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Keywords

its, system, vehicle, damper, control, mr, semi, application, suspension, active

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Technical Note

MR damper and its application for semi-active control of vehicle suspension system

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Abstract

In this paper, a semi-active control of vehicle suspension system with magnetorheological (MR) damper is presented. At first a MR damper working in flow mode is designed. Performance testing is done for this damper with INSTRON machine. Then a mathematical model, Bouc–Wen model, is adopted to characterize the performance of the MR damper. With optimization method in MATLAB[®] and experimental results of MR damper, the coefficients of the model are determined. Finally, a scaled quarter car model is set up including the model of the MR damper and a semi-active control strategy is adopted to control the vibration of suspension system. Simulation results show that with the semi-active control the vibration of suspension system is well controlled. © 2002 Elsevier Science Ltd. All rights reserved.

1. Introduction

Vibration control of vehicle suspension systems has been a very active subject of research, since it can provide a very good performance for drivers and passengers. For a long time, efforts were done to make the suspension system works in an optimal condition by optimizing the parameters of the suspension system, but for intrinsic limitation of passive suspension system the improvement is effective only in a certain frequency range. Compared with passive suspensions, active suspensions can improve the performance of the suspension system over a wide range of frequency. Semi-active suspensions were proposed in the early 1970s [1], which can be nearly as effective as fully active suspensions in improving ride quality. When the control

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system fails, the semi-active suspension can still work in passive condition. Compared with active and passive suspension systems, the semi-active suspension system combines the advantages of both active and passive suspensions; i.e. it provides good performance compared with passive suspensions and is economical, safe and does not require either higher-power actuators or a large power supply [2].

In early semi-active suspension, the regulating of the damping force can be achieved by adjusting the orifice area in the oil-filled damper, thus changing the resistance to fluid flow, but the changing of speed is much slow for using of mechanical motion. More recently, the possible applications of electrorheological (ER) and magnetorheological (MR) fluids in the controllable dampers were investigated by many researchers [3,4]. ER and MR fluids are two kinds of smart materials, which made by mixing fine particles into a liquid with low viscosity. The particles will be formed into chain-like fibrous structures in the presence of a high electric field or a magnetic field. When the electric field strength or the magnetic field strength reaches a certain value, the suspension will be solidified and has high yield stress; conversely, the suspension can be liquified once more by removal of the electric field or the magnetic field. The process of change is very quick, less than a few milliseconds, and can be easily controlled. The energy consumption is also very small, only several watts. Both ER and MR fluids were initially developed independently in the 1940s [5,6]. Initially it was ER fluids that received the most attention, but were eventually found to be not as well suited to most applications as the MR fluids. In their non-activated or “off” state, both MR and ER fluids typically have similar viscosity, but MR fluids exhibit a much greater increase in viscosity, and therefore yield strength, than their electrical counterparts. For ER fluid, the maximum yield stress is about 10 kPa; but for MR fluid, the maximum yield stress can reach about 100 kPa.

The application of ER damper in vibration control of vehicle suspension system was investigated by many researchers [7–10]. As for MR damper, a research group at Virginia Tech [11] evaluated the response of a MR vehicle suspension under different control schemes of sky-hook, ground-hook, and hybrid semi-active control. They evaluated the performance of semi-active MR suspension for a quarter car model test facility, as well as for a heavy truck on road.

In order to characterize the performance of the MR damper, several models were proposed by many investigators [12–14]. Spencer et al. [12] proposed a modified Bouc–Wen model to describe the MR damper behavior. This model can accurately capture both the force–displacement and the force–velocity hysteresis loops, which involves as many as 14 parameters. Kamath and Wereley [13] developed an augmented six-parameter model to accurately describe both the force–displacement and the force–velocity hysteresis cycles, which is constructed using a nonlinear combination of linear mechanisms. Li et al. [14] also proposed a nonlinear viscoelastic–plastic model to describe the dynamic behaviors of the MR damper, however this model cannot accurately capture the smooth transition from pre-yield to post-yield region.

In this paper, a MR damper is designed and fabricated first, and then a non-parametric model for the damper is constructed and parameter estimation is done for the MR damper based on the experimental results. The model results

are compared with those of experimental results. A half-scale quarter car model is established with the model of the MR damper and the governing equation is obtained for the suspension system. Finally a semi-active control strategy, sky-hook control is adopted to control the vibration of suspension system over random road excitation. Simulation is carried out and results are compared with those of passive suspension. The potential application of MR damper in vehicle suspension system is proved.

2. MR damper and parameters estimation

2.1. Design of MR damper and experimental setup

The prototype MR damper works in flow mode as shown in Fig. 1. The damper is 218 mm long in its extended position, and has ± 25 mm stroke. The main cylinder houses a piston, a magnetic circuit, an accumulator and MR fluid. MR 132 LD, which was obtained from Lord Corporation, is used in the damper [15]. The MR fluid valve is contained within the piston and consists of an annular flow channel with 1.5 mm gap. The magnetic field is applied radially across the gap, perpendicular to the direction of fluid flow. The total axial length of the flow channel is 6 mm which exposed to the applied magnetic field. Viscosity of MR fluid in the valve will be increased by increasing the electric current through the electromagnet, thus resisting the MR fluid flow through the valve and increasing the damping force of the MR damper. The resistance of the electromagnet coil is $19\ \Omega$.

To apply the MR damper in vibration control of vehicle suspension system, the property of the damper should be determined first and then a model must be developed that can accurately reproduce the behavior of the MR damper. An experimental test rig is set up to determine the property of the MR damper and to obtain

Fig. 1. Schematic drawing of MR damper.

the dynamic data necessary for estimating the parameters of the model. In this test rig, the MR damper is fixed in a computer-controlled INSTRON Test Machine (Model 8874). The INSTRON Test Machine incorporates a load cell and a displacement sensor to measure the force produced by the MR damper and the displacement of the piston. Two types of excitations, sinusoidal and triangular, are used. The excitation frequencies are 1, 2 and 4 Hz and the amplitudes of excitation are 1, 2 and 4 mm, respectively. The applied electric current is from 0 to 1 A with increment of 0.25 A. The force and displacement responses of the damper are sampled simultaneously by the computer via an A/D converter. The excitation signal is also produced by the computer and sent out to the hydraulic actuator via a D/A converter. Velocity response can be obtained by differentiating the displacement. All experiments are carried out at the room temperature of 23°C.

2.2. Experimental results

The responses of MR damper at 1 Hz excitation under five constant electric currents are shown in Fig. 2. The effect of magnetic field on the damping force is clearly shown in these figures. With the increasing of the applied electric current, the damping force will increase remarkably, however when the applied electric current is more than 0.75 A, the increase of the damping force is no longer significant. This means that saturation of the MR effect occurs at 0.75 A. It is also noted that the force produced by the damper is not exactly centered at zero. This is due in part to the presence of an accumulator in the MR damper, which is filled with highly compressed air, and in part due to the existence of air in the cylinder since the damper cannot be fully filled with MR fluid. The maximum force of MR damper at 1 A is approximately equal to eight times of that without electric field. Similar results can be obtained from experiment at the other excitation frequencies and amplitudes.

To obtain the relation of equivalent damping coefficient to velocity and electric current, experiments are done under the triangular excitation. The equivalent damping coefficient of the damper against velocity under various electric currents is shown in Fig. 3. It is seen that at low velocity, equivalent damping coefficient will increase markedly. As the velocity increases, the equivalent damping coefficient

Fig. 2. Responses of (a) force vs. time (b) force vs. displacement under different electric currents, 0, 0.25, 0.5, 0.75 and 1 A, $x_0 = 4$ mm, $f = 1$ Hz.

Fig. 3. Equivalent damping coefficient vs. velocity under various electric currents.

under high electric current decreases rapidly whereas that without electric current decreases slowly. At high velocity, the effect of current on equivalent damping coefficient is also not so significant. This phenomenon means that the MR damper cannot be treated as a viscous damper under high electric current.

From the experiments, it is seen that the designed MR damper has very large changeable damping force range under magnetic field, although the saturated magnetic field is not so big. An improvement should be proposed to increase the saturated magnetic field and to avoid the side effect of the accumulator. Furthermore since the MR damper cannot be treated as a viscous damper under high electric current, a suitable model is necessary to be developed to describe the MR damper.

2.3. Bouc–Wen model and parameters estimation

For application of MR damper in vibration control, the model of the MR damper should be continuous in all the ranges and be numerically tractable, and the Bouc–Wen model is adopted here. The schematic model is shown in Fig. 4. The force in this system is given by

$$F = c_0 \dot{x} + k_0 x + \alpha z, \quad (1)$$

Fig. 4. Bouc–Wen model.

Table 1
 c_0 and α against electric current

Displacement	Parameter	Current (A)				
		0	0.25	0.5	0.75	1
2 (mm)	c_0 (N s/mm)	0.78	2.40	3.477	4.12	4.5
	α (N/mm)	5.63	4.23	6.877	7.95	8.3
4 (mm)	c_0 (N s/mm)	0.65	1.65	2.2	2.50	2.9
	α (N/mm)	5.63	4.102	6.877	7.95	8.3

where the evolutionary variable z is governed by

$$\dot{z} = -\gamma|\dot{x}|z|z|^{n-1} - \beta\dot{x}|z|^n + A\dot{x}, \quad (2)$$

where γ , β and A are parameters related to the shape of hysteresis loop. By adjusting the parameters γ , β and A , of the model it is possible to control the linearity in the unloading and the smoothness of the transition from the pre-yield to the post-yield region.

To estimate the parameters of the model, an error function is introduced as an objective function

$$J = \sum_{i=1}^N (F_{ei} - F_{pi})^2, \quad (3)$$

where F_e and F_p are experimental damping force and estimated damping force as defined in Eq. (1) and N is the number of experimental data. The optimization is carried out using MATLAB[®] constraint optimization toolbox. In the optimization, the inputs to the system are measured displacement, velocity obtained by differentiating the displacement and measured force.

The estimated parameters at 1 Hz excitation are $r = 4.0 \text{ mm}^{-2}$ (no applied current) and $r = 0.1 \text{ mm}^{-2}$ (current applied), $A = 180$, $k_0 = 0$, $\beta = 0$ and $n = 2$, while c_0 and α are listed as in Table 1. From Table 1, it is seen that the damping coefficient c_0 and α increase with electric current. However, while α remain nearly constant for both displacements, c_0 decreases with the displacement.

Fig. 5 shows comparison of the estimated responses at 1 Hz, 4 mm and under various electric currents to the corresponding experimental responses. It is seen that the estimated model can capture the properties of the MR damper except at the region where the velocity is near zero. This may be due to the accumulator and the asymmetric property since the MR damper was not fully filled with MR fluid. So the Bouc–Wen model can be used to characterize the property of the MR damper.

3. Semi-active control of vehicle suspension system

3.1. Vehicle suspension model

In this analysis a simple quarter car model with a MR damper being installed in suspension system will be used as shown in Fig. 6. This is a two-degrees-of-freedom

Fig. 5. Comparison of estimated responses with corresponding experimental responses: (a) force vs. time; (b) force vs. displacement; (c) force vs. velocity.

Fig. 6. Simplified quarter car model.

system, mass m_2 represents the sprung mass while mass m_1 means the unsprung mass; k_2 represents the stiffness of suspension system and k_1 means the stiffness of tire. The property of the MR damper is determined by Eq. (1), Bouc–Wen model. x_0 is road

excitation. In this study, the road excitation is a stationary random process with zero mean value as described in [16].

According to dynamic analysis, the governing equation of quarter car model can be obtained as:

$$\begin{aligned} m_1 \ddot{x}_1 - c_0(\dot{x}_2 - \dot{x}_1) - k_2(x_2 - x_1) + k_1(x_1 - x_0) &= \alpha z, \\ m_2 \ddot{x}_2 + c_0(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) &= -\alpha z, \end{aligned} \quad (4)$$

where the evolutionary variable z is governed by

$$\dot{z} = -\gamma|\dot{x}_2 - \dot{x}_1|z|z|^{n-1} - \beta|z|^n(\dot{x}_2 - \dot{x}_1) + A(\dot{x}_2 - \dot{x}_1). \quad (5)$$

Let $\mathbf{x} = [x_1 \ x_2]^T$. Then Eq. (4) can be written in matrix form as

$$\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F}y, \quad (6)$$

where

$$\begin{aligned} \mathbf{C} &= \begin{bmatrix} \frac{c_0}{m_1} & -\frac{c_0}{m_1} \\ -\frac{c_0}{m_2} & \frac{c_0}{m_2} \end{bmatrix}, \\ \mathbf{K} &= \begin{bmatrix} \frac{k_1 + k_2}{m_1} & -\frac{k_2}{m_1} \\ -\frac{k_2}{m_2} & \frac{k_2}{m_2} \end{bmatrix}, \\ \mathbf{F} &= \begin{bmatrix} \frac{a}{m_1} & \frac{k_1}{m_2} \\ -\frac{a}{m_2} & 0 \end{bmatrix} \quad \text{and} \quad \mathbf{y} = [z \ x_0]^T. \end{aligned}$$

Let $\mathbf{X} = [\dot{\mathbf{x}} \ \mathbf{x}]^T$. Then Eq. (6) can be written in state space as

$$\dot{\mathbf{X}} = \mathbf{A}\mathbf{X} + \mathbf{B}\mathbf{u}, \quad (7)$$

where

$$\mathbf{A} = \begin{bmatrix} \mathbf{C} & \mathbf{K} \\ \mathbf{I} & \mathbf{0} \end{bmatrix}, \quad \mathbf{B} = \begin{bmatrix} \mathbf{F} & \mathbf{0} \\ \mathbf{0} & \mathbf{0} \end{bmatrix} \quad \text{and} \quad \mathbf{u} = [y \ 0 \ 0]^T.$$

3.2. Control strategy

A semi-active control method, sky-hook control is introduced in this study to demonstrate the application of MR damper in the vibration control of vehicle suspension system. The state variables are the relative velocity between the sprung mass and the unsprung mass, as well as the velocity of sprung mass.

Semi-active sky-hook control policy can be described using Eq. (8). When the relative velocity between sprung mass and unsprung mass is in the same direction of the velocity of sprung mass, an electric current is applied to the MR damper, otherwise no damping force is required. But for MR damper it is impossible to provide a

zero force, so we should minimize the semi-active damping force without any electric current.

$$\begin{cases} \dot{x}_2(\dot{x}_2 - \dot{x}_1) > 0, & F = \text{Max}, \\ \dot{x}_2(\dot{x}_2 - \dot{x}_1) < 0, & F = \text{Min}. \end{cases} \quad (8)$$

3.3. Simulation results

For example, an electric current 0.25 A is chosen as the current applied to MR damper. The simulation is carried out with SIMULINK of MATLAB®. For comparison, a constant control is also introduced in the simulation with a constant current of 0.25 A being applied to the MR damper all the time. The parameters used in the simulation of the quarter car model are $m_2 = 221$ kg, $m_1 = 31$ kg, $k_2 = 14.23$ kN/m, $k_1 = 122.5$ kN/m, which to simulate a half-scale quarter car. In the simulation, we assume that MR damper responds very fast and the time delay is ignored.

Fig. 7 shows the acceleration response of sprung mass under different control strategies. It is seen that under constant control the acceleration in the frequency range between body resonance and wheel hop is very big compared to those with passive control and semi-active control. Acceleration response around the body resonance is isolated effectively using semi-active control but that around the wheel hop increases slightly.

Fig. 8 shows suspension travel response under different control strategies. It is seen that suspension travel response around the body resonance is reduced significantly under constant control and semi-active control, but they are unable to reduce the suspension travel response around the wheel hop. Fig. 9 shows tire deflection response under different control strategies. It is seen that similar results can be obtained like those from Fig. 7. Under semi-active control the tire deflection around the body resonance is reduced very effectively, but that between body resonance and wheel hop increases slightly compared with that under passive control.

From the simulation results, it can be seen that the semi-active control provides improved performances compared with those of the passive control and constant control.

Fig. 7. Acceleration response of sprung mass.

Fig. 8. Suspension travel response.

Fig. 9. Tire deflection response.

4. Conclusions

From the experiment investigation for the MR damper, it has been shown that the MR damper has a very broad changeable damping force range under magnetic field and the damping coefficient increases with the electric current, but decreases with excitation amplitude. The MR damper will become saturated as the applied electric current reaches a certain value. Under electric current, the MR damper cannot be treated as a viscous damper, but the property of the MR damper can be described with the Bouc–Wen model.

From the simulation results for a quarter car with MR damper, it has been shown that with semi-active control the acceleration response of sprung mass, suspension travel and tire deflection are effectively controlled around the body resonance. The semi-active control strategy is superior to both the passive control and the constant control strategies. The possible application of the MR damper in semi-active vibration control of vehicle suspension system is demonstrated.

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