VALIDATION AND EXPERIMENTAL EVALUATION OF MAGNETORHEOLOGICAL BRAKE-BY-WIRE SYSTEM

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ABSTRACT

Magnetorheological brake is one of x-by-wire system which is performing better than conventional brake system. MR brake consists of a rotating disc that is immersed with magnetorheological fluid in an enclosure of an electromagnetic coil. The applied magnetic field will increase the yield strength of the MR fluid where this fluid was used to decrease the speed of the rotating shaft. The purpose of this paper is to develop a mathematical model to represent MR brake with a test rig. The MR brake model is developed based on actual torque characteristic which is coupled with motion of a test rig. Next, the experimental are performed using MR brake test rig and obtained three output responses known as angular velocity response,torque and load displacement. Furthermore, the MR brake was subjected to various loads and current. Finally, the simulation results of MR brake model are verified with experimental results.

KEYWORDS*: Magnetorheological brake, Magnetorheological fluid, Brake-by-wire, Angular velocity, Torque, Displacement*

1.0 INTRODUCTION

X-By-Wire is one of the system that has potential to improve performance by minimized the number of part used in the system. Usually, by wire systems have been employed in several segments such as steering system, suspension system, braking system and medical equipments (Poynor & Reinholtz, 2001; Wang & Gordaninejad, 2003; Diep et al., 2006; Gudmundsson et al. 2010; Karakoc et al., 2008). MR brake has introduced as an actuator of brake-by-wire system in automotive industries. MR brake employs with MR fluid where this fluid solidifies once applied with magnetic field. This fluid is also

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known as smart fluid where it solidifies by increasing the strength of magnetic. Once the fluid is free from magnetic field, it represents as a Newtonian fluid behaviour. The MR brake consists of a rotating disc immersed with Magnetorheological Fluid (MR fluid) in an enclosure of an electromagnetic coil. MR fluid is developed using micron sized suspensions measured between 20-50 microns (Carlson, 2001; Bossis et al., 2002). The rheological behaviour of MR fluid is similar with the carrier fluid when there is no external field is occurred in the fluid (Jolly et al., 1998; Park et al., 2006).

The application of a MR fluid in braking system is related with current research work. Li and Du (2003) presented the design and experimental evaluation of a Magnetorheological brake and introduced an amplifying factor to evaluate brake performance. Meanwhile, Park et al. (2008) presented a design optimization procedure using simulated annealing combined with finite element simulations involving magnetostatic, fluid flow and heat transfer analysis. MR fluid selection for MR brake application, such as magnetic circuit design and torque requirements for automotive application was also studied. Karakoc et al. (2008) focussed on the investigation of practical MR brake design criteria such as material selection, sealing, working surface area, viscous torque generation and MR fluid selection for basic automotive braking system. Additionally, Tan et al. (2007) studies braking response of inertia/load by using an electro-rheological (ER) brake for ER-robotic application in term of ER braking velocity response in order to halt the robot arm rapidly. In 2009, Nam and Ahn (2009) proposed the new structure of MR brake with the waveform boundary of rotary disk that generated more resistance torque compare to the conventional MR brake. Furthermore, the MR brake system had been implemented to other application such as joystick and prosthetic knee (Li et al., 2007; Gudmundsson et al., 2010).

This paper decribes briefly the modeling and validation of MR brake-by-wire. The experimental evaluation is also presented for practical application of brake-by-wire system.

2.0 EXPERIMENTAL APPARATUS

Figure 1 shows the testing equipment used in the experiment with the load of 50 N and 100 N. The inertia load will be attached to the load shaft and coupled with brake shaft to generate a constant or falling load that resultant the net torque produce by MR brake. The function of AC electric motor is to drive the MR brake shaft to desired velocity where it is coupled to the input shaft/rotor of MR brake via pulley and A-type V-belt. The speed from the motor is transmitted to the MR brake shaft using belt tensional and well fitted beside the electric motor. The pulley shaft is connected to the MR brake shaft using jaw coupling and same concept also was applied at the load shaft. The pulley shaft and load shaft used a pillow block bearing to support the rotating shaft where the inner bearing will allow the shaft to rotate in free direction.

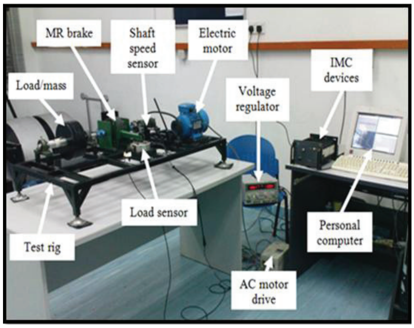


Figure 1. Mechanical assembly of the MR brake test rig

The MR brake housing is coupled to a load cell via an arm of length of 238 mm. In this equipment, the load cell is employed to measure the braking torque. The load cell was calibrated at 1 V: 53.4 Nm and the maximum torque measured using this sensor is 534 Nm. Next, the load cell is connected to the bridge amplifier which functions as the signal conditioning. Meanwhile, the rotational speed of the MR brake shaft was measured by using an ABS speed sensor. The MR brake test rig is equipped with an I/O device for data processing. Next, the Integrated Measurement and Control (IMC) device provides signal processing of the sensory system. The IMC device that is only capable to receive analogue voltage signal. Then, the signals are digitally processed and stored in a personal computer using FAMOS control software. IMC device is connected to a personal computer using NetBEUI protocol. A DC power supply manufactured by GWINSTEK is used to supply electric currents to the MR brake electromagnetic coil. All the measured data are displayed in Personal Computer (PC) for the further analysis.

3.0 MATHEMATICAL MODEL

The characteristics of MR fluid can be described by using a simple Bingham plastic model (Philips, 1969). The constitutive equation for a Bingham plastic fluid where the total shear stress (τ) is written as:

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

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 non-field viscosity of the fluid, and $\dot{\gamma}$ is the shear strain rate. Based on the Eq. (1), the braking torque generated by the friction of the interface between static and moving parts in the MR fluid inside the MR brake can be written as equations (Park et al., 2006; Karakoc et al., 2008):

$$T_b = 2\pi N \int_{r_i}^{r_o} (kH^\beta + \mu_p \frac{r w_s}{h}) r^2 dr \tag{2}$$

where r is the radius of the disk, w_s is the angular velocity of the rotating disk, h is the thickness of the MR fluid gap between rotor and enclosure, H is the magnetic field intensity corresponding with k and β. The values of k and β are constant by considering the relationship between the magnetic field intensity and the yield stress of the MR fluid.

An integration of Eq. (2) will determine the two types of components of braking torque which are torque generated due to applied magnetic fields (T_H) and torque due to friction and viscosity of the fluids (T_μ). Both torque elements are expressed as follows (Park et al., 2006; Karakoc et al., 2008).

$$T_\mu = \frac{\pi}{2h} N \mu_p (r_o^4 - r_i^4) w_s \tag{3}$$

$$T_H = \frac{2\pi}{3} N k \alpha (r_o^3 - r_i^3) i \tag{4}$$

Therefore, the total braking torque produced by MR brake can be written as follow

$$T_b = T_\mu + T_H \tag{5}$$

An effective MR brake torque generated when applied current to the magnetic coils. This will decelerate the dynamics of all inertia (J_{all}) that coupled rigidly to the MR brake shaft. The (T_{input}) term is the torque motor (T_m) is combined with loading torque (T_L) that is coupled rigidly on the MR brake shaft. The loading torque is generated based on the weight of the load (N) and the effective radius (r_L) of the load. Then, the mathematical expression in terms of (T_{input}) can be written in Eq. (6) as:

$$T_{input} = T_m + T_L = (P_m / w_m) + mgr_L \tag{6}$$

where P_m = power of electric motor and W_m = angular speed of electric motor pulley. Then, the equation motion of MR brake with a test rig is derived and stated in Eq. (7) as follow:

$$T_{input} - T_b - T_c = J_{all}\alpha_s \tag{7}$$

where T_c = viscous damping of bearing and α_s = angular acceleration of MR brake shaft. Then, the total inertia of MR brake (J_{all}) consists of rotor, shaft, four bearing inner parts, a sprocket, a pulley and the load can be written in Eq. (8).

$$J_{all} = J_{disk} + J_{sprocket} + J_{pulley} + 4J_{bearing} + J_{Load} + J_{shaft} \tag{8}$$

The total inertia is approximately about 0.25 Nm^2 for 50 N, 0.45 Nm^2 for 100 N and 0.65 Nm^2 for 150 N respectively. By substituting Eq. (6) into Eq. (7), the Eq. (9) can be written as follow:

$$((P_m / w_m) + mgr_L) - (T_\mu + T_H) - 4cw = J_{all}\alpha_s \tag{9}$$

4.0 MODEL VALIDATION

The MR braking response was tested by using a constant load of 150 N in order to energized MR brake when applied current to the brake coils. The response of the MR brake was investigated at four constant applied currents that are 1 A, 2 A, 3 A, and 4 A. These three responses that were obtained from the experiments were the shaft angular velocity, torque and load displacement. As shown in Figure 2. Based on the Figure 2(a), shaft angular velocity decreases very fast when applied current is increase. This is due to the maximum braking torque generated by MR brake that is shown in Figure 2(b). The MR brake torque increase proportionally when the current increases. Furthermore, the load displacement takes longer displacement when lower current is applied in this system which is shown in Figure 2(c). Basically, the load displacement responses are determined by integrating the MR brake shaft angular velocity.

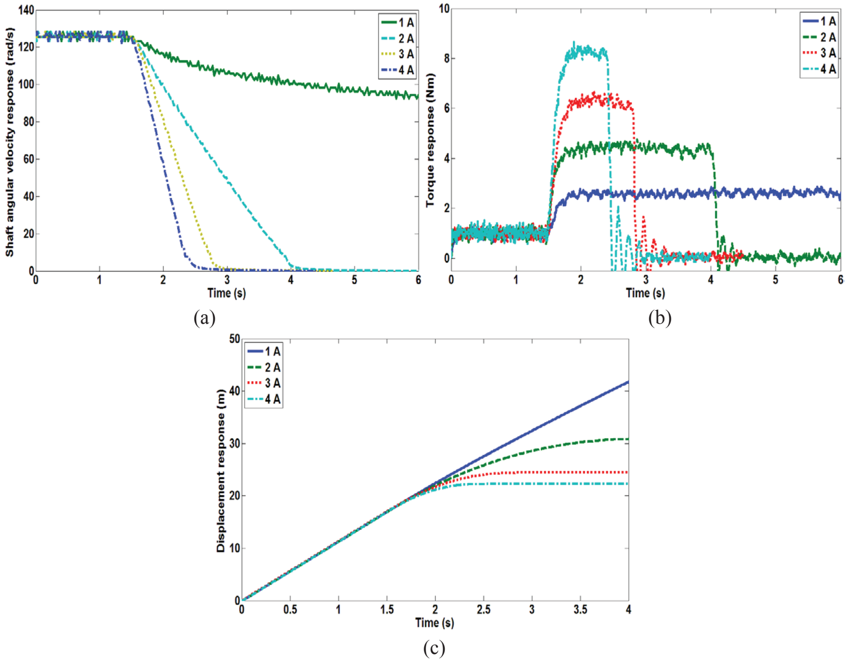


Figure 2. Measured responses at four constant currents; (a) Shaft angular velocity response, (b) Torque response and (c) Load displacement response

In this model validation, a constant load of 150 N is considered at various current. The current are varies from 1 A to 4 A with increment of 1 A. The triggering time for the electric current in modelling is 1.5 s which is same as experimental. Figure 3, Figure 4 and Figure 5 show the model validation result at various current. The simulation model was developed using MATLAB based on the equations of MR brake with test rig in the previous section and the parameters given in Table 1.

Table 1. MR brake parameters

Parameter	Value
Outer radius of disc, r_z	0.04 m
Inner radius of disc, r_i	0.01 m
Number of surfaces, N	2
MR fluid gap, h	0.0025 m

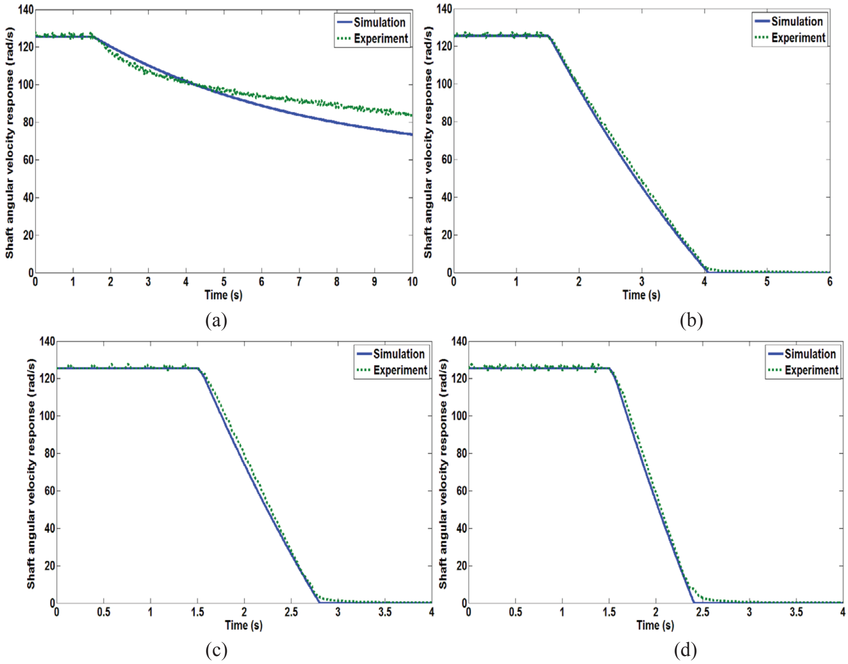
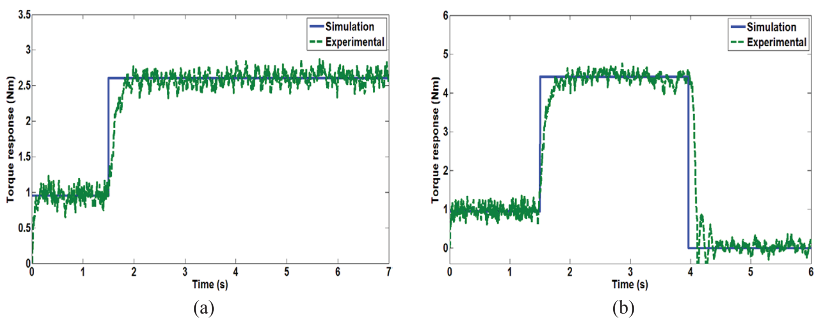


Figure 3. Comparison between model and experimental at several applied current for shaft angular velocity response; (a) 1 A, (b) 2 A, (c) 3 A and (d) 4 A

The model and experimental results are compared and it shows a close relationship between both results. This response indicates that the model of MR brake actuator with a test rig is valid. However, the response time of MR brake torque to reach constant steady-state has a delay time which is shown in Figure 4. This is due to the delay response of material and sensory system. Nevertheless, the trend is similar for experimental and model.



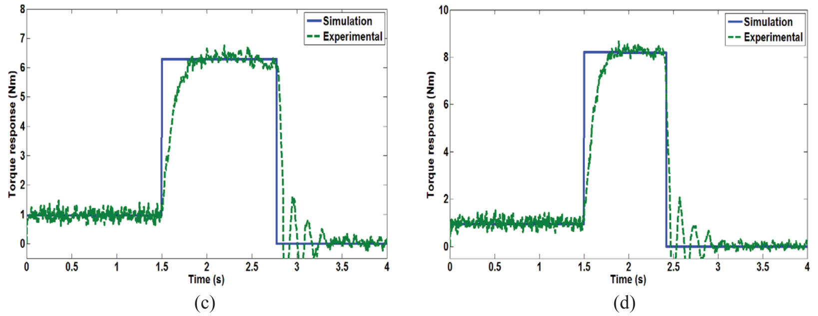


Figure 4. Characteristic of torque response between model and experimental; (a) 1 A, (b) 2 A, (c) 3 A and (d) 4 A

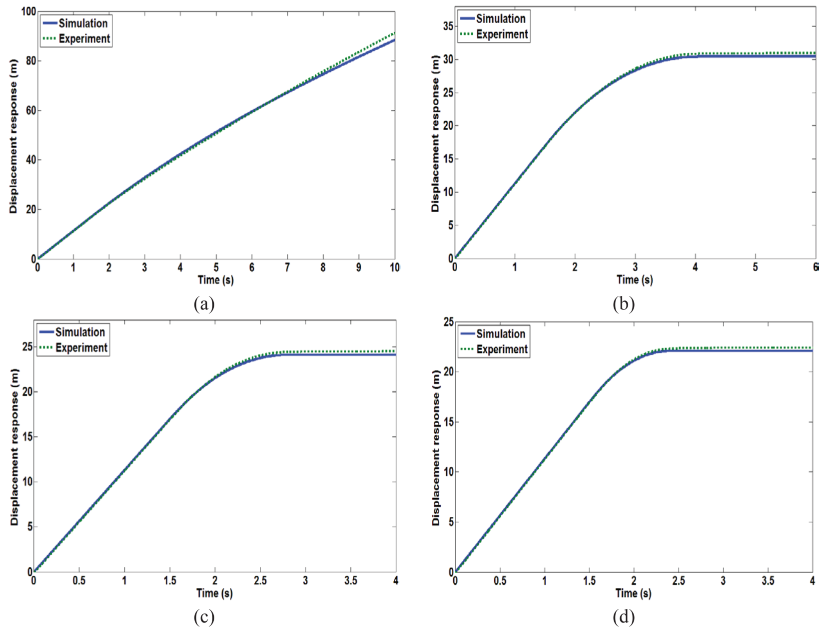


Figure 5. Comparison between model and experimental of load displacement response at several applied current; (a) 1 A, (b) 2 A, (c) 3 A and (d) 4 A

5.0 EXPERIMENTAL EVALUATION

In this section, the MR braking responses were experimentally evaluated at various loads and current. The behaviour of MR braking response in term of shaft angular velocity response, torque response and load displacement response are been compared. By observing the experimental procedure as discussed in previous section, various input parameter of MR brake is considered. The trend behaviour of the MR brake is influenced by load and current is shown in the results. The MR brake braking behaviour was divided into three sections. First section is described about shaft angular velocity response, second section is about torque response and last section is load displacement response.

5.1 Angular velocity

In this section, the main objective is to determine the time response or settling time of all inertia to decelerate when current is given. The current triggering time is 1.5 s for all cases. There are three load are tested in the MR braking process that is 50 N, 100 N and 150 N at various current. Figure 6 shows the shaft angular velocity response versus time at various current. The shaft angular velocity rotates equally for all masses at desired speed. The applied current at 1.5 s used to energize MR brake to produce braking torque and decelerates all inertia. At 1 A, 50 N of load decelerates slower and shows the trend of inertia is falling down until stationary. When increasing the load to 100 N and 150 N, the shaft angular velocity takes time over than 6 s to become static because of the fluids behavior turned to saturate condition based on applied magnetic. When the applied current is increased from 2 A until 4 A, the response time of the shaft angular velocity is decreased until stationary below 5 s. However, the effectiveness of MR brake reduces when the load is increased. This response is shown in Table 2 where the MR brake is effective at 50 N which is only takes 0.6 s to halt compared 100 N and 150 N takes 0.8 s and 1.06 s to halt the rotation of all the inertia.

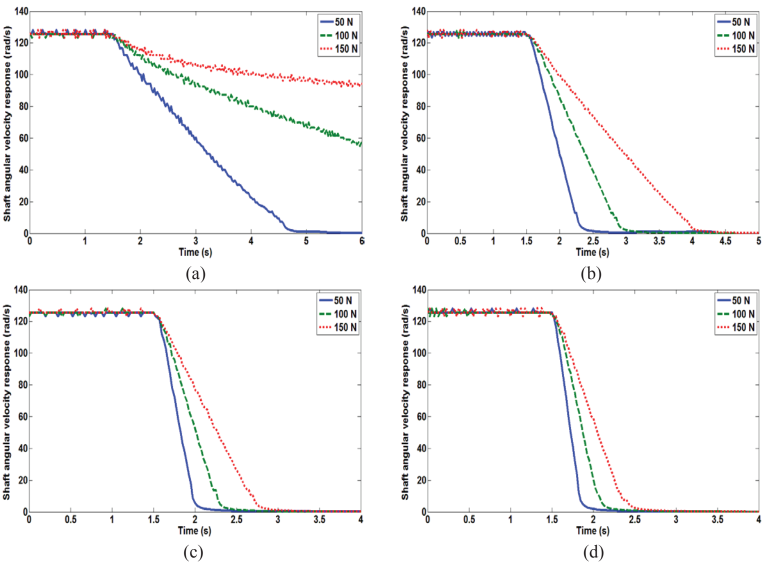


Figure 6. Shaft angular velocity response versus time at various current; (a) 1 A, (b) 2 A, (c) 3 A and (d) 4 A

Table 2. Stopping time

Load (N)	Current (A)			
	1	2	3	4
50	3.6 s	1.01 s	0.72 s	0.6 s
100	> 6 s	1.52 s	1.05 s	0.8 s
150	> 6 s	2.6 s	1.31 s	1.06 s

5.2 Torque

The effective of MR brake torque at the higher applied current. Figure 7 (a), Figure 7(b), Figure 7(c) and Figure 7(d) show the behavior of MR brake torque at various input applied current. Initially, the measured braking torque using load cell was numerically processes using Butterworth filter in order to reduce noise. The applied current is start from 1 A to 4 A with step increment 1 A is considered. It can be stated that the applied constant current to the MR brake which will overcome the brake torque 10 is also constant. The MR brake can produce more torque by increasing the current supplied. However, the MR brake cannot generated maximum torque at lower current and it shows the decreasing trend of torque response to halt all the inertia at 1 A for 50 N of load. Also, the time delay of MR fluid to be fully solidifies is 0.3 s using experimental method which is shown in Figure 7. The time delay is increases which +0.05 s when load increases with respect of lower load condition.

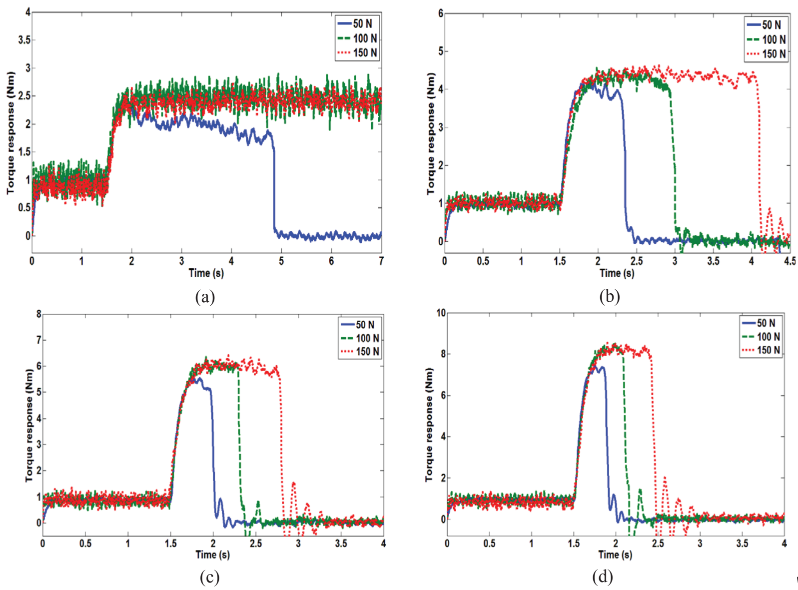


Figure 7. Torque response versus time at various current; (a) 1 A, (b) 2 A, (c) 3 A and (d) 4 A

Figure 7(a) and Figure 7(b) shows the longer steady-state of brake torque response using heavy load. However, the braking torque responses reduce when the load increases. Meanwhile, the torque is remains constant when the load increases from 100 N to 150 N where the maximum torque can be obtained at various current. From the Figure 7, the average of MR brake torque at various current is obtained. The MR brake net torque at 1 A, 2 A, 3 A and 4 A are 2.54 Nm, 4.46 Nm, 6.31 Nm and 8.1 Nm. Respectively this can be conclude that the maximum torque can be produced by MR brake is 8.1 Nm at 4 A because of the MR fluid had reaching the saturation point. Next section discussed about load displacement response at various load and current.

5.3 Load displacement

The aim of this section is to obtain the load displacement responses when current is applied to the MR brake. In this section, the load displacement response for each load is numerically processed by integrating the shaft velocity response. The shaft rotational speed is converted to revolution per second which is divided by 60 s and multiplies with 11 circumference of the load. Then, the load displacement response was obtained after integrations the shaft velocity response.

Figure 8 shows the load displacement response at various load and current. The applied current to the MR brake will decelerates the velocity of the load that coupled rigidly to the MR brake shaft. This will overcome the load displacement response which is reduced significantly based on applied current. At lower current, the load displacement response takes longer time to reach the constant steady-state displacement that can be seen in Figure 8(a) and Figure 8(b). The constant steady-state displacement means the load at zero velocity is constant at certain displacement. The fast response of the load to reach the steady-state displacement at the higher current that can be seen in Figure 8(c) and Figure 8(d). The different distance of load displacement response was captured and shown in Table 3. The smaller steady-state displacement at 4 A current which is at 50 N only takes 2.3 m meanwhile 100 N is 4.2 m and 150 N is 5.6 m.

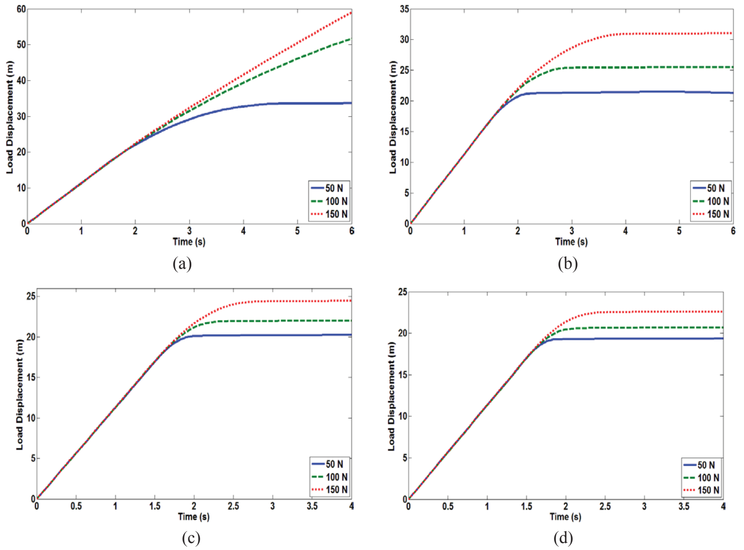


Figure 8. Load displacement response versus time at various current; (a) 1 A, b) 2 A, c) 3 A and d) 4 A

Table 3. Load displacement response

	Current (A)			
Load (N)	**1**	**2**	**3**	**4**
50	16.6 m	4.3 m	3.2 m	2.3 m
100	> 50 m	8.4 m	4.9 m	4.2 m
150	> 60 m	13.9 m	7.3 m	5.6 m

6.0 CONCLUSION

The model validation of MR braking response in term of shaft angular velocity responses, torque responses and load displacement responses was obtained and discussed in this paper. The MR brake with a test rig was developed which is contains some instrumentation sensors such as ABS speed sensor and load sensor. The MR brake model that been simulated by using MATLAB software to obtained the model result are validated with the experimental result. The results validation shows good agreement between both graphs in this study. Furthermore, the performance evaluation of MR braking response at various load and current are also presented and discussed. When increased the load, it will caused the MR brake takes longer settling time and the load displacement response becomes longer to constant steady-state displacement that been discussed in performance evaluation at various load and current. The heavier load will reduce the effectiveness of MR brake.

7.0 ACKNOWLEDGEMENT

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