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Modeling and Simulation of Magnetorheological Fluid Damper for Predicting the Saturation Limit to Applied Field Current.

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ABSTRACT

This study presents on electromagnetic analysis of magnetorheological fluid damper for predicting the saturation limit of applied field current. The performance of MR damper depends on its magnetic and hydraulic circuit design. In this paper a finite element model is used to examine and investigate the 2- D axisymmetric MR damper. The piston velocity and the input current to the coil are varied to evaluate the resulting change in magnetic flux density (B) at the MR fluid gap. The simulation results are used to find the MR damper force at different applied current using MATLAB that would be analysed that increasing the current increase the Damper force but increase becomes steady at higher excitation current that shows the magnetic saturation of the MR fluid.

KEYWORDS: Magnetorheological (MR) fluids, MR damper, Magnetic flux densit

1 Introduction:

Magneto rheological (MR) dampers are a new class of devices that including the necessity of having very low power requirements for vibration control in various areas like suspension system, building structure etc whose rheological properties can be rapidly changed by applying a magnetic field to the fluid domain. In the absence of the applied field it behaves like the Newtonian fluid. When the MR fluid is subjected to magnetic field, the induced dipoles cause particles to aggregate and form chain-like structures which restrict the flow of the fluid, thereby increasing the apparent viscosity of the fluid. The mechanical energy needed to cause the chain like structures to yield increases nonlinearly with increasing applied magnetic field, resulting in the field-dependent yield stress.

An MR fluid consists of micro-sized magnetically polarisable particles, such as iron particles, suspended in a carrier liquid such as mineral oil, synthetic oil, water or glycol. A typical MR fluid will contain 20 to 40 percent by volume of relatively pure iron particles around 3 to 10 micron diameter in size. A variety of proprietary additives, similar to those found in commercial lubricants, is commonly added.

When the MR fluid is employed in valve systems, the rheological change of the fluid causes a pressure drop of the control volume, which can be continuously tuned by controlling the intensity of the imposed magnetic field. This feature has triggered considerable research activities on the modeling and design of MR valve devices. A large amount of research on the modeling and design of MR valves and dampers has been performed [2–4]. The finite element method (FEM) has also been used in the modeling and design of MR valves and dampers [5, 9].

2. Mathematical Modeling of MR fluid valve:

In this section, the dynamic model is derived by considering the overall configuration of MR valves as shown in fig 1. The produced pressure drop is derived based on the assumption that the MR fluid exhibits Bingham behaviour on experiencing a magnetic field and the flow is fully developed in the ducts. By using a viscoplastic Bingham model, the shear stress τ of the MR fluid flow is given by

$\tau = \tau_y(\text{field})\,\text{sgn}() + \eta, \quad \tau \geq \tau_y$ (1)

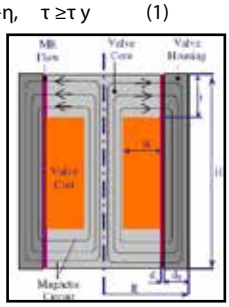


Fig. 1 MR valve configurations

where field is the magnitude of either the electric or the magnetic field, is the fluid shear rate, η is the plastic viscosity (at zero field) and τy is the yield stress.

Below the yield stress the material behaves Visco-elastically:

$\tau = G, \quad \tau < \tau_y,$ 2)

where G is the complex material modulus. Note that the complex modulus is also field dependent

In this study, the hydrocarbon-based MR fluid product (MRF-132DG) from Lord Corporation is used. By applying the least-squares curve fitting method to the fluid property specifications the yield stress corresponding approximate polynomial is given by:

$(\text{KPa}) = 52.962B^4 - 176.51B^3 + 158.79B^2 + 13.708B + 0.1442$ (3)

The total pressure drop in the damper is evaluated by summing the viscous component and yield stress component. which is approximated as

$$\Delta p = \Delta p_\eta + \Delta p_\tau$$

$$= \frac{6\eta L V_p (A_p + \pi R_1 d)}{\pi R_1 d^3} + 2c\frac{t}{d}\tau_y(H) \qquad (4)$$

Now for the damping force is governed by the damper's geometry, input electrical current to the electromagnet, the characteristics of MR fluids and the motion of the damper.

By assuming a quasi-static behaviour of the shock absorber and neglecting the frictional force, the damping force of the MR damper can be obtained as

(5) $$F_d = P_2 A_p - P_1(A_p - A_s) \qquad (5)$$

Where A_p and A_s are piston and piston shaft effective cross sectional areas respectively. P_1 and P_2 are the pressure in the upper and lower chamber of MR damper, respectively. The pressure in the gas chamber, Pa is approximated equal to the pressure in the lower chamber, P2. Thus, we may write

$$P_1 = P_a - \Delta P$$

$$P_2 = P_a \qquad (6)$$

In which the P is the pressure drop of the MR fluid flows through passing the orifice gap of the valve. The pressure drop P can be calculated by Eq. (4). The pressure in the gas chamber can be obtained by

$$P_a = P_0\left(\frac{V_0}{V_0 + A_s x_p}\right)^{\gamma} \quad (7)$$

In which P_0 and V_0 are the initial pressure and volume of the gas chamber, respectively. The parameter represents the coefficient of thermal expansion which varies between 1.4 and 1.7. Now considering Equation (5) and (6) the damping force can be described as:

$$F_d = P_a A_s + \Delta P(A_p - A_s) \quad (8)$$

Now by substituting Eq. (4) in Eq. (8) and considering Q =. Now separating the viscous and filed dependent parts, one may write

$$F_d = P_a A_s + C_{vis}\dot{x_p} + F_{MR}\,sgn(\dot{x_p}) \quad (9)$$

Where,

$$C_{vis} = \frac{6\eta L}{\pi(R - d_h - {}^{d}/_{2})d^3}\left(A_p - A_s\right)^2 \quad (10)$$

and

$$F_{MR} = \left(A_p - A_s\right)\frac{2ct}{d}\tau_y \quad (11)$$

The first term in Eq. (9) represents the elastic force due to the gas compliance; the second term is the damping force due to MR fluid viscosity and the third term represents the force due to the yield stress of the MR fluid. As it can be realized, the yield dependant term in Eq. (11) can be easily controlled by the applied magnetic field and as mentioned before, it is expected to have a large effect on the damping force.

Now MR fluid is the non-Newtonian fluid so it is important to model the viscosity as a function of the shear rate.

Shear rate, [1/sec], is defined as:

The shear rate ( ) is a function of the plastic viscosity (η). It was characterized by an exponential equation of the form:

3 Electromagnetic Analysis of MR damper

An MR damper is to be analyzed as a 2-D axisymmetric model as shown in fig 2. For a given current, we can determine the magnetic flux density at the Piston, MR Fluid and the Damper Housing. The flux leakage out of the engine and housing at the perimeter of the model is assumed to be negligible enough that no saturation of the material occurs. The ANSYS program includes a variety of elements that we can use to model electromagnetic phenomena. After some research, we determine that PLANE13 was the most suitable element for our model because it is a 2-D quadrilateral Coupled-Field-Solid, which contains four nodes.

Fig.2 2-D Axisymmetric MR Damper Piston in ANSYS

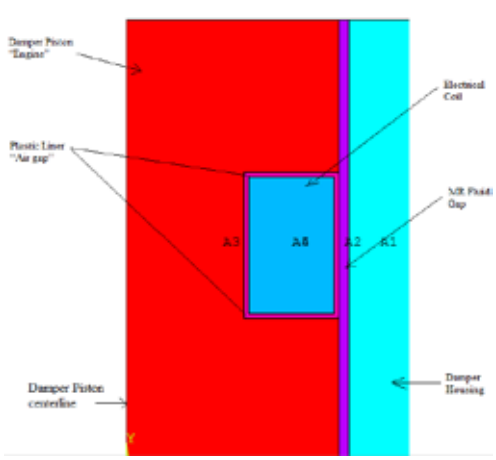


Table 1 Material used in this analysis.

Components	Material Type	Permeability	Saturation Flux density
Material Housing	Carbon steel	200	1.35
MR Fluid	MRF132 - DG	B-H curve	B-H curve
Piston	Carbon steel	200	200
Coil	Copper	1	-
Air gap	-	0.005	-

In these analysis applied current varies from 0 to 2 A, and results are taken in ANSYS APDL module. Results shows that increasing the current will increase the magnetic flux and that will described in Table 2. From that flux density it would be analysed yield stress at the MR fluid gap, according to that flux density shear stress, pressure drop and Damper force would be found out as per the equation (3), (4) and (9).

Table 2: Magnetic Induction results at different current

Current (Amps)	Magnetic Induction, B(f) (Tesla)
0.0	0.00
0.2	0.465
0.4	0.679
0.6	0.721
0.8	0.768
1.0	0.812
1.2	0.859
1.4	0.91
1.6	0.96
1.8	1.02
2.0	1.09

4 Results and Discussion

The assumption of no leakage at the perimeter of the model means that the flux will be acting parallel to this surface. This assumption is enforced by the flux parallel boundary condition placed around the model.In Figure 3, the flux path is surrounding the coil and if we look closely at the MR fluid gap, we can see the concentration of many flux lines. Additionally, the lines shift as soon as they hit the gap, therefore indicating the effect of the MR fluid on the electric circuit. Since the properties of the MR fluid are different than the engine and damper housing, the magnetic induction vectors are changing in directionsonce they hit the MR fluid gap.

Fig 3 Flux lines around the one stage coil

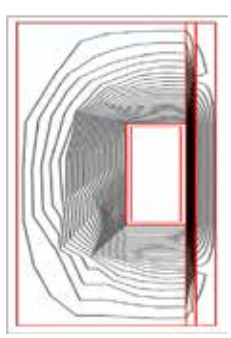
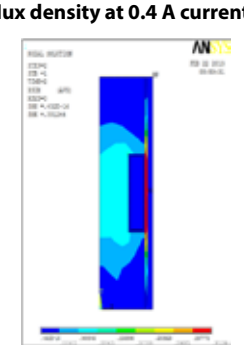


Fig 4 Matgnetic flux density at 0.4 A current



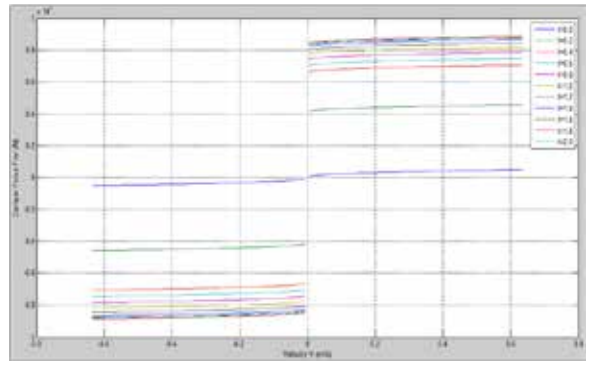
These magnetic flux lines represent the magnetic flux density vectors. So we have a high magnetic induction along the gap, as was observed in Figure 3. These concentration lines of magnetic flux will decrease by the time rheological saturation is reached.

Nodal solution is used to obtain results because it establishes the value at each node instead of looking at the element in general From the ANSYS model, one can generate the magnetic induction at the gap of the MR fluid. Each time we increase the electrical current to the coil, the magnetic induction will increase and we'll get a new set of data (Fig 4 shows: 0.4 A)

Now to calculate the force that is developed along the MR fluid gap, simply multiply the off-state pressure with the area of the piston, and the on-state pressure with the active fluid area.

$$F = \Delta P\eta \,.\, A\eta + \Delta P\tau \,.\, A\tau$$

Using MATLAB, can generate the force – velocity plot for the model designed in ANSYS for different velocity input (0-0.635 m/sec) as shown in Fig 5



5 Conclusions

Purpose of the study to find out the Magnetorheological saturation limit for applied current to damper system has been conclude that increasing the current would increase the magnetic force linearly up to 1.8 amp however this reaches at saturation stage after that The force provided at 1.8 Amps is almost the same as the one at 2.0 Amps (i.e., 8876 N). Beyond this current, we can barely generate any more force. So the maximum force that we can generate for the damper that was modeled for this analysis is approximately 8876 N.

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