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RESEARCH

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Optimal design and performance evaluation of a flow-mode MR damper for front-loaded washing machines

Quoc Hung Nguyen[1*], Ngoc Diep Nguyen[1] and Seung Bok Choi[2]

* Correspondence: nguyenhung2000vn@yahoo.com
[1]Department of Mechanical Engineering, Industrial University of Ho Chi Minh City, Hochiminh-12 NVBao, Vietnam
Full list of author information is available at the end of the article

Abstract

It is well known that the vibration of washing machines is a challenging issue to be considered. This research work focuses on the optimal design of a flow-mode magneto-rheological (MR) damper that can replace the conventional passive damper for a washing machine. Firstly, rigid mode vibration of the washing machine due to an unbalanced mass is analyzed and an optimal positioning of the suppression system for the washing machine is considered. An MR damper configuration for the washer is then proposed considering available space for the system. The damping force of the MR damper is derived based on the Bingham rheological behavior of the MR fluid. An optimal design problem for the proposed MR brake is then constructed. The optimization is to minimize the damping coefficient of the MR damper while the maximum value of the damping force is kept being greater than a required value. An optimization procedure based on finite element analysis integrated with an optimization tool is employed to obtain optimal geometric dimensions of the MR damper. The optimal solution of the MR damper is then presented with remarkable discussions on its performance. The results are then validated by experimental works. Finally, conclusions on the research work are given and future works for development of the research is figured out.

Keywords: Magneto-rheological; MR damper; Washing machine; Vibration control

Background

It is well known that the vibration of washing machines is a challenging issue to be considered. The vibration of the washing machine is mainly due to the unbalanced mass of clothes distributed in the washing drum. This occurs most frequently in the spin-drying stage, because the drum spins at a relatively high speed causing the clothes to be pressed against the inner wall of the spin drum, and these can become a large unbalanced mass until the end of the stage. Particularly, in a front-loaded washing machine (drum-type washing machine), the unbalanced mass of clothes easily occurs and very severe due to the effect of gravity. The vibration of the washing machine is transferred to the floor causing noises, unpleasant feeling for humans, and failure of the machine.

There are many researches on vibration control of washing machines which can be classified into two main approaches. The first approach is based on the control of the tub balance to eliminate the source of vibration [1,2]. In this approach, one type of dynamic balancer is used to self-balance the tub dynamics. A typical dynamic balancer is the hydraulic balancer containing salt water, which is attached to the upper rim of the basket. The liquid in the balancer moves to the opposite side of unbalance automatically due to the inherent nature of fluids when the rotational speed is higher than the critical speed of the spinning drum [1]. Another dynamic balancer that counteracts vibrations is to use two balancing masses. In this method, two balancing masses move along the rim of the basket. The rotation plane of the balancing masses can be easily chosen to be wherever judged suitable, always targeting at the reduction of the induced moments [2]. It is proved that the vibration of the washing machine can be significantly reduced by using a dynamic balancer. However, the complicated structure, high cost of manufacturing, and maintenance are a big obstacle for the wide application of this approach. In the second approach, the vibration of the washing machine is suppressed based on damping control of a suspension system [3]. It is noted that during the spinning process, the washing machine usually experiences the first resonance at quite low frequency, around 100 to 200 rpm. This results from the resonance of the washing drum due to the unbalanced mass. When the rotating speed exceeds 1,000 rpm, the side and rear panels of the frame may experience resonances which cause noises and vibration transferred to the floor. If a passive damper is used to reduce the vibration of the drum at low frequency, it will cause the vibration of the washing machine at high frequencies more severe. The reason is that more excitation force from the drum is transferred to the frame via the passive damper. Therefore, in order to effectively reduce the vibration of the washing machine at low frequency while the vibration of the machine at high frequencies is insignificantly affected, a semi-active suspension system such as a magneto-rheological (MR) damper should be employed.

Although there have been several researches on the design and application of MR dampers to control the vibration of washing machines [4-6], the optimal design of such MR dampers was not considered. The main objective of this study is to achieve the optimal design of the semi-active suspension system for washing machines employing MR dampers. Firstly, rigid mode vibration of the washing machine due to an unbalanced mass is analyzed and an optimal positioning of the suppression system for the washing machine is considered. An MR damper configuration for the washer is then proposed. The damping force of the MR damper is then derived based on the Bingham model of the MR fluid. An optimal design problem for the proposed MR brake is then constructed considering the damping coefficient and required damping force of the MR damper. An optimization procedure based on finite element analysis integrated with an optimization tool is employed to obtain optimal geometric dimensions of the MR damper. The optimal solution of the MR damper is then presented with remarkable discussions. The results are then validated by experimental works.

Methods

Vibration control of washing machine using MR damper

The washing machine object of this work is a prototype based on the LG F1402FDS washer manufactured by LG Electronics (Seoul, South Korea). A three-dimensional (3D) schematic diagram of the washer is shown in Figure 1. It is characterized by a suspended tub (basket) to store the water for washing linked to the cabinet by two springs

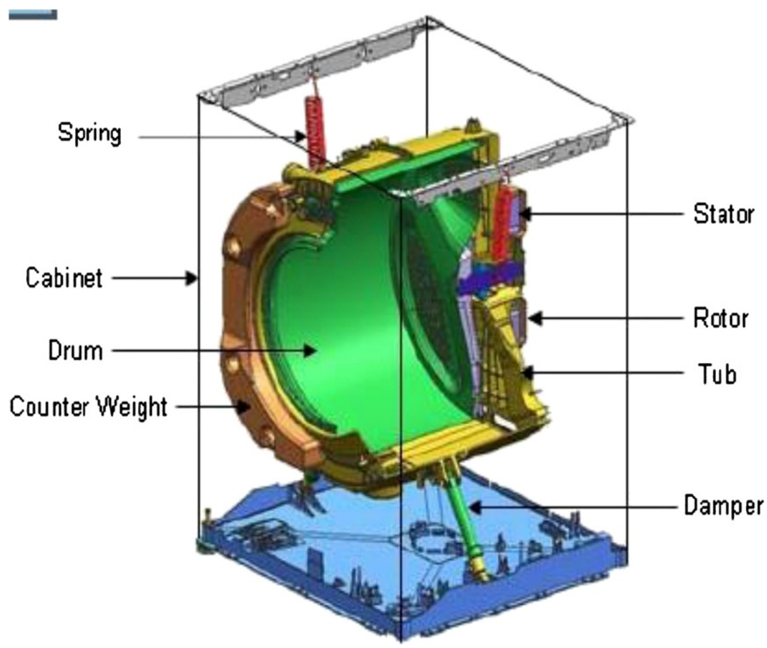

Figure 1 3D schematic of the prototype washing machine.

and two dampers. The rotor is directly connected with the drum which rotates against the tub while the stator is fixed on the back of the tub. When the drum is rotating, the unbalanced mass due to the eccentricity of laundry causes the vibration of the tub assembly. The vibration of the tub assembly is then transmitted to the cabinet and the bottom through the springs and the dampers. In Figure 2, a two-dimensional (2D) simplified schematic of the machine is depicted. From the figure, the following governing equation of the washing machine can be derived:

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

where m is the mass of the suspended tub assembly including the drum, laundry, shaft, counter weight, rotor, and stator. For the prototype washing machine, m is roughly estimated about 40 kg. c is the damping coefficient of each damper, and k is the stiffness of each spring which is assumed to be 8 kN/m in this study. φ is the angle of an arbitrary direction (u direction) in which the vibration is considered. F_u is the excitation force due to unbalanced mass in the u direction, $F_u = F_0\cos\omega t = m_u\omega^2 R_u\cos\omega t$, in which m_u and R_u are the mass and radius from the rotation axis of the unbalanced mass. From Equation 1, the damped frequency of the suspended tub assembly is calculated by

$$\omega_d = \omega_n\sqrt{1-\xi^2} \tag{2}$$

where $\omega_n = \sqrt{\frac{k\left[\sin^2(\varphi+\alpha_1)+\sin^2(\varphi-\alpha_2)\right]}{m}}$ and $\xi = \frac{c\left[\sin^2(\varphi+\beta_2)+\sin^2(\varphi-\beta_1)\right]}{2\sqrt{mk\left[\sin^2(\varphi+\alpha_1)+\sin^2(\varphi-\alpha_2)\right]}}$.

It is seen that the damped frequency and natural frequency of the tub assembly in the u direction are a function of φ. Therefore, in general, in a different direction of vibration, the tub assembly exhibits different resonant frequency. This causes the

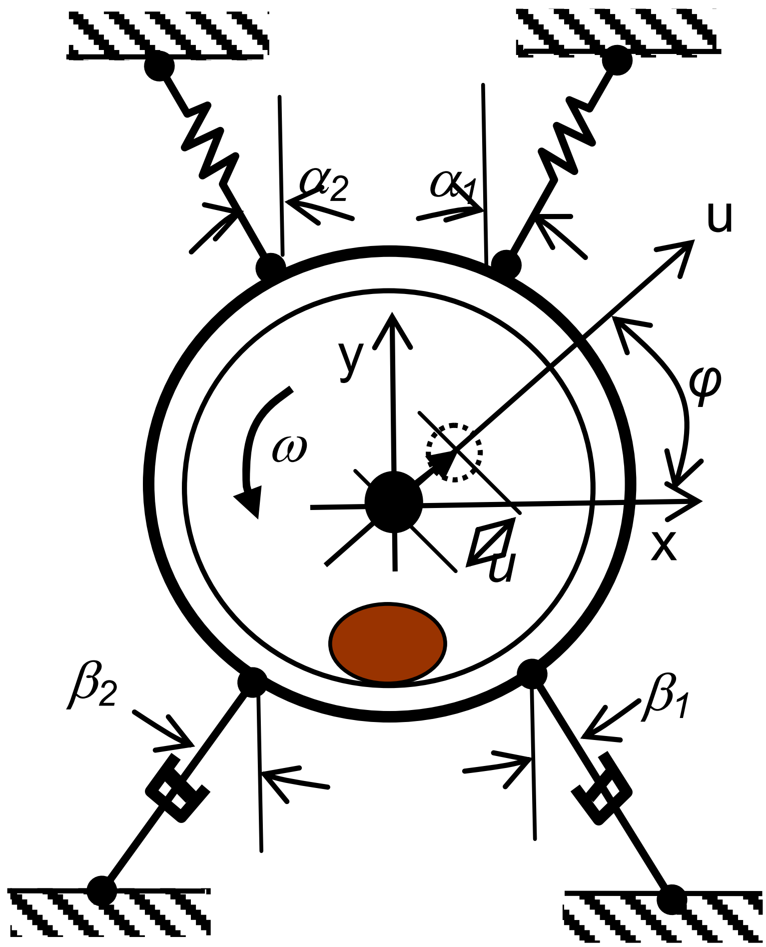

Figure 2 2D simplified schematic of the prototype washing machine.

vibration to become more severe and hard to control. In the design of the suspension system for the tub assembly, the frequency range of the resonance in all directions should be as small as possible. From the above, it is easy to show that, by choosing $\alpha_1 + \alpha_2 = 90°$ and $\beta_2 + \beta_2 = 90°$, Equation 1 can be simplified to yield

$$m\ddot{u} + c\dot{u} + ku = F_u \tag{3}$$

In this case, the damped frequency and natural frequency of the tub assembly do not depend on the direction of the vibration. We have

$$\omega_d = \omega_n\sqrt{1-\xi^2} \tag{4}$$

where $\omega_n = \sqrt{\frac{k}{m}}$ and $\xi = \frac{c}{2\sqrt{mk}}$.

An inherent drawback of the conventional damper is its high transmissibility of vibration at high excitation frequency. In order to solve this issue, semi-active suspension systems such as ER and MR dampers are potential candidates. In this study, two MR dampers are employed to control the vibration of the tub assembly.

Figure 3 shows the schematic configuration of a flow-mode MR damper proposed for the prototype washing machine. From the figure, it is observed that an MR valve structure is incorporated in the MR damper. The outer and inner pistons are combined to form the MR valve structure which divides the MR damper into two chambers: the upper and

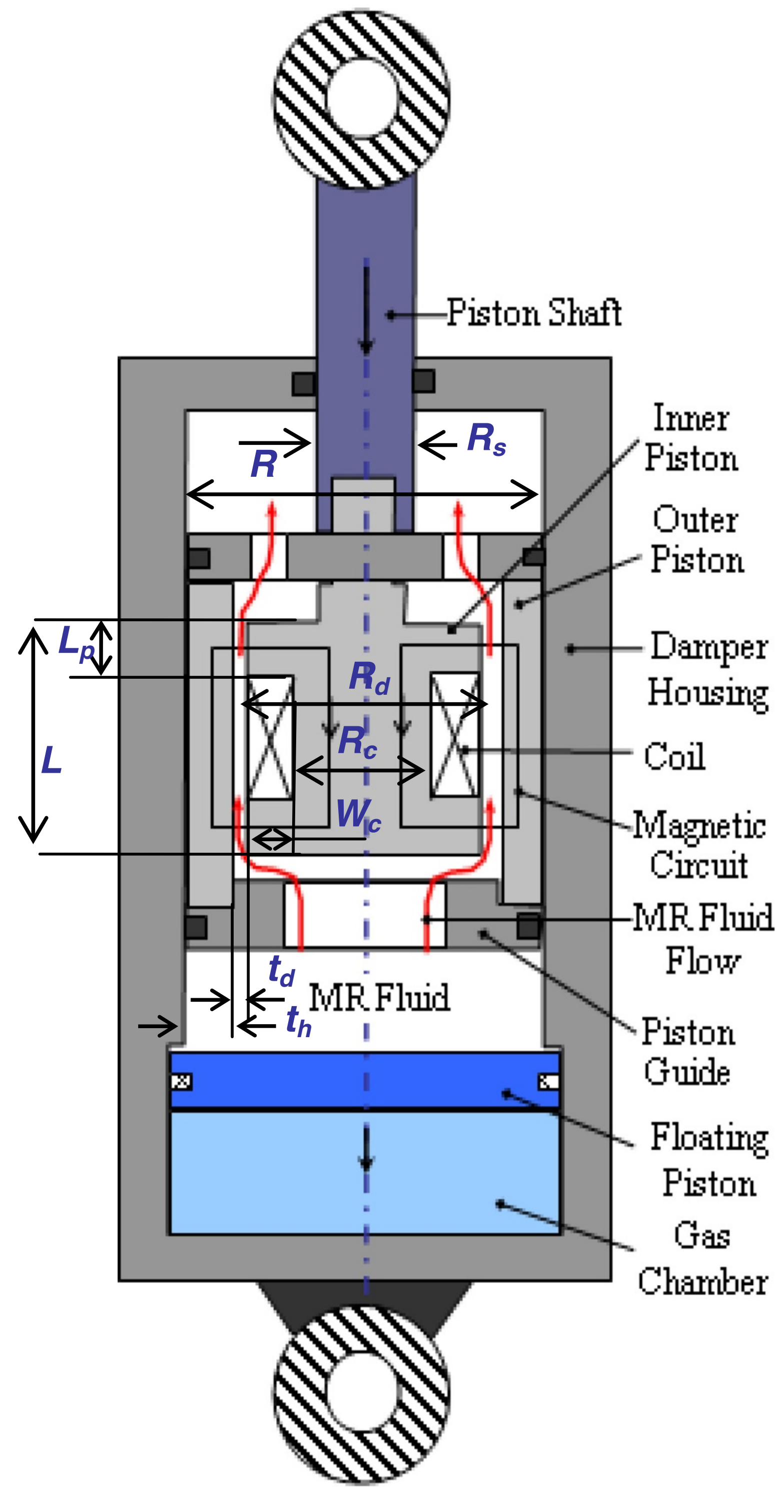

Figure 3 Schematic configuration of the MR damper.

lower chambers. These chambers are fully filled with the MR fluid. As the piston moves, the MR fluid flows from one chamber to the other through the annular duct (orifice). The floating piston incorporated with a gas chamber functions as an accumulator to accommodate the piston shaft volume as it enters and leaves the fluid chamber.

By neglecting the frictional force and assuming quasi-static behavior of the damper, the damping force can be calculated by [7]

$$F_d = P_a A_s + C_{\text{vis}} \dot{x}_p + F_{\text{MR}} \operatorname{sgn}(\dot{x}_p) \tag{5}$$

where P_a, c_{vis}, and F_{MR}, respectively, are the pressure in the gas chamber, the viscous coefficient, and the yield stress force of the MR damper which are determined as follows:

$$P_a = P_0 \left(\frac{V_0}{V_0 + A_s x_p} \right)^{\gamma} \tag{6}$$

$$C_{\text{vis}} = \frac{12\eta L}{\pi R_d t_d^3} \left(A_p - A_s \right)^2 \tag{7}$$

$$F_{\text{MR}} = 2\left(A_p - A_s \right) \frac{2.85 L_p}{t_d} \tau_y \tag{8}$$

In the above, τ_y is the induced yield stress of the MR fluid which is an unknown and can be estimated from magnetic analysis of the damper and η is the post-yield viscosity of the MR fluid which is assumed to be field independent. R_d is the average radius of the annular duct given by $R_d = R - t_h - 0.5t_d$. L and R are the overall length and outside radius of the MR valve, respectively. t_h is the valve housing thickness, t_d is the annular duct gap, and L_p is the magnetic pole length.

In this work, the commercial MR fluid (MRF132-DG) made by Lord Corporation (Cary, NC, USA) is used. The post-yield viscosity of the MR fluid is assumed to be independent on the applied magnetic field, $\eta = 0.1$ Pas. The induced yield stress of the MR fluid as a function of the applied magnetic field intensity (H) is shown in Figure 4.

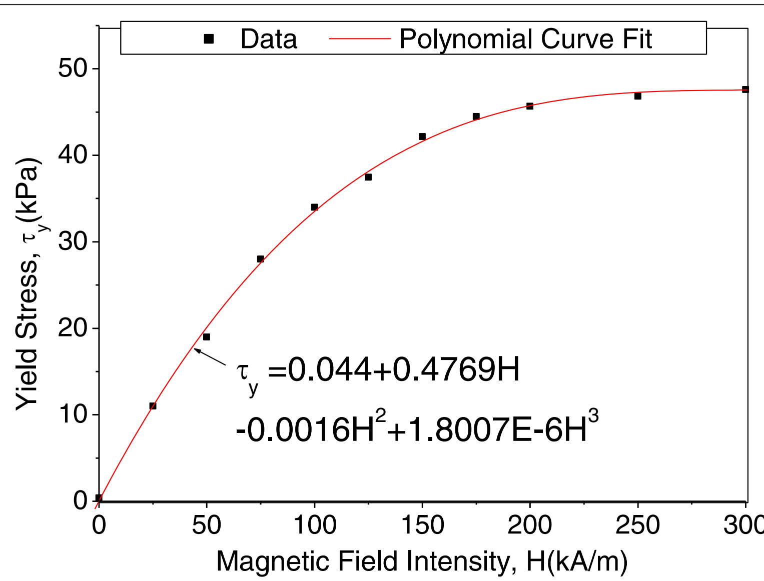

Figure 4 Yield stress of MR fluid as a function of magnetic field intensity.

By applying the least squares curve fitting method, the yield stress of the MR fluid can be approximately expressed by

$$\tau_y = C_0 + C_1 H + C_2 H^2 + C_3 H^3 \tag{9}$$

In Equation 9, the unit of the yield stress is kilopascal while that of the magnetic field intensity is kA/m. The coefficients C_0, C_1, C_2, and C_3 are respectively identified as 0.044, 0.4769, –0.0016, and 1.8007E-6. In order to estimate the induced yield stress using Equation 9, first the magnetic field intensity across the active MRF duct must be calculated. In this study, the commercial FEM software, ANSYS, is used to analyze the magnetic problem of the proposed MR damper.

Optimal design of the MR damper

In this study, the optimal design of the proposed MR damper is considered based on the quasi-static model of the MR damper and dynamic equation of the tub assembly developed in the 'Vibration control of washing machine using MR damper' section. From Figure 2 and Equation 3, the force transmissibility of the tub assembly to the cabinet can be obtained as follows:

$$\mathrm{TR} = \sqrt{\frac{1 + (2\xi r)^2}{(1-r^2)^2 + (2\xi r)^2}} \tag{10}$$

where r is the frequency ratio, $r = \omega/\omega_n$. The dependence of the force transmissibility on excitation frequency is presented in Figure 5. As shown from the figure, at low damping, the resonant transmissibility is relatively large, while the transmissibility at higher frequencies is quite low. As the damping is increased, the resonant peaks are attenuated, but vibration isolation is lost at high frequency. This illustrates the inherent trade-off between resonance control and high-frequency isolation associated with the design of passive suspension systems. It is also observed from the figure that when the

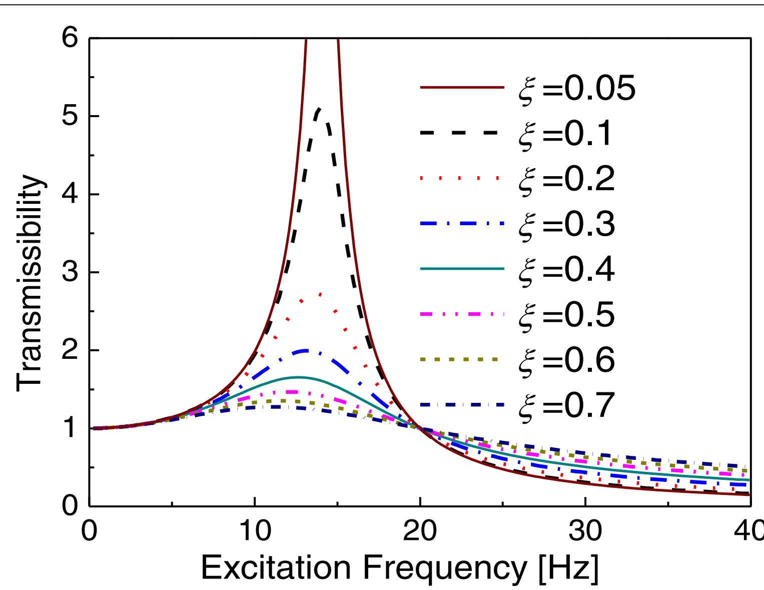

Figure 5 Transmissibility of the tub assembly.

damping ratio is 0.7 or greater, the resonant peaks are almost attenuated. Thus, the higher value of damping ratio is not necessary. It is noted that in Equation 5, the third term F_{MR} is much greater than the other and the behavior of the MR damper can is similar to that of a dry friction damper. By introducing an equivalent damping coefficient C_{eq} such that the work per cycle due to this equivalent damping coefficient equals that due to the yield stress damping force of the MR damper, the following equation holds [8]:

$$C_{\mathrm{eq}} = \frac{4|F_{\mathrm{MR}}|}{X\omega\pi} \tag{11}$$

or

$$|F_{\mathrm{MR}}| = \frac{X\omega\pi C_{\mathrm{eq}}}{4} = \frac{X\omega\pi\xi\sqrt{km}}{2} = \frac{kX\pi\xi r}{2} \tag{12}$$

In the above, X is the magnitude of the tub vibration which is determined by

$$X = \frac{F_0}{k}\sqrt{\frac{1}{(1-r^2)^2 + (2\xi r)^2}} = \frac{m_u r^2 R_u}{m}\sqrt{\frac{1}{(1-r^2)^2 + (2\xi r)^2}} \tag{13}$$

From Equations 12 and 13, the required value of F_{MR} can be determined from a required value of the damping ratio ξ as follows:

$$|F_{\mathrm{MR}}| = \frac{k\pi\xi m_u r^3 R_u}{2m}\sqrt{\frac{1}{(1-r^2)^2 + (2\xi r)^2}} \tag{14}$$

In this study, it is assumed that the spring stiffness is $k = 10$ kN/m, the mass of the suspended tub assembly is $m = 40$ kg, and the equivalent unbalanced mass is $m_u =$ 10 kg located at the radius $R_u = 0.15$ m. With the required damping ratio $\xi = 0.7$, at the resonance $r = \sqrt{1-\xi^2}$, the required value of F_{MR} can be calculated from Equation 14 which is around 150 N in this study.

Taking all above into consideration, the optimal design of the MR brake for the washing machine can be summarized as follows: Find the optimal value of significant dimensions of the MR damper such as the pole length L_p, the housing thickness t_h, the core radius R_c, the width of the MR duct t_d, the width of the coil W_c, and the overall length of the valve structure L that

$$\text{Minimize the viscous coefficient (objective function)}, \mathrm{OBJ} = c_{\mathrm{vis}} = \frac{12\eta L}{\pi R_d t_d^3}\left(A_p - A_s\right)^2$$

$$\text{Subjected to}: F_{\mathrm{MR}} = 2\left(A_p - A_s\right)\frac{2.85 L_p}{t_d}\tau_y \geq 150\ \mathrm{N}$$

In order to obtain the optimal solution, a finite element analysis code integrated with an optimization tool is employed. In this study, the first-order method with the golden section algorithm of the ANSYS optimization tool is used. Figure 6 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 pole length L_p, the housing thickness t_h, the core radius R_c, the width of the MR duct t_d, the width of the coil W_c, and the overall length

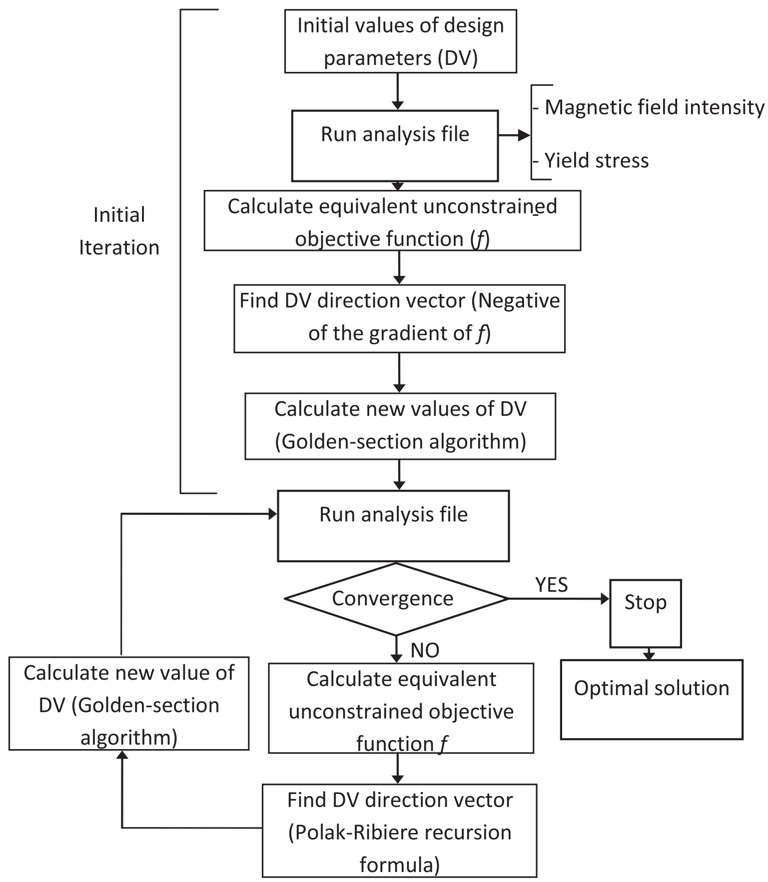

Figure 6 **Flow chart to achieve optimal design parameters of the MR damper.**

of the valve structure L must be coded as variables and initial values are assigned to them. The geometric dimensions of the valve 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 field intensity is not constant along the pole length, it is necessary to define paths along the MR active volume where magnetic flux passes. The average magnetic field intensity across the MR ducts (H_{mr}) is calculated by integrating the field intensity along the defined path then divided by the path length. Thus, the magnetic field intensity is determined as follows:

$$H_{\mathrm{mr}} = \frac{1}{L_p} \int_0^{L_p} H(s) ds \tag{15}$$

where $H(s)$ is the magnetic flux density and magnetic field intensity 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 field intensity is derived. Then the yield stress, yield stress damping force, and objective function are respectively calculated from Equations 9, 7, and 8. 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:

$$f(x) = \frac{\mathrm{OBJ}}{\mathrm{OBJ}_0} + \sum_{i=1}^{n} P_{x_i}(x_i) \tag{16}$$

where OBJ_0 is the reference objective function value that is selected from the current group of design sets. P_{x_i} 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

$$d^{(0)} = -\nabla f\left(x^{(0)}\right) \tag{17}$$

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

$$x^{(j+1)} = x^{(j)} + s_j d^{(j)} \tag{18}$$

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:

$$d^{(j)} = -\nabla f\left(x^{(j)}\right) + r_{j-1} d^{(j-1)} \tag{19}$$

$$\text{where } r_{j-1} = \frac{\left[\nabla f\left(x^{(j)}\right) - \nabla f\left(x^{(j-1)}\right)\right]^T \nabla f\left(x^{(j)}\right)}{\left|\nabla f\left(x^{(j-1)}\right)\right|^2}. \tag{20}$$

Results and discussion

In this study, an optimal design of the MR damper for the washing machine is performed based on the optimization problem developed in the 'Optimal design of the MR damper' section. It is assumed that the piston part and housing of the damper are made of commercial silicon steel, and the coil wires are sized as 21 gage (diameter = 0.511 mm) whose allowable working current is 2.5A. In the optimization, the applied current is assumed to be 2A. Figure 7 shows the optimal solution of the MR damper in case the pole length L_p, the housing thickness t_h, the core radius R_c, the width of the MR duct t_d, the width of the coil W_d, and the overall length of the valve structure L are considered as design variables. From the figure, it can be found that with a convergence tolerance of 0.5%, the optimal process is converged after 14 iterations and the solution at the 14th iteration is considered as the optimal one. The optimal values of L_p, t_h, R_c, t_d, W_d, and L, respectively, are 7.4, 2.5, 5.2, 2, 1.8, and 25 mm. It is noted that the optimal values of t_d and t_h are equal to their lower limits in this case. These limits are posed considering the stability and manufacturing cost of the damper. At the optimum, the viscous coefficient is significantly reduced up to 10 Ns/m from its initial value (290 Ns/m). When no current is applied to the coil, the damping ratio at the optimum is

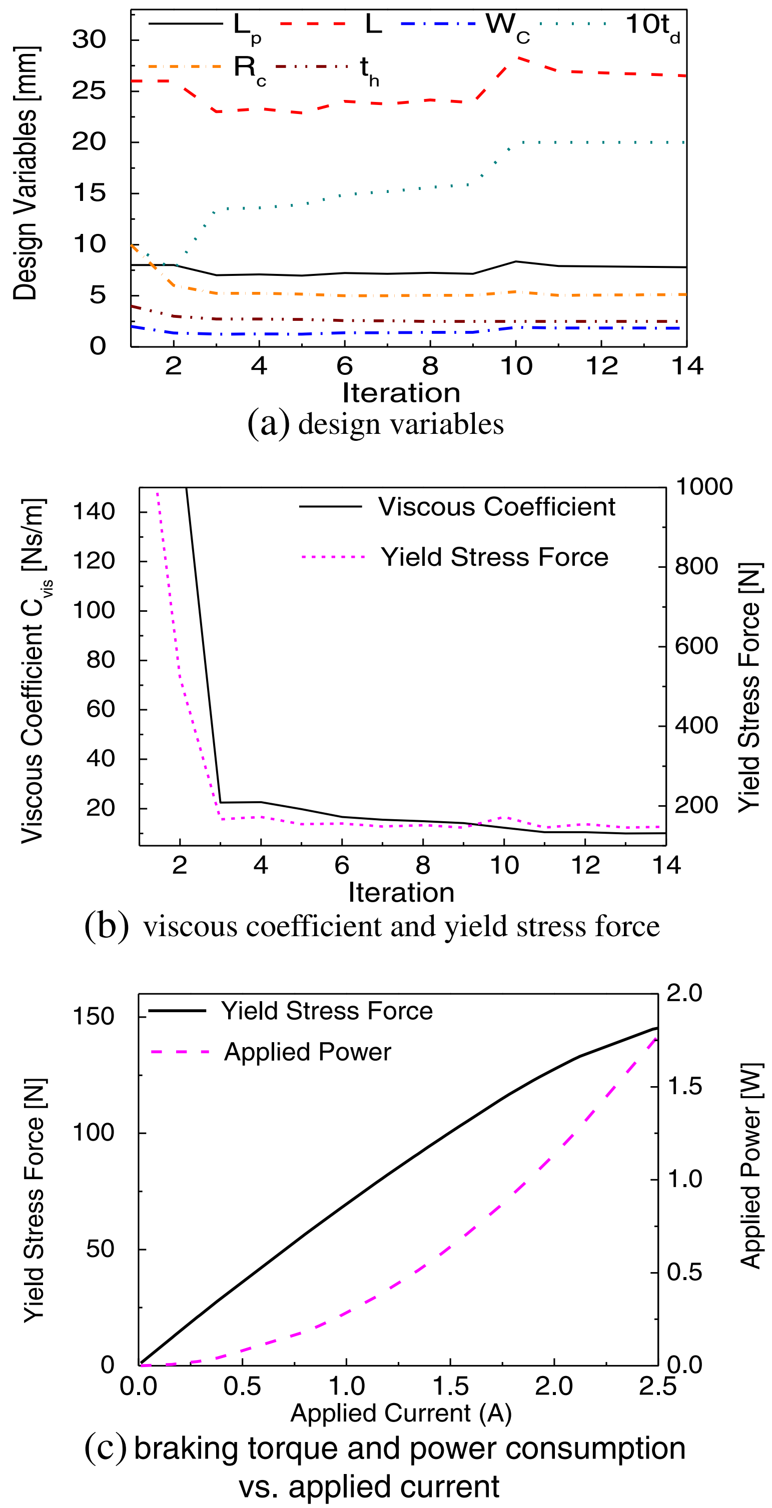

Figure 7 Optimal solution of the MR damper for the washing machine. (a) Design variables. **(b)** Viscous coefficient and yield stress force. **c)** Braking torque and power consumption vs. applied current.

around 0.01 which is very small so that a very small transmissibility of vibration from the drum to the cabinet at high excitation frequency can be achieved. It is also observed from the figure that the yield stress force of the damper at the optimum is

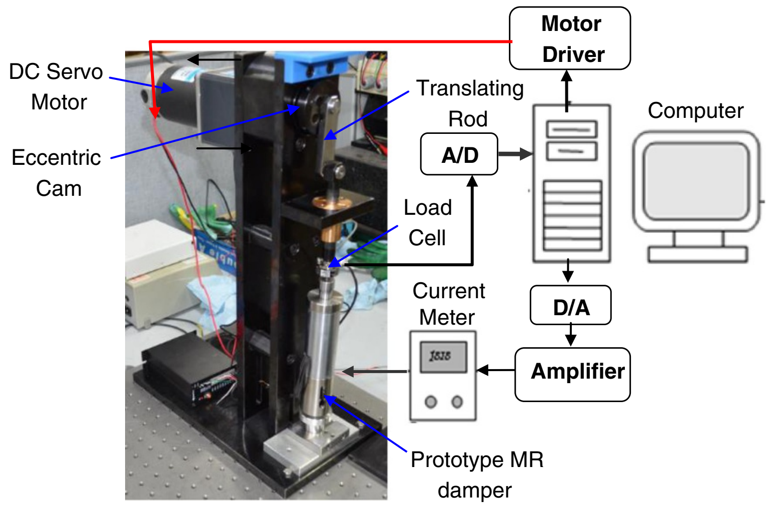

Figure 8 Experimental setup to test the performance of the optimized MR damper.

greater than the required value (150 N). The yield stress and power consumption as functions of the applied current of the optimized MR damper are shown in Figure 7c. From the figure, it is observed that the yield stress force almost increases proportionally to the applied current as the current varies from 0 to 2.5A.

In order to validate the above optimal results, experimental results of the optimized MR damper are obtained and presented. Figure 8 shows the experimental setup to test the performance of the optimized MR damper. In the figure, a crank-slider mechanism is employed to convert the rotary motion of the motor into the reciprocal motion of the damper shaft. The DC motor with a gearbox controlled by the computer is used to rotate the crank shaft at a constant angular speed of 0.5 rad/s. The damping force is measured by a load cell. The output signal from the load cell is then sent to the computer via the A/D converter for evaluation. Once the experiment process is stated, a step current signal from the computer is sent to the current amplifier. The output current from the amplifier, a step current of 2A, is applied to the coil of the damper.

Figure 9 shows the step response of the prototype MR damper. It is noted that in this case, the angular velocity of the motor is kept constant at 0.5π rad/s and the step response

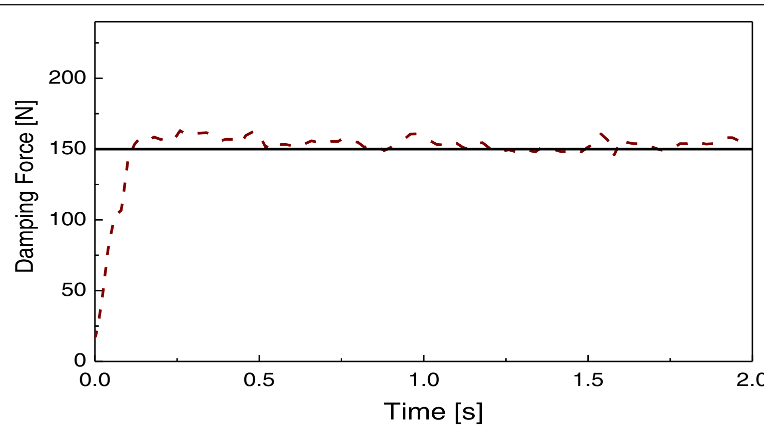

Figure 9 Experimental step response of the optimized MR damper.

of the MR damper is measured on the downward motion of the shaft. In the figure, the solid line presents the calculated damping force while the dash line presents the measured one. The results also show that, at the steady state, the average damping force is around 157 N which is a little greater than the calculated one (150 N). The difference may result from friction which is not taken into account in the modeling of the damper.

Conclusions

In this research, optimal design of a flow-mode MR damper to suppress the vibration of front-loaded washing machines was undertaken. After a brief introduction, the general governing equation of the washing tub assembly is derived. The position of the two springs and the two dampers to suppress the machine vibration was then optimally determined based on the governing equation. A configuration of the MR damper for the washing machine was then proposed, and the damping force was obtained based on the Bingham rheological model of the MR brake. The optimization problem for the damper was constructed such that the viscous coefficient of the damper is minimized and the yield stress force of the damper is greater than a required value that attenuates almost the resonant peak of the tub mechanism. The optimal design of the MR damper for a prototype washing machine was obtained based on the ANSYS finite element analysis of the MR damper magnetic circuit and first-order optimization method. The results show that performances of the MR damper are significantly improved with the proposed optimal design. A prototype of the optimized MR damper was manufactured, and experimental results on the performance of the prototype damper were obtained and presented. It was shown that the experimental results well agreed with the modeling one obtained from finite element analysis. As the second phase of this study, dynamics of the whole prototype washing machine equipped with the MR damper will be obtained and experimental results on the prototype washer will be conducted to evaluate the effectiveness of the optimized MR damper.

Competing interests
The authors declare that they have no competing interests.

Authors' contributions
QHN carried out the research background of the study, modeling of the system and optimal design of MR dampers. NDN carried out the experiment. SBC participated in configuration proposal of the MR damper, providing experimental data of the MR fluid and review of the manuscript. All authors read and approved the final manuscript.

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Author details
[1]Department of Mechanical Engineering, Industrial University of Ho Chi Minh City, Hochiminh-12 NVBao, Vietnam.
[2]Department of Mechanical Engineering, Inha University, Incheon 402-751, Korea.

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