Optimal Design of a Magnetorheological Fluid Suspension for Tracked Vehicle

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Abstract. This paper presents optimal design of controllable magnetorheological suspension system (MRSS) for a tracked vehicle. As a first step, a double-rod type MRSS is designed on the basis of the Bingham model of commercially available MR fluid, and its damping characteristics are evaluated with respect to the intensity of the magnetic field. Subsequently, the governing equation of motion of the MRSS featuring the MR valve is established. Then, the optimization problem to find optimal geometric dimensions of the MRSS is formulated by considering an objective function which is related to damping torque and control energy. The first order optimization method integrated with a commercial finite element method (FEM) software is adopted to obtain optimal solution of the system. The performance characteristics of the optimized MRSS are then evaluated and compared with initial ones.

1. Introduction
Recently, various semi-active suspension systems featuring MR fluid have been proposed and successfully applied in vehicle suspension systems [1]. However, most of previous researches have focused on wheeled vehicles such as passenger vehicles [2]. In general, tracked vehicles are exposed to severe working environment such as rough road and complex ride behaviors. Therefore, high performance suspension systems are required for tracked vehicles. The primary function of the tracked vehicle’s suspension system is to isolate the vehicle body and its contents from terrain induced vibrations to permit high vehicle speed and steering stability [3]. Consequently, the main contribution of this paper is to show how to design the magnetorheological suspension system (MRSS) which is incorporated with the optimized MR valve. A new double-rod type MRSS is modeled and two-coil annular MR valve is designed. The optimization is undertaken using ANSYS and solutions such as damping torque are presented.

2. Modelling of MRSS
One of representative passive type suspension systems for tracked vehicle is in-arm type hydropneumatic suspension unit (ISU). Some benefits of the ISU include weight savings, greater load levelling capability and easiness of height control. In this work, the passive double-rod-type ISU is
modified in order to devise the MRSS and its schematic configuration is shown in figure 1. It is seen that an MR valve is attached to the manifold part of the passive ISU. The manifold of the MRSS consists of the MR valve and two-coil annular valve. Above a certain pressure difference between actuating and accumulating cylinders, the MR valve needs to be activated by the magnetic field to achieve desired damping torque. To facilitate calculation of damping torque of the proposed MRSS, consider the coordinate system shown in figure 2. When the MR fluid flows through the MR valve, the pressure drop can be expressed by

\[ \Delta P_c = \Delta P_{MR} + \Delta P_{mv} \]  \hspace{1cm} (1)

where \( \Delta P_{MR} \) is the pressure drop due to the field dependent yield stress of the MR fluid, and \( \Delta P_{mv} \) is the pressure drop due to the viscosity of the MR fluid.

Figure 3 shows a simplified structure of the two-coil annular MR valve constrained specific volume. The valve geometry is characterized by the valve height \( L \), the valve radius \( R \), the valve housing thickness \( d_h \), the MR channel gap \( h \), the iron flange thickness \( t \) and the distance of between upper coil and lower coil \( a \). When the electric current is applied to the coil, a magnetic circuit appears as shown in the figure 3. The pressure drop of the yield stress and viscosity are calculated by [4]

\[ \Delta P_{MR} = 2ct \tau / h + ca \tau_a / h \]  \hspace{1cm} (2)
\[ \Delta P_{mv} = 6 \eta Q / \pi h^3 R_i \]  \hspace{1cm} (3)

where \( c \) is a coefficient which depends on the flow velocity profile, \( \eta \) is the base viscosity, \( Q \) is the flow rate through the MR valve and \( R_i \) is the average radius of the annular duct.

The pressure drop \( \Delta P_0 \) due to the orifice can be also obtained as

\[ \Delta P_0 = \rho g h_i \]  \hspace{1cm} (4)

where \( \rho \) is the density of the MR fluid, \( g \) is the acceleration due to gravity and \( h_i \) is the head loss including the major and the minor losses of the orifice. Thus, the damping torque at the rotational center of the wheel arm is expressed by

\[ T_d = l_{sh} \sin(87 - \alpha + \beta) F_d / \cos(\beta) \]  \hspace{1cm} (5)

where \( F_d \) is the damping force at the actuating piston given by

\[ F_d = \pi \rho \Delta P_c (\Delta P_c + \Delta P_0) \]  \hspace{1cm} (6)

3. Optimization of MRSS
In this study, ANSYS integrated with an optimization tool is used to obtain optimal geometric dimensions of the MR valve of MRSS in order to minimize an objective function. The optimal objective is to minimize the valve ratio defined by the ratio of the viscous pressure drop to the field-dependent pressure drop of the MR valve. This ratio has a large effect on the characteristics of the MR
valve. It is desirable that the valve ratio takes small value. From equations (2) and (3), the valve ratios of the two-coil MR valve is calculated by

\[ \lambda = \frac{\Delta P_m}{\Delta P_{MR}} = 3\eta LQ_0 / \pi h^2 R_c (t \tau_1 + 0.5a \tau_2) \]  

(7)

The yield stress of the MR fluid caused by the magnetic circuit is calculated from the approximate polynomial curve of the yield stress. This curve can be obtained as [5]

\[ \tau (kPa) = 52.962B^4 - 176.51B^3 + 158.79B^2 + 13.708B + 0.1442 \]  

(8)

where \( \tau \) is the yield stress caused by the applied magnetic field and \( B \) is the magnetic flux density of the applied magnetic field. To achieve optimal design parameters of the MR valve, the first-order method of the ANSYS optimization tool is used. The ANSYS optimization tool transforms the constrained optimization problem into the unconstrained one via penalty functions. The dimensionless, unconstrained objective function is formulated as follows;

\[ \lambda^*(y, p) = \lambda_i / \lambda_0 + \sum_{i=1}^{n} P_i(y_i) + q \sum_{i=1}^{m} P_x(g_i) \]  

(9)

where \( \lambda_i \) is the objective function, \( \lambda_0 \) is the reference objective function, \( q \) is the response surface parameter, \( P_i \) is the exterior penalty function and \( P_x \) is the extended-interior penalty function.

For the initial iteration, the search direction of design variables is assumed to be the negative of the gradient of the unconstrained objective function. The direction vector is calculated by

\[ d^{(0)} = -\nabla \lambda^*(y^{(0)}, 1) \]  

(10)

The values of design variables in the next iteration is calculated by line search parameter \( s_j \).

\[ y^{(j+1)} = y^{(j)} + s_j d^{(j)} \]  

(11)

In the subsequent iterations, the procedures are similar to those of the initial iteration except that the direction vectors are calculated according to the Polk-Pibiere recursion formula as follows;

\[ d^{(j)} = -\nabla \lambda^*(y^{(j)}, q) + u_{j-1}d^{(j-1)} \]  

(12)

\[ u_{j-1} = \frac{\{\nabla \lambda^*(y^{(j)}, q) - \nabla \lambda^*(y^{(j-1)}, q)\}^T \nabla \lambda^*(y^{(j)}, q) / |\nabla \lambda^*(y^{(j-1)}, q)|^2} \]  

(13)

4. Results and discussion

The valves are constrained in a cylinder of radius \( r = 23 \text{mm} \) and height \( L = 120 \text{mm} \). The base viscosity of the MR fluid is assumed to be constant, \( \eta = 0.092 \text{Pas} \) and the flow rate of the MR valves
is \( Q_0 = 3 \times 10^{-4} \text{m}^3\text{s}^{-1} \). In this study, the valve gap is chosen as \( 1\text{mm} \). The current density applied to the coils can be approximately calculated by

\[
J = \frac{I}{A_w}
\]

where \( I \) is current applied to the coils and \( A_w \) is the cross section of the coil wire. To evaluate power consumption of the valve coils, \( N \) is calculated by the following equation.

\[
N = I^2 R_w
\]

where \( R_w \) is the resistance of the coil wire which can be approximately calculated as follow;

\[
R_w = L_w r_w = V_c r_w / A_w
\]

where \( L_w \) is the length of the coil wire, \( r_w \) is the resistance per unit length of the coil wires and \( V_c \) is the volume of all coils of the MR valve. The valve core radius \( R_c \), the iron flange thickness \( t \) and the valve housing thickness \( d_h \) are calculated by the following equations.

\[
R_c = \frac{1}{2} \cdot \left\{ \sqrt{2R^2 - (W + h)^2} - (W + h) \right\}
\]

\[
t = R_c / 2
\]

\[
d_h = R - (W + h + R_c)
\]

Figure 4. Optimization results of the MR valve

Figure 5. Damping Characteristic of the MRSS
Figure 4 shows the optimal solution of a two-coil annular MR valve which is constrained in the specific volume when a current of 2.0A is applied to the valve coils. The valve ratio, pressure drop and power consumption at these initial values are $\lambda_0 = 0.087$, $\Delta P_{int} = 42.47\text{bar}$ and $N_o = 24.51\text{W}$, respectively. Optimal design variable are determined by $t_{opt} = 13.88\text{mm}$, $W_{opt} = 1.5\text{mm}$ and $d_{h,opt} = 5.53\text{mm}$. At these optimal design variables the power consumption is $N_{opt} = 21.08\text{W}$ and the difference between the magnetic flux density of the outer and inner duct is 0.0398T which is small and acceptable.

As shown in figure 5, the damping torque of MRSS is compared between initial one and optimized one. It is evident that damping torque is increased as the magnetic field increased. It is also observed that the damping torque of optimized MRSS is larger than initial ones for the same control input. This means that the optimized MRSS has good performance in terms of energy efficiency.

5. Concluding remarks
In this work, a controllable MRSS has been proposed for tracked vehicles. A double-rod-type MRSS was devised and its damping characteristics were analysed. An optimization procedure based on the ANSYS has been developed in order to find the optimal geometry of MR valve employed in the MRSS. Optimal solutions have been obtained and the performance of the MRSS such as damping torque has been analysed and presented.

References
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