Design of a squeeze film magnetorheological brake considering compression enhanced shear yield stress of magnetorheological fluid

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Abstract. A magnetorheological brake, consisting of rotating disks immersed in a MR fluid and enclosed in an electromagnet, is proposed to replace the conventional heavy weight low response hydraulic disk brake. The frictional characteristics of the proposed brake can be controlled by regulating the yield stress of the MR fluid as function of magnetic field and normal compressive force. The controllable yield stress retards the surfaces of rotating disks, thus MR fluid can be used as a brake lining material. The present research work attempts designing a squeeze film MR brake by accounting compression enhanced shear yield stress of magnetorheological fluid. Theoretical calculations indicate that the estimated braking torque of the six plate squeeze film MR brake, under compression, is in the order of 600Nm. To validate the theoretical design and its findings, a prototype of single-plate squeeze film MR disk brake has been developed. Experimental test setup helps to illustrate braking torque under different control currents (0.0 to 1.25 A).

1. Introduction

Conventional hydraulic disc brake, used in high torque application, wears out with time and deteriorates the braking performance. The magnetorheological fluid (MR) brake, having time invariant friction performance, may be considered as a good replacement for hydraulic disk brakes. MR fluid consists of micron sized magnetically permeable particles dispersed throughout the non-magnetic fluid carrier. Iron powder, having high saturation magnetization, is the most popular material to be used as magnetic particles. Under the presence of the magnetic field, magnetic dipole moment within particles induces, causing dipole interactions to form chains in the direction of flux paths. The formed particle-chains restrict fluid movement and increase yield strength of MR fluid. Rotational movement of disk (shear mode) and axial movement of pad (compression mode) affect the particle chains, and therefore the braking torque of MR brake.

Design of MR brake has been studied by many researchers. Huang et al. [1] presented theoretical design of a cylindrical MRF brake assuming annular space between circumference of disk and
electromagnet filled with MR fluid, and braking torque due to shearing of that MR fluid. Li and Du [2] experimentally investigated torque characteristics of designed disc MR brake operating at low speeds. Bydon [3] described the construction and operation of Lord’s MR brake, which offers maximum 5.65 Nm torque at operating speed 1000 rpm. The brake was restricted to be operated within temperature ranging between -30° to 70°C. Park et al. [4] presented a theoretical design of MRF brake required for automotive applications. Sukhwani and Hirani [5] designed a MRF brake for high rotational speed up to 2000 rpm. They considered the effect of MR gap on the performance of MR brake. Nguyen and Choi [6] considered minimization of zero-field friction heat in theoretical design of automotive magnetorheological brake to optimize the brake performance. In all aforementioned MRF brake configurations shear phenomenon of MR fluids was accounted. Although shear mode alone has been studied thoroughly by many researchers, shear mode under compression should be investigated to enhance the performance of MRF brake. The advantage of this mode is the attainment of high stresses as compared to shear mode under the same external field strength. 

The effect of compression on yield strength of MR fluid was considered by Tang et al.[7]. They did experiments on an iron-based MR fluid and found an increase in structure-enhanced static yield stress (~800 kPa) compared to yield stress without compression. Hongyun et al. [8] experimentally investigated the compressive and tensile performance of a MR fluid under different magnetic field and compared the test results with the shear yield stress of MR fluids. Mazlan et al [9] experimentally investigated the influences of the applied current on two types of MR fluids (MRF 241 ES and MRF 132 DG) under compression mode. Tian et al. [10] achieved ultra high yield stress in a general electrorheological fluid under compression. As per their study a lower compressive velocity results in a higher yield stress under the same electric field. The viscous force acting on particles decreases as the compressive velocity increases, while the force acting from the neighboring particles to squeeze particles out increases.

After comprehensive literature review it can be said that mathematical relation expressing compressive resistance of the MR fluid in terms of various variables is missing. There is a strong need to find such expression. Therefore, it has been decided to design compressive magnetorheological fluid brake considering single disc configuration and characterize brake at various speed and control currents to capture its behavior.

2. Theoretical study on squeeze film MR brake

The mechanical response of MR fluids, used in MR brakes, is often modeled using Bingham fluid equation [5]

$$\tau = \tau_{yd} + \eta \frac{\omega r}{h} \quad (1)$$

Where, $\tau$ is shear stress of MR fluid, $\tau_{yd}$ is yield shear stress of the MR fluid, $\eta$ is viscosity of the MR fluid, $\omega$ is angular velocity of the disk brake, $r$ is the radius of the brake and $h$ is the MR fluid gap. The MRF brake configuration illustrated in Figure 1 indicates that $h$ (MR fluid gap) acts as torque arm. For such configuration, the braking torque produced by MR fluid [2] can be given by

$$T = 2 \times F \times h \quad (2)$$

Where, $F$ is friction force ($F = \text{shear stress} \times \text{area}$). The expression for friction force becomes,

$$F = \int_{r_1}^{r_2} \tau \times 2\pi r dr \quad (3)$$

Substituting equation (3) in equation (2),
For multidisc, having $n$ disks (friction surface = $2n$) MR brake equation (4) becomes

$$T = 2 \int_{r_1}^{r_2} 2\pi h \left( \tau_{yd} + \frac{\eta \omega}{H} \right) dr$$

$$\approx 2\pi h \tau_{yd}(r_2^2 - r_1^2) + \frac{4}{3} \eta \pi \omega (r_2^3 - r_1^3)$$

(4)

Where, yield stress ($\tau_{yd}$) is a function of magnetic field intensity ($H$) and can be expressed [5] as

$$\tau_{yd} = \alpha_1 H^{1.5}$$

(6)

For a typical value of yield stress = 69 kPa at magnetic field intensity $H = 200$ kA/m [11], the value of $\alpha_1$ is

$$\alpha_1 = 7.714 \times 10^{-4} \text{ Pa (m/A)}^{1.5}$$

To express yield stress in terms of control parameter (current), one needs to express $H$ in terms of input current $I$, number of turns in electromagnetic coil $N$, and MR gap $h$. With such substitution, the yield stress can be expressed as

$$\tau_{yd} = 7.714 \times 10^{-4} \left( \frac{NI}{2h} \right)^{1.5}$$

(7)

Substituting equation (7) in equation (5), expression of torque becomes

$$T = 5.454 \times 10^{-4} \pi h \left( \frac{NI}{2h} \right)^{1.5} (r_2^2 - r_1^2) + \frac{4}{3} \pi \eta \omega (r_2^3 - r_1^3)$$

(8)

Figure 2 shows a configuration of multidisc MR brake considering six plates rotary disk. The model consists of low carbon steel six plates rotary disk brake, separator plates are attached to housing which is made of low carbon steel. Electromagnet has 1000 number of turns of 25 AWG wire. Two radial shaft seals (Nitrile Rubber) are being used in such a manner that they make contact with the separator plates of the housing. Two single row deep groove ball bearing (low carbon steel) support the MR brake structure, $w_0$ is disk width, $w_h$ is width of separator plate of housing, $h$ is MR fluid gap, $r_1$ is inner radius of the disk and $r_2$ is outer radius of the disk.

Figure 1. MR fluid gap acts a torque arm in MR brake.
The magnetic field, induced by electromagnet, generates dipole moment in iron particles of MR fluid. Attractive forces between magnetized particles align them along the field direction as shown in Figure 3.

If a rapid compression-assisted-aggregation process is applied to force MR fluids it will form a microstructure that is much stronger than the single-chain structure [12]. MR fluids greatly exceed the theoretical strength predicted from the single-chain structure. Figure 4 shows two zones of six plates MR brake configuration. In compression zone iron particles get compressed and form thick columns with strong and robust end wherever in tensile zone iron particles break their chain structure thus reducing the strength of the chain. The empirical expression of compression-assisted-aggregation model of MR fluid [12] is given by

\[ \tau_y(H) = \tau_{yd} + K_H P_e \]  

(9)

Where \( \tau_{yd} \) is the yield stress of the MR fluid without compression-assisted-aggregation. The slope \( K_H \) increases with the field \( H \). Yield shear stress \( \tau_y(H) \) increases with the normal stress \( P_e \).

**Figure 2.** Proposed configuration of six plates rotary disk MR brake.

**Figure 3.** MR fluid particles align along the field direction.

**Figure 4.** Two zones of six plates MR brake configuration.
The braking torque produced by the MR fluid is given by,

\[ T = \left( \int_{r_1}^{r_2} r h_f 2\pi n r \tau_t \, dh \right) + \left( \int_{r_1}^{r_2} r h_s 2\pi n r \tau_s \, dh \right) \]  

(10)

**Figure 4.** Compression and tensile zone of six plate rotary disk.

Where, \( \tau_t = \tau_y(H) + \eta \frac{\omega r}{h} \), \( \tau_t \) is the stress acting in the compression zone of the disk, \( h_i \) is the initial gap of MR fluid as shown in Figure 3 and \( h_f \) is the final gap of MR fluid in the compression zone as shown in Figure 4. \( \tau_s \) is the stress acting in the tensile zone of the disk. Here \( \tau_s = \eta \frac{\omega r}{h} \) because in the tensile zone the shear stress is only due to viscous force (\( \eta \), viscosity) of the MR fluid. This is gross approximation, but it will provide a conservation approach (actual torque produced will be more than estimated torque). \( h_s \) is the MR fluid gap in the tensile zone of the disk, \( h_s = h_i + h_f = 2h_i - h_f \).

In equation (9), normal stress \( P_e \) can be expressed [13] as

\[ P_e = \frac{D}{3h} \tau_{yd} \]  

(11)

Where, \( D = 2r \) is the diameter of the disk. Therefore equation (9) becomes

\[ \tau_y(H) = \tau_{yd} \left( 1 + K_H \frac{2r}{3h} \right) \]  

(12)

Substituting equation (12) and equation (7) in equation (10) the torque expression becomes

\[ T = 2\pi n \left[ \left( \frac{2.727 \times 10^{-4} N^{1.5} 1.5 \left( \frac{1}{\sqrt{h_f}} - \frac{1}{\sqrt{h_i}} \right) (r_2^2 - r_1^2) \right) \right] \right. 

+ \left. \left( \frac{0.101 \times 10^{-4} N^{1.5} 1.5 \left( \frac{1}{h_f^{1.5}} - \frac{1}{h_i^{1.5}} \right) K(H)(r_2^3 - r_1^3) \right) \right) + \left( \frac{\eta \omega (\log h_i - \log h_f)}{3} (r_2^3 - r_1^3) \right) \] 

\[ + \frac{2\pi n \eta \omega}{3} (\log h_s - \log h_i)(r_2^3 - r_1^3) \]  

(13)
When normal stress $P_e$ is equal to zero, yield shear stress $\tau_y(H)$ is the yield stress of the MR fluid without compression assisted aggregation ($\tau_{yd}$).

To study the braking torque of MR brake, let $r_1 = 55 \text{ mm}, r_2 = 196 \text{ mm}, h = 1 \text{ mm}, w_d = 3 \text{ mm}, w_h = 3 \text{ mm}, n = 6$, be considered. Viscosity of MR fluid ($\eta$) depends on the applied magnetic field ($B$) in the Bingham model. The approximate polynomial curve of viscosity of MRF-122-2ED [14] is given by

$$\eta = -3.904B^4 + 12.44B^3 - 15.85B^2 + 10.21B$$

(14)

Where, $\eta$ is in Pas and $B$ is in Tesla. In this theoretical analysis MRF 241 ES [11] is used as MR fluid medium. The magnetic field intensity ($H = \frac{NI}{2h}$) of the MR fluid is in kA/m. The approximate B-H polynomial curve [11] is given by

$$B = -10^{-11}H^2 + 6 \times 10^{-6}H + 0.156$$

(15)

In equation (12) slope $K_H$ varies with the magnetic field. The polynomial curve of $K_H$ [12] with respect to magnetic field intensity ($H$) is given by

$$K_H = 0.266 - 4 \times 10^{-7}H + 9 \times 10^{-13}H^2$$

(16)

In this theoretical analysis speed is kept as 500 R.P.M. To find out the braking torque of compressive MR brake $h_i = 1 \text{ mm}$ and final MR gap $h_f = 0.5 \text{ mm}$ is assumed. Substituting above values in equation (13), variation of torque $T$ for different magnetic field has been determined and plotted in Figure 5. Figure 5 shows that when the viscosity is function of magnetic field, torque increases with the current because magnetic field ($H$) varies with current.

To validate the results of braking torque experimentally, small scale (1:5) single plate MR brake configuration has been proposed. Figure 6 shows the schematic drawing of a single plate compressive MR brake and Figure 7 shows the detailed drawing of the compressive MR brakes. This MR brake has inner radius of the disk $r_1 = 10 \text{ mm}$, and outer radius of the disk $r_2 = 44 \text{ mm}$, initial MR gap $h_i = 1 \text{ mm}$ and final MR fluid gap $h_f = 0.5 \text{ mm}$ have been considered. Figure 8 and Figure 9 show the $h_i$ and $h_f$ of the compressive MR brake structure.

![Figure 5](image_url)

**Figure 5.** Variation of theoretical torque ($T$) for different control currents.
Figure 6. Schematic drawing of MR brake.

Figure 7. Detailed drawing of MR brake.

Figure 9. MR fluid gap $h_i$ of the MR brake before and after compression.

Figure 10. MR fluid gap $h_i$ and $h_s$ in MR brake compression.
3. Experimental study on squeeze film MR brake

In the proposed single disk squeeze film MR Fluid brake accommodating MR fluid in the annular space along the periphery as well as both sides of the rotor have been developed. The required compressive force is applied on the disc through the fluid when the electrical current is supplied to the brake coil. Removing power from the brake coil decompress the armature housing to its initial position. MR brake configuration has been made considering 1000 number of turns of wire on side electromagnet as shown in Figure 6. Figure 6 shows that small gap $h$ between rotor and housing is filled with MR fluid. For ease in manufacturing, shrink fit between brake rotor (radius, $r_2$) and shaft (radius, $r_1$) are used as shown in Figure 7. Wiper seals slide along the shaft. It has transition fit to the shaft with 0.8 µm tolerance. Radial ball bearing is fitted to the shaft with -13 µm tolerance interference fit. The bearing has interference fit of 6µm tolerance. Electromagnet1 is attached to the split housing (radius $r_7$) and electromagnet2 is attached to the stator. Housing is mounted on two wiper seals which will slide along the direction of attractive force by the stator when magnetic field is applied compressing six element springs, which are constrained between housing and stator. On the removal of magnetic field, spring will decompress and housing will go back to its initial position ensuring air gap is equal to 0.5 mm because preload of the springs is zero. Figure 11 shows the fabricated squeeze film MR fluid brake model.

Figure 12 shows the experimental test setup of squeeze film MR brake. It consists of 2 H.P and 1500 R.P.M. D.C. motor with speed controller. A flywheel (20 kg and 0.42 kg $m^2$ inertia) is connected between the D.C. Motor and MR brake through bearing bracket, jaw coupling and flexible coupling. A D.C. power supply is used to apply the D.C. to the electromagnet of squeeze film MR brake. A toggle switch is used to control the motor ON/OFF and MR Brake ON/OFF.

![Figure 11. Fabricated model of squeeze film MR brake.](image1)

![Figure 12. Experimental setup of squeeze film MR brake.](image2)
The experimental torque is conducted to explore the torque output of the prototype squeeze film MR brake. In this study, commercially available LORD MRF-241ES [11] is used. The output torque is determined by dividing the power loss of the DC Motor with the angular speed of the shaft measured at different applied coil current.

The transmitted torque is measured for angular speeds of 200 and 400 R.P.M for input currents up to 1.25 A as shown in Figure 13. It implies that the operating speed of squeeze film MR brake does not affect the torque performance, since the viscous torque is relatively small compared to the torque generated by the MR fluid effects.

![Figure 13. Experimental torque of squeeze film MR brake prototype for different velocities](image)

![Figure 14. Theoretical torque and experimental torque of squeeze film MR brake.](image)
The experimental and theoretical torque output results have been compared in Figure 14. Percentage difference between experimental torque and theoretical torque is shown in Table 1. At 0.25A input current there is no squeezing action due to the high friction force of wiper seal. Friction force of wiper seal plays a dominating role in the squeezing action which is not included in the theoretical calculation.

**Table 1.** Percentage difference between experimental torque (ET) and theoretical torque (TT).

<table>
<thead>
<tr>
<th>Current(A)</th>
<th>ET (Nm)</th>
<th>TT (Nm)</th>
<th>% Difference between ET and TT</th>
</tr>
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<tbody>
<tr>
<td>0.00</td>
<td>0</td>
<td>0</td>
<td>0.00</td>
</tr>
<tr>
<td>0.25</td>
<td>0</td>
<td>0.21</td>
<td>-100.00</td>
</tr>
<tr>
<td>0.50</td>
<td>2</td>
<td>0.58</td>
<td>239.55</td>
</tr>
<tr>
<td>0.75</td>
<td>4</td>
<td>1.09</td>
<td>264.43</td>
</tr>
<tr>
<td>1.00</td>
<td>6</td>
<td>1.76</td>
<td>240.63</td>
</tr>
<tr>
<td>1.25</td>
<td>7</td>
<td>2.62</td>
<td>167.14</td>
</tr>
</tbody>
</table>

At 0.5A and 0.75A input current percentage differences are 239.55 % and 264.43 % respectively. As current increases, attractive force by the stator increases. Therefore compressive velocity on the disk through the MR fluid increases. A higher compressive velocity results in a shortest test time, corresponding to a more solid like mechanical property of a material [15]. Therefore when the compressive velocity was increased, the compressive strength of a visco-elastic material like MR fluids would increase, which increased the braking torque. The compressive resistance may be described as $P = \eta \dot{\varepsilon} + k \varepsilon$ [16], where $\eta$ is viscous parameter [11] and $k$ is elastic parameter [9]. These parameters need to be determined experimentally. The $\varepsilon$ and $\dot{\varepsilon}$ are compressive strain and compressive strain rate respectively. Table 2 shows the theoretical torque of squeeze film MR brake including strain rate to improve the calculated values. These results have been plotted in Figure 15. On comparing the results of Figures 14 and 15, it can be said that including strain rate improves the results.

**Table 2.** Theoretical torque (TT) of squeeze film MR brake including strain rate.

<table>
<thead>
<tr>
<th>Current(A)</th>
<th>ET (Nm)</th>
<th>TT (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>0.25</td>
<td>0</td>
<td>3.83</td>
</tr>
<tr>
<td>0.50</td>
<td>2</td>
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</tr>
<tr>
<td>0.75</td>
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<td>4.76</td>
</tr>
<tr>
<td>1.00</td>
<td>6</td>
<td>6.01</td>
</tr>
<tr>
<td>1.25</td>
<td>7</td>
<td>7.77</td>
</tr>
</tbody>
</table>
Figure 15. Comparison between experimental torque and theoretical torque including strain rate.

Table 3 shows a comparison of the prototype (described in the present research) and high speed MR brake designed by Sukhwani and Hirani [5]. In both the designs MR gap is kept as 1 mm. The results listed in Table 3, clearly shows far better performance of squeezing action braking compare to high speed MR brake detailed in Sukhwani and Hirani [5]. In other words squeeze film action shall be used for high torque application.

<table>
<thead>
<tr>
<th></th>
<th>High speed MR brake</th>
<th>Squeeze film MR brake</th>
</tr>
</thead>
<tbody>
<tr>
<td>MR fluid</td>
<td>MRF 336AG</td>
<td>MRF 241ES</td>
</tr>
<tr>
<td>MR gap (mm)</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Torque (Nm) at I = 0.75A</td>
<td>1</td>
<td>4</td>
</tr>
<tr>
<td>Maximum torque (Nm)</td>
<td>4</td>
<td>7</td>
</tr>
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</table>

4. Conclusion
A squeeze film MR brake for high torque application is designed and manufactured. Performance of the prototype MR brake is studied using MRF 241ES both theoretically and experimentally. Experimental transient torque analyses show that squeeze film MR brake generates 7 Nm torque for input current of 1.25A which is applicable for high torque application. Dynamic torque is not sensitive to angular velocities up to 400 R.P.M. since the viscous torque is negligible compared to magnetic field torque.
References
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