Research on the Damping Device with Adjustable Inertia and Damping

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Abstract. In this paper, a new type of composite structure is proposed based on the original semi-active suspension of magnetorheological shock absorber and automobile suspension with inertia container, and the structure is analysed theoretically. The proposed new composite device is composed of a hydraulic cylinder, a magnetorheological damper and a hydraulic motor. The magnetorheological damper is compounded in the hydraulic cylinder and connected to the hydraulic motor through a hydraulic pipeline. The magnetorheological damper has the function of adjusting the damping, and the hydraulic motor achieves the purpose of adjusting the inertia by changing the mass. In order to analyse the device, the mechanical characteristics of magnetorheological valve and hydraulic motor are modelled and analysed. A quarter-vehicle suspension simulation analysis was performed on the overall operating characteristics of the system. The analysis results showed that the compound device has good variable damping inertia capacity characteristics, which is of great significance for improving the semi-active control suspension system.

1. Introduction
The suspension system, as an important part of the car, has an important influence on the handling, stability and comfort of the vehicle. Due to the fixed parameters of the traditional passive suspension, the performance cannot be changed in the driving process, so the comfort and controllability cannot be satisfied. The semi-active suspension damping system has lower energy consumption than the active suspension witness system, which has gradually been studied and applied in recent years.

Magnetorheological (MR) fluid is a kind of smart material. Because of its fast response speed, large damping range, reversibility, and low energy consumption, it is widely used in robots, automobiles and mechanical engineering [1]. The advantages of magnetorheological fluids have aroused great interest in the development of magnetorheological equipment [2][3]. The linear magnetorheological damper has gained popularity and has proven its potential to enhance the performance of the suspension system. The University of Nevada [4] developed a military high-mobility multi-purpose wheeled vehicle magnetorheological damper. The test results show that the magnetorheological semi-active suspension can improve the mobility of the vehicle under severe conditions; Dong Xiaomin of Chongqing University [5] has systematically studied automobile magnetorheological damping technology, including: magnetorheological effect, damper system theory, design method, experimental research and dynamic model of damper;

In 2002 [6], Professor Smith proposed the concept of "inertial container", which defined an inertial container as a device with two independent free end points, and the equal and opposite force at both ends of the device was proportional to the relative acceleration at both ends. Later, some scholars proposed the concept of inertial container-spring-damper semi-active suspension (ISD suspension)
[7,8], which promoted the further development of mechanical vibration isolation network, and many scholars applied it to semi-active suspension. In the study of the shelf.

This paper proposes a new type of compound damping device based on the adjustable inertia capacity and damping. The inertial container and the magnetorheological damping unit are used in the vibration damping system at the same time. Performance of semi-active damping system.

2. Analysis of mechanical characteristics of composite device

2.1. Hydraulic inertia container

![Figure 1. Hydraulic inertia container](image)

During the reciprocating movement of the piston in the hydraulic inertia vessel, the output shaft of the hydraulic motor drives the flywheel to rotate so as to transmit the inertial force at both ends of the piston rod to the flywheel:

$$F - f = A(P_2 - P_1) + m\ddot{x}$$  \hspace{1cm} (1)

Where: $F$ is the axial external force received by the piston; $f$ is the friction force on the inner wall of the piston cylinder; $m$ and $\ddot{x}$ are the mass and acceleration of the piston, respectively.

2.2. Magnetorheological damper

The working mode of magnetorheological fluid is flow mode. This paper uses the Bingham model to describe and predict the pressure drop loss of the magnetorheological damping unit. The hydraulic drop under different excitation frequencies is shown in Figure 2. As the current and excitation frequency increase, the hydraulic drop of the magnetorheological damping channel increases, and the energy consumption of the device increases.

![Figure 2. Hydraulic drop of magnetorheological fluid damping unit under different excitation currents.](image)
2.3. Magnetorheological damper

According to the formula, the force-displacement transfer function of the inertial vessel damper integrated device is derived:

$$F(s) = \frac{1}{\frac{A^2\eta v}{V^2}s^2 + K_0 A^2 s} + \frac{1}{X(s)} + \frac{f(s)}{X(s)}$$

According to the relationship, it can be seen that the inertia coefficient $b$ of the inertial vessel is only related to the effective area $A$ of the piston of the hydraulic cylinder, the displacement $V$ of the hydraulic motor, the moment of inertia $J$ of the flywheel, and the volumetric efficiency $\eta v$. The inertia coefficient can be adjusted by changing the mass of the flywheel.

3. Dynamic model establishment

Using the integrated original hydraulic inertial vessel-magnetorheological damper as the suspension structure, compared with the traditional passive suspension, a two-degree-of-freedom dynamic model is established. As shown in Figure 5, the suspended mass is $m_2$ and the vertical coordinate of the device is $x_2$. The suspended mass is $m_1$, the vertical coordinate is $x_1$, $q$ is the road input, the tire stiffness is $k_1$, the spring stiffness is $k_2$, the inertia coefficient is $b$, and the damping coefficient is $c$.

$$m_1\ddot{x}_1 + k_2(x_1 - x_3) + b(\dot{x}_1 - \dot{x}_2) = 0$$

$$m_2\ddot{x}_3 - k_2(x_1 - x_3) - c(\dot{x}_2 - \dot{x}_1) + k_1(x_3 - q) = 0$$

$$b(\dot{x}_1 - \dot{x}_2) = c(\dot{x}_1 - \dot{x}_2)$$
Laplace transform on the formula can be obtained:

\[
m_2s^2x_2(s) + bs^2(x_2(s) - x_3(s)) + k_2(x_2(s) - x_1(s)) = 0
\]

(6)

\[
m_1s^2x_1(s) + cs(x_1(s) - x_3(s)) + k_2(x_2(s) - x_1(s)) + k_1(x_1(s) - q(s)) = 0
\]

(7)

\[
bs^2(x_3(s) - x_3(s)) = cs(x_1(s) - x_3(s))
\]

(8)

Calculate the transfer function of car body (m2) acceleration relative to road input:

\[
H_A = \frac{ck_1bs^2 + k_1k_2bs + ckk_2}{\Delta s}
\]

(9)

The transfer function of suspension dynamic deflection relative to road input:

\[
H_{DE} = \frac{ck_1bs^2 + k_1k_2bs + ckk_2}{\Delta s} - \frac{(k_1(ck_2 + k_2sb + cs^2b + cs^2m_2 + s^3bm_2))}{\Delta s}
\]

(10)

The transfer function of tire dynamic load relative to road input:

\[
H_F = \frac{k_1^2(ck_2 + k_2sb + cs^2b + cs^2m_2 + s^3bm_2) - k_1}{\Delta s}
\]

(11)

Where: \(\Delta s = (ck_1k_2 + cks^2b + cks^2m_2 + cks^2m_1 + cks^2m_1 + cs^4bm_2 + cs^4bm_1 + k_2s^3bm_2 + k_2s^3bm_1 + k_1s^3bm_2 + cs^4m_2m_1k_1k_2sb)s^2\)

4. Simulation results

Taking a mature car suspension parameter, as shown in Table 1, the inertia coefficient \(b\) is 300 kg in the simulation.

| Model            | Unit | Value |
|------------------|------|-------|
| Sprung mass \((m_2)\) | kg   | 257   |
| Unsprung mass \((m_1)\) | kg   | 31    |
| Tire stiffness \((k_1)\) | N/m | 127976|
| Spring stiffness \((k_2)\) | N/m | 20213 |
| Damping coefficient \((c)\) | N/m | 1592  |

4.1. Semi-active and passive suspension comparison

Using 12 mm, 10 Hz sine excitation as input to compare Semi-active and Passive suspension, Figure 6 shows that the tandem suspension has better comfort compared to passive suspension.

![Figure 6. Car body vibration acceleration](image)

4.2. The effect of adjusting the damping coefficient on the suspension

Change the damping coefficient \(c\) of the series shock absorber to 0.3, 0.7, 1.0, 1.5, and 2.0 times of the \(c\) value. The simulation results are shown in Figure 7-9. With the decrease of the damping coefficient, the amplitude gains of the three indexes of body acceleration, suspension dynamic deflection and tire
Dynamic load become significantly larger at the first-order resonance and second-order resonance of the suspension. In the frequency range between the low-frequency and high-frequency resonance frequencies, reducing the damping coefficient can significantly reduce the sprung acceleration and the gain of the tire dynamic load, and has little effect on the suspension dynamic deflection.

Therefore, at low-frequency and high-frequency resonances, the damping coefficient of the suspension should be increased, and the damping coefficient should be appropriately reduced in the frequency range between the two resonance frequencies, which can enable the vehicle to obtain better ride comfort and driving safety. In the process of suspension design, the choice of damping coefficient should be compromised. If the damping coefficient of the suspension can be adjusted according to the vibration state, so that the vehicle can adapt to different road conditions during driving, and better driving can be obtained, performance. In this paper, the magnetorheological damper is applied to the parallel suspension. By controlling the magnitude of the excitation current, the purpose of adjusting the damping coefficient of the suspension is achieved, and the damping performance of the suspension is improved.

Figure 7. Car body acceleration gain.

Figure 8. Suspension dynamics gain.

Figure 9. The dynamic load gain.
4.3. The influence of adjusting the coefficient of inertia on the suspension

Change the inertia coefficient $b$ of the inertial container in the parallel suspension, respectively according to the 0.3, 0.7, 1.0, 1.5, 2.0 times of $b$, where the value of $b$ is 300kg, and the simulation results are shown in Figure 10-12. It can be seen from the figure that as the inertia coefficient gradually increases from 0.3$b$ to 2.0 times $b$, the sprung acceleration gain is significantly reduced at the low-frequency resonance of the vehicle body. However, in the high-frequency vibration stage, the inertia coefficient affects the suspension dynamics. The deflection has almost no effect, but a small coefficient of inertia will increase the sprung acceleration and tire dynamic load. In the low-frequency vibration stage, a small inertia coefficient can effectively reduce the sprung acceleration and tire dynamic load, which can improve vehicle comfort.

Therefore, at the first-order resonance and the second-order resonance, the inertia coefficient should be appropriately increased. In the low-frequency vibration range, the inertia coefficient should be reduced, which can keep the vehicle suspension in a good working condition. However, in the inertial vessel suspensions studied by most scholars, the inertia coefficient is usually a fixed value that cannot be adjusted, and the characteristics of the suspension cannot be adjusted according to vibration excitation. Therefore, this paper tries to propose a parallel suspension structure with adjustable inertia coefficient and apply it to the innovative design of semi-active suspensions in order to obtain good suspension vibration damping performance.

![Figure 10. car body acceleration gain.](image)

![Figure 11. suspension dynamics gain.](image)

![Figure 12. the dynamic load gain.](image)
5. Conclusion
The type of the series suspension structure is determined, the design theory analysis of the variable inertial capacity and variable damping compound device is completed, the hydraulic motor and hydraulic cylinder in the hydraulic inertial capacity unit are analysed, and the mechanical model of the hydraulic inertial container is derived. By analysing the working mode and constitutive relationship of the magnetorheological fluid, the damping force model of the magnetorheological damping unit is deduced; the influence of the spring damping coefficient and the inertial capacity coefficient of the series suspension on the performance of the series suspension system is discussed.

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