Investigation of an Automobile magnetorheological damper with asymmetric mechanical characteristics

Xianyu Song, Xiaomin Dong*, Maosen Yan and Xin Li

School of Mechanical Engineering, Chongqing University, Chongqing, Shapingba District, Chongqing 400044, China

*Corresponding author’s e-mail: xmdong@cqu.edu.cn

Abstract. In order to reduce the cost of sensors in automotive magnetorheological suspension systems and improve system reliability, a structural design scheme of automotive magnetorheological dampers with asymmetric mechanical characteristics is proposed. In this structure, the rebound damping coefficient with or without the action of the magnetic field will be greater than the compression damping coefficient, which solves the influence of the traditional magnetorheological damper limited by its own adjustable range, and continuously outputs the passive damper under different working conditions. According to the structural design plan, the mechanical output characteristics of the system are analysed theoretically, and the prototype machine is processed and tested. The research results show that the designed magnetorheological damper has the working characteristics of continuously outputting asymmetric damping force, which verifies the effectiveness of the design ideas and methods.

1. Introduction
Magnetorheological damper (MRD), as a typical semi-active suspension system control device, has the characteristics of simple structure, large damping force, wide adjustable range, fast response speed, and low power consumption. And the effect is similar to that of active suspension with lower cost [1]. Under normal circumstances, in order to ensure the driving safety and ride comfort of the car, the rebound damping coefficient of the traditional car suspension damper will be greater than the compression damping coefficient, but its characteristic parameters can only be designed for specific working conditions. When the actual working conditions deviate from the design working conditions, the vibration reduction effect will be greatly reduced [2]. M. Silveira et al. compared the performance of symmetrical (linear) and asymmetrical (non-linear) dampers, and the results showed that using an asymmetric system for vibration and shock absorption is a better choice for passenger cars [3,4].

The synthesis of optimal damper characteristics in compression and rebound has been a difficult task, which is mainly due to the coupling between different performance indicators and the complex dependence on various parameters of asymmetric multi-stage dampers [5]. MRD can continuously output the working characteristics of passive damper under different working conditions, which can not only improve the performance of the vehicle suspension, but also reduce the number of sensors and reduce the complexity of the control algorithm, thereby reducing costs and Improve reliability. Yu Jianqiang et al. proposed an automotive MRD with asymmetrical mechanical characteristics, which uses valve plates and springs to adjust the size of the entrance and exit of the annular passage between the piston and the cylinder during the rebound and compression movement to achieve asymmetric characteristics [2]. Michal Makowski et al. proposed a valve-type MRD to be applied to the Polish Warrior 2000 off-road vehicle. In numerical and experimental studies, a lower rate of change of wheel
force can be obtained through MRD and control algorithms to improve riding comfort [6]. Others mostly use algorithms to achieve asymmetric behaviour or consider temperature and other environments to produce asymmetric mechanical behaviour on the damper [7].

This paper proposes a structural design of an asymmetric MRD to adaptively adjust the asymmetric damping ratio to match the required damping characteristics of the suspension, which can improve car riding comfort and reduce traditional MRD’s asymmetric damping ratio cannot be further adjusted due to its own adjustable range. Furthermore, the feasibility of the design is analysed by developing prototypes and conducting experiments.

2. The principle and theoretical analysis of valve-type magnetorheological damper

2.1. Structural design and working principle

Based on the design of the vehicle's magnetorheological semi-active suspension damper in the laboratory, the magnetorheological damper structure shown in Figure 1 is designed. The function of the valve plate and the spring is to change the flow passage of the annular passage between the piston and the cylinder during the rebound and compression movement. The piston divides the cylinder into a rebound cavity and a compression cavity. During the compression movement, the MR fluid flows from the compression chamber to the rebound chamber through the annular channel. The pressure difference between the two chambers is greater than the pre-force of the spring. The spring is compressed, the valve plate opens, and the magnetorheological fluid flows through the middle of the valve cylinder. On the contrary, during the rebound movement, under the combined action of the spring and the pressure difference, the valve plate is reset, and the magnetorheological fluid flows through the throttle hole of the valve cylinder after the magnetorheological fluid is in the annular channel. Due to the increased head loss, the rebound force is greater than the compression force. The damping force is adjusted by changing the magnetic field at the annular channel. The piston compensation cylinder of the damper structure ensures that the magnetorheological vibration damping device does not have a sudden change in the damping force value at the turning point of compression and restoration, and realizes a smooth transition of the damping force value.

![Figure 1. Structural schematic diagram of magnetorheological damper](image)

2.2. Theoretical analysis of mechanical properties

The MRD uses the magnetic field generated by the coil current to change the flow behaviour of the magnetorheological fluid in the gap between the piston and the piston cylinder, thereby effectively controlling the damping force. Rheological experiments show that MR fluid behaves as a quasi-Newtonian fluid at very low shear rates. The Bingham model is used to describe its mechanical behaviour. Combining the plate model and the principle of fluid mechanics continuity, the effect of the valve structure can be obtained. Maximum damping force of annular channel:

\[
F_{\text{annu}} = F_\eta + F_v |\text{sgn}(v)|
\] (1)
where $F_\eta$ is the viscous damping force, and $F_\tau$ is the controllable damping force. Considering the fluid characteristics of a single flow channel as a fixed parallel plate gap flow, the viscous damping force can be obtained

$$F_\eta = \frac{12 \eta L A_p}{dd^2} v$$

(2)

And the controllable damping force obtained due to the magnetic shear yield stress is calculated as [8]

$$F_\tau = \frac{c L_m A_p}{dd} \tau_y$$

(3)

where $\eta$ is the viscosity of the liquid after yield, which is not related to the magnetic field, which is assumed to be a fixed value in the Bingham model; $c$ is the damping coefficient of the magnetorheological fluid, with a value of 2~3; $v$ is the velocity through the damping channel; $L_m$ is the working length of the magnetorheological effect corresponding to the channel; $\tau_y$ is the shear yield stress of the magnetorheological effect under the corresponding damping channel.

2.2.1. Theoretical Analysis of Compression Damping Force.

As shown in Figure 1, during the restoration stroke, the magnetorheological fluid in the restoration chamber is squeezed and flows through the annular channel. At this time, the valve plate is closed and flows to the compression chamber through the throttle hole structure at the valve barrel. The output damping force $F_{com}$ is expressed as:

$$F_{com} = F_{annu} + F_{big\_hole} + F_k$$

(4)

Where $F_{annu\_com}$, $F_{big\_hole}$, $F_k$ are the damping force due to MR fluid via the channel, the damping force induced by MR effect via the valve barrel, the damping force of the spring assembly respectively.

$$F_{big\_hole} = \rho g h_{big\_hole}$$

(5)

$$h_{big\_hole} = \left(\xi_i + \xi_e\right) \frac{v_{big\_hole}^2}{2g}$$

(6)

$$F_k = k \Delta x$$

(7)

where, $h_{big\_hole}$ is the loss head; $v_{big\_hole}$ is the velocity of the MRF via the valve barrel, $\xi_i$, $\xi_e$ is the loss coefficient of import and export respectively. Meanwhile, $k$ is the spring stiffness coefficient, $\Delta x$ is displacement of the spring.

2.2.2. Theoretical Analysis of rebound Damping Force.

As shown in Figure 1, during the rebound stroke, the magnetorheological fluid in the rebound cavity is squeezed, the valve plate is closed, and flows to the compression cavity through the orifice holes structure at the valve barrel. The output damping force $F_{reb}$ is expressed as:

$$F_{reb} = F_{annu} + F_{orifice}$$

(8)

Where $F_{orifice}$ is the damping force due to MR fluid via the valve’s orifice holes. And the head loss for orifice holes is represented as
orifice \( v \) is the velocity of the MRF via the orifice holes, \( \xi^i \), \( \xi^e \) is the loss coefficient of import and export respectively, which is different from \( \xi^i \), \( \xi^e \).

Since the size of the hole in the rebound stroke is much smaller than the size of the hole through the valve barrel during compression

\[
v_{\text{orifice}} \gg v_{\text{big hole}}, \quad h_{\text{orifice}} \gg h_{\text{big hole}}
\]

The compression damping force is less than the rebound damping force in the structure. Therefore, the damping force with or without current under compression and rebound, as shown in Figure 2.

![Figure 2. Damping force characteristics of MR damper](image)

3. Test Result and Discussion

3.1. Experimental set-up

In order to verify the performance of the magnetorheological damper, a prototype MR damper was manufactured, and a vibration test was carried out on the relationship of the damping force with different excitation speeds and currents. The experimental setup which is mainly composed of MTS System (figure 3), DC power, PC, etc. The test uses sinusoidal excitation, the test amplitude is 25mm, the speed is 0.1m/s, 0.3m/s, 0.6m/s, 0.7m/s, the excitation current is 0~1.6A, and the step size is 0.4A. Meanwhile, in order to avoid the influence of temperature on the test, air-cooled cooling is performed at intervals of about 15 minutes for every three groups of tests. Sufficient cycles have been measured for each single testing case to ensure the performance stability and uniformity. All the experiments were carried out at the room temperature of 25°C.
3.2. Result and Discussion

Figure 4 shows the damping force vs. displacement curve and damping force vs. velocity curve of the damper corresponding to different currents when the excitation velocity is 0.3m/s. It can be seen that the rebound force is greater than the compression force, and both the compression and rebound damping force values increase with the increase of the current (magnetic field), which has a certain controllability.

Figure 5 shows the force-displacement curve without current and the current is 0.8A. It can be found that there is obvious asymmetry with or without current. Figure 5(a) shows that even when the current fails, the MRD can still achieve asymmetry instead of the symmetrical behaviour of the traditional MRD. The experimental results are consistent with the design scheme.

![Figure 3. Experimental setup for testing the MRD](image)

(a) damping force vs. displacement  
(b) damping force vs. velocity.

Figure 4. Damping force with different excitation current
The asymmetry coefficient of the suspension damper refers to the ratio of the compression force to the rebound force. This value fluctuates greatly with the change of the working speed, but all are between 0 and 1. Figure 6 shows the variation of the asymmetry coefficient of the designed magnetorheological damper with current and frequency. It can be seen from Figure 6 that as the working speed increases, the asymmetry coefficient gradually decreases. This is because the asymmetric damping force is mainly affected by the throttling loss part of the viscous damping force. As the speed increases, the throttling of the compression stroke the increase in loss of specific gravity has a small effect on the throttling loss of the return stroke caused by the valve plate. As the current increases, the asymmetry coefficient will gradually increase. This is because with the increase of the current, the viscous damping force of the compression stroke and the rebound stroke will increase, so the asymmetric damping characteristic will become weaker. Although the asymmetry coefficient will decrease with the increase of frequency and current, its value is still between 0 and 1, which shows that the design method in this paper is reasonable.

4. Conclusions
This paper analyses the structure of the magnetorheological damper for automobile suspension with asymmetric mechanical characteristics from the perspective of structural design. A prototype was developed and a vibration test was carried out. The relationship between the damping force and the change of different excitation speeds and currents was tested. The results show that the rebound force is greater than the compression force, and both the compression and rebound damping force values increase with the increase of the current (magnetic field).

As the speed increases, the head loss caused by the size of the orifice increases quadratically, and the proportion of the asymmetric force affected by the head loss in the output damping force increases, and the asymmetric damping characteristics will become stronger; at the same time as the current increases, the viscous damping force of the compression stroke and rebound stroke will increase, so the asymmetric damping characteristics will become weaker, which verifies the effectiveness of the design ideas and methods.
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