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Response of A Magnetorheological Brake under Inertial Loads

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Abstract: The study is objected to investigate the response of a magnetorheological brake (MRB) system under thefree move inertial mass. The disk-type MRB comprises of a rotating disk immersed in magnetorheological fluids (MRFs) and surrounded by an electromagnet coil. The magnetized coil causes a solidification of the MR fluid so that the shear stress between the moving part and static part increases resulting in the decrement speed of the moving parts. The shear stress can be varied by applying different electric current to the coil. The study began with the part design using the3D modeling software, followingbythe magnetostatic analysis. The flux density across the magnetorheological fluid could be predicted through this finite element magnetic simulation. The quantity of magnetic flux was then used to predict the shear stress between static and moving parts. The fabricated MRB was integrated onto a test rig which employs load cell and speed sensor as well as completely instrumented with data acquisition.Since the MRB test rig performed a simple free rotation system, a linear second order differential equation was derived to model the stopping time and braking torque behaviors. The equation of motion was built in a Simulink model, and the simulation results were compared to the real measurement. The achievable braking torque was also presented based on theaverage value from the load cell.

Keywords: magnetorheological fluids; magnetorheological brake; stopping time; braking torque

1. Introduction

Vehicle performance, safety, and cost have been becoming a major focus in theautomotive industry for many years due its potential improvement. In terms of cost and safety, automotive engineers were struggled to develop high safety vehicle with low production cost [1,2]. A common effort to achieve this aim is by minimizing the number of parts used in avehicle. X-by-wire is one of thetopics as a solution for this recent automotive issue. In a vehicle, x-by-wire has been employed in several segments for example steering and braking systems. The x-by-wire means the replacement of conventional mechanical apparatuses by electrical systems [3,4]. This discussion relates to the braking system as one of x-by-wire implementation that is so called brake-by-wire system. The primary goal of this study is the development of an actuator for abrake-by-wire system that utilizes magnetorheological fluids (MRFs).

The brake-by-wire system can be realized by replacing the mechanical components that connect the brake units on each wheel and the brake pedal with electrical parts. According to [5], the conventional hydraulic brake system has been firstly used on the brake-by-wire system that still utilized for failure safety requirements. All the brake control functions are implemented in one main electronic control unit. Meanwhile, the hydraulic system is required for security reasons to certify braking in case of the electrical failures [6]. There are several benefits of implementing thebrake-by-wire system. The delay between the time brake pedals pressed by the driver and the corresponding brake response of a conventional hydraulic brake

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exhibits 200-300 ms[6]. The delay caused by the pressure growthis restricted by head losses within the hydraulic lines. The use of electric brake system has the potential to reduce this time delay drastically, resulting in a reduction in braking time and distance. Besides that, the properties and behaviors of the brake will be easy to adapt by simply changing software parameters and electrical outputs instead of adjusting mechanical components [7].

MRFsare suspension containing fine iron particles of 1-20 microns in diameter dispersed in carrier fluids typically mineral oil with surfactant. The portion of magnetizable particles within the fluid carrier achieves 40% volume fraction[8]. When the fluid is subjected to a magnetic field, the magnetizable particles become magnetically dipoles and start to align along the fluxes as shown in Figure 1. The particles within the MRFs liquid carriedare trapped between the dipoles; hence, the movement of the fluid is restricted by particle chains thus increasing its viscosity. The fluid stiffens in the presence of magnetic fields in a fraction of milliseconds and behaves as a non-Newtonian Fluid. When the magnetic field is released, it displays a Newtonian fluid behavior. This irreversibility is often used for semi-active devices that need hydraulic and rheological behaviors such as semi-active vibration absorbers [9,10].

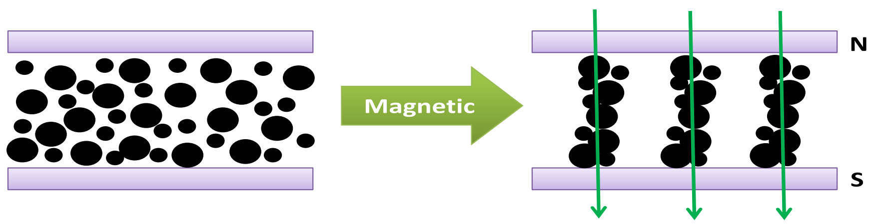


Figure 1. Aligning process of magnetizable particle within MRFs

It is important to note that this work utilizes MRFs as part of the magnetorheological brake-by-wire actuator. In fact, there is another type of fluids namely Electrorheological fluids (ERFs) which have the same principle work with the MRFs. ERFs is also a linear viscous liquid whose rheological behavior changes under the influence of an applied electric field, instead of a magnetic field. However, there are many drawbacks to ER fluid, including relatively small rheological changes and extreme property changes in temperature. Due to the drawbacks of ER fluids that require high control voltages, incapability in producing high shear forces and are susceptible to contaminants, they are not ideally suitable for automotive applications [10,11].

The rapid response of the MRFs has made this smart fluid becoming most alternative choice in many applications that need interface electromechanically such as clutch, damper, brakes, haptic device and so on [12]. Among those application, MRB is considered to be one of most interested topic in magnetorheological devices fields. The application of MRFs in the braking system is a relatively recent topic. MRB implements the most basic working mode of the MRFs namely shear mode. The braking effect is taken from the benefit of field dependent viscosity changing which causes frictions between static and moving parts. In order to enhance the performance of MRB, many types of MRB have been proposed and evaluated such as disc-type, drum-type and a combination of disc-type and drum-type MRBs as well as T-shaped MRBs [13,14]. In this work, disk-type MRB was proposed and investigated its performance.

Some previous works on MR brake can be found in many literatures. Park et al. presented a design optimization procedure using simulated annealing combined with finite element simulations involving magnetostatic, fluid flow, and heat transfer analysis [4]. In 2008, The same group [6] continued their studies on MRFs selection for MRB application, magnetic circuit design and torque requirements for automotive application. Followed by Jolly *et al.*[10], in which the work focused on investigation of practical design criteria for MRB such as material selection, sealing, working surface area, viscous torque generation, and MRFs selection for basic automotive braking system.

The dynamics of the MRB is not widely explored particularly the response of MRB against the free rotating inertial mass which relates with stopping time response and also its braking torque in variously applied static electric current. These two variables are important for speed and torque controls application. This article discusses the MRB behavior in terms of its stopping time and braking torque responses in various electric currents. The design of MRB is presented clearly covering its structure and finite element magnetic simulation. The MRB performance tests are conducted by rotating an inertial body with a certain mass at a desired velocity.Furthermore, the coupled MR brake is energized with a particular DC current to reduce the angular velocity of the body until zero speed rapidly by using the MRB net braking torque.

The outlines of thearticleare described as follows; in section 2, the description ofthe working principle and proposed thedesign of the disk-type MRB be presented. The discussion also covers the MRB design and materials. Section 3 is the mathematical modeling of the MRB,which covers field dependent braking torque model as well as linear second order system of the testing apparatus. Section 4 describes the simulation of themagnetic field using thefinite element software. Section 5 explains the test rig facility and measurement of the MR brake parameters. Section 6 discusses the stopping time responses of the MR brake under various input parameter. Section 7 explains the braking torque generated bydifferent constant electric currents. Finally, this paper is enclosed with conclusion and recommendation.

2. Design and Materials

This section describes the configuration of MR brake and its working principle. As it was introduced in Section 1, ER and MR fluids possess the property of changing their viscosity as an electric, or magnetic fields are applied. The changing viscosity is in turn proportional to the friction exerted on any moving body within the fluids. Hence, the brakes application consists of a disk immersed in MRFs. The braking torque exerted on the disk depends on the magnetic fields occurred within the MRFs. In anautomotive application, the magnetic fields are regulated according to the driver's pressure on the brake pedal. In the present design, an electromagnet is used to produce the magnetic field inside the MRB.

The schematic diagram of the cross-section MRB is presented in Figure 2. Next, the assembly of the MR brake is described in the following discussion. Thedesigned MRB consists of four main sections namely; a moving rotor (shaft and disk), a static body or housing (Wall), MRFs, and electromagnet unit. The wall and diskare mainly made of soft magnetic steel i.e. mild steel.The coil having 400 turns is a bronze wire as commonly used in DC motor coils. The bronze wire has a diameter of 0.45 mm. The rotor is rigidly fitted to the brake housing via two deep groove ball bearings. The radial gap between the static body and disk is 3 mm which is filled by the MRFs (MRF-132AD, Lord Corp.). The raw materials of each partare listed in Table 1.

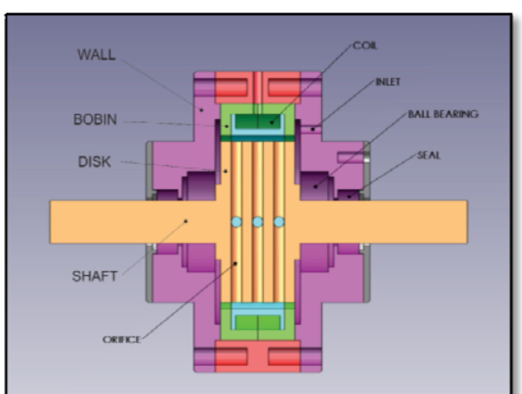


Figure 2. Design of the proposed MRB

Table 1. List of components

Num	Item	Quantity	Materials
1	Wall / Body	2 pcs	Mild steel
2	Shaft	1 pcs	Aluminium alloys A6xxx
3	Disk	1 pcs	Mild steel
4	Bobin	1 pcs	Aluminium alloys A6xxx
5	Coil	1 set	Bronze wire dia. 0.45 mm
6	MRF	unit volume	MRF-132AD
7	Deep groove ball bearing	2 pcs	standard
8	Oil seal	2 pcs	standard

Table 2. Properties of MRF-132AD

Property	Value/ limits
Base fluid	Hydrocarbon
Operating temperature	,-40 to 130 (°C)'
Density	3050 (kg/m^3)
Color	Dark gray
Weight percent solid	80.98 (%)
Specific heat at 25 (°C)	800 (J/kg K)
Thermal conductivity at 25 (°C)	0.25-1.06 (W/m K)
Flash point	>150 (>320)
Viscosity (slope between 800 and 500 Hz at 40 °C)	0.112 ± 0.02
k	0.269 (pa m/A0
β	1

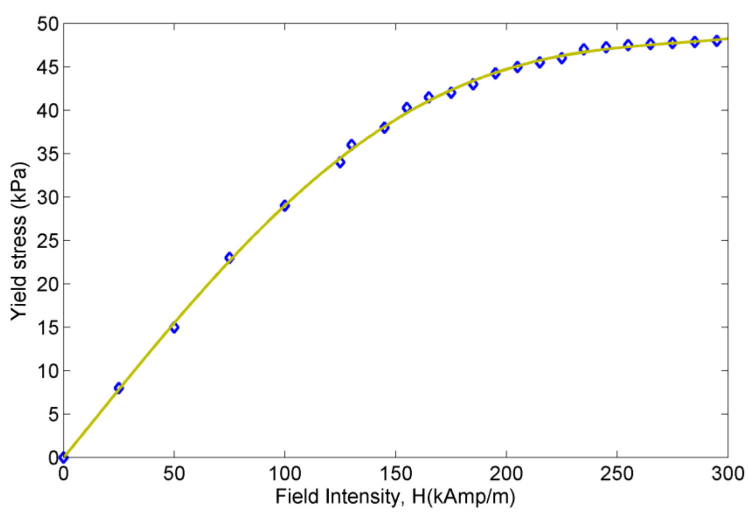


Figure 3. Yield stress versus magnetic field strength

The static body (MRB wall) has the primary function of enclosing the MR fluid inside the brake prototype. FourViton lip seals are placedbeside these ball bearings to avoid a fluid leaking. For measurement needs, the static body is designed to be freely swinging relatives to the rotor. In fact, the static wall is fixed to the load sensor during operations. The static body is supported by the two roller bearings that are mounted in the apparatus frame. On the other hand, the housing is used for attaching the electromagnetic coil as well as its base.

The samples of tested MR fluids in this study are hydro-carbon based MRF-132AD fluid manufactured by Lord Corporation. The properties of MRF-132AD can be seen on Table 2. This fluid has a nearly linear experimental stress rate curve that is well approximated by the Bingham model. The fluid type was chosen mainly due to its higher temperature resistance characteristics. The relationship between the shear yield stress and flux intensity of the MRF-132AD is depicted in Figure 3. This relationship is useful for MRB torque estimation through theoretical analysis.

3. Mathematical Model

The essential magnetic field dependent fluid characteristics of MR fluids can be described by a simple Bingham plastic model [8,15]. By using the constitutive equation for a Bingham plastic fluid, the total shear stress (τ) is stated as follows (Eq. 1),

$$\tau = \tau_H + \mu_p \dot{\gamma} \tag{1}$$

where τ_H is the yield stress due to the applied magnetic field H, μ_p is the constant plastic viscosity which is considered equal to the off state (no magnetic field) viscosity of the fluid, and $\dot{\gamma}$ is the shear strain rate. Here, the plastic viscosity is defined as the slope between the shear stress and shear stress rate, which is the traditional relationship for Newtonian fluids.

Based on Eq. (1) and the given geometrical configuration shown in Figure 2, the braking torque which is caused by the friction on the interfaces between the MR fluid and the solid surface within the MR brake can be written as [4,8];

$$T_b = 2\pi N \int_{r_w}^{r_z} \tau r^2 dr = 2\pi N \int_{r_w}^{r_z} (\tau_H + \mu_p \dot{\gamma}) r^2 dr \tag{2}$$

where$N$ is the number of surfaces of the brake disk in contact with the MR fluid, r is the difference between inner and outer radii of the brake disk. The yield stress and shear strain rate are defined as;

$$\dot{\gamma} = \frac{r\omega}{h} \text{and } \tau_H = kH^\beta \tag{3}$$

where ω is the angular velocity of the rotating disk, h is the thickness of the MR fluid gap, H is the magnetic field intensity, and k and β are constant parameters that approximate the relationship between the magnetic field intensity and the yield stress for the MR fluid. Then, Eq. (2) can be rewritten as;

$$T_b = 2\pi N \int_{r_w}^{r_z} \left(kH^\beta + \mu_p \frac{r\dot{\omega}}{h} \right) r^2 dr \tag{4}$$

The applied magnetic field H can be generated inside the MR brake when current i is presented to the electromagnet coil as,

$$H = \alpha i \tag{5}$$

where α is a proportional gain. By performing the integration in Eq. (3) and substituting Eq. (1), it can be understood that the resulting braking torque is contributed by two torque element. First is torque due to the yield stress induced by the applied magnetic field (T_H) and another is torque due to the friction and viscosity of the MR fluid (T_μ). Both torque elements are expressed as follows [16],

$$T_H = \frac{2\pi}{3} Nk\alpha({r_z}^3 + {r_w}^3)i = T_i i \tag{7}$$
$$T_\mu = \frac{\pi}{2h} N\mu_p({r_z}^4 + {r_w}^4)\dot{\theta} = T_v\dot{\theta} \tag{8}$$

where $\dot{\theta}$ is the rotational speed of the disk. r_w and r_z are the inner and outer radii of the brake disk, respectively. In other way, the total braking torque outputted by the MR brake can be written as follows,

$$T_b = T_H + T_\mu \tag{9}$$

Next, the net or effective MRB dynamic response is derived by using the free body diagram as shown in Figure 4. During rotation, the inertial load will generate a constant loading or falling load as stated by Tan *et al.*[17]. Refer to Figure 3, the T_L term is the loading torque that acts on the output rotor of the MR brake. This loading torque is generated due to the inertial effect of the rotating mass (m) about the radius (r_i) of the load cylinder. The T_L term is mathematically expressed in Eq. (8). Physically, the total output MR braking torque (T_b) is used to overcome the torque generated by the falling load (T_L). This will result in a net torque which is used to decelerate the all inertial loads (J) coupled rigidly to the drum shaft of the MR brake. The falling load (T_L) can be written as,

$$T_L = mgr_1 \tag{8}$$

By observing the free body diagram, the MR effective braking torque is shown in Eq. (9) as follows.

$$T_b - T_L = -J\alpha_{MR} \tag{9}$$

In Eq. (9), the total moment of inertias of the MR brake (J) consists of rotor drum shaft, four bearing inner parts, a sprocket, a pulley and the inertial load. Therefore, the total output inertia (J) of the MR brake is expressed in the Eq. (10).

$$J = J_{rotor} + J_{sprocket} + J_{pulley} + 4J_{bearings} + J_{load} \tag{10}$$

The total moment inertia is approximately about 0.932 kg m^2. By substituting Eqs. (7) and (8) into Eq. (9), the Eq. (11) can be written as follows.

$$mgr_1 - (T_H + T_\mu) = J\ddot{\theta} \tag{11}$$

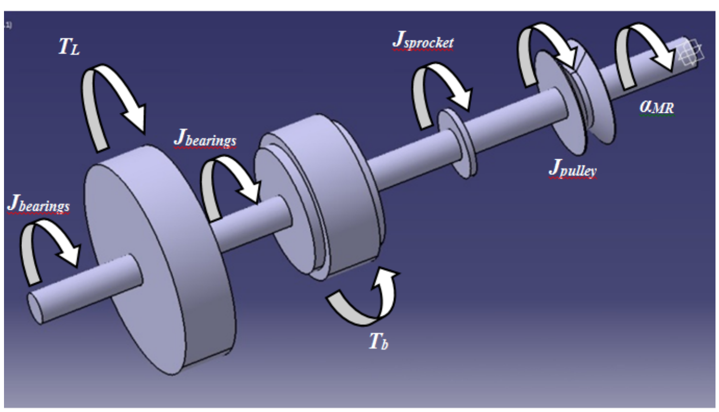


Figure 4. The free body diagram of MRB apparatus

4. Magnetostatic Analysis

To minimize the error calculation of the MR brake, a simulation based finite element model (FEM) is developed using FEMM software (the software can be obtained freely from the internet). The finite element analysis in this study consists of a magnetostatics simulation which gives the pattern of magnetic flux density and field intensity. This is useful for the beginning step of design in which the designer can estimate the magnetic field distribution within the MR brake. Furthermore, there is no record from the previous works that reported about real measurement of flux density within the MRB. This fact indeed one of limitation in MRFs based device design. Therefore, the magnetostatic simulation is very useful at the initial stage of MRB design. The value of flux density obtained from the simulation can then be used for theoretical prediction of the braking torque.

The first step in the finite element modeling is to define the brake geometry. Since the problem is axisymetric, meaning that the geometry, material properties and all loads are all consistent along the tangential direction, only the cross-sectional is modeled [4]. This way, the solution becomes a two-dimensional problem, allowing the use of FEMM plane elements. The method reduces the computational cost of each simulation. On the preliminary stage of simulation, the material assignment should be done carefully. In this case, the magnetic properties of the raw materials used in the MRB have been provided in the FEMM except MRF-132AD properties. The nonlinear magnetic saturation behavior of the MRFs can be referred to the Figure 5.

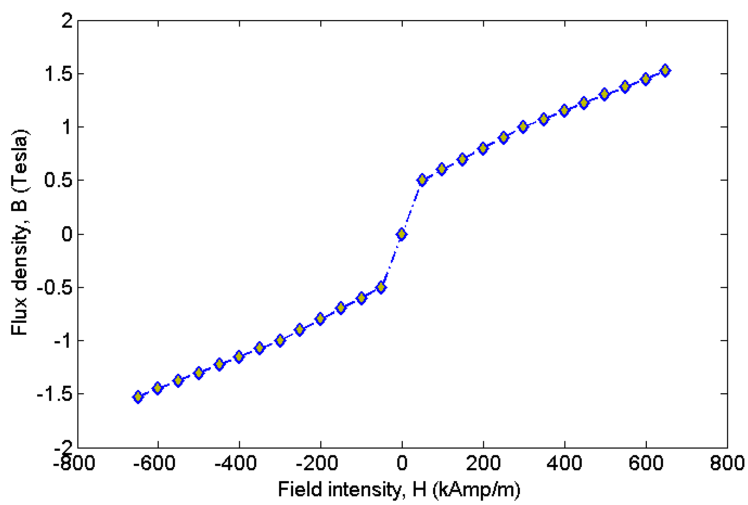


Figure 5. Flux density versus field intensity of MRF-132AD

Figures 6 and 7 present the preliminary result of the finite element simulation.Figure 6 shows the magnetic flux lines distribution in disk-type configuration. The lines represent the distribution of magnetic flux covered area within the MR brake. From the figure, the gap around the drum circumference can be influenced well by the electromagnet coil. The result is simulated based on 1 Ampere of applied electric current to the coil. From the figure, it can be seen that the flux lines become weaker when it is far from the magnet source. The color line shows the different value of the flux density. In that area, the Relatively Moveable Poles (direct-shear modes) of magnetorheological fluid are utilized because of the high concentration of the flux line.

The finite element simulation presents the magnetic flux density in effective braking area. The magnetic flux density is measured in Tesla (T). This parameter will influence the magnetic shear stress within the magnetized area. The more current applied to the coil, the bigger the magnetic flux density is produced. By applying 1 A DC current, the pattern of flux density value in MR fluids region is shown in Figure 7. It can be concluded that the most effective area of the designed MR Brake is in radial side better than annular area.

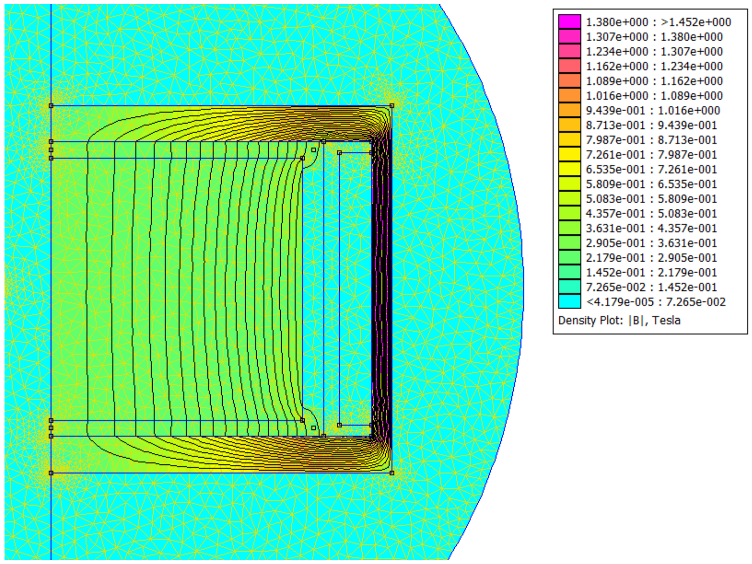


Figure 6. Flux lines distribution at 1 A current applied

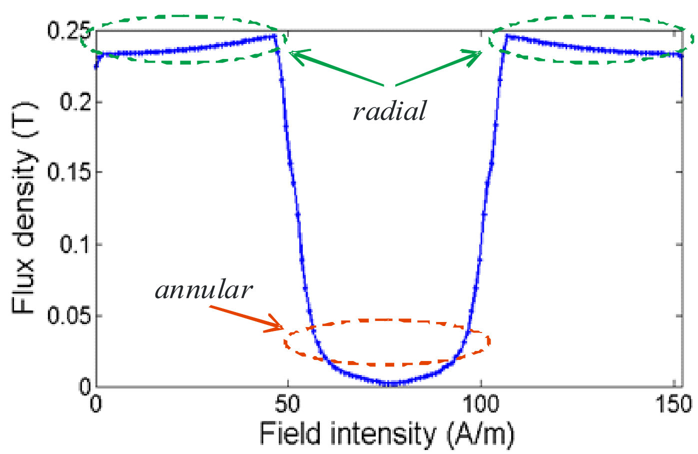


Figure 7. Flux pattern within the MRFs gap

5. Testing Facility

Figure 8 illustrates the MR brake test rig facility. An AC motor is coupled to the input shaft or rotor of the MR brake via an A-type V-belt. The belt is tensioned to drive the rotating system until reaching the desired velocity and is released when the electric current is applied. The housing of this MR brake is coupled to a load cell via an arm which has the length of 238 mm. In this equipment, the load cell is employed for braking torque measurement. A speed sensor that commonly used as ABS speed sensor is utilized for measuring drum rotational speed. The MR brake test rig is equipped with an I/O device for data processing. The Integrated Measurement and Control (IMC) device provides signal processing of the sensory system. These signals are digitally processed and stored in a personal computer using FAMOS control software. IMC device is connected to the personal computer using NetBEUI protocol. A DC power supply manufactured by GW-INSTEK is used for supplying static electric currents to the MR brake electromagnetic coil. The schematic configuration of the MRB apparatus is depicted in Figure 9.

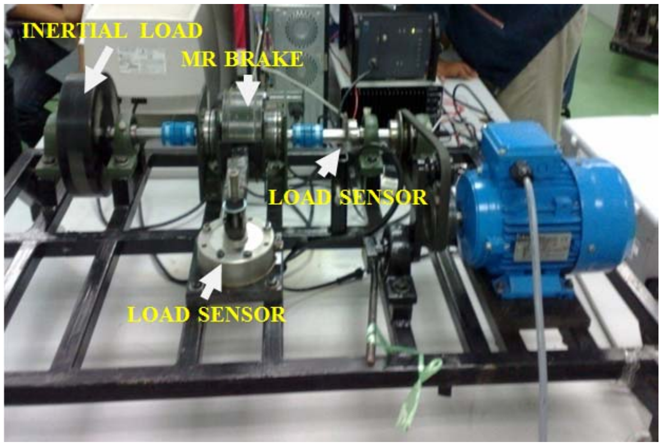


Figure 8. MRB Testing facility

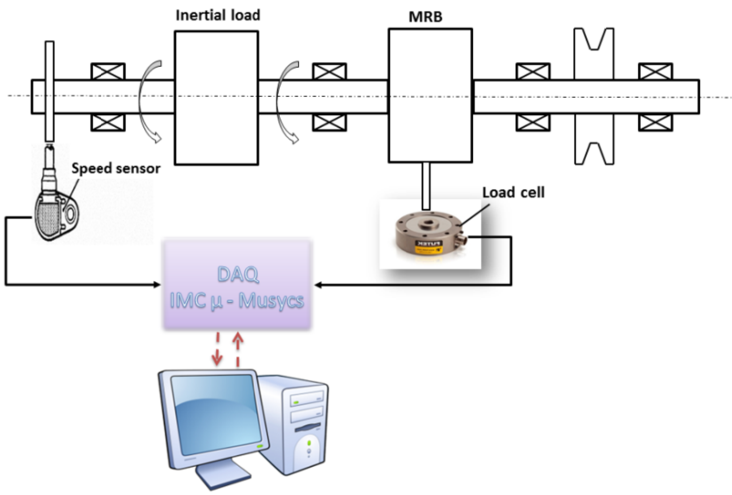


Figure 9. Schematic diagram of the test setup

6. Stopping Time Behavior

Table 3. Simulation parameters

Nomenclature	Value	Description
J	0.932 kgm2	total moment of inertia
N	2	number of surface
μ	0.09 Pa s	MR fluid viscosity η
r_z	0.0415 m	outer radius of disc
r_w	0.04 m	inner radius of disc
r_I	0.1 m	inertial load radius
k	0.269 Pa m/A	electric constant
n	0.00525 m	MR fluid gap
α	12500 m-1	proportional gain

In this section, the simulation and experimental results of the velocity responses of the MRB are presented. The simulation study was performed in MATLAB-Simulink^{TM} based on the governing equations. All parameters from the derived equations have been assigned in the MATLAB as listed in Table 3 below,

Figure 10 shows the sample of simulation results. The x-axis is time variable and the y-axis is rotating speed variable. This result is obtained by applying 0.5 Ampere electric current to the coil. The time needed to stop an inertial load of 10 kg from 2.5 rad/s until fully stop can be seen in that particular figure.

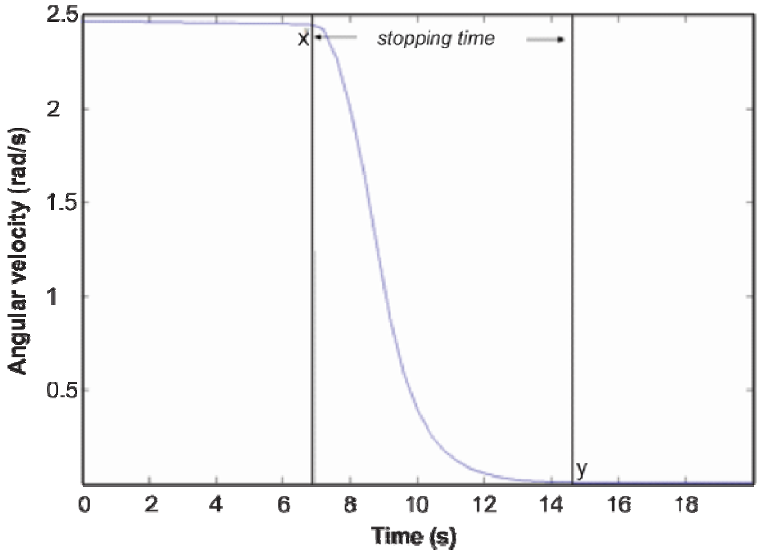


Figure 10. Velocity response resulted from simulation

Using the developed MRB, the experimental investigation was conducted as follows. Initially, the AC-motor drives the pulley attached on the input shaft of MR brake stationary. When the shaft rotates in a constant speed (indicated from speed sensor), a certain current is supplied to the MRB together with releasing the belt tensioner. The applied current will yield the magnetic field across the coil and thus the MR effect is generated. The load cell then measures the produced braking torque, while the speed sensor records the rotating velocity response. All these measured data are then displayedin a PC for further analysis.

Several amount of currents i.e. 0.25, 0.5, 1, 1.5, 2 and 2.5 Amperes were applied to the MR brake coil during experimental work. The testsare performedin room temperature of 25 – 30 ^{o}C. Figure 11 shows the rotating velocity responses under various currents. From the Figure, it can be seen that the magnitude of the time needed for stopping the load decreases proportionally with the increase of the current applied to the MR brake coils.

The simulation results obtained from the governing equation of motion need to be validated with the experimental results. This step is important for the future work. When the model is valid, it is ready to be integrated with a control system that will be proposed for ABS using MR brake. In this paper, the validation is provided in several currents applied namely: 0.5 Amp, 1 Amp, 1.5 Amp and 2 Amp. The results are shown in Figures 12 (a), (b), (c) and (d) respectively. In Figure 12, the closeness between the measured and modeled results indicates that the dynamic model of MR brake can be accepted for further control system implementation.

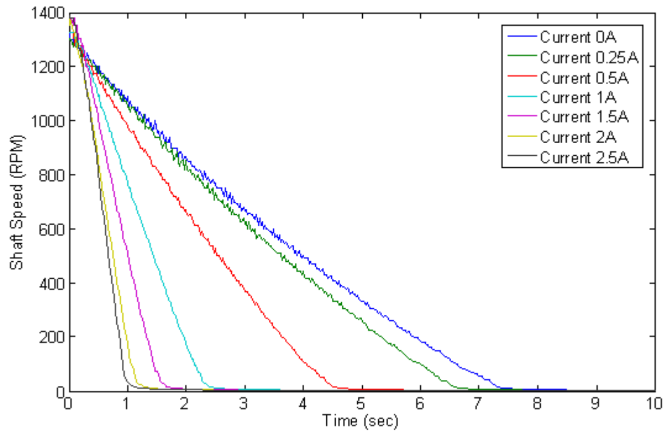
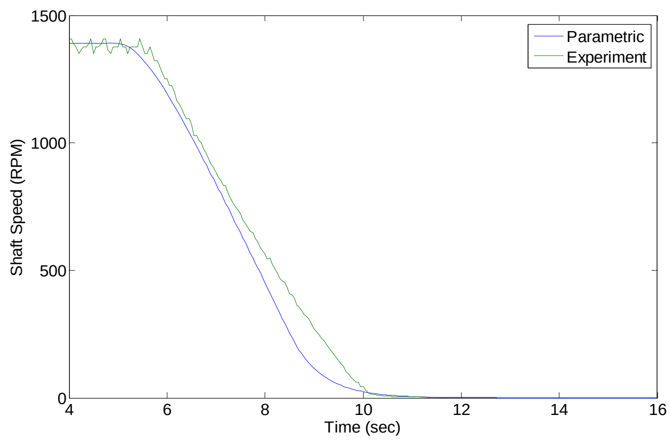
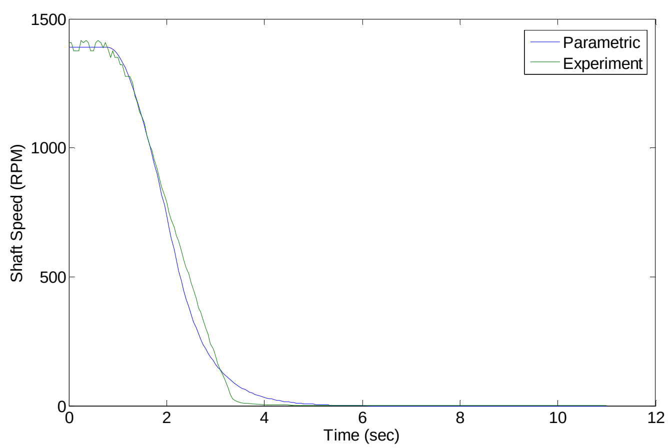


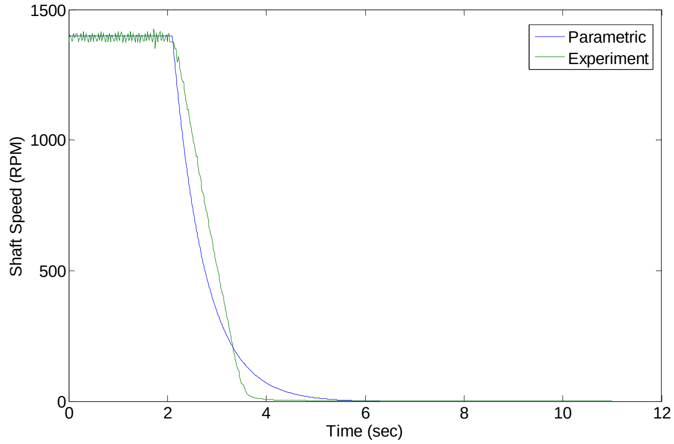
Figure 11. Responses of stopping time in various currents



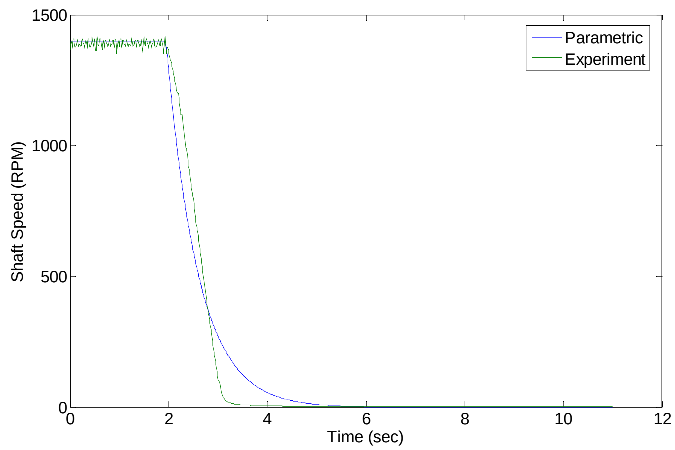
(a). Applied current 0.5 Amp



(b) Applied current 1 Amp



(c) Applied current 1.5 Amp



(d) Applied current 2 Amp

Figure 12. Comparison between simulation and experiment results of stopping time

7. Braking Torque Achievement

In this section, the behavior of the MRB torque responses are studied in order to know capability the of the MRB in generating torque. The torque recorded by the load cell is the total torque response generated by the MRB. The experimental results of retarding torque in time domain are displayed in Figure 13. From the figure, it can be noted that to get the shorter stopping time in constant inertial load which has initial rotating speed, the higher torque should be applied. On the other way, by increasing current applied to the MR brake coils, the bigger braking torque can be obtained. This fact agrees with the velocity response, in which the higher electric current allows the MRB to generate larger MR effective braking torques to decelerate the loads more rapidly. The average braking torque of each current supplied is also calculated. The relation between the average braking torque and current is shown in Figure 14. From the figure, it can be seen that the torque increases gradually as the increasing current.

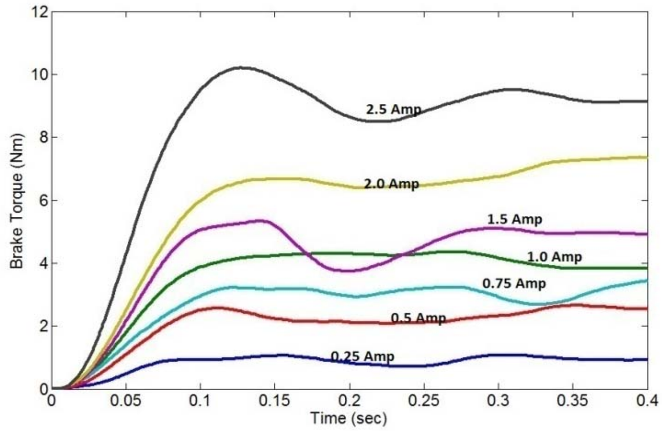


Figure 13. Braking torque in time domain

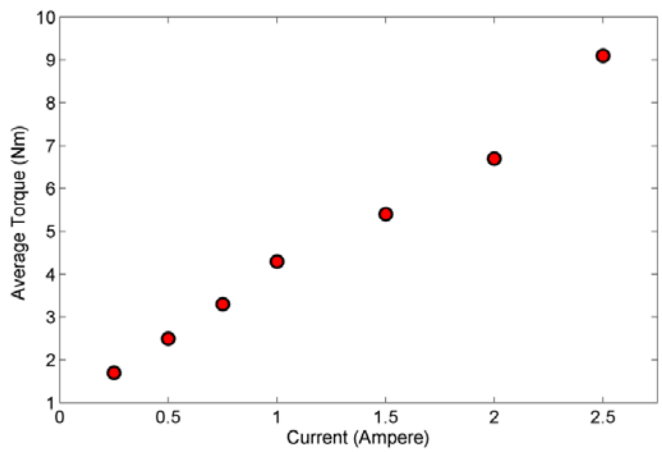


Figure 14. Average braking torque in various applied currents

8. Conclusion

A MRBtest rig whichwas instrumented with several sensors has been developed. The mathematical model of the braking system has also been derived based on the free body diagram. To study the influence of electric current to the magnetic flux, the finite element model simulation was performed using FEMM. From this modeling, the behaviors of magnetic flux lines, intensity and density have been obtained. The MRB behavior was studied in this paper through simulation and experimental works. By using the energized MRB to overcome the loading torques and inertias, the velocity and torque responses of the MRB could be measured by the speed and load sensors respectively. In this work, the simulation results of velocity responses have been compared with the experimental results. The closeness of the results indicated that the mathematical models of braking system were validated. Generally, it can be concluded that the increasing current applied to the MRB coils will shorten the stopping time as the effect of the increasing braking torque.

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