Heating of a High-torque Magnetorheological Fluid Limited Slip Differential Clutch

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ABSTRACT: Theoretical and experimental studies on heating of a high-torque, multi-plate magnetorheological (MR) fluid limited slip differential (LSD) clutch are presented. A lumped parameter system approach is assumed for theoretical heating analysis. The experimental study is conducted to examine the temperature rise of the clutch. Electric power input and slippage effects are investigated both theoretically and experimentally. The effect of temperature increase on the torque performance of the clutch is also examined. The results show that the transferred torque is insensitive to clutch temperature increase. For all cases, theoretical and experimental results are in good agreement.

Key Words: magnetorheological (MR) fluid, controllable clutch, limited slip differential clutch, heating, temperature.

INTRODUCTION

In recent years, magnetorheological (MR) and electrorheological (ER) fluids have received increased attention for use in various devices, such as shock absorbers, brakes, clutches, and engine mounts (Carlson and Weiss, 1994). Although there are numerous designs in controllable ER/MR fluid devices, only limited analytical and experimental studies have been conducted on heat transfer analysis of such devices. Mostly, MR and ER fluid shock absorbers were analyzed since the performance of such devices is strongly affected by the heating of the fluid. The temperature rise in MR or ER fluid shock absorbers reduces fluid viscosity, which affects the dynamic force range.

Makris et al. (1997) theoretically studied viscous heating of viscous and ER fluid dampers. Closed-form solutions for temperature rise in various types of dampers are presented and it is stated that the temperature rise of controllable fluid dampers can be estimated by combining the results for viscous and hysteretic dampers. Breese and Gordaninejad (1999) and Gordaninejad and Breese (1999) presented experimental and theoretical studies on heat generation and dissipation of controllable MR fluid shock absorbers. Various types of MR fluid dampers were tested under sinusoidal and random excitation. The damping coefficients were experimentally determined as a function of temperature and utilized in the theoretical model. A non-dimensional form of the governing equation was also developed.

Dogruoz et al. (2000, 2001, 2003) performed theoretical and experimental analysis for enhancing heat transfer from MR fluid dampers using fins. A lumped system analysis is performed to predict the temperature rise of the damper. Two prototype dampers, one finned and one unfinned, were designed and constructed. The installation of finned housing enhanced the convective heat transfer and consequently mitigated the decrease in damping force.

The literature on analytical and experimental heat transfer analysis on ER clutches is limited to Smyth et al. (1994), Oravsky (1998, 2000a,b, 2002), and Ellam et al. (2002). Smyth et al. developed a heat transfer model for a cylindrical ER clutch for different operational modes. The effects of gap thickness, rotor diameter, and operation speed on the steady state operating temperature of the ER fluid are examined. Oravsky (1998, 2000a,b) developed dimensional models for radial and cylindrical ER fluid clutches. Both conductive and convective heat transfer are taken into account in the models. The flow of ER fluid in gaps was also modeled. As a continuation to a more complicated model (Oravsky, 2000a), a simpler and universal non-dimensional mechanical and heat transfer model is developed by Oravsky (2000b). Ellam et al. (2002) and Oravsky (2002) also examined the rate of heat transfer from a radial electrostructured fluid clutch plate surface both theoretically and experimentally. In their study the upper plate of the clutch is driven and the lower plate is

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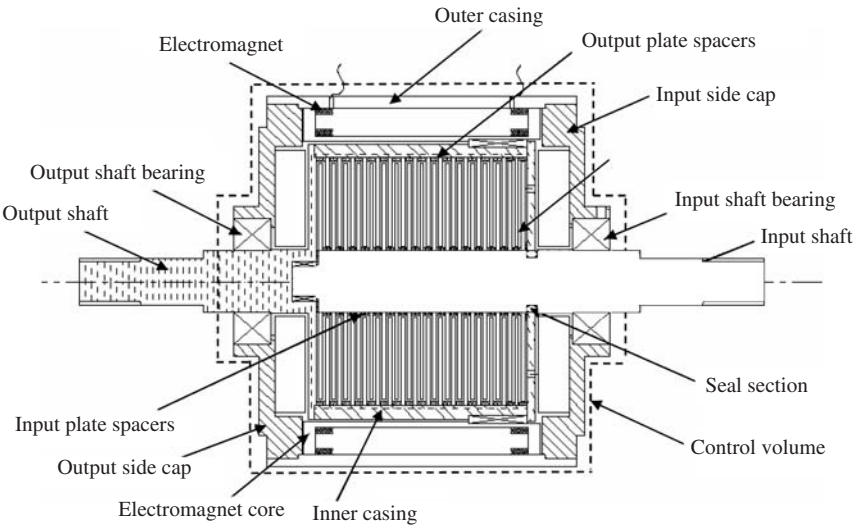

Figure 1. *Cross-sectional view of multi-plate MR fluid LSD clutch.*

stationary and thermally insulated. They developed a computational fluid dynamics (CFD) model for flow and temperature profiles. The electrostructured fluid was considered as a Newtonian fluid due to complexities in modeling heat transfer in non-Newtonian fluids. It was demonstrated that the heat transfer rate could be predicted effectively.

The present study focuses on the heating analysis of a high-torque, multi-plate MR fluid clutch for a limited slip differential clutch (LSD) application. A LSD clutch is a device that regulates the rotational speed of driven shafts of a vehicle in uneven road conditions. The design of the clutch is presented in detail by Kavlicoglu et al. (2003). Effects of different phenomena on heating of the clutch are examined theoretically and experimentally. The effect of electrical power input and slippage are evaluated separately and also a combined analysis is conducted. A theoretical model is developed using a lumped system approach. The transmitted torque is monitored as the temperature of the clutch is increased.

A 43-plate MR fluid clutch is designed and developed for this study. The maximum torque capacity of this clutch is 244 Nm for an input electric current of 3 A (Kavlicoglu et al., 2003). The clutch has an overall length of 152.4 mm and a diameter of 152.4 mm. A schematic of the clutch is given in Figure 1. MR fluid of 0.5 mm thickness is located between each set of input and output plates.

THEORETICAL ANALYSIS

A lumped system analysis is performed to obtain the temperature of the MR fluid clutch as a function of time. A lumped system approach can be justified if Biot (Bi) number is <0.1 (Incropera and DeWitt, 1996). Biot number is defined as:

$$\mathrm{Bi} = \frac{hL_c}{k} \tag{1}$$

where, h is the convective heat transfer coefficient, k is the effective thermal conductivity of the clutch, and L_c is the characteristic length of the clutch, $L_c = V/A_s$, where V is the volume of the system and A_s is the system's surface area. This assumption implies that the temperature gradients within the clutch boundary are negligible (Incropera and DeWitt, 1996).

For the multi-plate MR fluid clutch a control volume is selected, as shown in Figure 1. Since the materials used in the clutch are mainly steel, the overall thermal conductivity of the clutch is selected as 37.7 W/mK (Incropera and Dewitt, 1996). The lowest k value was selected from the provided thermal property tables for steel, which results in a more conservative Biot number calculation. The experiments were conducted with no forced flow of air around the clutch. Therefore, a nominal value of 28 $\mathrm{W/m^2K}$ was selected for the natural convective heat transfer coefficient. The Biot number was determined to be 0.011 for the values given in Table 1. This validates the lumped parameter approach. For the control volume shown in Figure 1, the energy balance equation can be written in the rate form as:

$$\dot{Q} - \dot{W} = \frac{\mathrm{d}U}{\mathrm{d}t} \tag{2}$$

where $\dot{Q}$ is the rate of heat transfer, $\dot{W}$ is the rate of work, and $\mathrm{d}U/\mathrm{d}t$ is the rate of internal energy change of the system. The constant specific heat values at room temperature are assumed for all components of the MR fluid clutch. With this assumption, the internal energy

Table 1. Physical properties and dimensions.

Property	Value
Convective heat transfer coefficient, h	$28\ \frac{W}{m^2K}$
Radius of the control volume, r_{cv}	76.2 mm
Length of the control volume, L_{cv}	152.4 mm
Surface area of the control volume, A_s	$0.109\ m^2$
Volume of the control volume, $V = \pi r_{cv}^2 L_{cv}$	$2.78 \times 10^{-3}\ m^3$
Characteristic length of the control volume, $L_c = \frac{V}{A_s}$	25.4 mm
Resistance of the electromagnet, R	34.1 Ω
Specific heat of steel, C_{Psteel}[a]	$0.444\ \frac{kJ}{kg^\circ C}$
Specific heat of stainless steel, C_{Pss}[a]	$0.477\ \frac{kJ}{kg^\circ C}$
Specific heat of MR fluid, C_{Pmrf}[b]	$0.80\ \frac{kJ}{kg^\circ C}$
Thermal conductivity of steel, k_{steel}[a]	$37.7\ \frac{W}{m^\circ C}$
Thermal conductivity of stainless steel, k_{ss}[a]	$14.9\ \frac{W}{m^\circ C}$
Thermal conductivity of MR fluid, k_{mrf}[b]	$0.25–1.06\ \frac{W}{m^\circ C}$
Total internal heat capacity, $\sum_n mc_p$	$9450\ \frac{Joule}{^\circ C}$

[a]Incropera and DeWitt (1996).
[b]Lord Corporation.

rate can be written as:

$$\frac{dU}{dt} = \sum_n mc_p \frac{d\theta}{dt} \tag{3}$$

where n is the number of elements that are affecting the internal energy of the clutch, $\sum_n mc_p$ is the total internal heat capacity, and $d\theta/dt$ is the rate of change of temperature of the clutch.

Since heat transfer is only from the outer casing to the surroundings, only convective heat transfer is considered. The conduction through both shafts are neglected. Therefore, the heat transfer rate equation can be written as:

$$\dot{Q} = -hA_s(\theta(t) - \theta_{amb}) \tag{4}$$

where $(\theta(t) - \theta_{amb})$ is the temperature difference between the clutch and the surroundings. The surrounding temperature is assumed to be constant at θ_{amb}. A_s is the surface area where convective heat transfer occurs.

The power for the MR fluid clutch for the given control volume is the difference between P_{in} and P_{out}. The input power, P_{in} is produced by the motor and the input electrical power to the electromagnet. Therefore:

$$P = P_{in} - P_{out}. \tag{5}$$

$$P = -(P_{motor} + P_{electrical}) + T_{out}\omega_{out}. \tag{6}$$

Electric power changes with the input current and is defined as:

$$P_{electrical} = I^2 R \tag{7}$$

where R is the resistance of the electromagnet coil and I is the supplied input current.

Substituting the heat transfer rate, $\dot{Q}$, the power, P, and the internal energy change of the system, dU/dt into Equation (2), the energy balance can be written as:

$$[-hA_s(\theta(t) - \theta_{amb})] - [-(P_{motor} + P_{electrical}) + T_{out}\omega_{out}] = \sum_n mc_p \frac{d\theta(t)}{dt}. \tag{8}$$

Rearranging the terms in Equation (8), and defining two constants X_1 and X_2, Equation (8) can be written as:

$$\frac{d\theta(t)}{dt} + X_1\theta(t) = X_2 \tag{9}$$

where:

$$X_1 = \frac{hA_s}{\sum_n mc_p}. \tag{10}$$

$$X_2 = \frac{P_{motor} + P_{electrical} - T_{out}\omega_{out} + hA_s\theta_{amb}}{\sum_n mc_p}. \tag{11}$$

For a given input electrical current both X_1 and X_2 are constant. Therefore, the rate of change of temperature is governed by a linear non-homogeneous constant coefficient differential equation. For $t = 0$, the initial condition is $\theta(t=0) = \theta_{amb}$, therefore, the temperature distribution can be written as:

$$\theta(t) = \left(\theta_{amb} - \frac{X_2}{X_1}\right)e^{-X_1 t} + \frac{X_2}{X_1}. \tag{12}$$

For theoretical temperature analysis three different cases are considered. The first case is the temperature increase from the electrical heating due to the power input to the electromagnetic solenoid only ($P_{motor} = 0$, $T_{out} = 0$). The second case is temperature increase due to slippage of the clutch plates only ($P_{electrical} = 0$), and the third case considers the simultaneous effect of slippage and electrical power input to the clutch.

EXPERIMENTAL STUDY

The clutch heating experiments are performed to examine the increase in the temperature of the MR fluid clutch under different conditions. In these tests, two K-type thermocouple modules are used. K-type thermocouples are best suited for measurements in the 0–480° (C) range. The positive conductor is made of chromel, and the negative conductor is made of alumel. The first thermocouple is attached to the outer casing of the clutch to obtain the temperature change on the radial direction. Another thermocouple is attached on the side cap of the clutch for the temperature change in the axial direction. Temperature increase in both locations is recorded simultaneously to determine if the temperature

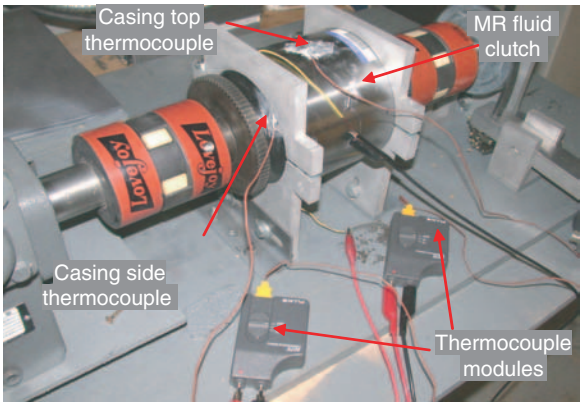

Figure 2. *Experimental setup and the locations of the two thermocouples.*

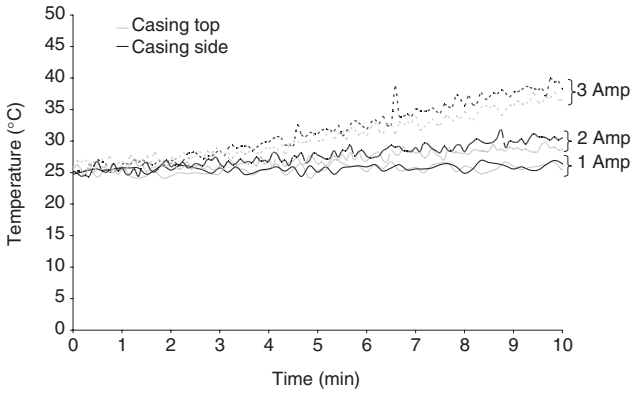

Figure 3. *Heating test results for 1 A input current for multi-plate MR fluid clutch.*

increase in the casing top and side cap is the same. Locations of the thermocouples are shown in Figure 2.

In order to explore the temperature change due to different reasons three cases are considered:

Case (1) Temperature increase due to electrical power input
Case (2) Temperature increase due to viscous slippage between clutch plates
Case (3) Temperature increase due to the combined effect of electrical power input and slippage between the clutch plates

An AC motor drives the MR fluid clutch and the angular velocity of the motor is controlled by varying the frequency of the input voltage using an AC inverter.

Effect of Electrical Power Input

In these tests, temperature increase in the MR fluid clutch due to electrical power input to electromagnet is considered. Electrical power is provided by two 60 V power supplies and the tests are performed for 1, 2, and 3 A input electric currents. Figure 3 shows the result of the 10-min heating tests for 1, 2, and 3 A input electric currents. As can be seen, the clutch is initially at the room temperature of about 25.2°C. At the end of the 10-min period, temperature of the clutch is increased to 26.8°C for 1 A input current. Results indicate that the temperature increase on the casing top and casing side are about the same for all test conditions. This means that the heat generated inside the clutch due to the electrical power at 1 A input electrical current is conducted in the axial and radial directions, equally. Similarly, the casing top and casing side temperature increases were very close to each other for 2 and 3 A input electric currents as in the 1 A case. The casing side and casing top temperature increase values are given in Table 2. Thermocouple reading at room temperature fluctuates about ±0.5°C.

Table 2. Temperature increase comparison for three different cases.

Case	ΔT (°C) Casing top	ΔT (°C) Casing side	ΔT (°C) Theoretical
Case (1) Viscous slippage[a] effect only	1.0	0.9	0.9
Case (2) 1 A electromagnet	1.1	0.8	1.1
Case (2) 2 A electromagnet	4.1	3.9	4.0
Case (2) 3 A electromagnet	16.0	15.8	15.2
Case (3) Viscous slippage + 1 A electromagnet	11.9	11.5	11.6
Case (3) Viscous slippage + 2 A electromagnet	19.5	18.9	18.3
Case (3) Viscous slippage + 3 A electromagnet	29.8	28.9	27.2

[a]The slippage rpm is 120 rpm between clutch plates.

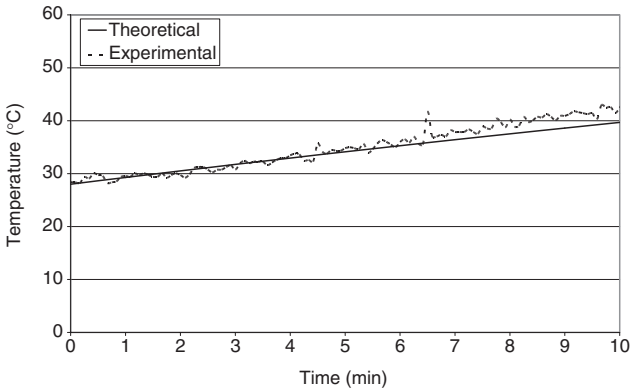

Figure 4. *Comparison of experimental and theoretical temperature distribution for electromagnet effect for multi-plate LSD MR fluid clutch for 3 A.*

Figure 4 compares the experimental and theoretical temperature histories for the case when 3 A input electrical current is supplied. The comparative results for the temperature increase on the casing surface of the

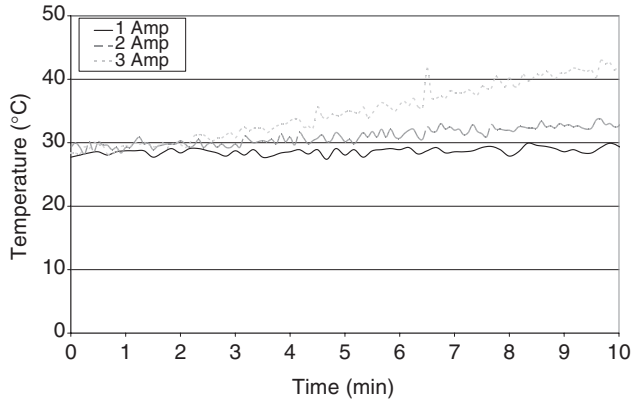

Figure 5. *Outer casing temperature increase comparison for different input current due to the electromagnet effect.*

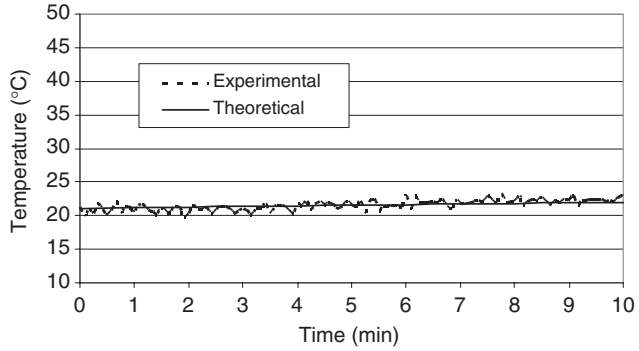

Figure 6. *Temperature test results for the slippage effect at 120 rpm relative velocity difference.*

clutch for different input currents can be seen in Figure 5. As the input current increases, heat generated by the electrical power increases, and as expected it results in faster temperature increase in the clutch.

Effect of Slippage

Another source of heating in this MR fluid LSD clutch is the slippage between clutch plates; therefore, temperature increase due to this effect is examined, separately. During these tests, the clutch plates are subjected to maximum slippage. The input side of the clutch is rotated at 120 rpm, while the rotation of the output shaft is prevented. For vehicle applications, when the clutch is functioning properly, it should allow limited slippage in cases where the vehicle is travelling on a curved or slippery road. However, these tests cover the worst-case situation, where the heating due to slippage is insignificant. The experimental and theoretical results of slippage effect at 120 rpm relative velocity difference are presented in Figure 6. The temperature increase on the casing top and casing side are recorded, simultaneously. Similarly, in order to determine the effect of slippage on the temperature increase, the clutch

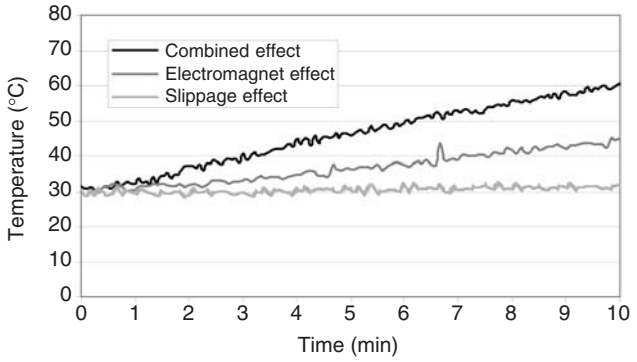

Figure 7. *Temperature increase comparison for 3 A input current and 120 rpm relative velocity difference between MRF LSD clutch plates.*

is operated for 10 min. After 10 min of operation, the temperature has increased 3°C both on the casing top and casing side.

Combined Effect of Electrical Power Input and Slippage

The final set of MR fluid clutch heating tests is conducted to examine the combined effect of electrical power input and slippage. During the tests, the MR fluid LSD clutch is subjected to slippage at 120 rpm and an input electric current is applied at the same time. Figure 7 shows the temperature increase comparison between cases of electrical power input only, slippage only, and combination of electrical power input and slippage for an input electrical current of 3 A. Results of the test demonstrate that the temperature increase due to the combined effects, ΔT_{comb}, is greater than the summation of temperature increase due to electrical power only, ΔT_{elec}, and temperature increase due to slippage effect only, ΔT_{vis}:

$$\Delta T_{\text{comb}} > \Delta T_{\text{elec}} + \Delta T_{\text{vis}}. \tag{13}$$

When the electrical power is supplied to the MR fluid clutch, the electromagnetic field is generated. Under the applied magnetic field, iron-particles in the MR fluid form chains (and or columns) between plates. Thus, when the clutch plates are slipping, it becomes difficult to break the chains due to increased surface shear stress. Therefore, the friction between the plates and MR fluid increases, and consequently, heat generated by the combined effect is more than the summation of the heat generated by the two separate effects discussed earlier.

By examining Figure 7 it can also be concluded that the heat generated due to the electrical power input is more effective than the heat generated due to the slippage. The temperature increase at the end of a 10-min time period due to the effect of electrical power input is only 16°C. On the other hand, it is 1°C due to the effect of slippage, and it is 29.8°C due to the combined effect of electrical power input and slippage. The combined effect tests are repeated with electrical

power input at 1 and 2 A input electric current while keeping the relative velocity difference at 120 rpm.

Table 2 summarizes the experimental and theoretical temperature increase for all cases after 10 min of testing. Theoretical and experimental values of temperature are compared in Figure 8 for the case of 2 A input electrical current and 120 rpm relative velocity difference. The theoretical prediction of the temperature increase is in good agreement with the experimental results.

Effect of Heating on Transmitted Torque

The torque transfer capability of the clutch at elevated temperatures is also examined. The transmitted torque is recorded simultaneously in the heating analysis. The torque transfer capability and temperature as a function of time is given in Figure 9 for the combined effect in the case of 3 A input electrical current and 120 rpm relative velocity difference. It is demonstrated that the torque capacity of the clutch did not reduce at elevated temperatures. This is due to the fact that the viscous torque is relatively small as compared to the MR torque; therefore, the reduction in viscous torque does not affect the overall torque performance. It is expected that the clutch would keep a constant level of transmitted torque unless the temperature exceeds the manufacturer specified operating temperature of 130°C (Lord Corporation). Beyond this temperature the MR fluid may not be stable, i.e., the yield stress reduces, which can result in a reduction in the torque transfer capacity of the clutch.

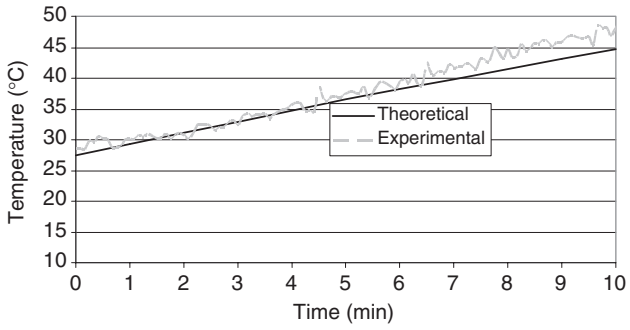

Figure 8. *Comparisons between experimental and theoretical temperature distributions due to combined effect for multi-plate MRF LSD clutch for 120 rpm slippage velocity and 2 A input current.*

SUMMARY AND CONCLUSIONS

Theoretical and experimental heating studies of a high-torque, multi-plate MR fluid clutch is conducted. The clutch is modeled as a lumped parameter system. The MR fluid clutch is tested under three different operating conditions. The effects of electrical power input and viscous slippage on temperature increase are examined, separately. The combined effect of slippage and electrical power input is also studied.

The temperature increase in the combined effect is higher than the summation of individual temperature increase due to electrical power and viscous slippage. This can be explained with the increased shear stress due to yield stress induced by the applied magnetic field. The clutch casing and clutch cap side temperatures are monitored, and it is demonstrated that the temperature increase in the casing and side of the clutch is the same. Theoretical predictions slightly underestimate the temperature increase, however, the difference does not exceed 8% in the temperature range tested. The results also show that the transmitted torque does not change with temperature due to the negligible contribution of the viscous torque.

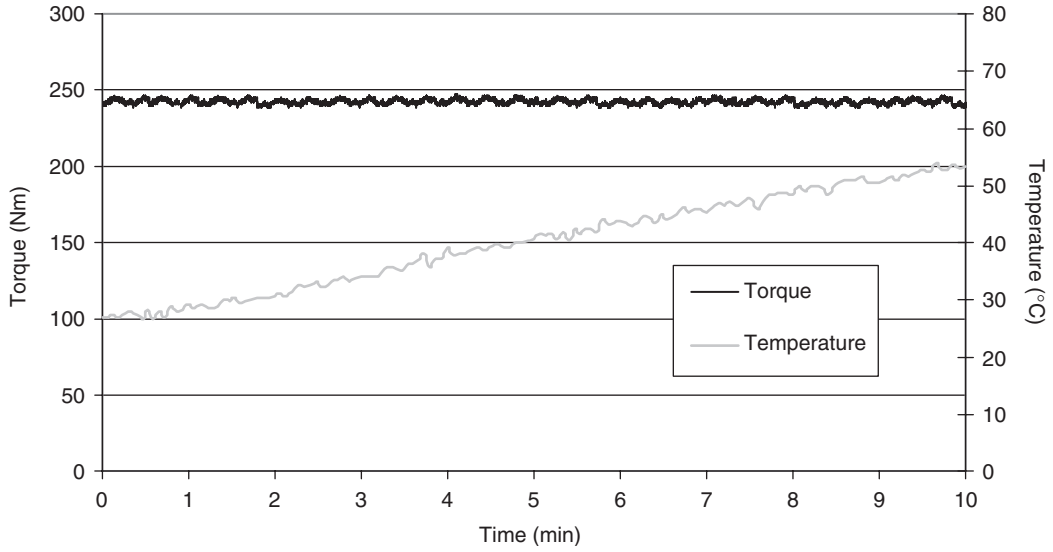

Figure 9. *The transmitted torque and temperature as a function of time at 3 A input current and 120 rpm relative velocity difference.*

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REFERENCES

Breese, D.G. and Gordaninejad, F. 1999. "Heating of Magneto-rheological Fluid Dampers: A Theoretical Study," *Smart Systems for Bridges, Structures, and Highways*, In: Liu, S.C. (ed.), *Proceedings of SPIE Conference on Smart Materials and Structures*, Vol. 3671, pp. 2–10, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, USA.

Carlson, D.J. and Weiss, K.D. 1994. "A Growing Attraction to Magnetic Fluids," *Machine Design*, 66(15):61–64.

Dogruoz, M.B., Gordaninejad, F. and Wang, L.C. 2000. "Heat Transfer of Magneto-rheological Dampers," *Smart Systems for Bridges, Structures, and Highways*, In: Liu, C. (ed.), *Proceedings of SPIE Conference on Smart Materials and Structures*, Vol. 3988, pp. 84–93, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, USA.

Dogruoz, M.B., Gordaninejad, F., Wang, E.L. and Stipanovich, A.J. 2001. "An Experimental Study on Heat Transfer from Fail-safe Magneto-rheological Fluid Dampers," *Damping and Isolation*, In: Inman, D.J. (ed.), *Proceedings of SPIE Conference on Smart Materials and Structures*, Vol. 4331, pp. 343–353, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, USA.

Dogruoz, M.B., Wang, E.L., Gordaninejad, F. and Stipanovic, A.J. 2003. "Augmenting Heat Transfer from Fail-safe Magneto-rheological Fluid Dampers Using Fins," *Journal of Intelligent Material System and Structures*, 14(2):79–86.

Ellam, D.J., Bullough, W.A. and Oravsky, V. 2002. "Heat Transfer from an ESF Radial Plate Clutch Surface," In: Bossis, G. (ed.), *Proceedings of the 8th International Conference on Electrorheological Fluids and Magnetorheological Suspensions*, pp. 104–110, World Scientific, London, England.

Gordaninejad, F. and Breese, D.G. 1999. "Heating of Magnetorheological Fluid Dampers," *Journal of Intelligent Materials, Systems and Structures*, 10(8):634–645.

Incropera, F.P. and DeWitt, D.P. 1996. *Fundamentals of Heat and Mass Transfer*, 4th edn, John Wiley & Sons, Hoboken, NJ, USA.

Kavlicoglu, B., Gordaninejad, F., Evrensel, C.A., Fuchs, F. and Korol, G. 2003. "A Multi-plate Magneto-rheological Fluid Limited Slip Differential Clutch," In: Anderson, E.H. (ed.), *Proceedings SPIE Smart Structures and Materials Conference, Industrial and Commercial Applications*, Vol. 5054, pp. 341–349, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, USA.

Lord Corporation, http://www.mrfluid.com/pdf/MRF_132AD_2002_15_0.pdf

Makris, N., Roussos, Y., Whittaker, A.S. and Kelly, J.M. 1997. Viscous Heating of Fluid Dampers During Seismic and Wind Excitations – Analytical Solutions and Design Formulae, Technical Report No. UCB/EERC-97/11, Earthquake Engineering Research Center, University of California Berkeley, November 1997.

Oravsky, V. 1998. "Dissipation of Energy in a Concentric ER Clutch and its Refined Quasi-static Model," In: Nakano, M. and Koyama, K. (eds), *Proceedings of the 6th International Conference on Electrorheological Fluids and Magnetorheological Suspensions and their Applications*, pp. 631–640, World Scientific, London, England.

Oravsky, V. 2000a. "Unified Dynamic Model of Concentric and Radial Electrorheological Clutched including Heat Transfer," In: Tao, R. (ed.), *Proceedings of the 7th International Conference on Electrorheological Fluids and Magnetorheological Suspensions*, pp. 827–838, World Scientific, London, England.

Oravsky, V. 2000b. "Nondimensional Common Thermo-Electro-Hydro-Mechanical Model of ER Clutches and Some its Approximate Solutions," In: Tao, R. (ed.), *Proceedings of the 7th International Conference on Electrorheological Fluids and Magnetorheological Suspensions*, pp. 839–848, World Scientific, London, England.

Oravsky, V. 2002. "Heat Transfer from an ESF Radial Plate Clutch Surface Part 2 – Theory," In: Bossis, G. (ed.), *Proceedings of the 8th International Conference on Electrorheological Fluids and Magnetorheological Suspensions*, pp. 673–679, World Scientific, London, England.

Smyth, R., Tan, K.H. and Bullough, W.A. 1994. "Heat Transfer Modeling of Cylindrical ER Clutch," *International Journal of Modern Physics B*, 8(20–21):3033–3050.