Geometric Optimal Design of MR Damper Considering Damping Force, Control Energy and Time Constant

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Abstract. This paper presents an optimal design of magnetorheological (MR) damper based on finite element analysis. The MR damper is constrained in a specific volume and the optimization problem identifies geometric dimensions of the damper that minimizes an objective function. The objective function is proposed by considering the damping force, dynamic range and the inductive time constant of the damper. After describing the configuration of the MR damper, a quasi-static modelling of the damper is performed based on Bingham model of MR fluid. The initial geometric dimensions of the damper are then determined based on the assumption of constant magnetic flux density throughout the magnetic circuit of the damper. Subsequently, the optimal design variables that minimize the objective function are determined using a golden-section algorithm and a local quadratic fitting technique via commercial finite element method parametric design language. A comparative work on damping force and time constant between the initial and optimal design is undertaken.

1. Introduction
The suppression of mechanical and structural vibration using semi-active control method has been actively worked by many researchers in last two decades. Recently, various semi-active suspension systems featuring MR fluid have been proposed and successfully applied in the real field, especially in vehicle suspension systems [1-3]. Generally, the modeling and design of MR dampers have been done based on the quasi-static assumption of the damper [4,5]. In order to accurately model MR dampers, various dynamic models considering hysteresis behavior of MR damper have been also proposed [6,7]. However, researches on optimal design in order to improve performance of the MR dampers such as damping force, dynamic range and inductive time constant are still considerably rare.

Consequently, the main contribution of this research work is to find optimal dimensions of vehicle MR dampers considering an advanced objective function which includes damping force, dynamic range and inductive time constant altogether. In this study, the cylindrical MR damper for vehicle suspension proposed by Sung [3] is considered. After describing the configuration of the MR damper, a quasi-static modelling of the damper is performed based on Bingham model of MR fluid. The initial geometric dimensions of the damper are then determined based on the assumption of constant magnetic flux density throughout the magnetic circuit of the damper [8]. Subsequently, the optimal design variables that minimize the objective function are determined using a golden-section...
algorithm and a local quadratic fitting technique via ANSYS parametric design language. A comparative work on damping force and time constant between the initial and optimal design is undertaken.

2. Quasi-static modelling of the MR damper
The cylindrical MR damper for vehicle suspension proposed by Sung [3] is presented in figure 1. The outer and inner pistons are combined to form a MR valve structure which divides the MR damper into two chambers: the upper and lower chambers. These chambers are fully filled with the MR fluid. As the piston moves, the MR fluid flows from one chamber to the other through the annular duct (orifice) of the valve structure. The floating piston and gas chamber function as an accumulator to accommodate the damper rod volume as it enters and leaves the fluid chamber. By neglecting the frictional force and assuming quasi-static behavior of the damper, damping force ($F_d$) and dynamic range ($\lambda_d$) of the MR damper can be expressed as follows:

$$F_d = P_a A_s + c_{vis} \dot{x}_p + F_{MR} \text{sgn}(\dot{x}_p) ; \quad \lambda_d = \frac{P_a A_s + c_{vis} \dot{x}_p + F_{MR,max}}{P_a A_s + c_{vis} \dot{x}_p}$$

(1)

where gas chamber pressure $P_a$, the viscous coefficient $c_{vis}$ and the yield stress force $F_{MR}$ are given by

$$P_a = P_0 (\frac{V_0}{V_0 + A_s x_p})^\gamma ; \quad c_{vis} = \frac{12\eta L}{\pi R_d t_d^3} (A_p - A_s)^2 ; \quad F_{MR} = \frac{2c L_p}{t_d} \tau_y$$

In the above, $P_0$ and $V_0$ are initial pressure and volume of the accumulator, $\gamma$ is the coefficient of thermal expansion which is ranging from 1.4 to 1.7 for adiabatic expansion, $x_p$ is the piston displacement, $A_p$ and $A_s$ are respectively the piston and the piston-shaft effective cross-sectional area, $\tau_y$ is the yield stress of the MR fluid induced by the applied magnetic field, $\eta$ is the post yield viscosity of MR fluid, $L$ is the length of the inner piston, $L_p$ is the length of the magnetic pole, $R_d$ and $t_d$ are the average radius and gap of the annular duct, $c$ is the coefficient which depends on flow velocity profile, $F_{MR,max}$ is the yield stress force with the maximum applied current. In this work, the commercial MR fluid (MRF132-LD) from Lord Corporation is used. The induced yield stress of the MR fluid can be expressed as a function of the applied magnetic field intensity ($H_{ml}$) as follows:

$$\tau_y = p(H_{ml}) = C_0 + C_1 H_{ml} + C_2 H_{ml}^2 + C_3 H_{ml}^3$$

(2)

In Eq. (2), the units of $\tau_y$ and $H_{ml}$ are kPa and KA/m, respectively. The coefficients $C_0$, $C_1$, $C_2$, and $C_3$ are respectively identified as 0.3, 0.42, -0.0012 and 1.05E-6. In order to calculate the damping force it is necessary to solve magnetic circuit of the damper. In general, for improving accuracy of the magnetic circuit solution, FEM is employed. In this study, the commercial FEM software ANSYS is used. The FE model to solve the magnetic circuit of the damper using 2D-axisymmetric coupled element (PLANE13) is shown in figure 2. It is noted that in FEM solution, the magnetic field intensity and flux density are not constant along the pole length. Therefore, the magnetic flux, the average magnetic flux density and magnetic field intensity across the MR ducts are determined as follows:
\[
\Phi = 2\pi R_d \left( \int_0^{t_p} B_{mr}(s) ds - \int_0^{t_p} B_{mr}(s) ds \right); \quad B_{mr} = \frac{1}{L_p} \int_0^{t_p} B_{mr}(s) ds; \quad H_{mr} = \frac{1}{L_p} \int_0^{t_p} H_{mr}(s) ds
\]

(3)

where \(B_{mr}(s)\) and \(H_{mr}(s)\) are the magnetic flux density and magnetic field intensity at each nodal point on the defined path along the MR active volume where magnetic flux passes. The inductive time constant and control energy of the MR damper can be calculated as follows:

\[
T = \frac{L_m}{R_w}; \quad N = I^2 R_w
\]

(4)

where \(L_m\) is the inductance of the valve coil given by \(L_m = N_c \Phi / I\), \(R_w\) is resistance of the coil wire and \(N_c\) is the number of coil turns.

3. Optimization of the MR damper

In this study, FEM integrated with an optimization tool is used to obtain optimal values of design variables (DV) such as the coil width \(w_c\), the pole length \(L_p\), the MR orifice gap \(t_d\) and the outer piston thickness \(t_{op}\) which minimize an objective function. For vehicle suspension design, ride comfort and suspension travel are two conflicting performance indexes to be considered. In order to reduce the suspension travel, high damping force is required. On the other hand, for improving ride comfort, low damping force is expected. Thus a large dynamic range is required. Furthermore, a fast time response MR damper is also required to improve controllability of the suspension system. Taking the above requirements into account, the following objective function is proposed:

\[
OBJ = \frac{F_{MR,r}}{F_{MR}} + \frac{\lambda_{d,r}}{\lambda_d} + \frac{T}{T_r}
\]

(5)

where \(F_{MR,r}\), \(\lambda_{d,r}\) and \(T\) are respectively the reference damping force, dynamic range and inductive time constant of the damper which determined from the solution of MR damper at initial values of design parameters (DV). Based on widely used MR dampers, the initial value of \(t_d\) is chosen by 1mm. The values of other DV are determined based on the assumption of constant magnetic flux density throughout the magnetic circuit of the damper as follows [8]:

\[
w_c = \sqrt{R^2 - 4L_p^2} - 2L_p; \quad t_{op} = R - w_c - 2L_p
\]

(6)

In the above, the initial value of \(L_p\) is determined such that the magnetic flux density does not exceed the saturation values of the valve core material (silicon steel) which is 1.5 Tesla in this study. For the valve structure constrained in a cylinder of radius \(R=23\,mm\) and length \(L=50\,mm\), the initial value DV can be determined as follows: \(L_p=7.75\,mm\), \(w_c=1.5\,mm\), \(t_{op}=5\,mm\). In this study, the first order method implemented in the ANSYS optimization tool is used to find the optimal values of the design variables that minimize the proposed objective function.

Figure 3 shows the optimal solution of the MR damper based on FEM. In the optimization problem, the post-yield viscosity of the MR fluid is \(\eta=0.092\,Pas\), the piston velocity is 0.4m/s, the coil wires are sized as 24-gauge and the maximum applied current is \(I=2A\). The initial values of the...
yield stress force, dynamic range, inductive time constant and objective function are 2078N, 6.66, 48.5ms, and 3, respectively. From the figure, it is observed that the solution converges after 7 iterations at which the minimal value of the objective function is 2.376. At the optimum, the yield stress force, dynamic range, and inductive time constant are 2498N, 10.22 and 44.26ms, respectively. The optimal values of $L_p$, $w_c$, $t_d$ and $t_{op}$ are 21mm, 1.92mm, 1.1mm and 4.45mm. The results show that the yield stress force, dynamics range and conductive time constant are significantly improved. The power consumption of the damper is 2.64W at the optimum, which is much smaller than that at the initial (8.68W). In order to evaluate performance characteristics of the optimized MR damper in the applied current range, the damping force and inductive time constant of initial and optimized dampers at different values of applied current are presented in figure 4. The figure shows that the yield stress force and inductive time constant of the optimized damper are much better than those of the initial one at any value of the current. The figure also shows that, at high applied current value, the yield stress force approaches to saturation while the time constant is reduced in proportion to the applied current.

4. Conclusion

In this study, geometric optimal design of vehicle MR dampers based on FEM solution was performed using ANSYS parametric design language. The optimization problem was to identify geometric dimensions of the valve structure employed in the MR damper that minimizes an objective function. The objective function was proposed considering the damping force, dynamic range and the inductive time constant of the damper and their reference values. The reference values were determined based on the assumption of constant magnetic flux density throughout the magnetic circuit of the damper. It was shown that, by minimizing the objective function, the yield stress force, dynamics range and conductive time constant are significantly improved at any value of applied current. The power consumption of the optimized damper was also significantly reduced.

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