A Design Strategy for Magnetorheological Dampers Using Porous Valves

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Abstract. To design a porous-valve-based magnetorheological (MR) damper, essential design parameters are presented. The key elements affecting the damper performance are identified using flow analysis in porous media and an empirical magnetic field distribution in the porous valve. Based on a known MR fluid, the relationship between the controllable force of the damper and the porous valve characteristics, i.e. porosity and tortuosity, is developed. The effect of the porosity and tortuosity on the field-off damping force is exploited by using semi-empirical flow analysis. The critical flow rate for the onset of nonlinear viscous damping force is determined. Using the above design elements, an MR damper using by-pass porous valve is designed and tested. The experimental damper force and equivalent damping are compared with the predicted results to validate this design strategy.

1. Introduction

In conventional MR devices, flow channels are straight and electromagnetic coils are configured so that the flow streamlines are normal to magnetic field. This requires that the magnetic coil be positioned in a bobbin that is concentric to a tubular flux return. The magnetic coil is usually immersed in the MR fluid, which can reduce the life of the coil and lead to difficulties in extracting heat generated by the coil and damping action. This magnetic coil configuration limits the active length of the flow channel to be far less than the valve length. Configuration of the flow channel and flux return is also constrained by magnetic field saturation and precise MR valve flow gaps. In comparison, an MR porous valve configuration, in which MR fluids are forced to flow through voids within a packed bed of particles, has the advantage of a larger active volume. In this case, the active length can be as long as the valve length. Configuration of the flow channel and flux return is also constrained by magnetic field saturation and precise MR valve flow gaps. In comparison, an MR porous valve configuration, in which MR fluids are forced to flow through voids within a packed bed of particles, has the advantage of a larger active volume. In this case, the active length can be as long as the valve length. Shulman [1] proposed spiral channels or packed beds of particles placed inside a solenoid. In [2], a hydraulic ram was used to quasi-statically investigate the resisting force as MR fluids flowed through porous media in which the magnetic field was parallel to the average flow. Cook et al. [3] experimentally evaluated the dynamic performance of an MR damper exploiting a by-pass porous valve. However, a reliable design methodology for porous-valve-based MR damper has not yet been proposed since the flow mechanisms of the MR fluid in the porous valve are not well understood.

A design strategy for developing a porous-valve-based damper is described in this paper. Firstly, based on the Darcy’s law of the flow in a porous media [4], the effect of the porosity and tortuosity of the valve on the flow resistance is described, in which the condition for the onset of a nonlinear viscous force term is included. The effect of the nonlinear viscous force on the performance of the adaptive damper is evaluated. By using an empirical magnetic field analysis for the MR fluid flowing through the porous valve, the effect of the porosity and permeability of the filler material on the field
distributions is presented. These analytical tools can be explored for damper performance evaluation. Using experimental data [3], the analytical damper behavior with different porous media structures is validated. Finally, based on the known material properties of an MR fluid and determined design parameters, an MR damper using a bypass porous valve is developed and the predicted damper performance is validated by the experimental results of the damper.

2. Damper configuration and testing setup
To verify the design strategy for porous valves, an MR damper with a bypass porous valve is developed, and the damper performance, i.e. damping capacity and controllability, is experimentally evaluated. As shown in Figure 1, the damper mainly consists of a hydraulic cylinder connected to a bypass porous MR valve. The bypass valve was fabricated around a stainless steel tube with an inner diameter of 9.4 mm and a length of 100 mm. This tube can be packed with filler materials, such as steel or plastic spheres and rods, and a magnetic coil with 50 ohms of resistance is wrapped around the outside of the bypass valve to provide magnetic filed in the valve. The piston diameter in the hydraulic cylinder is 38 mm (1.5 in) and the double-ended rod diameter is 25.4 mm (1 in). Sinusoidal displacement excitations (with amplitude at 12.7 mm and frequency up to 7 Hz) were applied to the damper on an MTS dynamic testing machine, and the applied current ranged from 0 to 1 A. The measured force and displacement were used to reconstruct hysteresis behavior, compute equivalent damping and other characteristics of the damper, i.e. yield force and viscous damping.

![Figure 1. MR damper with porous valve](image1)

![Figure 2. Equivalent flow analysis model](image2)

3. Design parameters for porous MR valve
Since the bypass valve is randomly packed with fillers, the flow of the MR fluid through the valve containing a porous media is evaluated using a semi-empirical methods. Three major parameters related to the porous media are used for fluid behavior characterization in this design strategy, that is, porosity, tortuosity and Reynolds number. The porosity, $\varepsilon$, is defined as the ratio of volume of the void space to the volume of the whole space in the valve. The tortuosity, $\xi$, is defined as the ratio of the length of an imaginary flow path in the porous media to the whole length of the valve. The Reynolds number, $Re$, is introduced to evaluate the onset condition of a nonlinear viscous force. To derive the flow motion, the flow path in the porous media is considered as a multiple-pipe system, in which a set of identical cylindrical tortuous channels is grouped in parallel as shown in Figure 2. For each channel, the ratio of the internal fluid volume to the internal wetted surface area is defined as a hydraulic radius, $R_h$, which can be related to the volume and surface area of the filler and the porosity as:

$$R_h = \frac{V_p}{S_p (1 - \varepsilon)} \quad V_p : \text{volume of filler}, \quad S_p : \text{surface area of filler}$$
The diameter and length of each channel are then given as \( 4R_h \) and \( \xi L \), respectively. The behavior of the MR fluid flowing through a cylindrical channel is well described by a chain model [2]. The behavior of the MR fluid in a uniform magnetic field (of intensity \( H \)) can be represented by the Bingham model [5]. By combining the nonlinear Darcy’s law, the pressure difference between the inlet and outlet of the porous valve as a function of the flow rate, \( Q \), as shown in Figure 2 can be empirically obtained by:

\[
\Delta P = P_1 - P_2 = \frac{8 \xi L}{3 \pi R_h} \tau_s \text{sgn}(Q) + \frac{2 \eta_2 \xi^2 L Q}{R_d \varepsilon} A \left[ 1 + \left( \frac{1 - \varepsilon + \varepsilon - \varepsilon_0}{\varepsilon - \varepsilon_0} \right) \frac{\text{Re}}{\xi \text{Re}_{cr}} \right]
\]

(2)

where \( \tau_s \), \( \eta \), and \( \rho \) are the yield stress, viscosity and density of the MR fluid, respectively, and \( A \) is the cross-sectional area of the porous valve. In equation (2), the first term is a controllable resistance determined by the yield stress of the MR fluid, and the second term includes a linear viscous resistance and a nonlinear viscous resistance. The Reynolds number is defined as \( \text{Re} = \left( \rho A R_h Q \right) / \left( \varepsilon \eta A \right) \) such that the nonlinear viscous term in equation (2) is similar to a nonlinear force derived from inertial terms in the Navier-Stokes equation [4]. The constant critical Reynolds number, \( \text{Re}_{cr} \), and nonlinear shape constants \( \varepsilon_0 \) and \( \beta \) are empirically determined, and \( \text{Re}_{cr} \) represents the onset of the nonlinear viscous force.

In addition, based on an empirical equation [6], the yield stress of the MR fluid is determined to be:

\[
\tau_s = 271700 \Psi^{1.5239} \tanh(6.33 \times 10^{-6} H_f)
\]

(3)

where, \( \Psi \) is the volume fraction of the iron particles in the MR fluid. The magnetic field transverse to the MR fluid, \( H_f \), is known to be [2]:

\[
H_f = \kappa(\xi) \frac{\mu_s - \mu}{\varepsilon (\mu_s - \mu_f)} H, \quad \text{where} \quad \mu = \left[ \mu_f^{\psi/3} + (1 - \varepsilon) \mu_f^{\psi/3} \right]
\]

(4)

In equation (4), \( \mu_s \) and \( \mu_f \) are the magnetic permeability of the fillers and MR fluid, respectively, and \( \mu \) is an average magnetic permeability. External magnetic field, \( H \), is known to be a function of the coil number and valve length of the porous valve, and the corrector \( \kappa \) is used to represent the effect of the tortuous channel on the orientation of the magnetic filed applied to the MR fluid, and then is a function of the tortuosity. Notably, the magnetic field transverse to the MR fluid is affected by the permeability of the filler materials and the porosity and tortuosity of the porous valve.

To design an MR damper with a porous valve, maximum damper force at a maximum piston velocity and controllable range (the ratio of the total force to the viscous force) are usually given as design objectives, and the MR fluid is chosen in advance. In the current design strategy, optimized \( \varepsilon \) and \( \xi \) are determined to maximize an achievable yield stress by using equations (3) and (4), and then \( R_h \) can be known by using (1). To reduce the effect of the nonlinear viscous force on the controllable range, the valve area, \( A_p \), will be chosen by maintaining a lower \( \text{Re} \) than \( \text{Re}_{cr} \). Thus, the valve length, \( L \), and the piston area, \( A_p \), can be determined from the required force and controllable range by using equation (2).

4. Parameter identification and validation

To determine \( \varepsilon_0 \), \( \beta \) and \( \text{Re}_{cr} \), measurements of the MR damper with 3.5 mm magnetic steel beads in the porous valve [3] were used, such that the analytical force obtained by using equation (2) matched the experimental force with the same displacement excitation. Figure 3 shows an example of the force-velocity hysteresis comparison between analytical and experimental results. To validate the identified parameters, the experimental data of the MR damper with 2.0 mm and 5.5 mm magnetic steel beads in the porous valve [3] were compared with the analytical results. Equivalent damping, which was obtained by equating the dissipated energy over a cycle to the energy dissipated by an equivalent viscous damper, was used to characterize the damping performance and to evaluate the analytical and measured results. As shown in Figure 4, the equivalent damping as a function of the peak velocity for
three different beads are grouped together, and the experimental results match the analytical results quite well.

Based on the design strategy and the determined parameters, an MR damper with a porous valve was developed as shown in Figure 1. To validate the design strategy, the predicted damper performance is compared with the experimental results as shown in Figures 5 and 6, where the porous media in the valve is 3.5 mm and 5.5 mm magnetic steel beads, respectively.

5. Conclusions
A design strategy for the MR porous valve was developed. The effect of the nonlinear viscous force on the damper performance was evaluated. The parameters of the analytical model were empirically determined, and the design strategy was validated using experimental data.

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