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SESSION 1

Active/Passive Devices for Civil Structures

Heating of Magneto-Rheological Fluid Dampers: A Theoretical Study

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ABSTRACT

This paper focuses on a theoretical model that predicts the temperature increase of Magneto-rheological (MR) fluid dampers experiencing a sinusoidal input motion. A theoretical model is developed to estimate the temperature rise based on the non-linear behavior of the MR fluid damper. This model is solved numerically, and the numerical solution is compared with a known linear solution and experimental results in order to validate the accuracy of the model. Also, a non-dimensional form of the governing equations are developed to examine the key parameters. The non-dimensional terms show the effect of external and internal parameters on the trends of heat dissipation as well as heat generation within the MR fluid damper.

1. INTRODUCTION

Semi-active energy dissipating devices are increasingly being investigated for various applications [1,2]. Controllable fluid dampers are of particular interest in this area. These fluids include MR and Electro-Rheological (ER) fluids, which change properties when exposed to magnetic and electric fields, respectively. Devices, specifically axial dampers, utilizing MR fluids are explored in this study. The primary purpose of a damper in any dynamic application is the dissipation of energy. This energy would otherwise remain within the system that may lead to premature failure of equipment. The damper dissipates the excess energy and primarily converts it to heat. This study focuses on the theoretical modeling of generation and dissipation of heat for MR fluid dampers.

Current research trends in this specific area are sparse. Some investigators have researched the effects, on a microscopic scale of heat transfer within the passages of a system containing MR Fluid [3,4]. The idea is to use MR Fluid to control heat transfer through a channel. This could be accomplished by filling an opening between two walls with MR Fluid. The heat flux across the MR Fluid could be partially controlled by varying the thermodynamic properties of the MR Fluid. The thermodynamic properties of MR Fluid can be changed through the application of a magnetic field to the fluid. Other research performed on MR fluid dampers included testing in an elevated ambient temperature scenario in order to ensure performance requirement [5]. Other attempts

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have been made to quantify the temperature increase due to energy absorption of viscous as well as ER fluid dampers [6].

In this study, a theoretical model is proposed to estimate the temperature of MR fluid dampers. The model accounts for the increased internal energies due to a sinusoidal type input as well as the heat generated through the use of an internal electromagnet. The model will account for the inherent non-linearity of MR fluid dampers. The theoretical model is compared to a known exact solution, and experimental data. In addition, dimensionless quantities are developed allowing insight into the effects of different contributing parameters.

2. THEORETICAL MODELING

The theoretical model to determine the temperature increase in MR fluid dampers utilizes a lumped system analysis. The lumped system approach assumes the difference in the temperatures of any point within the damper with other points (i.e, temperature gradient) is within 5%. A control volume is established to which an energy balance is applied, and the validity of the lumped system analysis is presented. Terms accounting for work entering and leaving the system, heat crossing the boundary and the internal energy of the system are represented. Non-linearity inherent in MR fluid dampers is represented through a constitutive law for energy dissipating devices. The energy balance is presented in rate form. Solving the energy balance provides temperature of the system at any given instant. The theoretical model is then manipulated to provide non-dimensional terms accounting for heat leaving the system as well as heat generated by the electromagnet.

2.1. Lumped System Analysis

In order to determine the validity of a lumped system analysis, the Biot numbers of the materials of construction were compared. A Biot value of 0.1 or less is recommended for an accurate lumped system analysis [7]. All Biot numbers in the present analysis were found to be a minimum of two orders of magnitude less than the recommended value.

First, the boundary for the control volume is established. This allows heat to leave the system through conduction only, while work is entering the control volume through mechanical and electrical forms. For stationary systems an energy balance yields

$$\dot{Q} - \dot{W} = \frac{dU}{dt} \tag{1}$$

where $\dot{Q}$ is the rate of heat transfer, $\dot{W}$ is the power and dU/dt is the rate of change of total energy of the system. The term $\dot{Q}$ is the rate of heat entering or leaving the lumped system. In this specific case no heat enters the system, while heat is leaving the system through convection with the surroundings. Therefore,

$$\dot{Q} = hA_s[\Theta(t) - \Theta_0] \tag{2}$$

where h is the convection heat transfer coefficient, A_s is the outer surface area of the damper, Θ_0 is the ambient temperature and $\Theta(t)$ is the surface temperature of the damper at any given time, t.

The second term in the LHS of Eq. (1), $\dot{W}$, represents the work entering or leaving the system which can be expressed as,

$$\dot{W} = -F(t)\frac{dX(t)}{dt} - I^2(t)R \tag{3}$$

where

$$X(t) = \bar{X}\sin(\omega t). \tag{4}$$

Here, $X(t)$ is a sinusoidal input displacement, I is the input electric current, R is the wire resistance of the electromagnet. In Eq. (3), $F(t)$ represents the work input to the system through reciprocation of the shaft. This analysis assumes that the input shaft work is sinusoidal. Also, $I^2(t)R$, is the electric power applied to the damper, shown as a function of time. For this analysis, however, the electrical work applied to the damper will be assumed constant with respect to time in order to simplify the analysis.

The nonlinear constitutive law for the MR fluid dampers is assumed to be

$$F(t) = C\left|\frac{dX(t)}{dt}\right|^{\alpha} \operatorname{sgn}\left(\frac{dX(t)}{dt}\right) \tag{5}$$

where C is the damping coefficient and α is a fractional exponent that accounts for the nonlinearity inherent in a MR fluid damper. It should be noted that both C and α are functions of input electric current, I, and temperature, Θ. Generally the range of α is $0 \leq \alpha \leq 1.5$. A value of one represents purely viscous behavior and a value of zero corresponds to purely rigid plastic behavior.

On the right hand side of Eq. (1), dU/dt is the rate of change of the internal energy of the system. Because this is a lumped system analysis, this quantity represents a sum of all internal energies of the material contained within the damper. The Biot number analysis shows that the temperature gradient within the damper is much less than 5% difference across any distance within the damper, therefore, the rate of change of temperature is assumed to be identical for all the materials within the control volume. The right hand side of Eq. (1) can be expressed as

$$\frac{dU(t)}{dt} = \frac{d\Theta(t)}{dt}\Sigma m\hat{c}_p \tag{6}$$

where $d\Theta(t)/dT$ is the rate of change of the temperature of the system and $\Sigma m\hat{c}_p$ is the summation of the internal energies of the materials contained within the system. Substitution of Eqs. (2-6) into Eq. (1) and rearranging the terms yields,

$$\dot{\Theta}(t) + \lambda(\Theta(t) - \Theta_0) = \frac{C\bar{X}\,\omega\cos(\omega t)}{\Sigma m\hat{c}_p}\left|\bar{X}\,\omega\cos(\omega t)\right|^{\alpha} \operatorname{sgn}\left(\frac{dX(t)}{dt}\right) + \eta \tag{7}$$

where

$$\lambda = \frac{hA_s}{m\hat{c}_p} \tag{8}$$

and

$$\eta = \frac{I^2 R}{\Sigma m\hat{C}_p}. \tag{9}$$

Equation (7) represents the differential energy balance for the entire lumped system. Due to the nonlinear nature of Eq. (7), it only can be solved numerically. An exact solution is available if α is either zero or one. For the case of a MR fluid damper α is a non-integer, therefore, this requires that an approximate solution for Eq. (7) be developed. Forward difference, utilizing a 4th order Runge-Kutta method, is employed to solve Eq. (7) for any value of α.

2.2 Dimensionless Form

Further generalization of the solution can be obtained by developing non-dimensional quantities from Eq. (7). Rearranging and substituting values in Eq. (7) yields,

$$\frac{\dot{\Theta}(t)}{\mathrm{K}\bar{\lambda}} + \Delta\bar{\Theta} = \frac{\cos^{\alpha+1}(\omega t)}{\bar{\lambda}} + \phi \tag{10}$$

where

$$\Delta\bar{\Theta} = \frac{\Delta\Theta(t)}{\xi\bar{X}^{\alpha+1}\omega^{\alpha}} = \frac{\Theta(t) - \Theta_0}{\xi\bar{X}^{\alpha+1}\omega^{\alpha}}, \tag{11}$$

$$\mathrm{K} = \xi\bar{X}^{\alpha+1}\omega^{\alpha}, \tag{12}$$

$$\xi = \frac{C}{\Sigma m\hat{c}_p}, \tag{13}$$

$$\bar{\lambda} = \frac{\lambda}{\omega}, \tag{14}$$

and

$$\phi = \frac{\eta}{\bar{\lambda}\mathrm{K}} \tag{15}$$

Equation (14) is a dimensionless quantity that represents the heat capacity and heat removal characteristics of the system. Equation (15) is a dimensionless quantity that represents the energy input to the system due to the electromagnet.

3. RESULTS AND DISCUSSION

First, a comparison is made between an exact and the numerical results to ensure the validity of the approximate solution. Figure 1 shows the comparison of the approximate and exact solutions of Eq. (7) for $\alpha = 1$. A step size of 25 was used to produce the results shown in Figure 1. The results indicate that the numerical simulation is a good approximation to the solution of Eq. (7) with minimal error.

Next, in order to produce the theoretical results for a MR fluid damper, the damping coefficient, *C*, as well as the fractional exponent α needed to be determined or approximated for a specific damper. Therefore, these coefficients are found through using a least squares curve-fitting technique applied towards actual experimental data. A typical fit of Eq. (5) to experimental data is shown in Figure 2. The MR fluid damper used for this purpose was designed, built and tested at the University of Nevada, Reno [8] for a front suspension of a modern mountain bicycle. The damper is approximately 15 cm in length and has an outside diameter of 3.2 cm. This damper was required to have a through shaft design due to geometric constraints. The through shaft design eliminates the need for an accumulator and accommodates a stroke of approximately 5.0 cm. The damper is extremely light in weight, at approximately 170 grams. For the results shown in Figure 2, the input to the damper was a sinusoid of 0.01m amplitude at a frequency of 2 Hz. Additional parameters used in producing results were surface area (A_s) of 0.01342 m^2, resistance (R) approximately 6.0 ohms, and a convection coefficient of 25 W/m^2-^oC.

Once the coefficients were determined as explained above, the theoretical simulation was used to predict the temperature response of an experimental MR fluid damper. The above method tended to produce results that over predicted experimental data. The reason for this arises from the fact that the performance of any damper changes with increases or decreases in temperature of the fluid. This results in the coefficients C and α not being constant. Using experimental data, temperature-dependent functions for C and α were developed. These functions were subsequently integrated into the approximate solution of the temperature problem. The functions developed were C=(-5.36)*(Temperature)+260 and α=(-0.0042)*(Temperature)+0.455. These functions indicate that the coefficient C varies significantly with change in temperature while the coefficient α is nearly constant. Figure 3 shows the effect in predicting the temperature increase of a MR fluid damper when including these temperature dependent coefficients compared with the previous coefficients that did not account for temperature changes in the coefficients. The final theoretical temperature increase for a MR fluid damper is compared to the actual experimental data in Figure 4 for 2Amp input electric current to the damper.

The theoretical model was extended to include the dimensionless quantities shown in Eqs. (14) and (15). Equation (14) is a measure of the amount of heat leaving the system. The value for $\bar{\lambda}$ was varied from 0 to 0.1, while the value of ϕ was set equal to zero. This is equivalent to removing the work provided by the electromagnet from the system and only analyzing the dissipation due to the input from the mechanical work. The lower limit of $\bar{\lambda} \approx 0$ corresponds to having nearly no convection present on the surface of the damper. At the other extreme when $\bar{\lambda} \approx 0.1$ corresponds to the damper in question having a very high heat transfer coefficient and/or a very large surface area. Figure 5 shows plots of $\bar{\lambda}$ plotted against dimensionless time (ωt) and $\Delta\bar{\Theta}$, which are presented in Eq. (11). The results show that the dimensionless temperature increase is greatly insensitive and non-linear with respect to the $\bar{\lambda}$ term. Only in the case of $\bar{\lambda} \approx 0.1$ there is a dramatic decrease in the temperature of the damper. It should be noted that $\bar{\lambda}$ = 0.00012 for the solution that corresponds to the experimentally tested damper.

Finally, the effect of the heat Equation (15) corresponds to the work added to the system through the electromagnet. Figure 6 shows plots of different values of ϕ plotted against dimensionless time (ωt) and $\Delta\overline{\Theta}$. For the plots shown $\overline{\lambda}$ was kept at a constant value of 0.00012. The value of ϕ was varied from 0 to 6000. When $\phi = 0$ this corresponds to the off state of the electromagnet. At a value of $\phi = 6000$, the electromagnet is at nearly full power. As expected, increasing ϕ increases the temperature of the system in a linear fashion.

4. SUMMARY AND CONCLUSIONS

A theoretical model was developed which provided the temperature increase of an MRF damper subjected to a sinusoidal input motion. The theoretical model allowed different power inputs to the electromagnet to be explored. Additionally, the theoretical model was numerically solved for any value of the fractional exponent. The model was then further enhanced by accounting for the change of performance of a MR fluid damper with large changes in temperatures. Comparisons were made between the theoretical model and experimentally determined data. This comparison validates the accuracy of the model as a useful prediction tool. Non-dimensional terms were extracted from the theoretical model that allowed the isolation of different input and output parameters to be accounted for. The non-dimensional parameters showed that the temperature increase was fairly insensitive to changes in the heat transfer coefficient and the surface area of the damper. On the other hand, the temperature increase was sensitive to the work increases caused by the electromagnet, showing a nearly linear increase.

5. ACKNOWLEDGEMENTS

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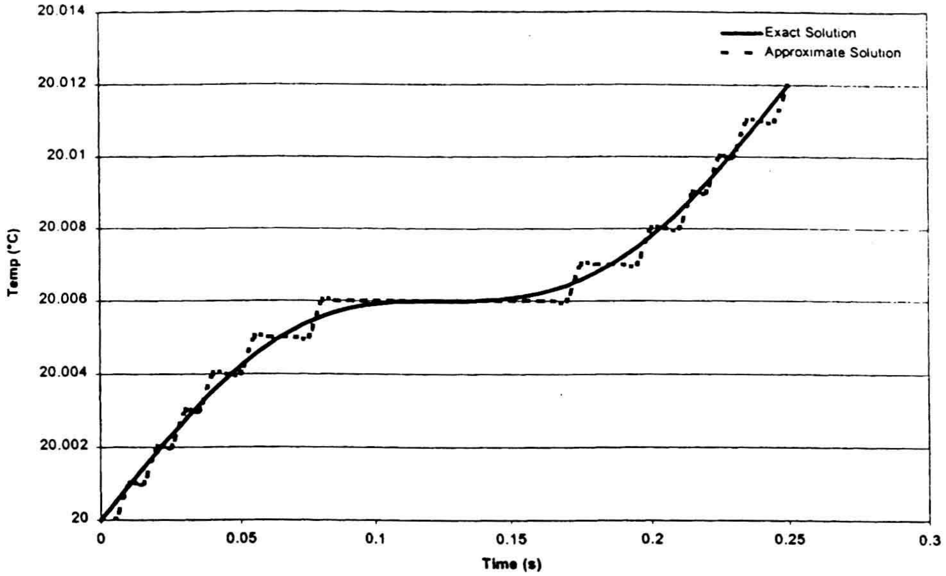

Figure 1. Comparison between the exact and the numerical solution for the case where α=1, C=2,537N-s/m, λ=0.00067, η=0, A_s = 0.039 m^2 ,h=25 W/m^2-°C, f= 2 Hz, and the amplitude of motion is 0.01m.

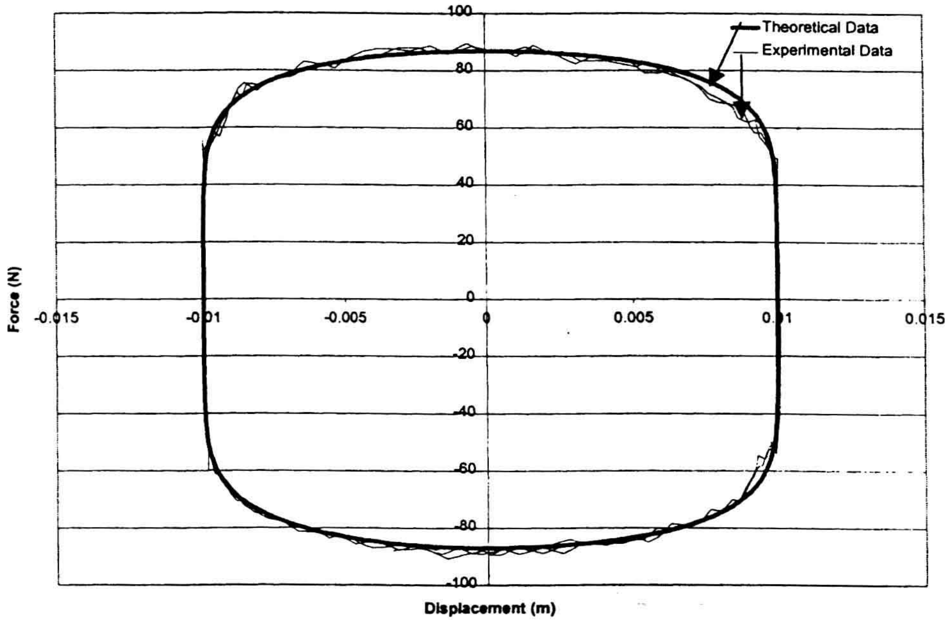

Figure 2. Least square curve fitting of the experimental results for evaluation of C and α.

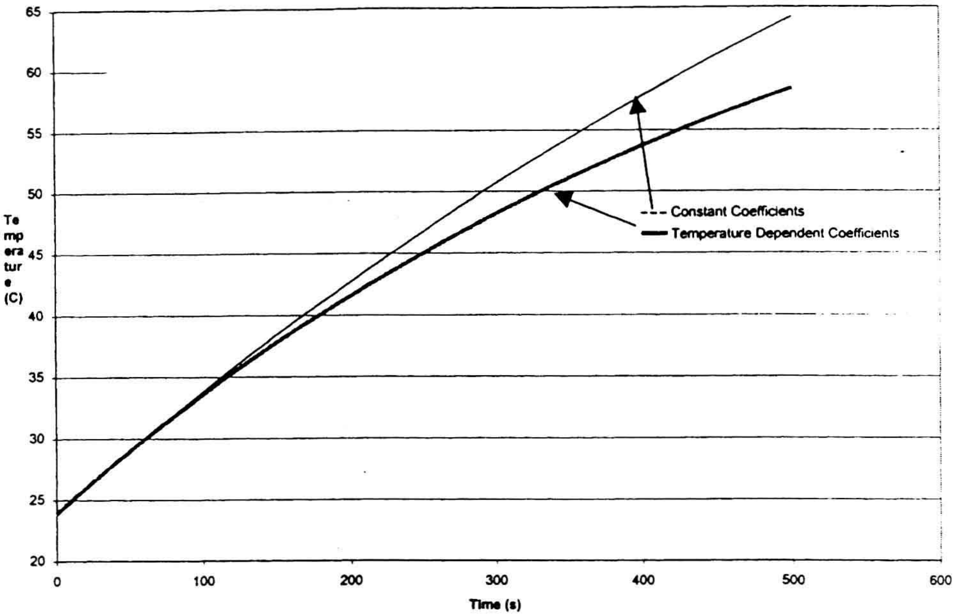

Figure 3. The effect of temperature dependency of constitutive coefficients (C and α) on the temperature rise of the MR fluid damper.

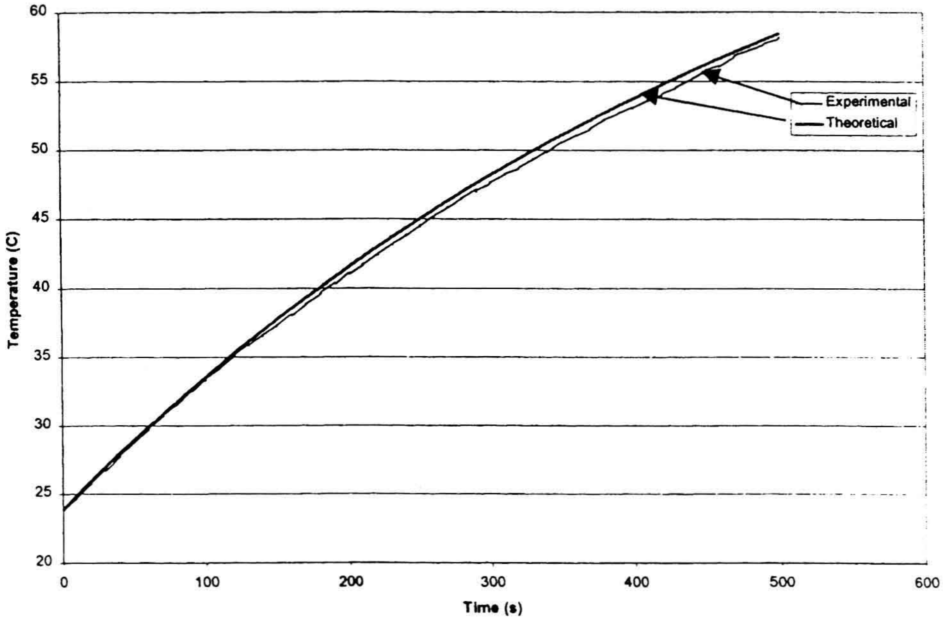

Figure 4. Comparison of theoretical and experimental results, using temperature-dependent constitutive coefficients model.