MAGNETORHEOLOGICAL FLUID BRAKE – BASIC PERFORMANCES TESTING WITH MAGNETIC FIELD EFFICIENCY IMPROVEMENT PROPOSAL

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A review of all magnetorheological brake types was presented. Based on overall braking torque analytical comparison for all magnetorheological brake types and other relevant parameters, the most promising design was selected. A test rig, utilizing selected brake type filled with magnetorheological fluid – Basonetic 5030 was manufactured and then tested. To analyze the effect produced by magnetic field on magnetorheological fluid and hence at overall braking torque, the authors used amplification factor. Results were discussed and the magnetic field efficiency improvements were proposed.

Keywords: magnetorheological brake, Basonetic 5030, test rig, amplification factor, efficiency improvements

Introduction

The conventional friction brake (FB) is the most commonly used brake type in almost any mechanical system today. However, it is characterized by drawbacks such as periodic replacement due to wear, large mechanical time-delay, bulky size etc. [13, 29], partially altered. Electromechanical brakes (EMBs) have potential to overcome some of these drawbacks and are a suitable FB replacement. Today EMBs are applicable in almost any mechanical system. Application of intelligent materials is the next step in the development of EMB.

Magnetorheological (MR) fluids belong to a class of intelligent materials that respond to applied magnetic field with fast, continuous, and reversible change in its rheological behaviour [3, 7, 28], partially altered. MR fluids are a type of suspensions, with carrier fluid usually mineral or synthetic oil, water, kerosene and micro size magnetizable particles dispersed in it. When exposed to external magnetic field particles form a chain-like structures thus changing the viscosity of the fluid. In this study, authors used BASF’s magnetorheological fluid, Basonetic 5030.

MR fluids have attracted extensive research interest in recent years since they can provide simple, quiet and fast response interface between electronic control and mechanical system [11, 20, 22]. A lot of work was done on MR fluid brakes modelling, properties investigation and control [3, 6, 11]. A wide range of MR fluid devices have also been investigated for their potential applications in different systems, such as: clutch system, vibration control, seismic response reduction, etc. [9, 10, 26].

MR fluid brakes have also been used in actuators due to their distinguished force control and power transmission features [5, 15]. By applying a proper control effort, viscosity with large varying range is achievable with the MR fluid brake. Currently, there are many solutions for MR fluid brake design. Some MR fluid brakes with attractive properties, such as high yield stress and stable behavior, have been developed and commercialized [4, 16].

The objective of this work was to compare overall braking torque analytical expressions and design complexity for all MR fluid brake types. Based on results obtained from these comparisons, MR fluid brake type with the most promising properties was manufactured and tested on a specially designed test rig. Results were discussed and a proposal for new MR brake design with higher magnetic efficiency was presented.

Magnetorheological effect

MR fluids are suspensions composed out of three major components: carrier fluid - usually mineral or synthetic oil, magnetizable particles - carbonyl iron powder and set of additives [2, 17], partially altered. When exposed to an external magnetic field (ON state), MR fluid acts as Newtonian fluid [7, 11] and can be described as:

\[ \tau = \eta \cdot \dot{\gamma} \]  

(1)

In (1) \( \tau \) represents shear stress, \( \eta \) the viscosity of the fluid and \( \dot{\gamma} \) shear rate. Often, for MR fluid brakes,
denoted as \( \dot{\gamma} = r \cdot \omega / g \), where \( r \) is rotor radius, \( \omega \) and \( g \) are angular speed and MR fluid gap length, respectively.

When in ON state, MR the rheological properties of MR fluid change. Magnetizable particles induce polarization and form chain-like structures in magnetic flux path direction, thus changing apparent viscosity of the fluid. ON state behavior of MR fluid is often represented as a non-Newtonian [1, 3, 5], having a variable yield strength. The usage of Bingham’s model (2), in this situation, gives reasonably good results, [1, 11, 20, 22]:

\[
\tau = \tau_B + \eta \cdot \dot{\gamma} ,
\]

where \( \tau_B \) is the yield stress, developed in response to the applied magnetic field. Its value is a function of the magnetic field induction \( B \).

When used in a device, MR fluid can be in one of four modes: shear, flow (pressure), squeeze and pinch, [8, 18, 28]. In brake i.e. torque transfer applications, MR fluid operates in shear mode [1]. Braking torque values are adjusted continuously by changing the external magnetic field strength.

**Magnetorheological brakes**

MR brake consists out of four main parts: rotor, housing i.e. stator, coil and MR fluid, Figure 1. The shape of the rotor is what differentiates MR brake types from each other. One needs the quantitative parameters of MR brake, to be able to determine its specific application suitability.

MR brake types, mechanical model, quantitative parameters comparison for all MR brake types are presented in next section.

**Magnetorheological brake types**

Through literature research [9, 14, 22, 23, 24], authors of this paper have identified five major MR fluid brake designs: drum brake, inverted drum brake, disk brake, T-shape rotor brake and multiple disks brake, Fig. 1.

Drum brake along with the disk brake is the easiest designs to manufacture. However, large inertia is its major drawback compared to disk brake design [1]. The disk brake design is more compact than all other designs but is also more complex to manufacture. Despite its advantages, this design is not so common in literature [22].

**Figure 1: Types of MR brakes: a) drum, b) inverted drum, c) disk, d) multiple disks, e) T-shaped rotor, [28]**

For all aforementioned MR brake types, the rotor has a cylindrical shape and the magnetic flux lines run in the radial direction, Fig. 1.

To author’s knowledge, an in-depth comparison of all these architectures is not yet available.

**Mechanical model**

The key objective in MR fluid brake design is to establish the relationship between the overall braking torque, magnetic field strength and design parameters. Interaction of MR fluid and inner surfaces of the brake will generate the braking torque. Based on Eq. (2) and the specific geometrical configuration of MR brake, for all MR brake types, it applies:

\[
dT = 2\pi N r \tau^2 dr ,
\]

where:
- \( N \) – number of surfaces of the rotor, perpendicular to the magnetic flux lines and in contact with MR fluid,
- \( r \) – the radius of the rotor.

The overall braking torque \( T_{\text{Overall}} \) consists of three components:
- the magnetic field induced component \( T_B \), due to the field-dependent yield stress,
- the fluid viscosity dependent component \( T_{\text{vis}} \) and
- the friction induced component \( T_{\text{fric}} \).

Thus, the overall brake torque:
The sum of the first two components $T_B$ and $T_{vis}$ i.e. the braking torque can be obtained by the following integral:

$$T_B + T_{vis} = 2\pi N \int_{R_i}^{R_o} \tau r^2 dr,$$

where $R_O$ and $R_i$ are the brake's rotor outer and inner radii respectively. Considering practical conditions, for all MR brake types, the value of the $R_i$ can be ignored because the $R_O$ is several order of magnitude of the $R_i$ (Figure 2).

**Figure 2: MR fluid disk brake simplified representation**

Based on Eq. (5), the authors formed the final analytical expressions for all five MR brake designs. Expressions were adopted from several different literature sources [1, 6, 12, 13, 21, 27, 29] and were partially altered in order to make the comparison easier. The last part of torque, $T_{fric}$ can be precisely obtained only by torque gauge.

$$T_{Overall} = \frac{4}{3} \pi R_O^3 + \frac{\eta g \rho h}{g} + T_{fric},$$  

(6)

$$T_{Overall} = \pi N \left( \frac{4}{3} \pi R_O^3 + \frac{\eta g \rho h}{g} R_O^4 \right) + T_{fric},$$  

(7)

$$T_{Overall} = 4 \pi h (\tau R_O^2 + \frac{\eta g \rho h}{g} R_O^4) + T_{fric},$$  

(8)

$$T_{Overall} = 8 \pi h (\tau R_O^2 + \frac{\eta g \rho h}{g} R_O^4) + T_{fric}.$$  

(9)

Variable $h$ is the height of the rotor.

It is now easy to distinguish components of overall braking torque in Eq. (6–9), for disk, multiple disks, drums and T-shaped rotor, respectively. The yield stress $\tau_B$ given in Eq. (2), varies with magnetic induction, but can reasonably be fitted with the third-order polynomial [12], as follows:

$$\tau_B = K_1 B + K_2 B^2 + K_3 B^3,$$

(10)

where $K_i$ represents coefficients of regression.

**Performances comparison**

MR brakes can be compared on several different aspects e.g. overall braking torque, dynamic range, mechanical simplicity, inertia, electric power consumption, torque to volume ratio, compactness, etc. Authors of this paper compared level of mechanical simplicity for all five MR brake types and the overall braking torque based on analytical expressions, given in Eq. (6–9), Table 1. In addition, the dynamic range and inertia were consider and presented in the same table.

**Table 1: Characteristics of various MR brake designs**

<table>
<thead>
<tr>
<th>Brake type</th>
<th>$T_{Overall}$</th>
<th>Level of mechanical simplicity</th>
<th>Dynamic range $T_B/T_{vis}$</th>
<th>Inertia</th>
</tr>
</thead>
<tbody>
<tr>
<td>(1) M H</td>
<td>$\frac{\tau_B g \eta^2}{\rho h}$</td>
<td>R</td>
<td>$1/2mR_O^2 *$</td>
<td></td>
</tr>
<tr>
<td>(2) M L</td>
<td>$\frac{4}{3} \tau B g \eta^2 R_O$</td>
<td>L</td>
<td>$1/2mR_O^2$</td>
<td></td>
</tr>
<tr>
<td>(3) H L</td>
<td>$\frac{4}{3} \tau B g \eta^2 R_O$</td>
<td>L</td>
<td>$1/2mR_O^2N_D$</td>
<td></td>
</tr>
<tr>
<td>(4) H H</td>
<td>$\frac{\tau_B g \eta^2}{\rho h} R_O$</td>
<td>H</td>
<td>$mR_O^2 - m_i R_i^2 / 2$</td>
<td></td>
</tr>
</tbody>
</table>

Explanations:

Brake types: (1) drum and inverted drum, (2) disk, (3) multiple disks, (4) T-shaped rotor.


* For inverted drum brake design, expression is the same as for the T-shape rotor design.

$N_D$ – the number of rotor disks, $m_i$ – missing inner mass of the disk.

Dynamic range represents ratio of field induced component $T_B$ and viscous component $T_{vis}$ (friction torque component not considered).

T-shaped rotor brake performs best dynamic and can offer biggest overall braking torque, however it has a large inertia, and is considered difficult to manufacture. The other four designs are globally less compact than the T-shaped rotor design. The two drum based designs have comparable performances but are burdened with even larger inertia for same rotor radius. Finally, the disk and multiple disks designs offer a good alternative, being mechanically the simplest, with smaller inertia and giving reasonable dynamic range and the overall braking torque.

**Test rig**

Based on test rigs literature research [10, 15, 20, 27] and MR brake properties comparison (Eq. (6–9) and Tab. 1),
the authors of this paper selected the most promising MR brake design and manufactured it. With lowest inertia, good dynamic range and the highest mechanical simplicity, the disk type promised the biggest potential.

For performance evaluating of the selected MR brake type a rig was set up. The test rig was designed and manufactured at Faculty of technical sciences, Novi Sad – Serbia, and is presently at its Laboratory for engines and vehicles. The test rig with its parts is depicted in Figure 3 and 4, and consists of four main parts:

- drive,
- power supply,
- MR brake and
- measuring and data acquisition equipment.

An 8-pole AC squirrel cage motor with 0.75 kW and nominal speed of 700 rpm, model 5 AZ 100 LA – 8 (Končar) was placed at one end of the support-frame, of the test rig. The inverter - Micro Master (Siemens), Fig. 4, position 2, controls the direction and variable-speed of the AC motor. Speed range was from 150 rpm to 750 rpm with 50 rpm increment. These two elements form a drive part of the test rig. The flexible coupling connects AC motor and the shaft of the MR brake. MR brake rests on two self-aligning ball bearings into housings P203 (FK Bearing group). To avoid leakage of MR fluid, Nitrile Rubber lip seals, suitable for MR type application, have been used.

Torque transfer from the MR brake to a measurement device was done indirectly. A load arm, connected to the MR brake housing at one end, rests on top of the load cell on the other end, Fig. 3. Thus, by measuring the force on the load cell, the value of transmitted torque was obtained. Load cell was internally calibrated by calibration weights. The capacity of the load cell, 1030 (Tedea), was 15 kg.

An optical encoder, model AMT102-V-REV-C (CUI Inc.), was connected to the shaft of the MR brake, at the opposite side of the AC motor and rotated at the same speed sample rate was 2048 per rotation. The signals were processed by universal amplifier, KWS 673.A2 (HBM), Fig. 4, position 3. The DC power supply, EA PS 2016-100 (Elektro-automatik), was connected to the leads of the coil to provide flux generation. This was the control current, with range of 0 A to 2 A and 0.2 A increment. The coil, made of copper wire with diameter of 1 mm (18 gauges) has been coiled on outer radius of the MR brake housing. The MR fluid used in this experiment was Basonetic® 5030, from BASF® [17]. It is a carbonyl iron powder based MR fluid.

Experimental results and improvement proposal

The goal of this experiment was to determine overall braking torque capabilities of selected MR brake design. The experiment was conducted on a specially designed test rig, with different control currents and speed sets. To eliminate the effects of previous observations, different combinations of control current and rotational speed were set for each reading. To bring repeatability in the reading, every speed set was carried out twice at different instance of time. For every reading, approximately 1 min time, before torque data recording, has been allotted to distribute the carbonyl iron particles uniformly in the MR fluid and form a stable structure.

Experimental results

The experiment itself consisted out of tree parts. The first part was to determine the influence of the supporting ball bearings and seals, without MR fluid inside the brake and no control current applied. This was a friction braking torque component. Second part of the experiment had the same setup but it included MR fluid inside the brake. Viscous torque data was then
recorded, assuming that bearings and seals did not change their friction characteristics in time.

Aforementioned recordings were needed in order to get clear and precise information about field induced component. This was the third part of the experiment and it included MR fluid inside the brake and application of the control current.

The same speed sets were used for the friction and the viscous torque component measurements were repeated. For each speed set, a 2 minute recording time was used. Some field induced component results are depict in Figure 5. Magnetic field influence is apparent.

![Figure 5: Samples of overall braking torque results: a) 0.2 A at 150 rpm, b) 1.6 A at 300 rpm](image)

Because of the large number of data obtained in this experiment, authors decided to use amplification factor to determine the effect produced by magnetic field. Amplification factor (AF) represents relation between overall braking torque and sum of friction and viscous torque, i.e. relation between the ON and OFF state of the MR fluid.

\[
AF = \frac{T_{\text{Overall}} \text{ at current}}{T_{\text{Overall}} \text{ at zero current}}
\]

Amplification factor curves for all 13 speed sets are plotted in Figure 6. This figure shows linearity in amplification factor with increase in control current. This was expected, since the higher the current, the higher the field induced torque should be. Results are presented with rotational speed variation as well.

Figure 6 indicates that the amplification factors for speeds 150, 200, 250, 300 and 350 were much larger than the ones for 450, 500, 550, 600, 650, 700 and 750 rpm.

![Figure 6: Variation in amplification factor with control current](image)

**MR brake improvement proposal**

In order to increase the overall braking torque of any MR brake type its geometry needs to be optimized [14]. State of the art MR brake designs base their torque generation on the inner section of the coil [4, 10, 13]. These designs are rather inefficient, because the magnetic field is not used to its full potential and active MR area is small. Flux mostly travels through the housing of the MR brake. In this configuration, magnetic field is concentrated on small area of the MR fluid.

![Figure 7: Novel MR brake design with enhanced magnetic field efficiency](image)

By altering earlier MR brake designs, it is possible to concentrate magnetic field effect. One such design has been presented in Figure 7.

Compared to the dimensions of the tested MR disk brake, radius of the coil was reduced so that it could be placed in a more suitable position inside new MR brake design. New coil position allows for magnetic flux to be
used more efficiently. Now, the magnetic flux inside MR brake passes through MR fluid several times.

Based on analytical comparison, presented in Eq. (6–9) and Tab. 1, new design features two disks in order to create more MR fluid active area. Greater number of disks creates more MR fluid gaps thus the volume of MR fluid can be increased.

**Conclusion**

Authors compared all MR brake types on several different aspects. Based on this information, the most promising MR brake type was selected, manufactured and tested on a specially designed test rig.

The MR brake produced desirable results, which coincide with literature sources. Approximately linear relation between the overall braking torque and the control current intensity was observed. To present results in more readable manner, the amplification factor was introduced.

The experiment showed that the tested MR brake has potential for practical applications due to easiness and accuracy of control. However, the value of the overall braking torque is still small. To increase it, better utilization of the existing magnetic field is needed. The authors suggested different approach in comparison to the conventional MR brake design that would increase the overall braking torque by increasing the magnetic field efficiency and the contact area of the MR brake fluid. By multiplying the number of the disks in contact with MR fluid, value of the overall braking torque will multiply as well, Eq. (7).

In order to maximize the potential of the proposed MR brake, further investigations on magnetic field propagation is needed as well as design optimization. Finite element method model of the novel MR brake is the next step in this process.

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