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The impact of bobbin material and design on magnetorheological brake performance

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Abstract

In this work, a new configuration of magnetorheological brakes (MRBs) is developed in order to improve the compactness, manufacturing accuracy and cost of conventional ones. In the conventional configuration of MRBs, the coil is normally wound on a nonmagnetic bobbin which is placed on the stationary housing. This causes difficulties in manufacturing and the bottle-neck problem of the magnetic circuit of the MRBs. In the proposed configuration, the nonmagnetic bobbin is eliminated and the coil is wound directly on a magnetic bobbin which is a part of the housing. In this case, the magnetic bobbin part should be designed with a contractive cross-section in order to prevent magnetic flux going through and thus forcing the magnetic flux across the MR fluid (MRF) duct. After proposing the new configurations of MRBs, the modelling of the MRBs is performed based on the Bingham rheological model of the MRF. An optimal design of the proposed MRBs and conventional MRBs is then performed based on finite element analysis of the magnetic circuit of the MRBs. A comparative work between the optimal parameters of the proposed MRBs and the conventional MRBs is conducted and the advanced performance characteristics of the proposed MRBs are then investigated. In addition, experiments on both the conventional and the proposed MRBs are performed to validate the advanced performance characteristics of the proposed MRBs.

(Some figures may appear in colour only in the online journal)

1. Introduction

Magnetorheological fluid (MRF) is a non-colloidal suspension of magnetizable particles that are of the order of tens of microns (20–50 $\mu$m) in diameter. Generally, MRF is composed of oil, usually mineral or silicone based, and varying percentages of ferrous particles that have been coated with an anti-coagulant material. When inactivated, MRF exhibits Newtonian-like behaviour. When exposed to a magnetic field, the ferrous particles that are dispersed throughout the fluid form magnetic dipoles. These magnetic dipoles align themselves along lines of magnetic flux which impede the MRF from deformation. The time span in which this change occurs lies within a few milliseconds and the effect is completely reversible. Although similar in operation to electro-rheological fluid (ERF) and ferro-fluid, MRF is capable of much higher yield strengths when activated. Due to this advantage, many MRF-based mechanisms have been developed such as MR dampers, MR brakes, MR clutches, MR valves, etc, and some of them are now in commercial use.

There has been a great deal of research on the development of MRB brakes (MRBs) in various applications, and several configurations of MRBs have been reported in...
A new configuration for MRBs in which the coil is placed on the housing is preferred. Therefore, in cases of rotary-shaft MRBs, the former approach (i.e. the coil is placed on the housing) is preferred. However, this inverted configuration is only convenient for rotary, such as automotive brakes. In this configuration, the coil is wound on a nonmagnetic bobbin. In this case, the magnetic flux across the MRF is distributed only at the two flanges of the housing (two poles). The significant dimensions of the MRB are also presented in the figure. It is noteworthy, from a manufacturing perspective, that it is hard to achieve a high accuracy for the above MRBs, especially in assembling the bobbin into the housing. In order to facilitate a manufacturing process which reduces the MRBs’ cost, a new configuration without the nonmagnetic bobbin, shown in figure 1(b), is proposed. As shown in the figure, in this configuration, the coil is wound directly on the magnetic bobbin which is a part of the housing. The magnetic bobbin is designed with a special shape that almost resists the magnetic flux going through it and thus forces the magnetic flux to go across the MRF gap. The significant dimensions of the MRB are shown in the figure as well. By assuming that MRFs rheologically behave as Bingham plastic fluids and by the assumptions of the linear velocity profile in the MRF ducts of the brake, the braking torque and the off-state torque of the above MRBs are respectively determined by [9]

\[ T_d = \frac{\pi \mu_0 R_1^4}{d_o} \left[ 1 - \left( \frac{R_1}{R_0} \right)^4 \right] \Omega + \frac{4 \pi \tau \tau_o}{3} (R_0^3 - R_1^3) + 2 \pi R_1^2 L_x \left( \tau_y + \mu \frac{\Omega R_0}{d} \right) + 2 T_{or} \]  

(1)

\[ T_{d0} = \frac{\pi \mu_0 R_0^4}{d_o} \left[ 1 - \left( \frac{R_1}{R_0} \right)^4 \right] \Omega + \frac{4 \pi \tau \tau_o}{3} (R_0^3 - R_1^3) + 2 \pi R_0^2 L_x \left( \tau_y + \mu_0 \frac{\Omega R_0}{d} \right) + 2 T_{or} \]  

(2)

In this section, configurations of rotary MRBs with a magnetic bobbin are proposed based on conventional configurations of MRBs using a nonmagnetic bobbin. Figure 1(a) shows the conventional configuration of a drum-type MRB. In this configuration, the coil is wound on a nonmagnetic bobbin. In this case, the magnetic flux across the MRF is distributed only at the two flanges of the housing (two poles). The significant dimensions of the MRB are also presented in the figure. It is noteworthy, from a manufacturing perspective, that it is hard to achieve a high accuracy for the above MRBs, especially in assembling the bobbin into the housing. In order to facilitate a manufacturing process which reduces the MRBs’ cost, a new configuration without the nonmagnetic bobbin, shown in figure 1(b), is proposed. As shown in the figure, in this configuration, the coil is wound directly on the magnetic bobbin which is a part of the housing. The magnetic bobbin is designed with a special shape that almost resists the magnetic flux going through it and thus forces the magnetic flux to go across the MRF gap. The significant dimensions of the MRB are shown in the figure as well. By assuming that MRFs rheologically behave as Bingham plastic fluids and by the assumption of the linear velocity profile in the MRF ducts of the brake, the braking torque and the off-state torque of the above drum-type MRBs are respectively determined by [9]
The o-ring friction torque is the friction torque between the o-ring and the shaft of the brake. In the case of MRBs with a magnetic bobbin, the length of the MRF in the annular duct of the brake which depends on exerted magnetic flux density across the duct, \( L \), \( \tau \) and \( d \) are respectively the yield stress of the MRF, \( \mu \) and \( \mu_e \) are respectively the post-yield viscosity and the yield stress of the MRF in the annular duct of the brake which depends on exerted magnetic flux density across the duct, \( L_a \) is the length of the annular duct of MRF, and \( L_e \) is the effective length of the MRF annular duct. In the case of MRBs with a nonmagnetic bobbin, \( L_e = 2\pi R \), where \( \pi R \) is the pole length. In the case of MRBs with a magnetic bobbin, \( L_e = L_a \), \( T_{or} \) is the friction torque between the o-ring and the shaft of the brake. The o-ring friction torque \( T_{or} \) can be approximately calculated by

\[
T_{or} = (f_c L_e + f_o A_t) R_{or}
\]

where \( L_e \) is the length of the seal rubbing surface (the shaft circumference at the sealing), \( f_c = 2\pi R_{or}, R_{or} \) is the shaft diameter at the o-ring seal, \( f_c \) is friction per unit length of the shaft circumference due to o-ring compression depending on percentage of seal compression and hardness of the o-ring material, \( f_o \) is the o-ring friction force due to fluid pressure acting on a unit projected area of the o-ring, and \( A_t \) is the seal projected area. It is noted that MRBs operate in the shear-mode, so the pressure due to MRF acting on the o-rings is very low, which can be neglected, \( f_o \equiv 0 \). In this study, a 70-durometer rubber o-ring is used and the compression of the o-ring is set at 10%. In this case the coefficient \( f_c \) is around 122.5 N m\(^{-1}\).

Recently, a new configuration of MRBs, in which the MRF is energized both at the end-faces and the outer cylindrical face of the rotor, was proposed \([8, 9]\). This configuration is referred to as a hybrid-type MRB. Figure 2(a) shows the conventional configuration of the hybrid-type MRBs with significant dimensions. As shown in the figure, when a current is applied to the coil, a magnetic circuit is generated and the magnetic flux goes across both the annular duct and the radial ducts (the end-face ducts) of MRF. Therefore, the magnetic circuit results in induced stresses of MRF in both the annular duct and the radial ducts between the rotor and the stator of the brake. Similarly to the drum-type MRBs, in the conventional hybrid MRB, a nonmagnetic bobbin is used for the coil which causes difficulties and high cost in manufacturing. In order to facilitate the manufacturing process and reduce the cost of the hybrid MRB, a new configuration of the MRBs without the nonmagnetic bobbin (using the magnetic bobbin as a part of the housing) is proposed and shown in figure 2(b). The significant dimensions of the MRB are also presented in the figure. For the hybrid MRBs, the braking torque and off-state torque can be respectively determined as follows:

\[
T_h = \frac{\pi \mu_e R_{ho}^4}{d_{ho}} \left[ 1 - \left( \frac{R_{hi}}{R_{ho}} \right)^4 \right] \Omega + \frac{4\pi \tau_y e}{3}
\]

\[
× \left( R_{ho}^4 - R_{hi}^4 \right) + 2\pi R_{ho}^2 L_{ch} \left( \tau_y + \mu_e \frac{\Omega R_{ho}}{d_h} \right) + 2T_{or} \quad (4)
\]

\[
T_{ho} = \frac{\pi \mu_o R_{ho}^4}{d_{ho}} \left[ 1 - \left( \frac{R_{hi}}{R_{ho}} \right)^4 \right] \Omega + \frac{4\pi \tau_y o}{3}
\]

\[
× \left( R_{ho}^4 - R_{hi}^4 \right) + 2\pi R_{ho}^2 L_h \left( \tau_y o + \mu_o \frac{\Omega R_{ho}}{d_h} \right) + 2T_{or}. \quad (5)
\]

In the above, \( R_{hi} \) and \( R_{ho} \) are the inner and outer radius of the rotor of the hybrid MRB, \( d_{ho} \) and \( d_{ha} \) are respectively the gap size of the annular and the end-face MRF ducts, \( L_h \) and \( L_{ch} \) are the length of the rotor and the effective length of the annular duct respectively, \( \mu_e \) and \( \mu_o \) are respectively the post-yield viscosity of the active MRF in the end-face and the annular duct, and \( \tau_y e \) and \( \tau_y o \) are respectively the yield stress of the active MRF in the end-face and annular ducts. It is noteworthy that \( \mu_e, \mu_o, \tau_y e \) and \( \tau_y o \) are functions of the exerted magnetic field.

### 3. Linear MR brakes with magnetic bobbin

The main difference between a linear and a rotary MRB is that in the linear MRB, the rotor is replaced by a shaft which linearly moves against its housing. The linear MRBs are also referred to as shear-mode MR dampers, which are often used when a low damping/braking force is required \([11]\). Figure 3 shows two configurations of single-coil linear MRBs. In the first configuration, shown in figure 3(a), a nonmagnetic bobbin is used for the coil. In the same manner as the rotary MRBs, in order to improve manufacturing accuracy and cost, a configuration with a magnetic bobbin is proposed in this

![Figure 2. Schematic configuration of single-coil hybrid MRBs. (a) With nonmagnetic bobbin. (b) With magnetic bobbin.](image-url)
Figure 3. Schematic configuration of single-coil linear MRBs. (a) With nonmagnetic bobbin. (b) With magnetic bobbin.

Figure 4. Schematic configuration of two-coil linear MRBs. (a) With nonmagnetic bobbin. (b) With magnetic bobbin.

study. Figure 3(b) shows the configuration of the proposed linear MRB with a magnetic bobbin designed as a part of the housing. The significant dimensions of the brakes are also shown in the figure.

In order to increase magnetic flux density across the MRF gap, especially when the housing length is significantly greater than its outer radius, a configuration with two or more coils can be used. Figure 4(a) shows the two-coil linear MRB with a nonmagnetic bobbin while the two-coil linear MRB with a magnetic bobbin is shown in figure 4(b). By neglecting the friction force due to post-yield viscosity of the MRF in the duct between the shaft and the housing, the damping force and the off-state force of the linear MRBs can be respectively estimated by

\[ F_b = 2\pi R_s L_e \tau_y + 2F_{or} = 2(\pi R_s L_e \tau_y + F_{or}) \]  
\[ F_{b0} = 2\pi R_s L_d \tau_{y0} + 2F_{or} = 2(\pi R_s L_d \tau_{y0} + F_{or}) \]

where \( R_s \) is the shaft radius, \( \tau_y \) is the field-dependent yield stress of the active MRF in the duct, \( L_d \) is the length of the MRF duct, \( L_e \) is the effective length of the active MRF in the duct, and \( F_{or} \) is the coulomb friction force between the shaft and the o-ring which can be approximately calculated by \[ F_{or} = f_c L_o + f_h A_r \]

where \( L_o \) is the length of the seal rubbing surface (the shaft circumference), \( L_e = 2\pi R_s \), \( f_c \) is friction per unit length of the shaft circumference due to o-ring compression depending on percentage of seal compression and hardness of the o-ring material, \( f_h \) is the o-ring friction force due to fluid pressure acting on a unit seal projected area, and \( A_r \) is the seal projected area. This is different from rotary MRBs in which the radius of the shaft at the o-ring sealing can be chosen as small as possible regardless of the MRF duct radius. For linear MRBs, the radius of the shaft at the o-ring sealing should be equal to that of the MRF duct. Therefore, an increase of the braking force of the linear MRBs by increasing the MRF duct radius will result in a corresponding increase of off-state force. This trade-off benefit should be taken into account in the design of the linear MRBs.

4. Optimal design of MR brakes

In this section, the optimal design of the conventional MRBs with a nonmagnetic bobbin and the proposed MRBs without a nonmagnetic bobbin is considered. In design of MRBs, the most important performance parameter is the maximum braking torque/force. Even when the MRBs are constrained within a specific volume, it is expected that the maximum braking torque/force produced by the MRBs will be as high as possible. It is noteworthy that instead of maximizing the braking torque/force of the MRBs, in this study the following
Table 1. Magnetic properties of the MRB components.

<table>
<thead>
<tr>
<th>Material</th>
<th>Relative permeability</th>
<th>Saturation flux density</th>
</tr>
</thead>
<tbody>
<tr>
<td>Silicon steel</td>
<td>( B-H ) curve (figure 9(a))</td>
<td>1.55 T</td>
</tr>
<tr>
<td>Copper</td>
<td>1</td>
<td>( \times )</td>
</tr>
<tr>
<td>MRF-140-CG</td>
<td>( B-H ) curve (figure 9(b))</td>
<td>1.8 T</td>
</tr>
<tr>
<td>Nonmagnetic steel</td>
<td>1</td>
<td>( \times )</td>
</tr>
</tbody>
</table>

corresponding objective functions are minimized.

\[
\text{OBJ}_T = \frac{1}{T_b} \quad (9)
\]

\[
\text{OBJ}_F = \frac{1}{F_b} \quad (10)
\]

where \( T_b \) is the braking torque of the rotary MRBs determined by equations (1) and (4) respectively for the drum-type and the hybrid MRBs.

Another important issue that should be accounted for in the design of MRBs is the off-state torque/force. Generally, the smaller the gap size of the MR duct, the greater is the braking torque/force produced. However, the small gap size of the MRF duct is limited due to difficulties encountered in the manufacturing process. In addition, the small gap size of the MRF duct can result in a high value of the off-state torque/force, that reduces the performance of the MRBs, such as in high dissipated energy, overheating, and small dynamic range which degrades controlled performance of the MRBs. In this research, the off-state torque/force of MRBs is evaluated by introducing the MRB torque/force ratio—the ratio of the maximum braking torque/force and the off-state torque/force—these ratios can be mathematically expressed by

\[
\lambda_T = \frac{T_{b_{\text{max}}}}{T_0} \quad (11)
\]

\[
\lambda_F = \frac{F_{b_{\text{max}}}}{F_{b0}} \quad (12)
\]

where \( \lambda_T \) and \( \lambda_F \) are respectively the torque ratio of the rotary MRBs and the force ratio of the linear MRBs. \( T_{b_{\text{max}}} \) is the maximum value of the braking torque of the rotary MRBs determined by equations (1) and (4) respectively for the drum-type and the hybrid MRBs, \( T_0 \) is the off-state torque of the MRBs calculated by equations (2) and (5) for the drum-type and the hybrid MRBs, respectively, and \( F_{b_{\text{max}}} \) is the maximum value of the braking force of the linear MRBs.

In this research, finite element (FE) analysis integrated with the optimization toolbox in ANSYS software is employed to find the optimal solution. It is pointed out that the FE analysis is used to solve the magnetic circuits of the MRBs, from which the braking torque/force of the MRBs can be calculated. The results are sent to the optimization toolbox and the optimal solutions are obtained by using the first order method with the golden section algorithm. In the Figure 5. Magnetic properties of silicon steel and MR fluid. (a) \( B-H \) curve of silicon steel. (b) \( B-H \) curve of MRF-140-CG.

5. Results and discussion

In this section, the optimal solutions of the abovementioned MRBs are obtained and the performance characteristics of the optimized MRBs are evaluated. It is assumed that all magnetic parts of the MRBs are made of commercial silicon steel, and the coil wires are sized as 21-gage whose allowable working current is 2.5 A. In this study, the high yield stress MRF made by the Lord Corporation, MRF-140-CG, is used. The Magnetic properties of the brake components are given in table 1 and figure 5. In the field-dependent Bingham model, the rheological properties of MRFs can be estimated by the following equation [12, 5]:

\[
Y = Y_{\infty} + (Y_0 - Y_{\infty})(2e^{B_{\text{max}} \alpha_{Sf}} - e^{-2B_{\text{max}} \alpha_{Sf}}) \quad (13)
\]

where \( Y \) stands for the rheological parameters of MR fluid such as yield stress and post-yield viscosity. The value of parameter \( Y \) tends from the zero applied field value \( Y_0 \) to the saturation value \( Y_{\infty} \). \( \alpha_{Sf} \) is the saturation moment index.
Figure 6. Maximum braking torque and torque ratio of rotary MRBs featuring MRF-140-CG. (a) $R = 30$ mm. (b) $R = 60$ mm.

Figure 6 shows the maximum braking torque of the rotary MRBs obtained from optimal solutions as a function of the brake volume ratio ($L/R$), where $L$ is the overall length and $R$ is the outer radius of the MRBs. The angular velocity of the brake shaft is set by $\Omega = 20\pi$ rad s$^{-1}$ (600 RPM). The torque ratio of the MRBs is also presented in the figure. In addition, the maximum braking torque of the disc-type MRBs is presented in the figure as a reference. It is noted that the disc-type MRB is very popular because of its advantages such as high compactness, simplicity and low cost. However, in cases where the volume ratio of the MRBs is greater than 0.4 ($L/R > 0.4$), the braking torque produced by disc-type MRBs is smaller than that produced by the hybrid ones [8].

In figure 6(a), the radius of the MRBs is set by $R = 30$ mm and the brake length $L$ is changed from 10 to 90 mm (the brake volume ratio is changed from 0.33 to 3.00). It is noted that the smaller the value of the contracted section thickness, the more magnetic flux is forced to run across the MRF duct. In this case, the gap size of the MRF ducts, the thickness of the nonmagnetic bobbin, the thickness of the contracted section of the magnetic bobbin, and the minimum thickness of the two end housings are respectively set at 0.6, 0.4, 0.5 and 3 mm, considering the manufacturing advantages. From the results, it is observed that the maximum braking torque produced by the proposed MRBs is greater than that produced by the conventional ones. As the brake volume ratio increases up to 3.00, the maximum braking torques generated by the proposed and the conventional MRBs are almost converged. The main reason is that when the length of the brake is significantly greater than its radius, the width of the coil ($w_c$) is very small compared to the length of the annular duct, and thus the active lengths of the MRB ducts are almost similar. From the torque ratio graph, it can be found that the torque ratio is reduced when the brake volume ratio is greater than around 1.50. The reason is that when the volume ratio is greater than 1.5, the maximum braking torque of the MRBs is approaching its saturation while the off-state torque is continuously increased. Noteworthily, in the above, it is assumed that the coils are fully wound in coil-sectional area. Therefore, it is found from the optimal results that the number of turns of the proposed MRBs with a magnetic bobbin is smaller than that of the conventional ones with a nonmagnetic bobbin. With a smaller number of turns, it is expected that the power consumption of the MRBs with a magnetic bobbin is smaller than that of the conventional ones. This will be mentioned in more detail later when the prototypes are manufactured.

In figure 6(b), the radius of the MRBs is set by $R = 60$ mm and the brake length $L$ is changed from 20 to 180 mm (the brake volume ratio is changed from 0.33 to 3.00). In this case, the gap size of the MRF ducts, the thickness of the nonmagnetic bobbin, the thickness of the contracted section of the magnetic bobbin, and the minimum
Figure 7. Maximum braking force and force ratio of linear MRBs featuring MRF-140-CG as function of volume ratio $L/R$.
(a) $R = 20$ mm. (b) $R = 30$ mm.

case, the gap size of the MRF ducts, the thickness of the nonmagnetic bobbin, the thickness of the contracted section of the magnetic bobbin, and the minimum thickness of the two end housings are respectively set at 0.8, 0.5, 0.5 and 4 mm, considering the manufacturing advantages. From the maximum braking torque graph it is found that almost the same results as those in figure 6(a) are obtained. From the torque ratio graph, it is seen that the brake volume ratio, at which the torque ratio of the MRBs has maximum value, is a little smaller than that in figure 6(a).

Figure 7 shows the optimal results of the linear MRBs as a function of the brake volume ratio. In figure 7(a), the radius of the MRBs is set by $R = 20$ mm and the brake length $L$ is changed from 20 to 100 mm (the brake volume ratio is changed from 1.00 to 5.00). In this case, the gap size of the MRF ducts, the thickness of the nonmagnetic bobbin, and the thickness of the contracted section of the magnetic bobbin are respectively set at 0.5, 0.4 and 0.5 mm, considering the manufacturing advantages. For single-coil MRBs, it is observed that at a small value of the brake volume ratio the maximum braking force produced by the proposed linear MRBs is greater than that produced by the conventional ones. As the brake volume ratio increases up to 2.50, the maximum braking force generated by the proposed MRBs becomes smaller than that of the conventional ones. In case of two-coil MRBs, it is observed that the value of volume ratio at which the maximum braking force generated by the proposed MRBs becomes smaller than that of the conventional ones is about 5.00 (nearly twice as high as that in the case of single-coil MRBs). In figure 7(b), the radius of the MRBs is set by $R = 30$ mm and the brake length $L$ is changed from 30 to 150 mm (the brake volume ratio is changed from 1.00 to 5.00).

In this case, the gap size of the MRF ducts, the thickness of the nonmagnetic bobbin, and the thickness of the contracted section of the magnetic bobbin are respectively set at 0.6, 0.4 and 0.5 mm. The figure also shows the same trends as figure 7(a) does.

In order to validate the above optimal results, experimental results of the optimized MRB prototypes are obtained and presented. In the above, many types of MRB with different dimensions are considered and experiments for all the MRBs cannot be covered. In this study, experiments are performed for only four prototypes: the hybrid rotary MRBs constrained by $R = 60$ mm, $L = 40$ mm (both nonmagnetic bobbin and magnetic bobbin) and the two-coil linear MRB constrained by $R = 20$ mm, $L = 60$ mm (both nonmagnetic bobbin and magnetic bobbin). The significant parameters of the prototype MRBs are shown in tables 2 and 3. From the tables, it is noted that the coil wire is fully wound in the coil-sectional area. The number of coil turns in the case of the proposed MRBs with a magnetic bobbin is generally a little smaller than that of the conventional MRBs with a nonmagnetic bobbin. Figure 8 shows the magnetic flux density of the prototype MRBs obtained from finite element analysis. The figure shows that the magnetic at the contracted section of the magnetic bobbin is reached to the magnetic saturation of the housing material. Therefore, the magnetic flux is forced to run across the annular MRF duct. It is also observed that the magnetic
Figure 8. Magnetic density of the prototype MRBs obtained from finite element analysis. (a) Rotary MRB with nonmagnetic bobbin. (b) Rotary MRB with magnetic bobbin. (c) Linear MRB with nonmagnetic bobbin. (d) Linear MRB with magnetic bobbin.

Table 2. Parameters of the prototype rotary MRBs.

<table>
<thead>
<tr>
<th>Parameters (mm)</th>
<th>MRB with nonmagnetic bobbin</th>
<th>MRB with magnetic bobbin</th>
</tr>
</thead>
<tbody>
<tr>
<td>MRF gaps $d_0, d$</td>
<td>0.8, 0.8</td>
<td>0.8, 0.8</td>
</tr>
<tr>
<td>Coil height $h_c$</td>
<td>3</td>
<td>2.6</td>
</tr>
<tr>
<td>Coil width $w_c$</td>
<td>6.6</td>
<td>8.5</td>
</tr>
<tr>
<td>Coil chamfers $c_{h1}, c_{h2}$</td>
<td>0, 0</td>
<td>3, 1.6</td>
</tr>
<tr>
<td>Housing thickness $t_h$</td>
<td>9</td>
<td>9.6</td>
</tr>
<tr>
<td>Side housing thickness $t_s$</td>
<td>4.6</td>
<td>4.6</td>
</tr>
<tr>
<td>Rotor radius $R_i, R_o$</td>
<td>15, 47</td>
<td>15, 46.5</td>
</tr>
<tr>
<td>Rotor length $L_a$</td>
<td>26.8</td>
<td>29</td>
</tr>
<tr>
<td>No. of coil turns</td>
<td>92</td>
<td>80</td>
</tr>
</tbody>
</table>

The flux of the proposed MRB is distributed almost along the annular duct of MRF with an average magnitude which is a little smaller than that in the conventional ones. In order to clearly show the magnetic flux inside the MRB, the magnetic flux density vectors of the prototype MRBs obtained from finite element analysis are shown in figure 9.
Figure 9. Magnetic flux density vector of the prototype MRBs obtained from finite element analysis. (a) Rotary MRB with nonmagnetic bobbin. (b) Rotary MRB with magnetic bobbin. (c) Linear MRB with nonmagnetic bobbin. (d) Linear MRB with magnetic bobbin.

Table 3. Parameters of the prototype two-coil linear MRBs.

<table>
<thead>
<tr>
<th>Parameters (mm)</th>
<th>MRB with nonmagnetic bobbin</th>
<th>MRB with magnetic bobbin</th>
</tr>
</thead>
<tbody>
<tr>
<td>MRF gaps (d_0, d)</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>Coil height (h_c)</td>
<td>2.2</td>
<td>2</td>
</tr>
<tr>
<td>Coil width (w_c)</td>
<td>5</td>
<td>8</td>
</tr>
<tr>
<td>Coil chamfers (ch_1, ch_2)</td>
<td>0, 0</td>
<td>3, 0.7</td>
</tr>
<tr>
<td>Housing thickness (t_o)</td>
<td>4.3</td>
<td>4.8</td>
</tr>
<tr>
<td>Shaft radius (R_s)</td>
<td>12.6</td>
<td>12.6</td>
</tr>
<tr>
<td>No. of coil turns</td>
<td>100</td>
<td>96</td>
</tr>
</tbody>
</table>

Figures 10(a) and (b) respectively show the experimental setup to test the performance of the optimized rotary and linear MRBs. In figure 10(a), a DC motor with a gear-box controlled by the computer is used to rotate the shaft of the rotary MRB at a constant angular speed of \(2\pi\) rad s\(^{-1}\). The torque generated by the MRB is measured by a torque...
Figure 10. Experiment setup to test performance of the optimized MRBs. (a) Experiment setup for rotary MRBs. (b) Experiment setup for linear MRBs.

Figure 11(a) shows the step response of the hybrid rotary MRBs featuring MRF-140-CG. The results show that the steady torque obtained from the experiment is a little smaller than that obtained from the static modelling based on FE analysis. This mainly comes from the loss of magnetic field to the ambient and at the contact between the magnetic parts of the MRB. The average value of the braking torque of the proposed MRB at the steady state is 26.77 N m which is around 96.4% of the modelling value (27.76 N m), and that of the conventional MRB is 31.36 N m which is around 96.2% of the modelling value (32.6 N m). Hence, a good correlation between experimental results and static modelling based on FE analysis can be assumed. By measuring the power consumption of the MRBs at steady state, it is found that the power consumption of the proposed MR brake is 13 W while that of the conventional one is 15.6 W.

Figure 11(b) shows the step response of the two-coil linear MRB featuring MRF-140-CG. It is noted that in this case the angular velocity of the motor is kept constant at 0.5π rad s⁻¹ and the step response of the MRB is measured on the downward motion of the brake shaft. The results also show a good correlation between experimental results and static modelling based on FE analysis. In this case, at steady state, the power consumption of the proposed linear MR brake is 4.7 W while that of the conventional one is 5 W.
Figure 11. Experimental step responses of the optimized MRBs featuring MRF-140-CG. (a) Hybrid rotary MRBs, $R = 60$ mm, $L = 40$ mm. (b) Two-coil linear MRBs, $R = 20$ mm, $L = 60$ mm.

From the above results, it can be stated that, by optimal design, even though the number of coil turns (also the power consumption) of the proposed MRBs (using a magnetic bobbin) is smaller than that of the conventional MRBs (with a nonmagnetic bobbin), the performance characteristics of the proposed MRBs can be better than those of the conventional MRBs. It is also noted that by using the magnetic bobbin configuration, the manufacturing accuracy of the MRBs can be improved and the manufacturing cost can be reduced.

6. Conclusions

In this research, a new configuration of magnetorheological brakes (MRBs) was developed in order to improve the compactness, manufacturing accuracy and cost of conventional ones. In the proposed configuration, the nonmagnetic bobbin is eliminated and the coil is wound directly on a magnetic bobbin which is a part of the housing, to improve the manufacturing accuracy and cost, and to reduce the bottle-neck problem of the magnetic circuit of the MRBs. The magnetic bobbin of the housing was designed with a contractive cross-section in order to prevent magnetic flux going through and thus forcing the magnetic flux across the MR fluid (MRF) duct. Based on the Bingham rheological model of the MRF, the braking torque/force and off-state torque/force of the MRBs were derived. An optimal design of the proposed MRBs and conventional MRBs was then performed based on finite element analysis of the magnetic circuit of the MRBs. Based on optimal results, the performance characteristics of the proposed MRBs and the conventional MRBs were evaluated and the advanced performance characteristics of the proposed MRBs were demonstrated via both simulations and experiments.

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References