EXPERIMENTAL EVALUATION OF MAGNETORHEOLOGICAL DISK BRAKE

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INTRODUCTION

The convectional 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. [29, 13]. 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 EMB's development.

Magnetorheological fluids (MR fluids) belong to a class of intelligent materials that respond to applied magnetic field with fast, continuous and reversible change in their rheological behaviour [7, 28, 3]. MR fluids are type of suspensions. Carrier fluid is usually mineral or synthetic oil, water, kerosene etc. with dispersed micro size ferromagnetic particles. When exposed to external magnetic field, particles form a chain-like structures thus changing fluid's viscosity. In this research, authors used BASF's MR 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 [22, 20, 11]. A lot of work was done on MR fluid brakes modelling, properties investigation and control [6, 11, 3]. A wide range of MR fluid devices have also been investigated for their potential applications in different systems, such as: clutch systems, 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 behaviour, have been developed and commercialized [16, 4].

The objective of this work was to evaluate overall braking torque for experimental MR fluid brake model. Based on literature research [1] and earlier authors' works, MR disk brake was manufactured and tested on a specially designed test rig. Results were presented as amplification factors and discussed latter on.

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 [17, 2]. When exposed to an external magnetic field (ON state), change in MR

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fluid's viscosity occurs. In the absence of an external magnetic field (OFF state), MR fluid acts as Newtonian fluid [7,11] and can be described as:

$$\tau = \eta \cdot \dot{\gamma} \tag{1}$$

where: τ represents shear stress, η - fluid`s viscosity and $\dot{\gamma}$ - shear rate, often, for MR fluid brakes, denoted as $\dot{\gamma} = \frac{r\omega}{g}$, where r is rotor radius, ω and g are angular speed and MR fluid gap length, respectively.

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

$$\tau = \tau_{\mathrm{B}} + \eta \cdot \dot{\gamma} \tag{2}$$

where: τ_{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 of four main parts: rotor, housing (stator), coil and MR fluid, Figure 1. The rotor's shape is what differentiates MR brake types from each other. One needs MR brake's quantitative parameters, to be able to determine its specific application suitability.

Magnetorheological brake types

Through literature research [22, 24, 9, 14, 23], 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, Figure 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 the most common MR brake design found in literature today and was the first one to be investigated [25]. It is the easiest one to manufacture and gives reasonably good results in terms of weight and compactness [14]. There are some variations in MR disk brake design such as: the use of two coils instead of one in order to increase the magnetic pole area and/or relocation of the coil on top of the disk in order to reduce its external diameter [22], but the basics remain the same. It is also interesting to note that the MR disk brake design is currently the only one commercially available as a standard product, manufactured by Lord Corporation [16] and that it was used in several studies [7, 21, 19].

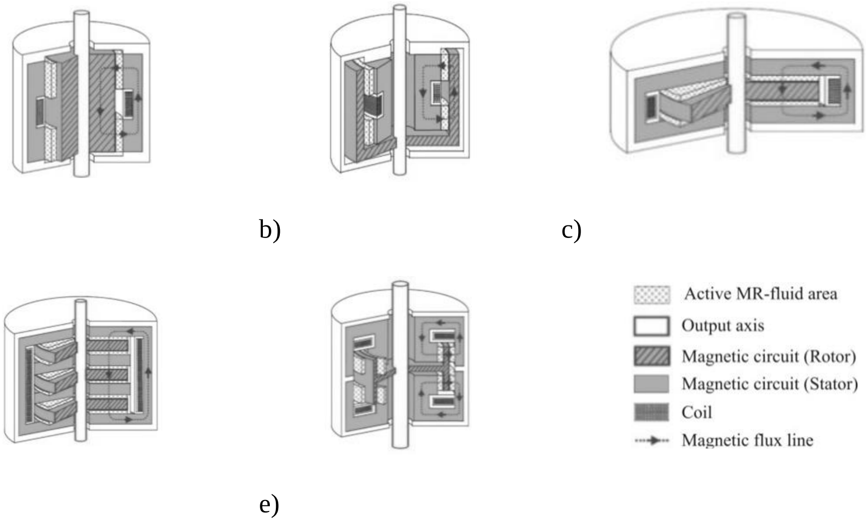

Figure 1 Types of MR brakes, a) drum, b) inverted drum,

c) disk, d) multiple disks, e) T-shaped rotor [29]

In order to increase compactness of the MR disk brake design, several disk-shape rotors can be used instead of one, with segments of stator located between each rotor disk, Figure 3, d). This multiple disk design is very popular in literature and was used in several applications that required high torque in limited space and weight [22, 13, 23]. The equations describing this particular design are very similar to those of the single disk brake.

The T-shape rotor brake design, Figure 1, e), is more compact than all other designs, but it is also more complex to manufacture. Despite its advantages, this design is not so common in literature [22].

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

To authors' 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. Change in MR fluids viscosity and its interaction with the inner surfaces of the brake will generate the overall braking torque. Based on Equation (2) and MR brake's specific geometrical configuration, for all MR brake types, it applies:

$$dT = 2\pi N \tau r^2 dr \tag{3}$$

Where: N is number of surfaces of the rotor, perpendicular to the magnetic flux lines and in contact with MR fluid and r is the rotor's radius.

The overall braking torque, $T_{Overall}$, consists of three components:

- the magnetic field induced component, T_B, due to the field-dependent yield stress,

- the fluid viscosity dependent component, T_{vis} . and
- the friction induced component, T_{fric}

Thus, the overall brake torque is:

$$T_{Overall} = T_B + T_{vis} + T_{fric} \tag{4}$$

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

$$T_B + T_{vis} = 2\pi N \int_{R_i}^{R_o} \tau r^2 dr \tag{5}$$

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 R_o is several order of magnitude of R_i . The final MR disk brake overall braking torque expression is:

$$T_{overall} = T_B + T_{vis} + T_{fric} = \frac{4}{3}\pi\tau_B R_o^3 N_d + \pi\eta\frac{\omega}{g} R_o N_d + T_{fric} \tag{6}$$

where: N_d , ω , g are number of disks in use, angular speed and MR fluid's gap, respectively.

Aspect of commercial availability of materials was considered, so authors decided to use AISI 1018 chemical equivalent, Č1221. The $B-H$ curves for Č1221 and Basonetic 5030 are presented in Figure 2. The first step in deterring the MR brake's analytical braking capacities was to determine the operating point couples for materials in use. The higher values were preferable but also require larger coil. Two coils were manufactured for this experimental purpose, with 250 and 500 coils.

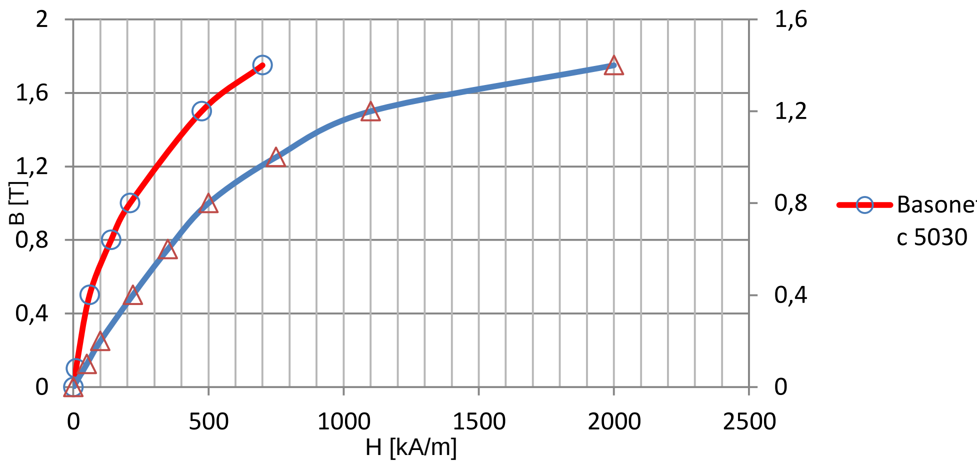

Figure 2 B-H curves for Basonetic 5030 and AISI 1018

In this configuration, magnetic field induction was low and around 0.1 T, placing material's operating points at the very root of $B-H$ curves. This low value magnetic field produced low value shear stress in the MR fluid. Placing above mentioned brake's characteristic and fluid's ON and OFF values in Equation 6, estimated overall braking torque was around 0.2 Nm

Test rig

Based on test rigs literature research [10, 27, 20, 15], the authors of this paper selected the most promising MR brake test rig design and manufactured it. With lowest inertia, good dynamic range and the highest mechanical simplicity, the disk type promised the biggest potential.

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 Figures 3 and 4, and consists of four main parts:

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

An 8-pole AC motor, 5 AZ 100 LA – 8 model (*Koncar*), with power of 0.75 kW and maximum speed of 750 rpm, Figure 3, position 2, was placed at one end of the support-frame of the test rig. The inverter –VLT 5400 (*Danfoss*), Figure 3, position 1, controls the AC motor's direction and variable-speed. Speed was changed with increment of 50 rpm, in a range from 50 rpm to 700 rpm. These two elements form a drive part of the test rig. The flexible coupling, Figure 3, position 3, connects AC motor and the MR brake's shaft. MR brake rests on two self-aligning ball bearings, 6000 (*Fag*). 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, Figure 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, PW6CC3MR (*HBM*), was 7 kg.

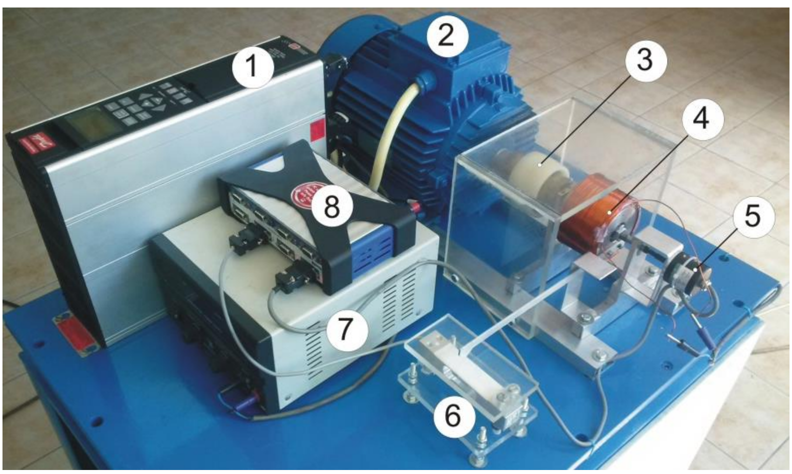

Figure 3.Test rig for magnetorheological brake performance evaluating:

1. inverter, 2. AC motor, 3.coupling, 4. MR fluid brake, 5. optical encoder, 6. load cell, 7. DC power supply,8. data acquisition card

The optical encoder, model AMT102-V-REV-C (*CUI Inc.*), was connected to the MR brake's shaft, 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 data acquisition card, Quantum MX840A *(HBM)*, Figure 4, position 8. 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 5 A or 10 A, depending on coil configuration. Increment was 0.2 A. The coil, made of copper wire with diameter of 1 mm (18 gauges) has been coiled on outer radius of the MR brake housing, ϕ 73 mm.

The MR fluid used in this experiment was Basonetic® 5030, from BASF® [17]. It is a carbonyl iron powder based MR fluid.

A typical testing procedure was as follows. Firstly, the MR brake's shaft was set to a certain speed for 1 min as an initial condition, which stirred the MR fluid in the brake to distribute it uniformly. The desired magnetic field was then applied by setting the coil current (control current) and waiting for 1 min. This ensured the forming of MR fluid's stable structure. The load cell detected transmitted torque. Finally, the signal from the load cell was processed and recorded.

EXPERIMENTAL RESULTS

The goal of this experiment was to determine overall braking torque capabilities of the 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, before torque data recording, approximately 1 min time has been allotted to uniformly distribute the carbonyl iron particles in the MR fluid and form a stable structure.

The experiment 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 component of braking torque. The 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 that were used for measurements of the friction and the viscous torque components were repeated. For each speed set, there was a 2 minutes recording time. Some field induced component results are presented in Figure 4. Magnetic field influence is apparent.

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

$$\text{Amplification factor} = \frac{T_{\text{Overall}} \text{ at current I}}{T_{\text{Overall}} \text{ at zero current}} \tag{7}$$

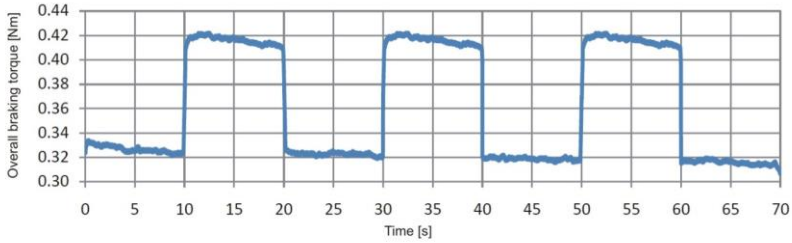

Figure 4 Sample of overall braking torque results

Amplification factor curves for all 14 speed sets are plotted in Figure 5. 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.

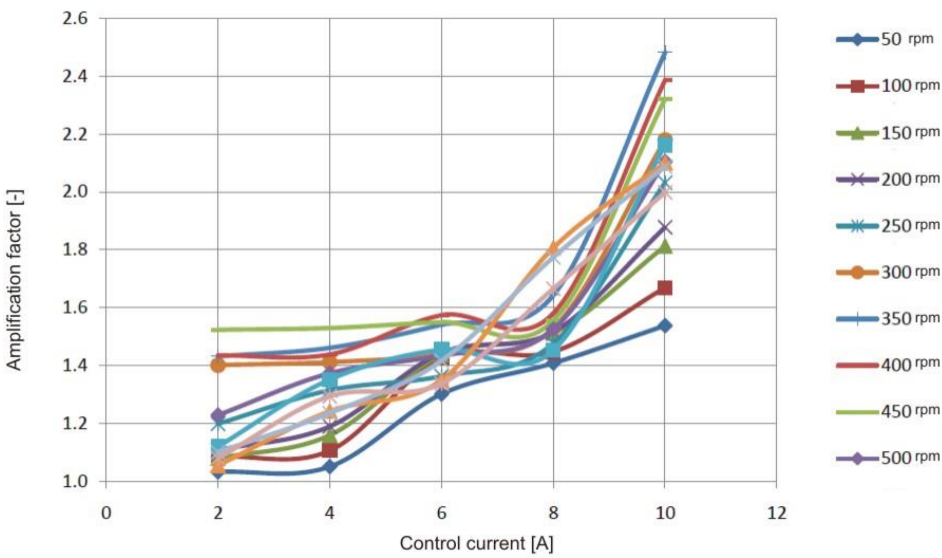

Figure 5 Variation in amplification factor with control current

Transition control current value for this MR brake model was 8 A. This was somewhat opposite to the torque predicted by equation (6) where T_{Overall} should increase more linearly with control current. This was not the case here. However, other authors have also reported the same trend - reduction of amplification factor with increment of speed and control current [27, 20]. This may be due to the possible shear thinning effect of the MR fluid at high shear rates. This reduces the effectiveness of MR brake at higher speeds. Therefore, to make MR brake effective at higher speed operation, one needs to think of „shear stable MR fluid“.

CONCLUSIONS

Based on literature research, the authors have decided for the most promising MR brake type, manufactured it and tested it on a specially designed test rig.

The MR brake produced desirable response and predicted 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 experiments have shown 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 MR brake's fluid contact area. By multiplying the number of the disks in contact with MR fluid, value of the overall braking torque will multiply as well, Equation (6).

In order to maximize proposed MR brake's potential, further investigations on magnetic field propagation are 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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