Article

Development of a Magnetorheological Damper with Self-Powered Ability for Washing Machines

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Abstract: Magnetorheological (MR) dampers have been widely investigated and proposed for vibration mitigation systems because they possess continuous variability of damping coefficient in response to different operating conditions. In the conventional design of MR dampers, a separate controller and power supply are required, causing an increment of complexity and cost, which are not suitable for home appliances like washing machines. To solve these issues and to reuse wasted energy from vibration of washing machines, in this study, a self-powered shear-mode MR damper, which integrates MR damping and energy-harvesting technologies into a single device, is proposed. The MR damper is composed of an inner housing, on which magnetic coils are wound directly, and an outer housing for covering and creating a closed magnetic circuit of the damper. The gap between the inner housing and the moving shaft is filled with MR fluid to produce the damping force. The energy-harvesting part consists of permanent magnets fastened together on the shaft and induction coils wound directly on slots of the housing. The induced power from the induction coils is directly applied to the excitation coils of the damping part to generate a corresponding damping force against the vibration. In order to achieve optimal geometry of the self-powered MR damper, an optimization for both the damping part and the energy harvesting part of the proposed dampers are conducted based on ANSYS finite element analysis. From optimal solutions, a prototype of the proposed damper is designed in detail, manufactured, and experimentally validated.

Keywords: magnetorheological fluid; self-powered; MR damper; shear-mode MR damper; suspension system; washing machine vibration

1. Introduction

Semiactive suspension systems have been widely used in the field of vibration control [1]. Since the 1990s, magnetorheological (MR) dampers have been promisingly studied for these systems due to their attractive features such as high damping force, good adaptability, continuous controllability,
and high reliability. MR dampers can be increasingly applied in industrial environments of civil structures [2], automobiles [3], and precision machines [4]. One of the particularly interested applications of MR dampers is to attenuate vibration of washing machines. The distribution of the laundry in the washing drum is often out of balance causing the vibration of washing machines. It is found that during spinning process, the washing machine usually experiences the first resonance at quite low frequency, around 100–200 rpm [5–7]. This resonance comes from the rigid vibration mode of the washing drum. When the spindle speed exceeds 1000 rpm, such as in spinning stage, resonances can appear at the rear and side frame panels bringing about noises, uncomfortable feelings, and the shortening of the washing machine life span. The dampers in conventional suspension systems could provide passive damping forces to well eliminate the vibration at low frequency of resonance but raise the force transmissibility at high excitation frequencies because the damping coefficient could not be changed. Therefore, to effectively mitigate the vibration of washing machines at low frequency of resonance while it is still well isolated at high excitation frequencies, semiactive suspension systems that can continuously control the damping coefficient such as MR dampers-based vibration control systems should be employed (Figure 1).

![Figure 1](image.png)

**Figure 1.** Force transmissibility of washing machines using different vibration control systems.

Although several studies on MR dampers have been performed to control the vibration of washing machines [8,9], optimal design of these MR dampers was not considered. Nguyen et al. [10] conducted optimal design of a small MR damper in flow-mode for washing machines. The results showed that size of the optimized MR damper is well fit for integration in washing machines to replace conventional ones. The maximum damping force was up to 150 N, which is greater than expected damping force for MR damper (from 80 to 100 N). However, off-state force of the damper was quite high, around 25 N. Later, Nguyen et al. [11] proposed a new shear-mode configuration of MR damper for washing machines. The optimal results indicated that the proposed shear-mode MR damper was able to generate a damping force of about 120 N with much smaller off-state force, around 10 N. It was also pointed out that the proposed damper can be well installed in washing machines to replace conventional dampers. In addition, no bobbin for coils is used, which facilitate the manufacturing. However, no experimental works was conducted in this research. Recently, Bui et al. [12] performed a research on optimal design of MR dampers for front-loaded washing machines, considering expected damping forces, assembly space, power consumption, and cost as an extension of study carried out by [11]. In addition, experimental works on performance characteristics of the damper and effectiveness of implementation of the damper in a prototype washing machine was also implemented. The research results showed a good correlation between the experimental performance and theoretical simulation. Similarly, Ulasyar and Lazoglu [13] developed a MR sponge damper for washing machines with design optimization of geometric parameters; however, stability and lastingness of the sponge are considerable issues. Although the above semiactive systems can well solve the problem of washing machine vibration control, they require auxiliary equipment such as sensors, control units, and power sources to enable the operation. This makes the systems more complicated and difficult for maintenance with high
Another drawback of the semiactive vibration control systems is that the vibration energy is wasted on wearing and heating rather than recovering it to power the dampers. Inspired by those issues, there has been a great deal of research efforts in exploring the area of energy harvesting for MR dampers. Cho et al. [14] proposed a smart passive control system consisting of an MR damper and an electromagnetic induction (EMI) device. Then, Choi et al. [15] and Kim et al. [16] studied the realization of this system by experimental works. However, in this smart passive system, the energy-harvesting part is designed apart from the MR damper, so the whole size of the system would be increased and may not be available for small installation spaces such as those in washing machines.

Consequently, this paper aims at the investigation of a self-powered MR damper for the suspension system of washing machines. The self-powered MR damper integrates MR damping part and energy-harvesting part into a single device in order to enable the MR damper to reuse the wasted mechanical energy from vibration for self-power, and also, it can be well installed in washing machines. Configuration of the damping part is similar to that in previous research [11,12]. The energy-harvesting part is based on electromagnetic induction featuring permanent magnets and arrays of coils. With the energy-harvesting part added to the MR damper, the greater the vibration of the damper shaft is, the higher the power is generated, and the damping level is automatically commanded. As a result, the proposed control system can smartly adapt to external excitations by itself without any sensors and controllers, which can improve complexity and cost of the suspension system. After the configuration of the self-powered MR damper is proposed, an optimization procedure is conducted for both the damping part and the energy harvesting part of the damper. From the optimal solutions, a prototype of the proposed damper is designed and manufactured. The performance of each part and the whole damper are then investigated by both simulation and experiment.

2. Configuration and Operating Principles

MR dampers typically operate in three modes: shear, flow, and mixed modes (the combination of the first two modes). The flow and mixed modes of MR dampers can provide high damping because of the pressure generated in the lower and upper chambers. However, they are more complicated and have higher cost than the shear-mode due to a large amount of MR fluid used in the chambers. It is noteworthy that washing machines do not require a so high damping force, but the zero-field friction force needs to be as small as possible to suppress the force transmissibility at high frequencies. Consequently, the shear-mode is implemented in our study. The schematic of the proposed self-powered MR damper in shear-mode is shown in Figure 2. As shown in the figure, the shaft of damper moves reciprocally due to the vibration of washing machines, developing damping force via direct shearing of the MR fluid. With this configuration, the interaction of the magnetic field between the damping and energy-harvesting parts could be diminished.

Figure 2. Schematic of the proposed self-powered shear-mode magnetorheological (MR) damper.
Compared with conventional MR dampers, the self-powered shear-mode MR damper is able to adapt itself based on the mechanical energy produced from its working process via an energy-harvesting vibration absorber. In general, the self-powered MR damper consists of two main components: the MR damping part and the energy-harvesting one. The MR damping part is composed of an inner housing with a thin wall, on which the exciting coils are wound directly, and an outer housing for covering and creating a closed magnetic circuit of the damper. The MR fluid is fully injected into the gap between the shaft and inner housing. The cross-section area of the thin wall is built to be as small as possible for a quick magnetic saturation of the flux lines. As a result, the magnetic flux is compelled to move across the MR fluid gap. In addition, appropriate chamfers are added to the cross-section of the coils to maximize this magnetic flux. When a magnetic field is applied, the solidification of the MR fluid occurs, and the damping force is produced from the friction between the housing and the shaft. In some researches, the MR damper could be designed with more exciting coils to improve the damping effectiveness. However, this lengthens the damper, which may not satisfy the installation space in washing machines. Furthermore, the more coils are used, the more exciting powers are demanded, which leads to high heat emission and more maintenance cost. Considering the above problems, the design with two exciting coils is proposed for the MR damping part.

The energy-harvesting (EH) part consists of magnetized permanent magnets and pole spacers alternately fastened together on the moving shaft and a slotted stator core covered by an outer housing. In this paper, the linear multipole type is proposed for the configuration of the EH part. One magnet and one adjacent pole spacer are grouped into one pole pair. The magnets are ring-shaped and possess axial magnetization; thus, they are placed in cross-pole positions to force the flux going through the spacers and across the air gap. The inducing coils are wound directly on the slots of the stator core and connected to the exciting coils of the damping part. Under linear excitations caused by the vibration of washing machines, the relative movement between the magnets mounted on the shaft end and the stator core appears. This results in an induced electric power supplied to the MR damping part.

3. Energy-Harvesting Part

In this section, the EH part is designed to power the MR damping part. Based on the principle of electromagnetic induction, the EH part will convert the kinetic energy resulting from vibration into electricity for the exciting coils of the MR damping part.

3.1. Modeling of the Energy-Harvesting Part

Figure 3 shows the schematic of the EH part. The pole spacers and stator core are made of commercial C45 steel, which is a common material with high magnetic permeability. The NdFeB grade N35 ring-shaped magnets are fitted on an aluminum shaft end alternately with the pole spacers. The magnet has an outer diameter of 28 mm, an inner diameter of 6 mm, and a length of 7 mm. In our research, the prototype front-loaded washing machine is the WF8690NGW washing machine produced by Samsung Electronics Co., Ltd., Seoul, Korea. Considering the available assembly space of dampers in the washing machine, the number of magnets and coils on the stator core are chosen to be 2 and 7, respectively. In this way, the maximum number of coils in the electromagnetic working state will be 4 out of 7 coils.
Figure 3. Schematic of the energy-harvesting part.

By neglecting the reluctance of pole spacers and stator core, the magnetic flux of the cylindrical air gap between the magnets and the stator core $\Phi_g$ is given as [17–20]

$$\Phi_g = \lambda \frac{B_{rem} l_m H_{ccoe} A_{gm}}{2r_{gm} B_{rem} + 1_m H_{ccoe} (A_{gm} / A_m)}$$  \hspace{1cm} (1)

where $\lambda$ expresses the efficiency of the magnetic flux considering leakage effect, $B_{rem}$ characterizes the magnet remanent flux density, $H_{ccoe}$ denotes the magnetic coercivity of the magnet, $\mu_0$ represents the relative magnetic permeability and is equal to $4\pi \times 10^{-7}$ N/A$^2$, $lm$ is the length of the magnet, $t_{gm}$ is the thickness of the air gap, $A_{gm}$ is the cylindrical surface area of the air gap, and $A_m$ is the sectional area of the magnet. $A_{gm}$ and $A_m$ can be determined by

$$A_{gm} = 2\pi \left( r_o + t_{gm} \right) \left( \frac{p_m - l_m}{2} \right)$$  \hspace{1cm} (2)

$$A_m = \pi \left( r_o^2 - r_i^2 \right)$$  \hspace{1cm} (3)

where $r_i$ and $r_o$ are the inner and outer radii of magnet, respectively, and $p_m$ is the pitch of the pole pair. The induced voltage $E$ in the inducing coil is defined as

$$E = -N \Phi_g \frac{\pi}{p_m} \sin \left( \frac{\pi}{p_m} x + \phi_0 \right) \frac{dx}{dt}$$  \hspace{1cm} (4)

where $x$ and $\frac{dx}{dt}$ are the displacement and velocity of the shaft, respectively, and $\phi_0$ is the initial phase angle of the inducing coil. The number of winding turns of each inducing coil $N$ is calculated by [17]

$$N = \frac{2A_c}{\sqrt{3} d_w^2}$$  \hspace{1cm} (5)

where $d_w$ is the diameter of the copper wire and $A_c$ is the cross-sectional area of the coil determined by the product of the coil height $h_{cm}$ and the coil width $w_{cm}$. Since the pitch of the pole pair $p_m$ is designed to double the pitch of the coil slot $p_{nm}$, the phase angle is $\pi/2$ between each nearby coil. Accordingly, the induced voltages in four active coils are

$$E_i = -N \Phi_g \frac{\pi}{p_m} \sin \left( \frac{\pi}{p_m} x \right) \frac{dx}{dt}$$  \hspace{1cm} (6)
\begin{align}
E_2 &= -N\Phi g \frac{\pi}{p_m} \cos \left( \frac{\pi}{p_m} x \right) \frac{dx}{dt} \\
E_3 &= -E_1 \\
E_4 &= -E_2
\end{align}

(7)

In case of low excitation frequencies, by neglecting the coil inductance, the coil is supposed to work as a resistor. The power of four active coils \( P \) is obtained by

\[ P = 2 \left( P_1 + P_2 \right) = 2 \left( \frac{E_1^2}{R_c} + \frac{E_2^2}{R_c} \right) = 2 \left( \frac{N\Phi g \frac{\pi}{p_m}^2}{R_c} \right) v^2 \]

(10)

where \( P_1 \) and \( P_2 \) are the powers of coils 1 and 2, respectively; \( R_c \) is the resistance of each coil; and \( v \) is the velocity of the shaft. Equation (10) describes the generated power of four active coils as a quadratic function of the excitation velocity. From the equation, it is also observed that geometric dimensions of the stator core and the pole spacers considerably affect the power magnitude. Consequently, to achieve the best performance of power generation, an optimization procedure for the design of the EH part should be implemented.

3.2. Optimal Design of the Energy-Harvesting Part

As mentioned above, the generated power in Equation (10) should be as large as possible to maximize the efficiency of the EH part. In the design of the EH part, the height of coil \( h_{cm} \), the width of coil \( w_{cm} \), the pitch of coil \( p_c \), the pitch of pole pair \( p_{m} \), and the thickness of housing \( t_{om} \) are significant dimensions and thus set as design variables. Another problem that should be considered is the available assembly space in the washing machine. Since the total length of the damper in equilibrium is approximately 230 mm and the desired maximum damper stroke is established by 40 mm, the length of the shaft end carrying the pole pairs \( L_s \) is restricted to be smaller than 30 mm. From the aspects of compactness and low cost, it would not be ideal to improve the possibility of generating power by increasing the EH part size. Therefore, although the outer radius of the EH part has no restriction, it should not be so much larger than that of commercial passive dampers and conventional MR dampers. In this optimization, the value of 22 mm is chosen to be the upper limit of radius \( R \) of the EH part. The final concern is the possibility of machining and assembling the stator core without warping, for which the thickness of the housing \( t_{om} \) is set to be 2 mm or greater. In summary, the design optimization of the EH part can be expressed: Determine the optimal geometry of the proposed EH part that maximize the generated power, subjected to the outer radius \( R \) is smaller than 22 mm and the length of the shaft end \( L_s \) is smaller than 30 mm.

In this work, the finite element model of the EH part is implemented using the 2D-axisymmetric couple element (plane 13) of ANSYS. It is noteworthy that the geometry of the EH part will continuously vary during the optimization process, so the meshing size is defined by the constant number of elements per line rather than the size of each element. Figure 4 shows the finite element model of the EH part. In order to obtain the optimal design of the EH part, the golden section algorithm combined with first-order method of the ANSYS optimization tool is utilized. The procedure in detail based on finite element analysis (FEA) has been presented in some researches [21,22].
The optimal solution of the proposed EH part is specified in Table 1. It is noted that the thicknesses of the air gap $t_{gm}$ and thin wall $t_{wm}$ should be as small as possible. In this study, both are set by 0.8 mm considering the manufacturing cost. Figure 5a,b shows the magnetic flux distribution and magnetic flux density of the optimized EH part, respectively. From Figure 5a, it is observed that the magnetic flux passes through the air gap between the magnets and the stator core. As a result, the induced voltages could be generated in the inducing coils of the EH part to power the exciting coils of the MR damping part. Besides, there is almost a saturation of magnetic flux toward the south poles of the magnets, which is marked in Figure 5b.

| Parameter (mm)   | Value | Parameter (mm)   | Value |
|------------------|-------|------------------|-------|
| Coil height $h_{cm}$ | 4.4   | Air gap thickness $t_{gm}$ | 0.8   |
| Coil width $w_{cm}$ | 4.56  | Thin wall thickness $t_{wm}$ | 0.8   |
| Coil pitch $p_c$  | 6.74  | Housing thickness $l_{om}$   | 2     |
| Magnet length $l_m$ | 7     | Outer radius $R$            | 22    |
| Pole pair pitch $p_m$ | 13.48 | Generated power $P$ (W)    | 19.3  |

**Figure 4.** Finite element model of the energy-harvesting part.

**Figure 5.** Finite element analysis of the optimized energy-harvesting part: (a) magnetic flux lines and (b) magnetic flux density.
4. MR Damping Part

4.1. Modeling of the MR Damping Part

The MR damping part is coaxial with and in front of the EH part. Figure 6 shows the schematic diagram of the shear-mode MR damping part. The shaft and the housing are made of C45 steel. The O-rings made of 70-durometer NBR rubber are used to seal the MR fluid.

\[ F_d = 2\pi R_s L_e (\tau_y + \eta \frac{v}{t_g}) + 2F_{or} \]  
(11)

\[ F_0 = 2\pi R_s L (\tau_y + \eta \frac{v}{t_g}) + 2F_{or} \]  
(12)

where \( R_s \) is the shaft radius; \( t_g \) is the MR fluid gap thickness; \( v \) is the velocity between the housing and the shaft; \( \tau_y \) and \( \eta \) are the yield stress and post-yield viscosity of the active MR fluid in the gap, respectively; \( L \) is the MR fluid gap length; and \( L_e \) is the effective length of the active MR fluid in the gap. In this configuration, \( L_e \approx L \). The Coulomb friction force \( F_{or} \) between the O-ring and the shaft can be determined by [23]

\[ F_{or} = f_c L_e + f_h A_e \]  
(13)

where \( L \) is the seal surface length and in this case, equals to the circumference of the shaft; \( f_c \) is the friction per unit length caused by the compression of the O-ring; \( A_e \) is the projected area of the seal; and \( f_h \) is the friction force caused by the fluid pressure on a unit projected area of the O-ring. It is noted that in the configuration of the shear-mode, the pressure on the O-rings could be ignored since it is almost negligible, \( f_h \approx 0 \). Moreover, the O-rings should be compressed moderately so that the zero-field friction force is not so high while the sealing ability is still guaranteed during the working process of washing machines. Accordingly, a compression of 15% is set for the O-rings and \( f_c \) is found to be 175.1 N/m.

For our research, the 132-DG MR fluid produced by Lord Corporation is employed, and the Bingham model is used to represent the general behavior of the MR fluid. Despite the inconsistency with experimental responses at low shear rate, the Bingham model is beneficial to the design of MR damper-based devices due to its simplicity and quick modeling. Based on this model, the dependence of MR fluid rheological properties on the applied magnetic field are given by [24]
\[ Y = Y_\infty + (Y_0 - Y_\infty)(2e^{-B\alpha SY} - e^{-2B\alpha SY}) \]  

(14)

where \( B \) is the applied magnetic density, \( Y \) expresses one of the MR fluid rheological properties such as the postyield viscosity \( \eta \) and the yield stress \( \tau_y \), and \( \alpha_{SY} \) is the saturation moment of the \( Y \) property. The \( Y \) value varies from the value of off-state \( Y_0 \) to the value of saturation \( Y_\infty \). The curve-fitting method is used to estimate the values of \( Y_0 \), \( Y_\infty \), and \( \alpha_{SY} \) from experimental data, and the results are shown in Table 2.

Table 2. Rheological properties of the magnetorheological (MR) fluid 132-DG.

| Parameters | \( \eta_0 = 0.1 \text{ Pa.s} \) | \( \tau_{0y} = 15 \text{ Pa} \) | \( \alpha_{SY} = 4.5 \text{ T}^{-1} \) |
|------------|---------------------------------|------------------|-----------------|
| \( \eta_\infty = 3.8 \text{ Pa.s} \) | \( \tau_{\infty y} = 40,000 \text{ Pa} \) | \( \alpha_{SY} = 2.9 \text{ T}^{-1} \) |

4.2. Optimal Design of the MR Damping Part

In order to improve the performance of the MR damping part during the operating process of washing machines, the maximum damping force (the damping force when a magnetic field is applied) determined by Equation (11) reaches to a required value at low resonance frequency whereas the zero-field friction force in Equation (12) is minimized at high excitation frequencies. Therefore, the dynamic model of washing machines should be first analyzed. The 2D mechanical modeling of the washing machine Samsung WF8690NGW is shown in Figure 7.

![Figure 7. A 2D mechanical modeling of the washing machine.](image)

The vibration equation of the washing machine shown in Figure 7 can be represented [10] as

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

(15)

where \( k \) is the spring stiffness, \( c \) is the damping coefficient, \( m \) is the total mass of the tub assembly, \( \varphi \) is the angle from the \( x \) axis of an arbitrary direction \( u \) of the vibration analysis, and \( \Delta u \) is the displacement of the center of the tube in the \( u \) direction. The exciting force \( F_u \) due to an imbalanced mass in the \( u \) direction is defined by

\[ F_u = F_{u0} \cos (\alpha x) = m_u \omega_u^2 R_u \cos (\alpha x) \]  

(16)

where \( F_{u0} \) is the magnitude of the exciting force and \( R_u \) and \( m_u \) are the radius and mass from the rotary axis of the imbalanced mass, respectively. From Equation (15), the damped frequency \( \omega_d \), natural frequency \( \omega_n \), and damping ratio \( \xi \) of the suspended tub assembly are given by
The resonance in all directions of vibration should be attenuated as much as possible. For this, by designing $\alpha_1 + \alpha_2 = 90^\circ$ and $\beta_1 + \beta_2 = 90^\circ$, Equations (15), (18), and (19) can be simplified to be independent of the angle $\varphi$ as follows

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

$$\omega_n = \sqrt{\frac{k}{m}}; \quad \xi = \frac{c}{2\sqrt{mk}}$$

In this case, the force transmissibility $T_f$ from the tub assembly to the cabinet is derived by

$$T_f = \frac{1 + (2\xi r_\omega)^2}{\sqrt{(1 - r_\omega^2)^2 + (2\xi r_\omega)^2}}$$

where $r_\omega = \omega/\omega_n$ is the frequency ratio. Figure 8 presents the relation between the force transmissibility and the excitation frequency under different damping ratios. From the figure, it is observed that the resonant peaks are nearly eliminated as the damping ratio $\xi$ reaches to 0.7. Thus, it is not necessary to get the greater value of $\xi$.

According to [10], the expected damping force for the MR damping part $F_r$ can be determined by

$$F_r = \frac{kX \pi \xi r_\omega}{2}$$

where the tub vibration amplitude $X$ is given by

$$X = \frac{F_{u0}}{k} \sqrt{\frac{1}{(1 - r_\omega^2)^2 + (2\xi r_\omega)^2}} = \frac{m_\omega r_\omega^2 R_\omega}{m} \sqrt{\frac{1}{(1 - r_\omega^2)^2 + (2\xi r_\omega)^2}}$$

Figure 8. Relation between the force transmissibility and the excitation frequency under a damping ratio of 0.1–0.7 with an increment of 0.1.
In this study, it is assumed that the total mass of the tub assembly \( m \) is 40 kg, the spring stiffness \( k \) is 10 kN/m, and the imbalanced mass \( m_u \) is 7 kg placed at the radius from the rotary axis \( R_u = 0.125 \) m. For the desired damping ratio \( \xi = 0.7 \) at the resonance frequency ratio \( r = \sqrt{1 - \xi^2} \), the expected damping force \( F_r \) in Equation (23) is calculated as 78.7 N. Consequently, the damping force object for the MR damping part is established by 80 N.

In the design optimization of the MR damping part, the design variables are the radius of shaft \( r_s \), the height of coil \( h_c \), the width of coil \( w_c \), the dimensions of chamfer \( h_{ch} \) and \( w_{ch} \), the length of pole \( l_p \), and the thickness of housing \( t_h \). Considering the installation space in the washing machine, the available length of the MR fluid gap \( L \) is approximately 60 mm. For simple structure, the outer radius of the MR damping part is set to equal to that of the EH part. The final restriction comes from, similar to the EH part, the ability of machining and assembling, for which the thickness of the housing \( t_h \) are set to be greater than 2 mm. The design optimization problem of the MR damping part, in summary, can be expressed: Determine the optimal geometry of the proposed MR damping part that minimize the off-state friction force \( F_0 \), subjected to the length of the MR fluid gap \( L \) is smaller than 60 mm and the maximum damping force \( F_d \) is higher than 80 N (the expected damping force).

Figure 9 shows the finite element model of the MR damping part created on the ANSYS software. First, the magnetic circuits of the MR damping part are solved based on FEA to calculate the magnetic density \( B \). The postyield viscosity \( \eta \) and yield stress \( \tau_y \) are then estimated using Equation (14). Finally, these parameters are substituted into Equations (11) and (12) to obtain the damping force and zero-field friction force, respectively.

Table 3. Optimal parameters of the MR damping part.

| Parameter (mm)         | Value | Parameter (mm)         | Value |
|------------------------|-------|------------------------|-------|
| Coil height \( h_c \)  | 8.57  | Thin wall thickness \( t_w \) | 0.8   |

Figure 9. Finite element model of the MR damping part.
Coil width $w_c$ 13.81 | Housing thickness $t_o$ 3.58
Coil height chamfer $h_{ch}$ 3.05 | Shaft radius $r_s$ 8.25
Coil width chamfer $w_{ch}$ 5.9 | Outer radius $R$ 22
Pole length $l_p$ 6.88 | Max. current intensity $I$ (A) 0.34
MR fluid gap length $L$ 55.13 | Max. damping force $F_d$ (N) 80.2
MR fluid gap thickness $t_g$ 0.8 | Zero-field friction force $F_o$ (N) 18.4

Figure 10. Finite element analysis of the optimized MR damping part: (a) magnetic flux line and (b) magnetic flux density.

5. Experimental Validation

Figures 11 and 12 show the prototype of the optimized self-powered MR damper and experimental setup, respectively, for verifying the optimal design results and evaluating the performance of the damper. In this test system, a servo motor (MSMD022S1T, Panasonic) with gearbox of ratio 15:1 is employed to provide a constant rotary speed for the crank shaft. The rotary motion of the motor is converted to the linear motion of the damper shaft by a crank-slider mechanism. The damping force is measured by a bidirectional force sensor produced by Forsentek Co., Limited, Shenzhen, China (FFG-200N, maximum measured force: 200 N) while the instantaneous velocity and displacement are measured by a linear variable differential transformer (LVDT) produced by RPD Electrosense (ACT1000A, measurement range: +/-25 mm). In order to compare the self-power MR damper with the MR damper using external power, a programmable power supply (PPW-8011, TWINTEX) is employed to supply power to the coils of the damping part. The power supply is controlled by the computer through the DAQ (myRIO 1900, National Instruments) to supply desired currents. It is noted that signals from the force sensor and LVDT are also transferred to the computer through the DAQ.
Figure 11. Prototype of the optimized self-powered MR damper (a) and its disintegrated components (b).

Figure 12. Experimental setup for testing the performance of the prototype self-powered MR damper.

5.1. Performance of the Energy-Harvesting Part

From Equations (6)–(9), it is shown that the induced voltages of two contiguous coils express the properties of the EH part. To validate the efficiency of the power generation, a harmonic excitation of 2.4 Hz frequency and 30 mm amplitude is applied to the prototype self-powered MR damper. This is the frequency when the first resonance often appears in most commercial washing machines. Figure 13 presents the comparisons between the theoretical modeling and experimental measurement of the induced voltages of coils 1 and 2. As shown in the figure, it is observed that the experimentally measured induced voltages are slightly lower than those obtained from theoretical analysis. This mostly results from the measurement noises and magnetic flux leakage. The average value of the produced voltages of coil 1 and coil 2 are about 93.7% and 93.8% of the simulated value, respectively. Thus, the experimental data matches well with the simulation results. Moreover, numbers of peaks are found in one excitation period of the generated voltages, due to the effect of the induced voltage frequency multiplication [17]. This would increase the overall harvested power and thus is very advantageous to the EH part.
Figure 13. Comparisons between the experimental and modeling results of the induced voltages of coils under 2.4 Hz frequency and 30 mm amplitude: (a) coil 1 and (b) coil 2.

Figure 14 shows the comparison between the numerical analysis and experimental results of the generated power with respect to the velocity excitation. As shown in the figure, the produced power almost exhibits a parabolic relation to the velocity excitation. The average error between the theoretical and experimental generated powers is about 8.4%. As a result, the experimental data of the generated power matches well with the simulation results.

5.2. Damping Force under Constant Magnetic Field

The damping performance is tested when the EH part is open and different constant currents are applied to the exciting coils of the MR damping part by an external power supply. Figure 15 shows the force-velocity and force-displacement curves under the often-resonant frequency of 2.4 Hz, amplitude of 30 mm, and different currents of 0, 0.2, 0.34 (the maximum value in Table 3 obtained from theoretical modeling), and 0.4 A. It is noted that the damping part of the self-powered damper in this research are similar to that of the MR damper in [12]; therefore, the experimental results in Figure 15 are almost the same as those in [12]. From the figure, it can be realized that the damping force significantly depends on the external magnetic field. When there is no current applied to the exciting coils \((I = 0)\), the force-displacement curve is almost elliptic and the force-velocity relation is approximately linear, which indicate that the behavior of the MR damper is nearly similar to that of a pure viscous damper. However, the forces in this case are not concentrated...
in the vicinity of zero due to the presence of the frictional forces between the shaft and the O-rings. There is a slight difference between the zero-field force obtained from the experiment and that obtained from the theoretical modeling. The average value of the zero-field force at the steady position (the position relatively far from the stroke ends) is 19.5 N, which is around 106% of the modeling value (18.4 N). This results from the inaccurate estimation of the friction force of the O-rings. It is believed that the accuracy would be improved with further tests on the O-rings.

When the magnetic field intensity increases ($I > 0$), the area of the force-displacement loop increases considerably, meaning that additional damping forces are produced. As shown in the figure, when the current of 0.34 A is applied to the coils, the maximum damping force increases from 19.5 to 76.2 N, which is around 95% of the simulated one (80.2 N). The difference mainly comes from the loss of magnetic field to the ambient and at the contact between the magnetic parts. However, when the current is increased to 0.4 A, no more significant change in force is found. Thus, it is proved that the damping force almost reaches to saturation at 0.34 A of the applied current.

![Figure 15. Experimental responses under a frequency of 2.4 Hz, an amplitude of 30 mm, and different currents of 0, 0.2, 0.34 (continuous lines from inside to outside), and 0.4 A (dash line): (a) force vs. velocity and (b) force vs. displacement.](image)

Moreover, as shown by the force–velocity responses in Figure 15a, the self-powered MR damper experiences two particular rheological regions: the preyield (rectangular mark) and postyield (the rest of the loop) regions. The preyield region exposes a strong hysteretic phenomenon while the postyield one almost exhibits a linear relationship. It is observed that in the preyield region, a sudden decrease/increase in value of damping forces corresponding to small velocities appears. This roll-off effect has been investigated in many researches.

5.3. Self-Adaptability of the Damping Force
Experiments are implemented to verify the self-adaptability of the damping force when the EH part is close. Figure 16 shows the comparison between the damping force with the EH part and that under constant currents when an excitation of 2.4 Hz frequency and 30 mm amplitude is applied to the prototype self-powered MR damper. As shown in the figure, the damping force with the EH part varies between the levels of 0 and 0.34 A constant currents. The maximum damping force with the EH part is 75.1 N, which is almost the same as that under the 0.34 A constant current (76.2 N) since the difference is only about 1.4%. It is also observed from Figure 16a that the force-velocity loop of the MR damper powered by the EH part experiences a hysteresis in the preyield region, but not a linear relationship in the postyield region. The explanation can be easily found on the continuous variation of the voltages generated by the EH part. Besides, in Figure 16b, it is realized that the area of the force-displacement loop with the EH part is smaller than that under the 0.34 A constant current. The generated work of the MR damper in case of the 0.34 A constant current is about 8.7 J, while the MR damper with the EH part handles only about 7.4 J. In other words, the MR damper supplied by the external power source with the constant current dissipates more energy than that by the EH part.

Figure 16. Experimental results of the MR damper with the energy-harvesting (EH) part and under constant external currents in the case when the excitation is 2.4 Hz frequency and 30 mm amplitude: (a) force vs. velocity and (b) force vs. displacement.
The same conclusions could be drawn from Figures 17 and 18 when the prototype damper is tested under a 2 Hz frequency and a 21 mm amplitude and a 2.2 Hz frequency and a 26 mm amplitude, respectively. These are also the excitations that often appear when washing machines are entering the resonance region. Although the maximum damping forces produced by the MR damper with the EH part decrease due to the reduction of the excitation velocities, they are still high enough to attenuate the vibration of washing machines that has not reached to resonance peak yet. Consequently, the feasibility of the proposed self-powered shear-mode MR damper for washing machines has been demonstrated.

![Figure 17](image1.png)

**Figure 17.** Experimental results of the MR damper with the EH part and under constant external currents in the case when the excitation is 2 Hz frequency and 21 mm amplitude: (a) force vs. velocity and (b) force vs. displacement.
Figure 18. Experimental results of the MR damper with the EH part and under constant external currents in the case when the excitation is 2.2 Hz frequency and 26 mm amplitude: (a) force vs. velocity and (b) force vs. displacement.

6. Conclusions

This paper focused on the design and evaluation of a self-powered shear-mode MR damper for washing machines to dissipate the vibratory energy due to an imbalanced laundry mass occurring in the washing drum. In order to improve the performance of the proposed damper, an optimization procedure employing the ANSYS FEA with golden section algorithm was performed considering expected damping force, installation space, size, and production cost. Based on the optimal solutions, a prototype of the proposed self-powered MR damper was designed and manufactured. Theoretical analyses and experimental studies on the EH part were performed. The induced voltage of each coil was about 94% of the predicted value and the appearance of the frequency multiplication effect would be beneficial to the overall harvested energy. The generated power almost represented a quadratic function of the velocity with the average error of about 8.4%. Accordingly, a good correlation between the modeling based on FEA and experimental results of the EH part was achieved.

For the damping ability under constant magnetic fields, the performance of the MR damping part was tested when the EH part was open. The results showed that the proposed MR damper could provide and control effectively the damping force. The damping force almost reached to
saturation at 0.34 A current, which agreed well with the theoretical simulation. The zero-field friction force was a little greater than the modeling value. It was advised that further experiments on the O-rings should be implemented to better evaluate their friction forces.

For the self-adaptability of the damping force, the MR damper with the closed EH part was investigated and discussed via both simulation and experiment. The experimental results indicated that the proposed self-powered MR damper could generate the same maximum damping force as under corresponding constant currents but dissipated less energy. This would reduce the power consumption and heat emission as compared with the conventional MR dampers.

With the EH part integrated into the MR damper, the system could be powered by itself from wasted mechanical energy. The complication and cost of the system are considerably reduced since the proposed damper could be self-adaptable to external excitations. However, comparing with MR damper working on external power, damping force obtained from the self-powered damper has more fluctuation. As the second phase of this research, further efforts will concentrate on experimentally studying the friction force of the O-rings and validating the proposed self-powered MR damper in prototype washing machines.

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Nomenclatures

\[ \begin{align*}
A_c & \quad \text{Sectional area of the inducing coil} \\
A_{gm} & \quad \text{Cylindrical surface area of the air gap} \\
A_m & \quad \text{Sectional area of the magnet} \\
A_r & \quad \text{Projected area of the seal} \\
B & \quad \text{Applied magnetic density} \\
B_{rem} & \quad \text{Magnet remanent flux density} \\
c & \quad \text{Damping coefficient} \\
d_w & \quad \text{Diameter of the copper wire} \\
E & \quad \text{Induced voltage in the inducing coil} \\
E_i & \quad \text{Induced voltages in the } i \text{ active coil} \\
EH & \quad \text{Energy-harvesting} \\
F_0 & \quad \text{Zero-field friction force} \\
F_d & \quad \text{Damping force} \\
F_{or} & \quad \text{Coulomb friction force between the shaft and the O-ring} \\
F_r & \quad \text{Required damping force for the MR damping part} \\
F_u & \quad \text{Exciting force due to an unbalanced mass in the } u \text{ direction} \\
F_{u0} & \quad \text{Magnitude of the exciting force} \\
f_c & \quad \text{Friction per unit length caused by the compression of the O-ring} \\
f_h & \quad \text{Friction force caused by the fluid pressure on a unit projected area of the O-ring} \\
H_{coe} & \quad \text{Magnetic coercivity of the magnet} \\
h_c & \quad \text{Height of the exciting coil} \\
h_{ch}, w_{ch} & \quad \text{Chamfer dimensions of the exciting coil}
\end{align*} \]
\( h_{cm} \) Height of the inducing coil
\( I \) Applied current
\( k \) Stiffness of each spring
\( L \) Length of the MR fluid gap
\( L_e \) Effective length of the active MR fluid in the gap
\( L_s \) Length of the seal rubbing surface
\( L_o \) Length of the shaft end
\( l_m \) Length of the magnet
\( l_p \) Length of the pole of the MR damping part
\( MRF \) Magnetorheological fluid
\( m \) Mass of the suspended tub assembly
\( m_u \) Unbalanced mass
\( N \) Number of winding turns of each inducing coil
\( P \) Power of four active coils
\( P_1, P_2 \) Powers of coil 1 and coil 2
\( p_s \) Pitch of the coil slot of the EH part
\( p_{mp} \) Pitch of the pole pair of the EH part
\( R \) Outer radius of the self-powered MR damper
\( R_e \) Resistance of each inducing coil
\( R_o \) Radius of the shaft
\( R_u \) Radius from the rotation axis of the unbalanced mass
\( r_i \) Inner radius of the magnet
\( r_o \) Outer radius of the magnet
\( r_s \) Radius of the shaft
\( r_o \) Frequency ratio
\( T_f \) Force transmissibility from the tub assembly to the cabinet
\( t \) Time
\( t_g \) Thickness of the MR fluid gap
\( t_{gm} \) Thickness of the air gap
\( t_o \) Thickness of the outer housing of the MR damping part
\( t_{om} \) Thickness of the housing of the EH part
\( t_w \) Thickness of the thin wall of the MR damping part
\( t_{om} \) Thickness of the thin wall of the EH part
\( v \) Velocity of the shaft
\( w_c \) Width of the exciting coil
\( w_{cm} \) Width of the inducing coil
\( X \) Amplitude of the tub vibration
\( x \) Displacement of the shaft
\( Y \) MR fluid rheological property such as postyield viscosity or yield stress
\( \alpha_1, \alpha_2 \) Angle between the spring and the y-axis
\( \alpha_{SY} \) Saturation moment of the Y property
\( \beta_1, \beta_2 \) Angle between the damper and the y-axis
\( \Delta u \) Displacement of the center of the tube assembly in the u direction
\( \eta \) Postyield viscosity of the active MR fluid in the gap
\( \lambda \) Efficiency of the magnetic flux considering leakage effect
\( \mu_0 \) Relative magnetic permeability
\( \xi \) Damping ratio
\( \tau_y \) Field-dependent yield stress of the active MR fluid
fluid in the gap

$\Phi_g$ Magnetic flux of the air gap between the magnets and the stator core

$\varphi$ Angle of the $u$ direction

$\phi_0$ Initial phase angle of the inducing coil

$\omega_d$ Damped frequency

$\omega_n$ Natural frequency

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