Optimization Analysis of a New Vane MRF Damper

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Abstract: The primary purpose of this study was to provide the optimization analysis certain characteristics and benefits of a vane MRF damper. Based on the structure of conventional vane hydraulic damper for heavy vehicle, a narrow arc gap between clapboard and rotary vane axle, which one rotates relative to the other, was designed for MRF valve and the mathematical model of damping was deduced. Subsequently, the finite element analysis of electromagnetic circuit was done by ANSYS to perform the optimization process. Some ways were presented to augment the damping adjustable multiple under the condition of keeping initial damping forces and to increase fluid dwell time through the magnetic field. The results show that the method is useful in the design of MR dampers and the damping adjustable range of vane MRF damper can meet the requirement of heavy vehicle semi-active suspension system.

1. Introduction

MR fluids undergo reversible and rapid changes in material characteristics when subjected to a magnetic field. This change is primarily observed as a significant increase of the yield shear stress of MR fluids and can be continuously controlled by tuning the magnetic field intensity [1]. These characteristics have created potential applications in semi-active vibration isolation and motion damping systems in industrial application. In vehicle engineering, energy-dissipating devices of the said systems include mainly piston-cylinder dampers and vane hydraulic dampers. Being compact structure and perfect heat-dissipating ability, the latter are usually adopted in some heavy vehicles. Today, a piston-cylinder MRF damper has been used in the real life applications due to its minimal power dissipation and a board range of capacities [2, 3]. For example, when it is utilized in automotive primary suspension, these automotive primary suspensions can offer combined advantages of both passive and active control systems and show less vehicle motion and superior ride quality in rough road [4, 5]. However, the applications of vane MRF dampers are still few. In this study, we proposed a new vane MRF damper. A phenomenological model that can accurately capture its dynamic behavior is presented. By optimization analysis, dynamic force range of the damper is extended to meet the performance requirement for certain heavy vehicle.

2. Vane MRF damper

Up to now, the structure of the conventional hydraulic damper is already perfect, so it is considered as basic structure of vane MRF damper. The striking dissimilarity between the both is that flow through the orifice valve (i.e. MR valve) of the vane MRF damper is exposed to magnetic field. A partly assembled vane MRF damper is shown in Figure1.
Because the yield stress of the MR fluid increases significantly with an increase of magnetic flux density, magnetic field is expected stronger and to plumb up the flow as possibly. So, to expand room for a coil assembly, clapboard is designed to be hollow and the coil with coil core is placed in this hollow. Correspondingly, an arc MR valve is placed between clapboard and rotary vane axle which one rotates relative to the other and they act as two magnetic poles respectively. So the MR valve consists of solenoid core, magnetic coil and rotary vane axle and it controls the flow rate and pressure drop through the gap, and so controls damping level. To evaluate the performance of this MR valve, a generalized MR valve model is developed.

3. Theoretical Modeling

According to the pressure drop equation deduced by Philips and his colleagues (1969) based on parallel-plate model between two paramagnetic poles [6],

$$\Delta P = \frac{12\eta Q}{wh^3} + \frac{3L\tau}{h}$$

(1)

Where $Q$ is the volume flow rate due to pressure drop, $\tau$ is the controllable yield stress of a MR fluid, $L$ is the length of MR valve, $w$ is the width of the plates, $\eta$ is viscosity of the fluid, and $h$ is the gap between the plates. To apply the above formula for the flow through the MR valve, in this case the following correspondences of symbols are made

$L \rightarrow \theta \cdot \bar{r}_{MR}$

Where $\theta$ and $\bar{r}_{MR}$ are radian and average radius of MR valve. Then the pressure drop is obtained as,

$$\Delta P = \frac{12\eta Q}{wh^3} \theta_1 \cdot \bar{r}_{MR} + \frac{3\tau}{h} \theta_2 \cdot \bar{r}_{MR} = \Delta P_{vis} + \Delta P_{MR}$$

(2)

Where $\theta_1 \cdot \bar{r}_{MR}$ is the arc length of the flow channel, $\theta_2 \cdot \bar{r}_{MR}$ is the arc length of the MR valve, $\Delta P_{vis}$ is pressure drop due to viscous orifice, and $\Delta P_{MR}$ is pressure drop due to MR effect. Suppose that there is no leakage gap and fluid flows only through the MR valve. Conservation of mass for an incompressible fluid yields:

$$\Delta V = \frac{\alpha}{2\pi} \cdot \pi (R^2 - r^2) b$$

(3)

$$Q = \frac{dV}{dt} = \frac{b(R^2 - r^2)}{2} \frac{\alpha}{2}$$

(4)

where $b$ is the width of the vane, $r$ is inner radius of the vane, $R$ is outer radius of the vane and the
rotary angle of the vane around its axis, $\alpha$, and the rotary velocity $\dot{\alpha}$, are reconstructed by

$$\alpha = \alpha_0 \sin \omega t \quad \dot{\alpha} = \alpha_0 \omega \cos \omega t$$

(5)

where $\alpha_0$ is the maximum angle of the vane around its axis, and $\omega$ is frequency.

To ensure the fluid sufficient amount of dwell time to reach its full yield stress, the high velocity behavior of MR fluid subjected to a magnetic field should be evaluated. Dwell time is defined as the amount of time the fluid spends in the presence of a magnetic field. In other words, dwell time is the time it takes the fluid to flow through the MR valve [7]. So dwell time is calculated from

$$t_{\text{dwell}} = \frac{\theta_{\text{z}} \cdot \bar{F}_{\text{MR}}}{v}$$

(6)

Where $v$ is the mean fluid velocity thro is the arc length of through the flow channel, as,

$$v = \frac{Q}{wh}$$

(7)

4. Optimization of the Structure

In terms of its feasibility for engineering applications, force characteristics of the damper have to be optimized.

4.1. Dynamic force range Analysis

A broad dynamic force range is important for force characteristics. Define parameter $K$ [8]

$$K = \frac{\Delta P_{\text{MR}}}{\Delta P_{\text{vis}}} = \frac{\tau_{\text{z}} \theta_{\text{z}} w^2}{2\eta \theta_1 b (R^2 - r^2) \dot{\alpha}} = k_1 k_2 \frac{\tau_{\text{z}} h^2}{2\eta \dot{\alpha}}$$

(8)

Where $k_1 = \frac{\theta_{\text{z}}}{\theta_1}$ is called magnetic field efficiency coefficient, $k_2 = \frac{w}{b (R^2 - r^2)}$ is the geometric parameter of vane MRF damper. It can be seen from equation (8) that the damping adjustable multiple of MRF damper is related to magnetic field efficiency coefficient $k_1$, geometric parameter $k_2$, height of gap $h$ and performance parameters of MRF ($\tau_{\text{ymax}}$, $\eta$).

4.1.1. The relation between $K$ and $k_1$

$K$ is directly proportional to $k_1$ which depends on the distribution of magnetic field. In the case of meeting the requirement of damper structure, the proper augmentation of $k_1$ may enhance $K$ in MRF damper.

4.1.2. The relation between $K$ and $k_2$

When the structure and the main geometrical parameters (radius of vane $R$, radius of axis $r$, width of vane $a$) of MRF damper are constants, $k_2$ has a direct ratio relationship with the width $w$ of the MR valve. The initial damping has a ratio relation with $1/w$, $K$ and $k_2$ present direct ratio. According to the geometrical parameters in vane hydraulic damper, the relation curve of damping adjustable multiple $K$ and $w$ is determined at different dynamic viscosities of MRF damper (Figure 2). Therefore, in the case of meeting the requirement of the initial damping, the proper augmentation of the width $w$ of the MR valve may enhance $K$. However, $w$ should be restricted by the width of magnetic field.
4.1.3. The relation between $K$ and $h$
For the vehicle semi-active suspension system, the MRF damper should have the fail-safe capability, which is referred to retain a minimum required damping capacity in the event of a power supply or electronic system failure. So the initial damping of the vane MRF damper should reach or approach to that of vane hydraulic damper. The pressure drop $\Delta P$ due to viscous orifice has a ratio relationship with $1/h^3$, $K$ has a direct ratio relationship with $h^2$. From Figure 3, we know that the gap height $h$ cannot be designed too large while keeping the initial damping of the vane MRF damper invariable. Besides, $h$ vs. the magnetic field intensity in the gap should be taken into account.

4.1.4. The relation between $K$ and the performance parameters of MRF
In order to make $K$ as large as possible, the MRF with the higher shear yield stress and the lower initial viscosity should be chosen. Commonly, with the augmentation of the particle volume percent, the shear yield stress and the viscosity are enlarged.

4.2. Magnetic Circuit Analysis
The magnetic field distribution of magnetic circuit is evaluated using ANSYS finite element analysis (FEA) software. A three-dimensional model developed by SolidWorks software, which can be seen in Figure 4, is imported into ANSYS software for specified boundary conditions that include material properties, electromagnetic activation current, and element insulation. A relationship between electromagnet activation current and magnetic field strength is determined using this software. Being an assembly model by SolidWorks software, the sizes of the model are very exact figures. The materials adopted in this model are as followings: coil core-DT4, vane and shell-steel45, constant relative permeability $\mu_r=700$ for steel-45, applying current-2 Amp, turns of coil-1000.

Figure 5 shows the distribution of magnetic field inside the MR valve. From the results presented in Figure 5(a), the magnetic flux density values of the MRF through MR valve is only 0.128–0.12T, which is too small to MRF, and there is no significant change in the magnetic flux occurs when the applying current is change. Obviously, the cause is that the coil core has become saturated with magnetic flux and magnetic flux of entire circuit is limited to alter further, as presented in Figure 5(b).
The geometrical parameters optimized in the electromagnetic analysis part can be seen in Figure 5. They are the coil turns, coil core radius. Especially, a bell-mouthed cone is designed in coil core. The optimization of each part is complete when no significant change in the magnetic flux occurs when the dimensions of said part are altered further. Furthermore, since the clapboard of the damper is one of the design constraints, coil radius needs to be optimized in this limited space. Because different MR thickness values give different magnetic flux density values inside the MR valve, a thickness between 0.8-1.2mm, is selected as a desired value of MR disk thickness.

Using the final design parameters, electromagnetic analysis is performed for 2A current inputs and 800 turns coil. Figure 6 shows the magnetic flux density values inside magnetic circuit. Obviously, 0.6–0.7T and 1.6–1.7T are achieved inside the MRF through MR valve, which is enough for MR effect, and coil core respectively. In addition to this, the magnetic flux density vectors on the MR surface are uniformly distributed due to the symmetry of the MR valve.

By using the dimensions obtained from the electromagnetic analysis, electromagnet coil resistance is 4 ohm, which is a low resistance. This allows applying currents as high as 2 or 3 Amp and therefore, a low electrical power.

5. Conclusions

A new controllable vane MRF damper was developed to determine the viability of implementation of this technology into some heavy vehicles. It has been demonstrated that it is desirable for a semi-active suspension system because of softer damping, fail-safe behaviour, lower dispersion of the damping force, lower heat generation and consuming less power compared to the conventional MR damper. It is believed that improvements of ride quality in rough road should be seen in the positive direction with further optimization of the semi-active algorithm.

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