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Research Article

Design and simulation of a new self–adaptive MR damper for washing machines featuring shear–mode and radial permanent magnets

Quoc Duy Bui[1,2], Quoc Hung Nguyen[3,*]

[1]*Faculty of Civil Engineering, HCMC University of Technology and Education, Ho Chi Minh City, Vietnam*

[2]*Faculty of Mechanical Engineering, Industrial University of Ho Chi Minh City, Ho Chi Minh City, Vietnam*

[3]*Faculty of Engineering, Vietnamese–German University, Thu Dau Mot City, Binh Duong Province, Vietnam*

Correspondence

Quoc Hung Nguyen, Faculty of Engineering, Vietnamese–German University, Thu Dau Mot City, Binh Duong Province, Vietnam

Email: hung.nq@vgu.edu.vn

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ABSTRACT

This research investigates a new magneto–rheological (MR) damper for washing machines which can effectively replace conventional passive dampers in the field of vibration control. Against traditional MR dampers and self–powered ones, the new damper is extremely cost–effective for commercialization coming from its compact structure and self–adaptive ability to external mechanical vibration for reasonable damping without any equipment or control. By adding a layer of MR fluid (MRF) between the damper shaft and housing, the damping coefficient can be adjusted according to applied magnetic field intensity. Radial ring permanent magnets are installed on the housing and the middle shaft part is made of non–magnetic material for the inactive state of the damper. Under great vibration of washing machines, the magnetic shaft parts move into the region of MRF and magnets, which generates closed magnetic circuits. Subjected to applied magnetic fields, the MRF undergoes a transition from free flowing to semi solid, thereby experiences increases in its rheological characteristics such as yield stress and viscosity. The relative movement between the shearing shaft and the MRF in the gap is then resisted and additional damping force is produced. The higher the oscillatory amplitude is the more magnetic shaft parts contact the MRF, which results in stronger induced magnetic fields and a larger damping force to suppress the oscillation. This displacement–based damping characteristic is especially proper for vibration suppression of washing machines. In order to enhance oscillation isolation efficiency and at the same time meet criteria of assembly space, size and cost, optimization is performed for essential geometrical dimensions of the proposed damper. Then, the advantages in performance characteristics of the proposed damper are figured out from the optimization results with discussions.

Key words: radial permanent magnet, washing machine, vibration control, shear–mode, self–adaptive, MR damper, MR fluid

INTRODUCTION

Vibration of washing machines is always a big challenge that has received much attention from scientists and manufacturers. Imbalanced clothes mass often occurs in spin–drying stage leading to centrifugation and excited vibration. Especially, vibration is more severe with respect to front–loaded (horizontal spindle axis) washing machines due to gravity effect. In conventional passive dampers, the constant damping coefficient can isolate most vibration at resonant frequency (about 100 to 200 rpm), but causing the exciting force to be transferred greater from the washing drum to the panels and ground at high frequencies (usually 1000 rpm or more), which results in noises and walking of washing machines. Therefore, it is expected to effectively and continuously control the damping level of suspension systems for vibration attenuation at all frequencies.

An interesting approach to deal with the above issue is magneto–rheological (MR) dampers–based suspension systems. MR fluid (MRF) is a smart material that can solidify when placed under an applied magnetic field, whereby generates damping force via friction against adjacent parts. By altering the magnetic intensity, the damping force can be controlled accordingly. Relying on continuous, rapid and reversible responses, MR dampers have shown great promise in the field of vibration and/or shock absorption. Up to now, there have been an increase in studies of MR dampers.

LITERATURE REVIEW AND RESEARCH CONTRIBUTIONS

Many scholars have performed studies of MR dampers for washing machines such as Carlson[1], Chrzan and Carlson[2], Spelta[3], Aydar *et al.*[4], Nguyen *et al.*[5], Ulasyar and Lazoglu[6], Wang *et al.*[7]. However,

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flow–mode configuration of these dampers leads to high cost since they demand a large amount of MRF. Furthermore, their maximum damping forces are pretty greater than desired one for washing machines (about 80–100 N) while the off–state forces are relatively high (about 25–50 N), which are not sufficient to prevent force transmission at high spindle speeds. Thus, shear–mode configuration is more appropriate for MR dampers of washing machines[8–12]. The research results showed the feasibility of shear–mode MR dampers; however, inherent disadvantages of complexity and high cost resulting from associated sensors, power supply and controller still exist.

Recently, area of energy–harvesting has been exploited to combine with MR dampers into self–powered ones[13–15]. Choi and Wereley[16], Ferdaus *et al.*[17] developed self–powered MR dampers, integrating MR damping, dynamic sensing and energy harvesting technologies into one device. Chen and Liao[18], Sapinski[19], Zheng *et al.*[20], Hu *et al.*[21] implemented researches on energy harvesting linear MR dampers with linear multi–pole electromagnetic generator. Xinchun *et al.*[22], Chen *et al.*[23] devised a novel self–powered MR damper using DC generator combined with ball–screw mechanism. Recently, Bui *et al.*[24,25] have developed shear–mode MR dampers with self–powered component for washing machines. The dampers have integrated structure, low cost and have shown feasibility in vibration suppression through experimental works. Generally, self–powered MR dampers can adapt to external responses on their own by converting waste vibration energy into electrical one to power themselves for corresponding damping. Since no equipment or control system is required, cost of these dampers is also reduced. Nevertheless, complicated structure coming from the integrated energy–harvesting part, coils winding and connection still impedes manufacturing and maintenance. In addition, velocity–based damping characteristic of self–powered MR dampers may not be suitable for operating conditions of washing machines.

It is well–known that displacement and velocity of the drum are both high at resonance, so high damping level is sensible. At high spindle speeds, however, the velocity is still high but the displacement may not, which results in unnecessary active–state and force transmission of self–powered MR dampers. Therefore, the damping of MR dampers for washing machines should be adjusted according to vibratory excitation amplitude rather than excitation velocity. From the above analyses, this study work proposes a new MR damper in shear–mode with stroke–by–activated ability for washing machines. By replacing exciting coils (usually found in conventional MR damper configurations) with radial permanent magnets and assigning non–magnetic material to the middle shaft part, damping force is self–adaptable to exterior mechanical vibration. In other words, the higher the vibratory amplitude is the more magnetic shaft parts move into the region of MRF and magnets, and the higher the damping force is achieved. In this way, the damper's performance is based on displacement, which especially conform to washing machines. The design without winding coils and control systems also reduces the damper cost, which facilitates commercialization. To improve damping performance and effectiveness and at the same time meet criteria of assembly space, size and cost, optimization procedure is implemented for main geometrical dimensions of the proposed damper. Then, the advances in performance of the proposed damper are figured out from the optimization results with discussions.

Contributions of this research work can be listed as follows

- A new MR damper with self–adaptive ability to accommodate damping level to external mechanical ossilation.
- The compact structure and extremely low cost coming from no requirements of coils winding, sensors, power supply or control system, which enhance feasibility of the proposed damper in practice.
- The concept and application of controlling damping force according to vibratory excitation amplitude, which is very adequate to operation of washing machines.
- The optimized designs of three configurations that improve performance and applicability of the damper to washing machines with different loading capacities.

APPROACHING METHOD

Modeling of self–adaptive MR damper

Schematic of off– and on–states of the single–magnet self–adaptive shear–mode MR damper is sketched in Figure 1. In the damper, a layer of MRF is added between the damper shaft and inner cylindrical surface of housing. Pole pieces and a radial ring–shaped permanent magnet are installed into alternative positions on the housing. The middle shaft part is made of non–magnetic material for the damper off–state.

In Figure 1(a), the damper off–state is maintained whenever vibratory amplitudes are low since no magnetic shaft part contacts the MRF yet and hence no close magnetic circuit across the MRF is created. Under greater oscillation, as presented from Figure 1(b), more magnetic shaft parts start to move into the region of MRF and magnets, which applies magnetic fields to the MRF. With radial magnetism of the magnet, the magnetic flux leaves the north pole, passing through the gap of MRF, then the magnetic shaft part, and comes back to the south pole through the pole pieces. Penetrating the thin wall between the magnet and MRF is the prior way of the flux. Therefore, a magnetic saturation here should be quickly reached to drive the flux across the MRF gap, for that the thin wall size is desired to be possibly smallest. This configuration enables the self–adaptable ability of the damper, which increases the damping force with the external vibratory amplitude. As a result, at resonant frequency of rigid mode, a high damping level is produced (due to large drum displacement) to eliminate the washing machine oscillation while a small damping force is maintained at higher frequencies (due to small drum displacement) to prevent the force transmissibility to the floor. In other words, the damper is self–adaptable to mechanical oscillation without any control.

For improvement of the damper performance, configurations with two or more magnets are also studied and discussed. Figure 2 shows the proposed shear–mode MR dampers in configurations of two and three magnets, respectively. It is noted that the pole pieces and magnets are symmetrically distributed about a symmetric plane in optimal design of dampers with multiple magnets.

Main geometry of the single–magnet adaptive MR damper is shown in Figure 3 (two and three–magnet dampers are in the same manner). Materials are assigned to the damper components as follows: NdFeB grade N35 for the radial ring–shaped permanent magnets, NBR rubber with 70–durometer hardness for the O–rings sealing the MRF gap, commercial C45 steel for the magnetic shaft part, housing and pole pieces while aluminium for the non–magnetic shaft part.

The damping forces in active–state F_d and off–state F_0 (also referred as zero–field friction force) are obtained by

$$F_d = 2\pi r_s \left(\tau_y + \eta \frac{v}{t_g}\right) + 2F_{or} \tag{1}$$

$$F_0 = 2\pi r_s L \left(\tau_{y0} + \eta_0 \frac{v}{t_g}\right) + 2F_{or} \tag{2}$$

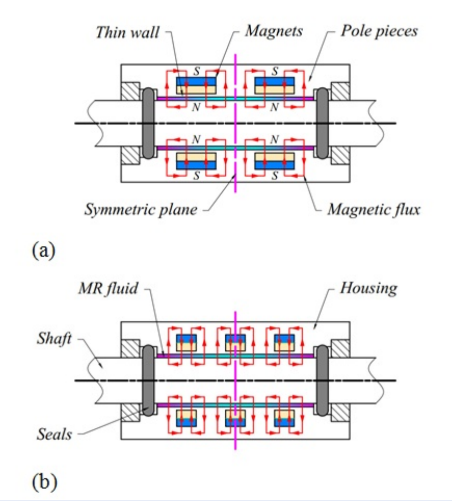


Figure 2: Schematic configurations of MR dampers with multiple magnets: (a) two–magnet and (b) three–magnet.

where h and t_y are the post yield viscosity and the yield stress of the MRF, respectively, v is the velocity, r_s is the shaft radius, and t_g is the MRF gap size. By using radial magnets, the effective length of the MRF gap L_{ef} can be approximated to the whole gap length L. In this design, $L = n_m l_m + n_p l_p$, where l_m, l_p denote the lengths of each magnet and pole piece; n_m, n_p are the numbers of magnets and pole pieces, respectively. By neglecting pressure of MRF applying on the O–rings due to shear operating mode, we can write the friction force F_{or} between the shaft and each O–ring referring to [26]

$$F_{or} = f_l L_r \tag{3}$$

where L_r is the sealing face length and f_l presents the friction for each unit length. Compression ratio of each O–ring is set by 15%, then 175.1 N/m is the calculated value of f_l.

According to [27], applied magnetic density B–based rheological characteristics can be estimated as follows

$$Y = Y_\infty + (Y_0 - Y_\infty)\left(2e^{-B\alpha_{SY}} - e^{-B\alpha_{SY}}\right) \tag{4}$$

Here Y presents one of the rheological properties of the MRF (t_y, h). Y_∞ and Y_0 are the saturated and off–state Y values, respectively. α_{SY} indicates the saturated Y moment index. The 140–CG MRF produced by Lord Corporation is used in this study and its rheological characteristics are given as follows[9]: $\tau_{y\infty}$ = 52000, τ_{y0} = 25 Pa, h_∞ = 4.4 Pa• s, h_0 = 0.29 Pa• s, $\alpha_{S\tau y} = 3\ \mathrm{T}^{-1}$, $\alpha_{Sh} = 5\ \mathrm{T}^{-1}$.

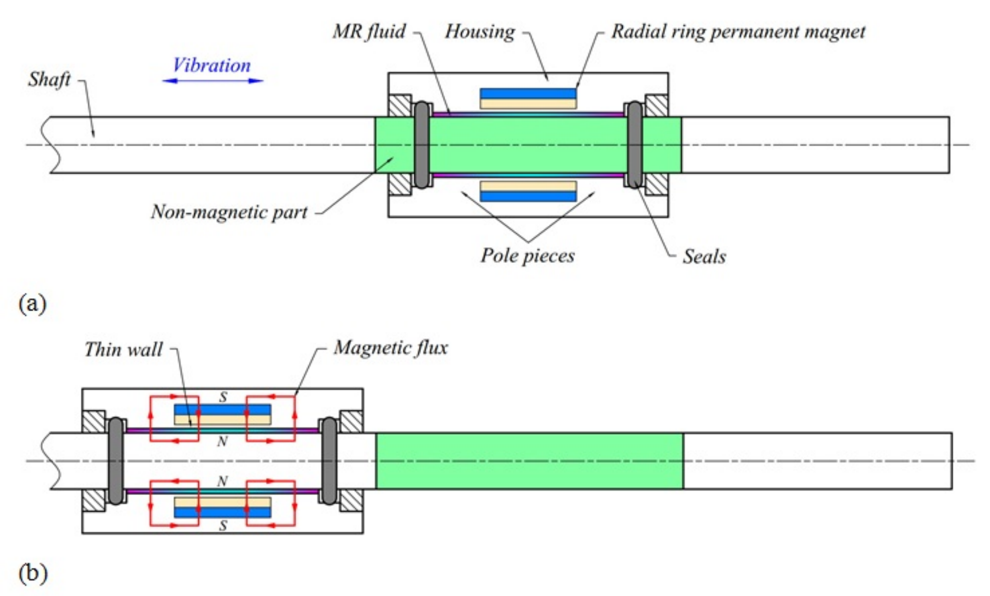


Figure 1: Configuration of the single–magnet MR damper: (a) off–state and (b) on–state.

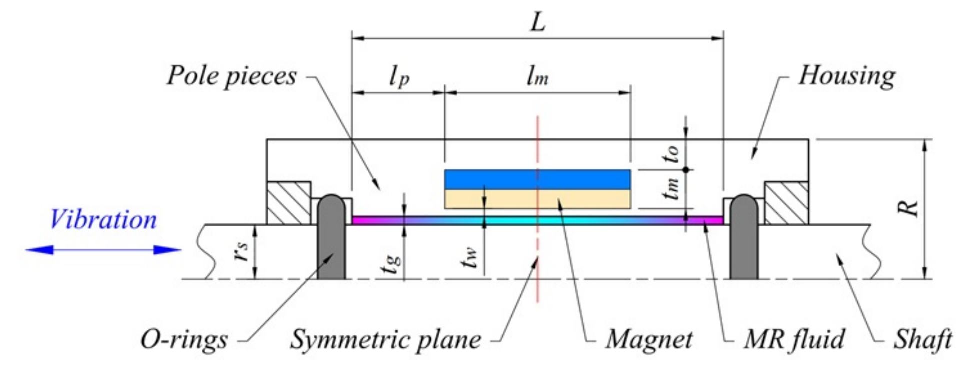


Figure 3: Main geometry of the single–magnet adaptive MR damper.

Optimal design of self–adaptive MR damper

In this work, optimization process for the proposed damper design is carried out to obtain best damping performance. Besides the maximum active damping force, the off–state force is also a parameter that should be under consideration. Obviously, the maximum damping force F_d should be high enough to attenuate oscillation at resonant frequency. The larger the damper size the greater the maximum damping force increases, but the off–state force F_0 may at once increase which intensifies the force transmission to the panels and ground at high frequencies. These indicate the conflict in the optimization objective function between the maximum active and off–state forces. In this study, the optimization objective is to find optimal essential geometry of the proposed damper which can generates a given expected maximum damping force while its off–state force is kept as small as possible. As mentioned in [10,24,25], the expected maximum damping force takes the value 80 N for the prototype 7 kg front–loaded washing machine labeled WF8690NGW produced by Samsung Electronics Co., Ltd.

In this configuration, the magnet length l_m, the pole length l_p, the magnet thickness t_m, the housing thickness t_o, and the shaft radius r_s are considered as design variables (DVs). It is noticeable that although small thicknesses of the thin wall and MRF gap are desirable to improve damping force, this impedes manufacturing process and raises production cost. Therefore, these two parameters are considered as constants and empirically take the value 0.8 mm from experience. Two constraints are also added to the optimization problem as follows. The first one is the length constraint of the operating region of the MRF to meet the occupied space condition of the washing machine. A restriction of 35 mm is set to the MRF gap length *L* considering the whole eye–to–eye length of the proposed damper, which takes an approximate value 200 mm. The second constraint is the size one coming from competition with commercial passive dampers for compactness, lightness, favourableness and cost–effectiveness. In that way, the damper radius *R* should not exceed 15 mm. With this constraint value, the damper radius is expected to be significantly smaller than radii of traditional and self–powered MR dampers.

In summary, the optimal design problem of the new self–adaptive MR dampers can be addressed: Find optimal main geometrical dimensions of the dampers so that the off–state force F_0 is minimized while the maximum active damping force should be greater than 80 N, the MRF gap length is limited by 35 mm, and the total radius is constrained not to exceed 15 mm.

In this work, the commercial ANSYS software is utilized and the finite element models based on 2D symmetric couple element (PLANE 13) of the three damper configurations are shown in Figure 4. In order to obtain the optimal results of the damper, the first–derivative (first–order) method combined with golden–section algorithm of ANSYS optimization tool[28,29] is employed. Figure 5 shows the flow diagram to achieve optimal solution. First, an analysis ANSYS file to solve the magnetic circuit and calculate the objective function is created with ANSYS parametric design language. The DVs in this file is encoded and assigned with initial values. Rather than element length, using the number of elements for each line to specify the meshing size is more reasonable because of the continuous geometric variation during the optimization phase. The geometrical dimensions of the damper are varied during the optimization process; the meshing size therefore should be specified by number of elements per line rather than element size. It is also noted that the paths along the active MRF should be defined to calculate the magnetic flux crossing the MRF gap. By this way, the average density of magnetic flux can be obtained

$$B=\frac{1}{L}\int_0^L B_p(s)\,ds \tag{5}$$

in which $B_p(s)$ expresses the density of magnetic flux per node on the path.

As shown in the figure, the process starts from the DVs initial values. The density of magnetic flux is firstly calculated by running the analysis file. Based on Equations 4, 1 and 2, the rheological parameters such as yield stress t_y and post–yield viscosity *h*, active force and off–state force (objective function) can be obtained, respectively. Via external penalty functions P_{xi} for the design variable x_i, the optimization problem with unconstrained objective function *f(x)* is transformed from that with constrained DVs

$$f(x)=\frac{OBJ}{OBJ_0}+\sum_{i=1}^{n}P_{x_i}(x_i) \tag{6}$$

where *OBJ* symbolizes the objective function and the subscript *0* determines the reference value chosen from the current design sets. It is supposed that the negative gradient of *f(x)* is assigned to the DVs search direction for the first iteration (*j* = *0*) as follows

$$d^{(0)}=-\nabla f\left(x^{(0)}\right) \tag{7}$$

Then the DVs values for the next iteration (*j* + *1*) are

$$x^{(j+1)}=x^{(j)}+s_j d^{(j)} \tag{8}$$

in which s_j is the line search parameter that is derived based on the local quadratic fitting technique in company with golden–section algorithm. The DVs take new values and the analysis file runs again, also the objective function is examined for convergence. If it is right, the process stops and the optimum solution can be found at this iteration. Otherwise, the following iterations will be implemented the same as the above steps, but with an exception to the search direction that is obtained by Polak–Ribiere recursion expression

$$d^{(j)}=-\nabla f\left(x^{(j)}\right)+r_{j-1}d^{(j-1)} \tag{9}$$

where

$$r_{j-1}=\frac{\left[\nabla f\left(x^{(j)}\right)-\nabla f\left(x^{(j-1)}\right)\right]^T\nabla f\left(x^{(j)}\right)}{\left|\nabla f\left(x^{(j-1)}\right)\right|^2} \tag{10}$$

RESULTS AND DISCUSSIONS

In this section, optimal solutions of the proposed self–adaptive MR dampers in shear–mode are achieved based on the abovementioned design optimization problem. Optimization process of the single–magnet damper is shown in Figure 6. It can be observed from the figure that the solution converges after 53 iterations, at which the objective function F_0 is 16.3 N.

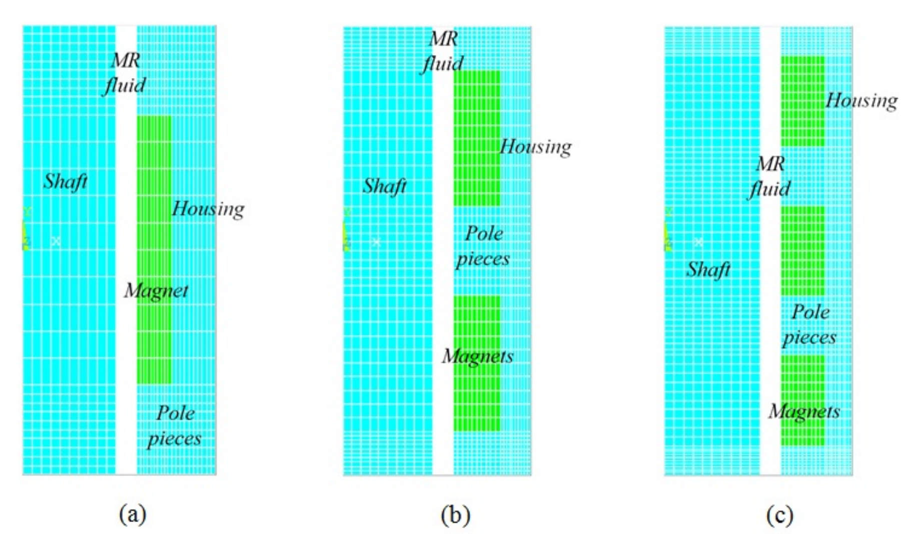


Figure 4: Finite element models of the proposed self–adaptive MR dampers: (a) single–magnet, (b) two–magnet and (c) three–magnet.

The optimal solution of the single–magnet damper is presented in Table 1. It is seen from the table that the active damping force can reach to 80 N as constrained for the resonance region while the off–state force is kept at 16.3 N which is small enough to prevent force transmissibility to the ground at high frequencies. Remarkably, the outer radius of the damper (14.6 mm) is much smaller than those of conventional MR dampers[10] (18.7 mm) and MR dampers with self–powered ability[24] (22 mm) in the same design problem, which is a highly considered business criterion.

FEA solution of magnetic flux of the optimized single–magnet self–adaptive MR damper is illustrated in Figure 7. The figure shows that the magnetic saturation occurs in the shaft and thin wall. By optimizing design, the magnetic flux crossing the MRF gap can be almost disposed throughout the gap length. This increases the solidification rate of the MRF in the gap and hence improves the damping force considerably.

Optimization process of the two–magnet damper is illustrated in Figure 8. The solution convergence occurs at 41^{st} iteration, where the objective function F_0 takes the value 15 N. The optimal solution of the single–magnet damper is summarized in Table 2 and the FEA solution of the optimized damper is shown in Figure 9. From the results, it is seen that as compared with those of the single–magnet damper, the outer radius and maximum damping force of the two–magnet damper are almost the same, but the off–state force is smaller (15 versus 16.3 N). This can be explained as follows. In the configuration of two magnets, the magnetic flux density across the MRF gap increases compared with the configuration of single magnet, which can be realized by the density scale towards yellow in the gap as presented in Figure 9(b). Thus, with the same objective maximum damping force of 80 N, the optimization process of the two–magnet configuration can further reduce the shaft radius against the single–magnet one (6.7 versus 7.3 mm), which leads to smaller off–state force. As a result, by increasing the number of magnets, the performance of the proposed damper can be meaningful improved.

In the same manner, optimization process of the three–magnet damper is conducted and presented in Figure 10. As shown in the figure, the solution convergence occurs at 41^{st} iteration, where the objective function F_0 takes the value 16.9 N. The optimal solution of the three–magnet damper is summarized in Table 3 and the FEA solution of the optimized damper is shown in Figure 11. From the results, it is seen that as compared with the single– and two–magnet dampers, the outer radius and maximum damping force of the two–magnet damper are almost the same, but the off–state force is greater (16.9 versus 16.3 and 15 N). This mainly comes from the gap length constraint of 35 mm, which are not enough for the three–magnet damper to achieve its best performance. As shown in Figure 11(b), the density scale in the MRF gap of the three–magnet damper is mostly green and

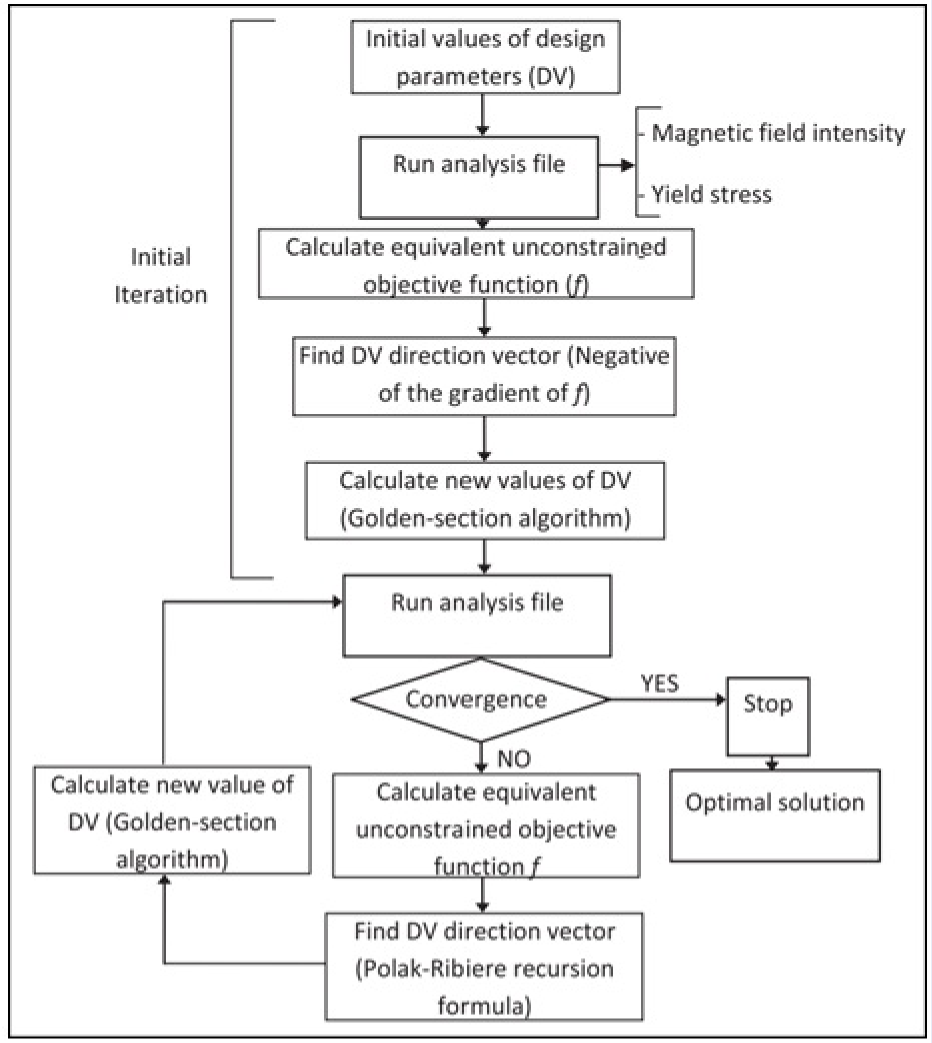


Figure 5: Flow diagram of the optimization process for the MR damper design.

Table 1: Optimal solution of the single–magnet self–adaptive MR damper.

Design variables (mm)	Performance characteristics (N)
Magnet: length l_m = 20.2, thickness t_m = 2.4, pole l_p = 7	Max. damping force F_d = 80
Shaft: radius r_s = 7.3	Off–state force F_0 = 16.3
MRF gap: length L = 34.1, thickness t_g = 0.8	
Housing: thin wall t_w = 0.8, thickness t_o = 3.4, outer radius R = 14.6	

Table 2: Optimal solution of the two–magnet self–adaptive MR damper.

Design variables (mm)	Performance characteristics (N)
Magnets: length l_m = 10.6, thickness t_m = 3.8, pole l_p = 3.4	Max. damping force F_d = 80
Shaft: radius r_s = 6.7	Off–state force F_0 = 15
MRF gap: length L = 34.9, thickness t_g = 0.8	
Housing: thin wall t_w = 0.8, thickness t_o = 2.4, outer radius R = 14.5	

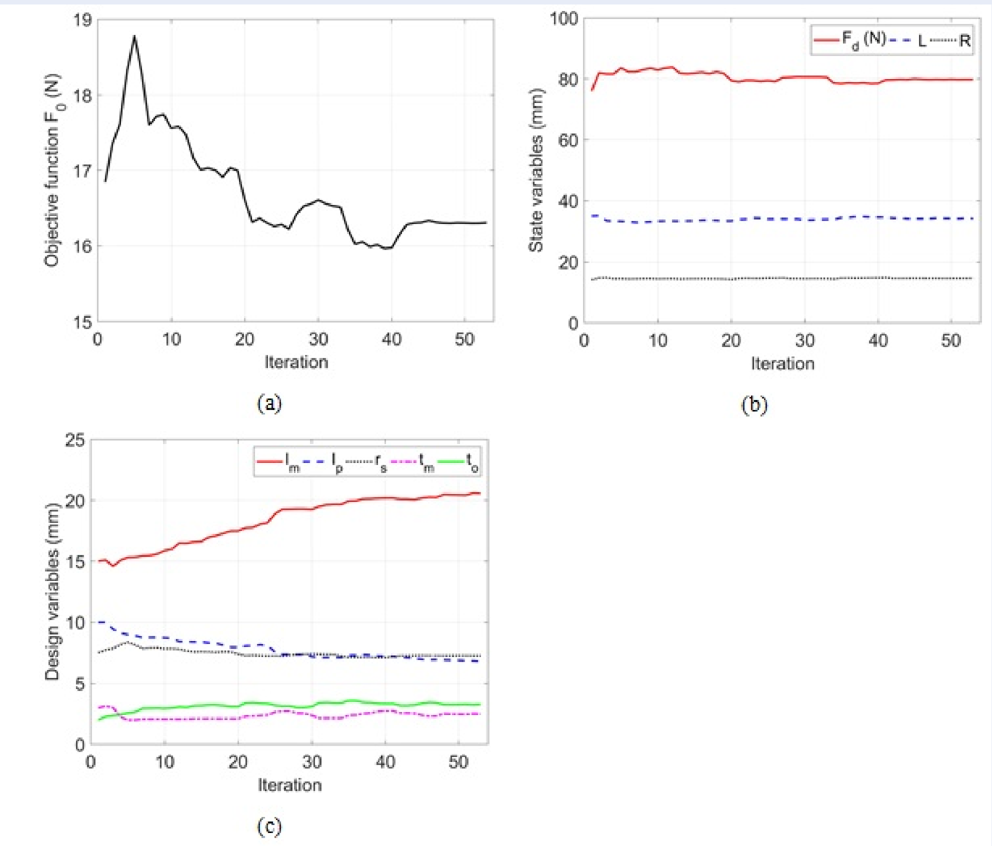


Figure 6: Optimization process of the single–magnet self–adaptive MR damper: (a) objective function, (b) state variables and (c) design variables.

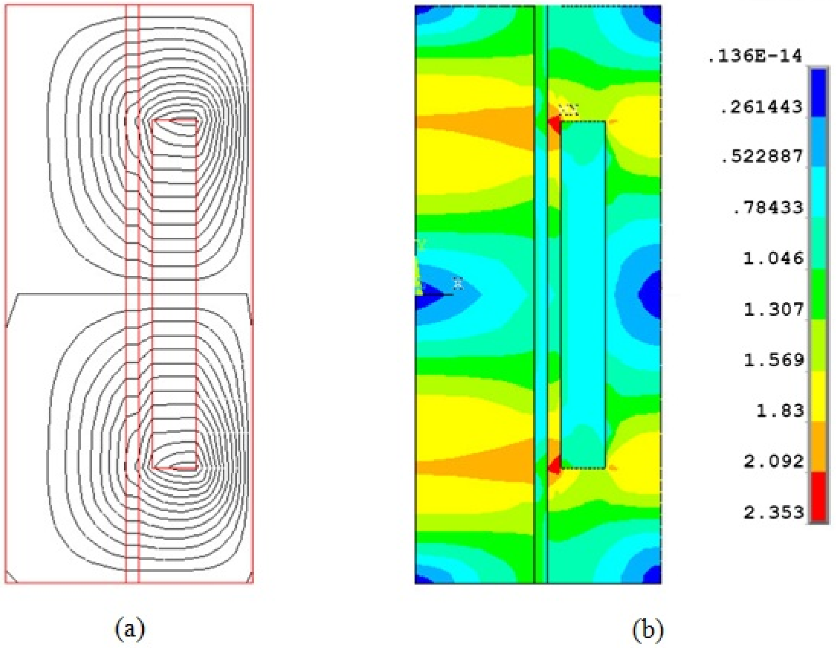
Table 3: Optimal solution of the three–magnet self–adaptive MR damper.

Design variables (mm)	Performance characteristics (N)
Magnets: length l_m = 7.1, thickness t_m = 3.2, pole l_p = 2.3 Shaft: radius r_s = 7.5 MRF gap: length L = 35, thickness t_g = 0.8 Housing: thin wall t_w = 0.8, thickness t_o = 2.1, outer radius R = 14.4	Max. damping force F_d = 80 Off–state force F_0 = 16.9

cyan, implying lower density of magnetic flux against that of the two–magnet damper. It is believed that the configurations of more magnets are in the same manner. Thus, the two–magnet self–adaptive MR damper is the best configuration in this study. However, when the available length of the damper increases such as dampers in 10 kg–load washing machines, the configurations of three or more magnets can be effective and feasible.

CONCLUSIONS

This research work concentrated on a new shear–mode MR damper which can substitute effectively for commercial dampers of washing machines in the field of vibration control. The proposed damper can adjust damping level by itself in accordance with exterior oscillatory excitations. Against traditional MR dampers and self–powered ones, the new damper requires no coils winding or control equipment, which facilitates manufacturing, maintenance and extremely reduces production cost. Furthermore, the damping characteristic based on displacement of the proposed damper is highly compatible with working principle of washing machines. These can deeply impress household engine producers and increase the damper commercialization. In this study, three damper configurations of single, two and three magnets were considered, also an optimization process for essential



(a) (b)

Figure 7: FEA results of the optimized single–magnet self–adaptive MR damper: (a) magnetic flux lines and (b) magnetic flux density.

geometrical dimensions of the dampers were implemented. The optimal solutions showed that the two–magnet configuration damper is best for this damper application. However, the configurations of three or more magnets could be more effective and feasible for washing machines with greater maximum load value. To continue this study work in the next phase, a prototype of the two–magnet MR damper will be designed, manufactured and assessed on a prototype washing machine.

ACKNOWLEDGEMENTS

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LIST OF NOTATION

B Density of magnetic flux crossing the MRF gap

B_p Magnetic flux density at each nodal point on the defined path

d Direction vector

f Dimensionless, unconstrained objective function

f_l Friction for each unit length caused by compression of O–rings

F_0 Off–state force

F_d Damping force

F_{or} Coulomb friction force between each O–ring and damper shaft

j Iteration

l_m Length of magnet

l_p Length of pole piece

L Length of MRF gap

L_r Seal surface length

n_m Number of magnets

n_p Number of pole pieces

OBJ Objective function

OBJ_0 Reference objective function

P_{xi} Exterior penalty function

r_m Radius of magnet

r_s Radius of damper shaft

R Radius of damper

s_j Line search parameter

t_g MRF gap thickness

t_m Magnet thickness

t_o Thickness of housing

t_w Thickness of thin wall

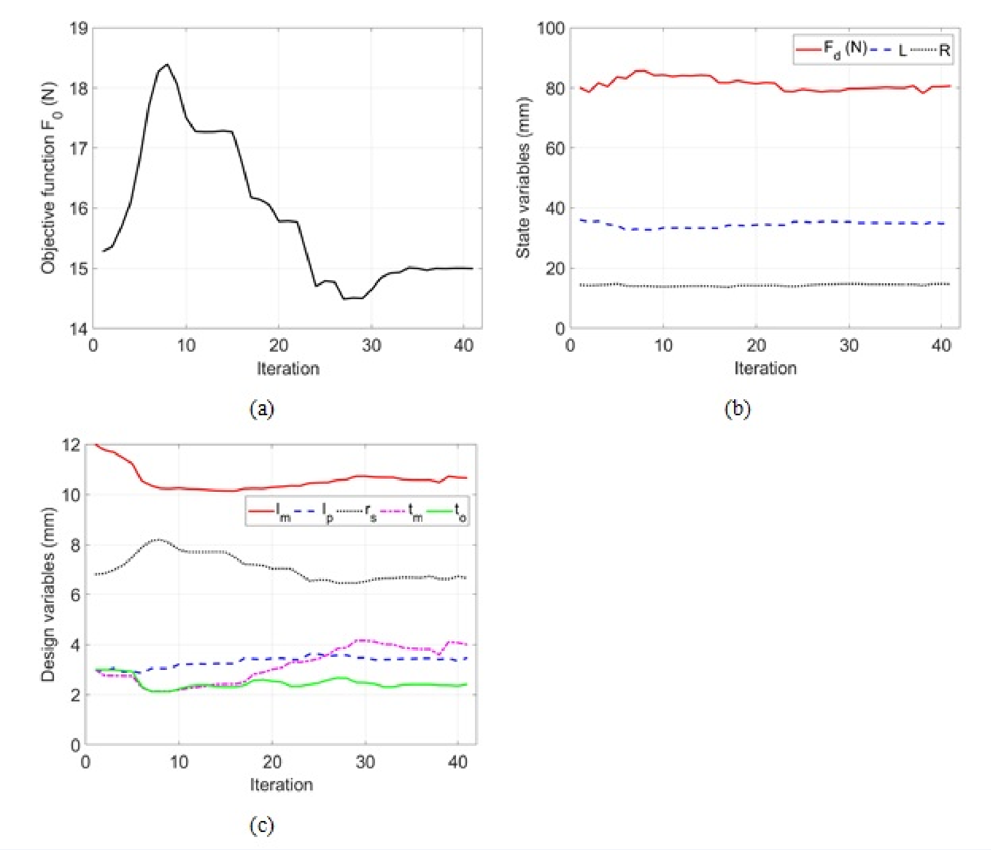
v Velocity of damper shaft

x_i Design variable

Y Rheological parameters of the MRF (yield stress, post yield viscosity)

Y_0 Off–state Y value

Y_∞ Saturated Y value



(a)

(b)

(c)

Figure 8: Optimization process of the two–magnet self–adaptive MR damper: (a) objective function, (b) state variables and (c) design variables.

α_{SY} Saturated Y moment index

h Post yield viscosity of the effective MRF

t_y Yield stress of the effective MRF

LIST OF ABBREVIATIONS

DV: Design variable

MR: Magneto–rheological

MRF: Magneto–rheological fluid

FEA: Finite element analysis

COMPETING OF INTERESTS

The author(s) declare that they have no competing interests.

AUTHORS' CONTRIBUTIONS

Q. D. Bui is working on this research topic. He is in charge of configuration design, program coding, formal analysis, evaluation and write the first draft of the manuscript; Q. H. Nguyen is the corresponding author of this research work. He is responsible for conceptualization, methodology, planning, supervision and final editing of the manuscript; All authors have read and agreed to the published version of the manuscript.

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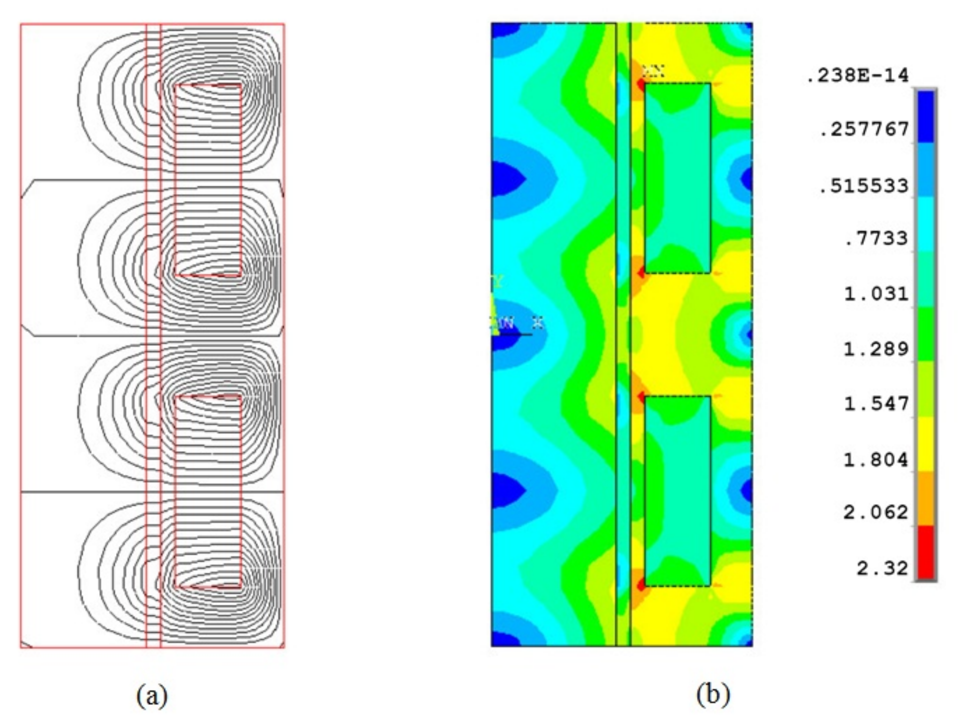

Figure 9: FEA results of the optimized two–magnet self–adaptive MR damper: (a) magnetic flux lines and (b) magnetic flux density.

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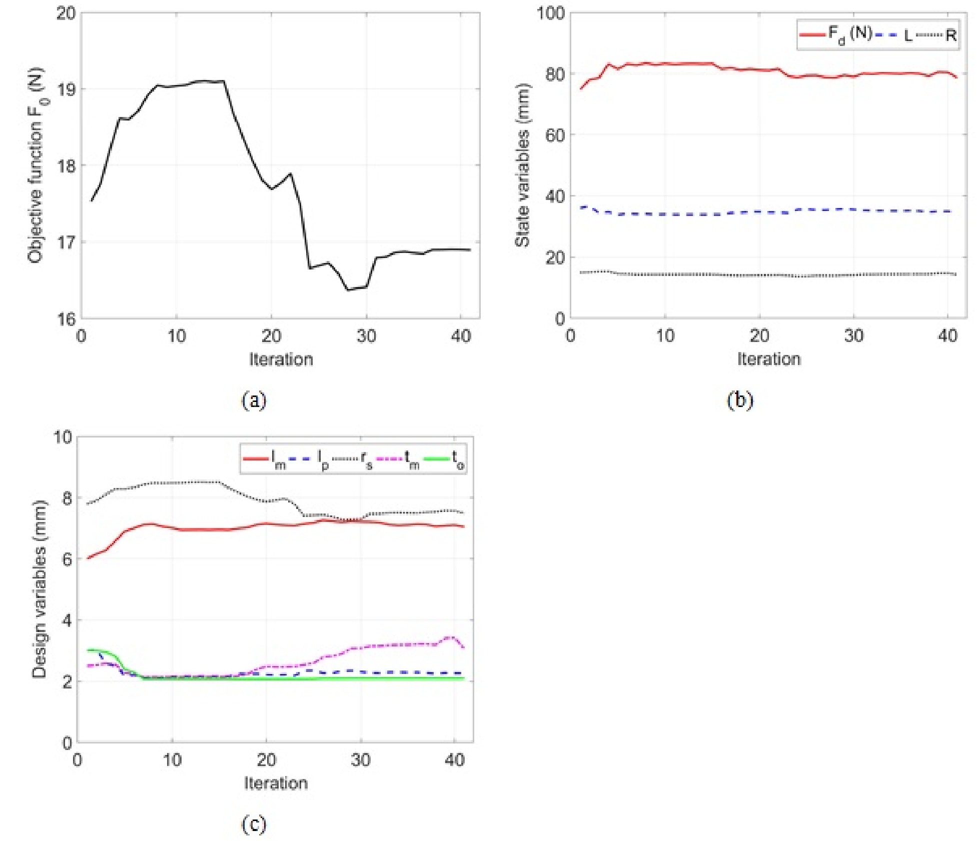

(a)

(b)

(c)

Figure 10: Optimization process of the three–magnet self–adaptive MR damper: (a) objective function, (b) state variables and (c) design variables.

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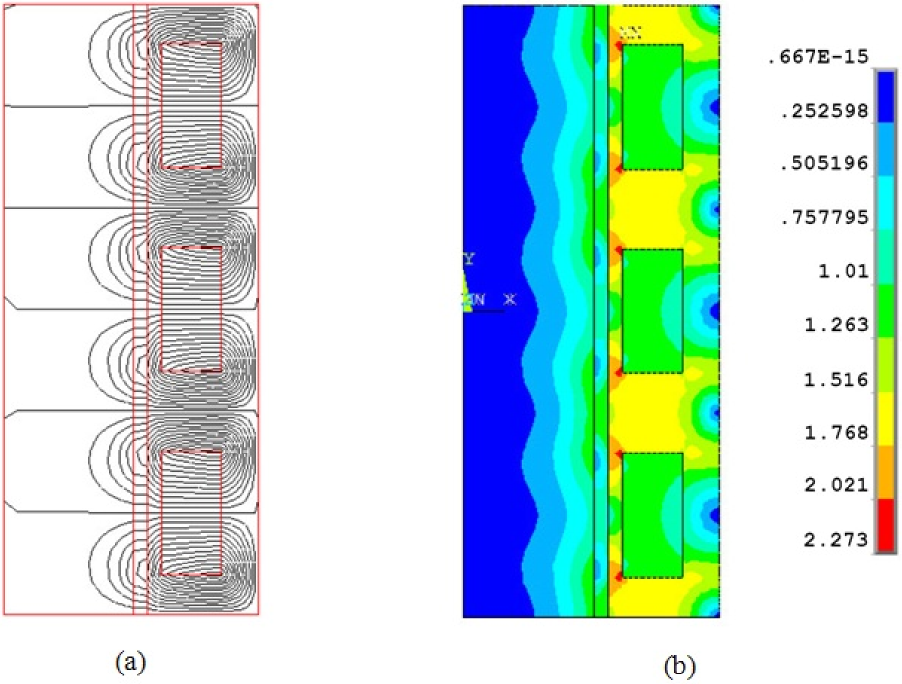

Figure 11: FEA results of the optimized three–magnet self–adaptive MR damper: (a) magnetic flux lines and (b) magnetic flux density.

Tạp chí Phát triển Khoa học và Công nghệ – Kĩ thuật và Công nghệ, 4(3):1105-1118

Bài nghiên cứu

Thiết kế và mô phỏng một giảm chấn MR tự đáp ứng mới cho máy giặt sử dụng kiểu trượt và nam châm hướng kính

Bùi Quốc Duy[1,2], Nguyễn Quốc Hưng[3,*]

TÓM TẮT
Bài báo nghiên cứu một giảm chấn lưu chất từ biến (MR) mới cho máy giặt có thể thay thế hiệu quả giảm chấn bị động thông thường trong lĩnh vực kiểm soát rung động. So với các giảm chấn MR truyền thống và tự cấp năng lượng, giảm chấn mới có chi phí rất thấp cho mục tiêu thương mại hóa nhờ kết cấu chắc gọn và khả năng tự đáp ứng mức giảm chấn phù hợp với dao động cơ học ngoài mà không cần bất kỳ công cụ điều khiển nào. Bằng cách thêm vào một lớp lưu chất MR (MRF) giữa trục và vỏ, hệ số giảm chấn có thể được điều chỉnh theo cường độ từ trường áp đặt. Các nam châm vĩnh cửu hình nhẫn với từ tính hướng kính được lắp vào vỏ, và phần trục ở giữa được làm bằng vật liệu phi từ tính để tạo trạng thái không kích hoạt của giảm chấn. Khi rung động của máy giặt tăng mạnh, các phần trục từ tính di chuyển vào vùng chứa MRF và nam châm, hình thành các mạch từ kín. Dưới tác dụng của từ trường, MRF chuyển đổi từ chảy tự do sang trạng thái bán rắn, các đặc tính lưu biến của nó như ứng suất chảy và độ nhớt tăng lên. Điều này khiến cho chuyển động tương đối giữa trục trượt và MRF trong khe hở bị cản trở, qua đó lực giảm chấn được gia tăng. Biên độ rung động càng lớn, càng nhiều phần trục từ tính tiếp xúc với MRF, các từ trường được sinh ra càng mạnh và lực giảm chấn được tạo ra càng lớn để dập tắt rung động. Đặc tính giảm chấn dựa trên chuyển vị này rất tương thích với sự giảm rung của máy giặt. Nhằm nâng cao hiệu quả cách ly rung động đồng thời thỏa mãn các tiêu chí về không gian lắp đặt, kích cỡ và chi phí sản xuất, quá trình tối ưu hóa được thực hiện cho các kích thước hình học thiết yếu của giảm chấn đề xuất. Sau đó những ưu điểm về đặc tính hoạt động của giảm chấn đề xuất được nhận xét và thảo luận từ các kết quả tối ưu.

Từ khoá: nam châm vĩnh cửu hướng kính, máy giặt, kiểm soát rung động, kiểu trượt, tự đáp ứng, giảm chấn MR, lưu chất MR

[1] *Khoa Xây dựng, Trường Đại học Sư phạm Kỹ thuật Thành phố Hồ Chí Minh, Thành phố Hồ Chí Minh, Việt Nam*

[2] *Khoa Công nghệ Cơ khí, Trường Đại học Công nghiệp Thành phố Hồ Chí Minh, Thành phố Hồ Chí Minh, Việt Nam*

[3] *Khoa Kỹ thuật, Trường Đại học Việt Đức, Thành phố Thủ Dầu Một, Tỉnh Bình Dương, Việt Nam*

Liên hệ

Nguyễn Quốc Hưng, Khoa Kỹ thuật, Trường Đại học Việt Đức, Thành phố Thủ Dầu Một, Tỉnh Bình Dương, Việt Nam

Email: hung.nq@vgu.edu.vn

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