_{y0} = 15 pa	$\alpha_{s\mu}$ = 4.5 T^{-1}
μ_∞ = 3.8 pa•s	$\tau_{y\infty}$ = 40,000 pa	$\alpha_{s\tau y}$ = 2.9 T^{-1}

For the self-power component, the stator core and the spacers are made of C45 steel. The permanent magnets made of NdFeB grade N35 and the spacers are mounted on a non-magnetic aluminium shaft end. The length of magnets and spacers are both 7 mm. The number of magnets and coils on the stator core is designed to be, respectively, 2 and 7 based on the available assembly space of dampers in the washing machine. By this way, the maximum number of coils in the electromagnetic working state will be 4 out of 7 coils. Assuming the reluctance of stator core and spacers are neglected for their high magnetic permeability, the magnetic flux of the air gap between the magnets and the stator core Φ_g is given as [10, 11]

$$\Phi_g = \frac{B_{rem} l_m \mu_0 H_{coe} A_{gm}}{2t_{gm} B_{rem} + l_m \mu_0 H_{coe} \left(A_{gm} / A_m \right)} \quad (11)$$

in which B_{rem} is the remanent flux density of magnet, H_{coe} is the coercive magnetic field intensity of magnet, μ_0 is the relative magnetic permeability and equals 4μ*10^{-7} N/A^2, l_m is the length of the permanent magnets, t_{gm} is the thickness of air gap, A_{gm} is the surface area of cylindrical air gap, and A_m is the cross-section area of magnet. A_{gm} and A_m can be determined by, respectively

$$A_{gm} = 2\pi \left(r_{m1} + \frac{t_{gm}}{2} \right) \left(\frac{p_m - l_m}{2} \right) \quad (12)$$

$$A_m = \pi \left(r_{m1}^2 - r_{m2}^2 \right) \quad (13)$$

where r_{m1} and r_{m2} are, respectively, the outer and inner radii of magnet, and p_m is the pitch of pole pair. The induced voltage E in the inducing coil is defined as

$$E = -N\Phi_g \frac{\pi}{p_m} \sin\left(\frac{\pi}{p_m} x + \varphi_0 \right) \frac{dx}{dt} \quad (14)$$

in which N is the number of turns of inducing coil, x and dx/dt are, respectively, the displacement and velocity of shaft, and φ_0 is the intinial phase angle of inducing coil. Since the pitch of pole pair p_m is double the pitch of coil slot p_{cm}, the phase angle is π/2 between each nearby coil. Therefore, the induced voltages in four working coils are

$$E_1 = -N\Phi_g \frac{\pi}{p_m} \sin\left(\frac{\pi}{p_m} x \right) \frac{dx}{dt} \quad (15)$$

$$E_2 = -N\Phi_g \frac{\pi}{p_m} \cos\left(\frac{\pi}{p_m} x \right) \frac{dx}{dt} \quad (16)$$

$$E_3 = -E_1 \quad (17)$$

$$E_4 = -E_2 \quad (18)$$

In order to increase the efficiency of generating power, the coils of phase angle 0 and π/2 are connected together and feed the first coil of the damping component. Similarly, the π and 3π/2 phase coils are connected together and feed the second coil. It is noted that the current applied to two coils of the damping component must be cross to make sure the magnetic flux goes correctly, thus bridge rectifier is employed in this design.

Another problem should be under consideration in the MR damper design is the available space of the washing machine. From assembly aspects, with the required maximum stroke of damper set by 40 mm, the available length of MR duct is approximately calculated by 50 mm. Despite no strict constraint on the outer radius of MR damper, it should be as small as possible in order to reduce its cost and weight. Since the outer radius of conventional damper is around 20 mm, the one of MR damper is restricted to be smaller than this value. For the possibility of machining the bushing cylinder without warping, the height of coil grooves should not be so small. In this case, it is set to be larger than 5.45 mm. Furthermore, the 0.34 mm-diameter coil wires are employed for both damping and self-power components of MR damper due to their popularity and good ability of filling slots. In order to ensure the durability of these coils, the generated current is limited to 1 A. In summary, the optimization of MR damper design for washing machine can be expressed as follows: Find optimal values of significant geometric dimensions of the proposed MR damper that maximize the damping force F_{sd}, subjected to the length of MR duct L_d is smaller than 50 mm, the outer radius of the damper R is smaller than 20 mm, the height of coil grooves h_c is greater than 5.45 mm, and the generated current is smaller than 1 A.

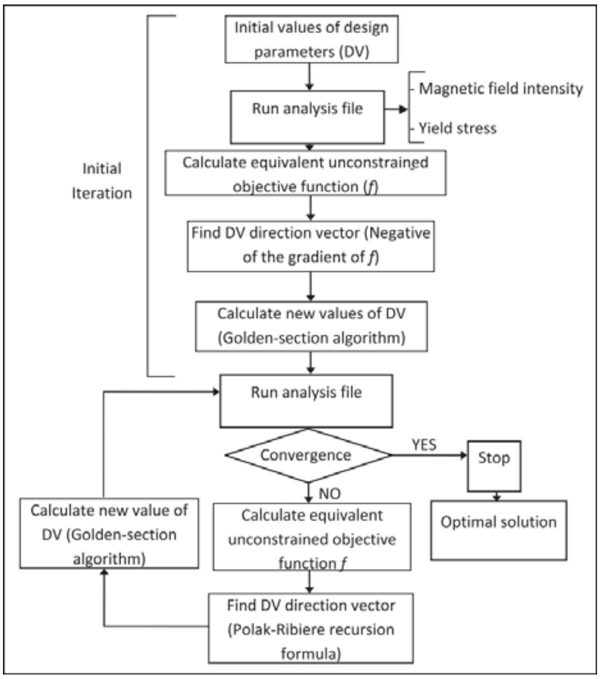

Figure 4. Flow chart to achieve optimal design parameters of the MR damper

In order to obtain the optimal solution, a FEA code integrated with an optimization tool is employed. In this study, the first order method with golden section algorithm of ANSYS optimization tool is used. The detailed procedure to obtain the optimal solution of MR fluid devices based on FEA has been mentioned in several researches [12, 13].

Figure 4 shows the flow chart to achieve optimal design parameters of the MR damper. Firstly, an analysis ANSYS file for solving the magnetic circuit of the damper and calculating the objective function is built using ANSYS parametric design language (APDL). In the analysis file, the design variables (DVs) such as the length of MR duct L_d, the outer radius R, the radius of shaft R_s, ... must be coded as variables and initial values are assigned to them. The geometric dimensions of the damper structure are varied during the optimization process; the meshing size therefore should be specified by the number of elements per line rather than the element size. Because the magnetic density is not constant along the duct length, it is necessary to define paths along the MR active volume where magnetic flux passes. Based on the induced voltages obtained from Equations 15-18, the average magnetic density across the MR ducts B_{mr} is calculated by integrating the magnetic density along the defined path then divided by the path length. Thus, the magnetic density is determined as follows:

$$B_{mr} = \frac{1}{L_d} \int_0^{L_d} B(s) ds \qquad (19)$$

where $B(s)$ is the magnetic flux density at each nodal point on the defined path.

From the figure, it is observed that the optimization is started with the initial value of DVs. By executing the analysis file, first the magnetic density is derived. Then the yield stress, post-yield viscosity, and objective function are respectively calculated from Equations 10 and 7. The ANSYS optimization tool then transforms the optimization problem with constrained design variables to an unconstrained one via penalty functions. The dimensionless, unconstrained objective function f is formulated as follows:

$$f(x) = \frac{OBJ}{OBJ_0} + \sum_{i=1}^{n} P_{x_i}(x_i) \qquad (20)$$

where OBJ_0 is the reference objective function value that is selected from the current group of design sets. P_{xi} is the exterior penalty function for the design variable x_i. For the initial iteration ($j = 0$), the search direction of DVs is assumed to be the negative of the gradient of the unconstrained objective function. Thus, the direction vector is calculated by

$$d^{(0)} = -\nabla f\left(x^{(0)}\right) \qquad (21)$$

The values of DVs in the next iteration ($j + 1$) is obtained from the following equation:

$$x^{(j+1)} = x^{(j)} + s_j d^{(j)} \quad (22)$$

where the line search parameter s_j is calculated by using a combination of the golden section algorithm and a local quadratic fitting technique. The analysis file is then executed with the new values of DVs, and the convergence of the objective function is checked. If the convergence occurs, the values of DVs at this iteration are the optimum. If not, the subsequent iterations will be performed. In the subsequent iterations, the procedures are similar to those of the initial iteration except for that the direction vectors are calculated according to the Polak-Ribiere recursion formula as follows

$$d^{(j)} = -\nabla f\left(x^{(j)}\right) + r_{j-1} d^{(j-1)} \quad (23)$$

where $$r_{j-1} = \frac{\left[\nabla f\left(x^{(j)}\right) - \nabla f\left(x^{(j-1)}\right)\right]^T \nabla f\left(x^{(j)}\right)}{\left|\nabla f\left(x^{(j-1)}\right)\right|^2} \quad (24)$$

5. Results and discussions

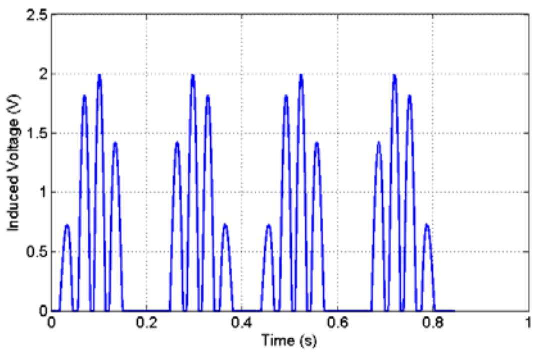

Figure 5. The voltage generated in each coil of the self-power MR damper at the resonance excitation and frequency

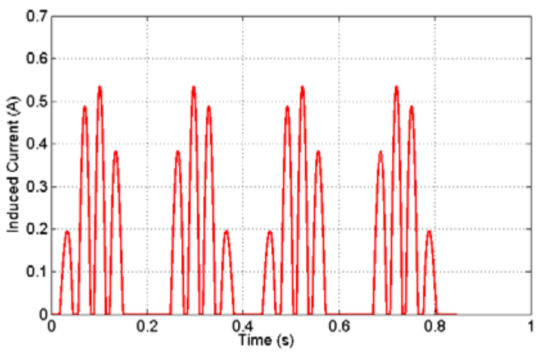

Figure 6. The current generated in each coil of the self-power MR damper at the resonance excitation and frequency

In this section, by using a MATLAB code, the results of induced voltage and current applied to the coils of the damping component are presented based on the analysis in the "Optimal design of MR damper" section. Figures 5 and 6 show the voltage and current generated in the self-power MR damper at the resonance excitation and frequency, respectively. The optimal results of the self-power MR damper are then obtained based on these results and the optimization problem developed above. From commercial aspects, the C45 steel is employed for magnetic components of the MR damper. The MR fluid gap size and air gap size are set from 0.5 mm to 1 mm based on the consideration of manufacturing, amount of MR fluid and size of o-rings. The optimal solutions of the proposed MR damper are summarized in Table 2, and the magnetic flux distribution of the optimized damper is shown in Figure 7. It is observed from the figure that the magnetic flux density in the thin wall of the damping component reaches to saturation, which agrees well to the theoretical analysis.

Table 2. Optimal parameters of the proposed shear-mode MR damper

Parameters			
Off-state force F_{s0} (N)	37.34	Coil width w_{c0} (mm)	2.5
Max damping force F_{sd} (N)	72.3	Axial chamfer ch_l (mm)	3.12
MR duct length L_d (mm)	50	Shaft radius R_s (mm)	8.25
Outer radius R (mm)	20	MR duct gap t_g (mm)	0.8
Coil groove height h_c (mm)	5.62	Thin wall t_i (mm)	0.8
Coil height h_{c0} (mm)	3.93	Sliding housing t_0 (mm)	4.53
Radial chamfer ch (mm)	1.7	Coil to centerline t_{fi} (mm)	8.13

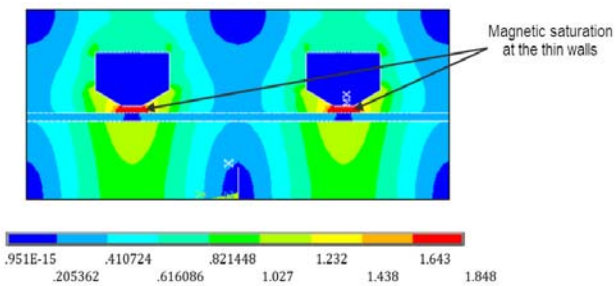

Figure 7. Magnetic flux density of the proposed optimized MR damper

5. Conclusions

This paper focused on the design a shear-mode self-power MR damper for suspension system of front-loaded washing machine to eliminate vibration due to an unbalanced laundry mass occurring in the washing drum. Firstly, a suppression system for washing machine featuring MR dampers was introduced and the configuration of the self-power MR damper was proposed. Optimization of the proposed MR damper was then performed considering required damping force, off-state friction force, size, manufacturing and low cost. From the optimal results, simulated performance of the optimized MR damper was obtained and a detailed design of the MR damper was then conducted. From the results, it was shown that there was a good correlation between simulation results and theoretical analysis. It is finally remarked that as the second phase of this research, a prototype self-power MR damper will be manufactured and applied to suspension system of a front-loaded washing machine for more specific and complete experimental evaluation.

Acknowledgement

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