Simulation Research on Damping Control of Vane Damper in Active Arm-Type Torsion Suspension

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Abstract. Based on the structure of the active arm type torsion suspension, this paper proposes an optimal control algorithm for the damping force of the vane damper which integrates the road surface information. Firstly, the structural characteristics and working principle of the active arm-type torsional suspension and vane damper are introduced. Then, the mathematical model of the damper and suspension system is proposed, and the dynamic characteristics of the arm suspension system are analyzed. Finally, based on the fuzzy PID control algorithm, a system simulation model based on road surface information is built. The simulation results show that the control strategy of the vane damper can improve the ride comfort of the suspension.

Keywords: active arm-type torsion suspension, vane damper, ride control strategy.

1. Introduction
The trailing arm suspension has a comparative advantage in optimizing space allocation [1]. In order to further optimize the suspension structure, a vane damper is used to replace the traditional telescopic damper, which is beneficial to improving the structural compactness of the arm suspension system. In order to further study the impact mechanism of the vane damper on suspension performance on this basis, this paper proposes a damping force optimization control algorithm that integrates road surface information, and adjusts suspension performance by actively identifying changes in road surface information.

2. Active arm type torsion suspension system and active adjustable vane damper
2.1. Active arm type torsion suspension system
As shown in figure 1, the active arm type torsion suspension system is mainly composed of a body adjustment motor, a worm gear, a coupling, a torsion bar spring, a vane damper, a solenoid valve, a trailing arm, oil circuit, positioning nut. The suspension is provided with a power module, a
transmission module and a damping module. In the power module, the motor provides power for the entire system. In the transmission module, the transmission device transmits the power provided by the motor in turn to the trailing arm through the worm gear, the coupling, the motor output shaft, the cylinder block, and the torsion bar spring. In the damping module, the vane damper and the torsion bar spring cooperate. When the suspension is excited, the road surface is transmitted to the trailing arm through the tire, which drives the trailing arm to rotate. The rotation of the trailing arm drives the torsion bar spring and the vane damper sleeve to rotate, thereby generating corresponding elastic moments and damping moments to slow the suspension vibration.

Figure 1. Overall structure of the suspension.

2.2. Active adjustable vane damper
The active adjustable vane damper is mainly composed of a cylinder block, a vane damper sleeve, an oil pipe, a solenoid valve, a damping oil and sealing rubber sleeves. The design structure of the vane damper is a two-cavity type, as shown in figure 2.

Figure 2. Vane damper structure.

The cylinder block is provided with two oil passage openings, and the oil flows to other chamber through an external solenoid valve connected to the oil pipe or a gap inside the damper. The damping force of the damper is produced by the throttling effect of the hole and gap. When the trailing arm is rotated by the impact of the road, the vane rotates accordingly. The trailing arm is vibrated to cause the hydraulic pressure difference between the left and right chambers, the hydraulic pressure forces the damping fluid in the cavity to pass back and forth through the solenoid valve or the gap. Finally, the vibration energy received by the trailing arm is converted into the thermal energy of the damping oil to achieve the effect of absorbing vibration.

The rubber strips are arranged at the contact position of the cylinder block and the sleeve and the contact position of the vane with the inner wall of the casing to prevent oil from flowing out of the gap. However, there are special definitions for the seal pressure and assembly clearance, so that it can maintain a proper assembly clearance. As a long through-hole between each cavity of the vane damper,
the clearance has an important influence on the calibration of the basic damping of the system. [2] As shown in figure 3:

Figure 3. Internal gap of the vane damper.

3. Mathematical model establishment

3.1. Hydraulic mathematical model of vane damper

Based on the structure of the suspension system and the structural characteristics of the double-cavity damper, the damping coefficient of the damper is simplified to viscous damping according to the theory of fluid mechanics. The expression is:

\[ F = cv \]  

(1)

Among them, \( F \) represents the damping force of the damper, \( v \) represents the moving speed of the damping oil, and the constant \( c \) represents the damping size of the damper, which is called a damping coefficient.

In order to calculate the value of the damping coefficient \( c \) of the vane damper, the relational formula needs to be obtained. According to the conclusion of reference [3], three relationships can be obtained: the relationship between the damping force of the damper and the different hydraulic pressure of the two chambers \( \Delta p = f(F) \), the relationship between the damper flow rate and the speed \( Q = f(v) \), and the relationship between the damper flow rate and the different hydraulic pressure of the two chambers \( Q = f(\Delta p) \).

a) The relation of \( \Delta p = f(F) \)

\[ M = F \cdot l \]  

(2)

\[ M = L\Delta P \frac{D_a - D_b}{2} \frac{D_a + D_b}{4} = \frac{L\Delta P}{8} \left( D_a^2 - D_b^2 \right) \]  

(3)

b) The relation of \( Q = f(v) \)

\[ Q = \int_0^{\frac{D_b}{2}} \omega rL dr = \frac{L(D_a^2 - D_b^2)v\cos\alpha}{8l} \]  

(4)

c) The relation of \( Q = f(\Delta p) \)

\[ \Delta p_2 = \left( \frac{Q_1}{C} \right)^2 \frac{L}{2} \]  

(5)

\[ \Delta p_1 = \Delta p_3 = \frac{128\mu l Q_1}{\pi d_a^4} \]  

(6)
\[ \Delta p = \Delta p_1 + \Delta p_2 + \Delta p_3 \] (7)

The above relations can obtain the relation \( Q = f(\Delta p) \).

In addition to the vibration damping oil of the vane damper flowing through the solenoid valve and the pipeline, it will also flow through the gap of the damper. The gap mainly exists in these four positions: between the sleeve vane and the inner wall of the cylinder, between the cylinder bulkhead and the sleeve, at the end faces on both sides of the bulkhead, and on the one end face of the sleeve. The total flow at the gap is expressed by \( Q_2 \). According to fluid mechanics, we can get:

\[ Q_2 = Q_{gap1} + Q_{gap2} + Q_{gap3} + Q_{gap4} = C_g \sqrt{\frac{2\Delta P}{\rho}} \frac{\pi d^2}{4} \] (8)

\[ Q = Q_1 + Q_2 \] (9)

Then we can obtain the relational expressions \( Q = f(\Delta p) \) by synchronizing the above expressions.

In summary, the relationship \( F = f(v) \) can be obtained by combining the all above relationships. The obtained relational formula is imported into MATLAB software for simulation and calculation. The obtained gradient of the curve is shown in figure 4, and the value of the damping coefficient \( c \) of the damper is 1422N/(m/s).

![Figure 4. Simulation of damping coefficient of the vane damper.](image)

3.2. Mathematical model of the suspension system

Due to the effect of adjustable damping of the damper, the duty ratio of the solenoid valve of the damper can be adjusted in real time according to the information of the road surface. Here the 1/2 suspension model is considered as adding an additional damping force on the basis of ordinary damper. The suspension torsion bar spring, damper, unsprung mass and tire are all equivalent to spring. Dynamic analysis of the suspension model using the two DOF vehicle model yields:

![Figure 5. Two DOF vibration model of suspension.](image)
Establish equation of state for suspension mechanical vibration model:

\[ m_2 \ddot{z}_2 = k_2 (z_1 - z_2) + c (\dot{z}_1 - \dot{z}_2) + U(t) \]  \hspace{1cm} (10)

\[ m_1 \ddot{z}_1 = -k_2 (z_1 - z_2) - c (\dot{z}_1 - \dot{z}_2) + k_1 (q - z_1) - U(t) \]  \hspace{1cm} (11)

\( m_1 \) is the sprung mass; \( m_2 \) is the unsprung mass; \( k_1 \) is the stiffness of the suspension spring; \( k_2 \) is the stiffness of the tire; \( c \) is the damping coefficient of the vane damper; \( z_1 \) is the sprung mass displacement; \( z_2 \) is the unsprung mass displacement; \( q \) is the ground vertical displacement; \( U(t) \) is the active control force, in which the main power can be adjusted in real time according to the road surface excitation. [4]

The collection of suspension road information mainly uses a displacement sensor, which is fixed to the trailing arm. When the tire is excited by the road, the tire jumps up and down to drive the trailing arm to rotate, thereby generating corresponding ground vertical displacement \( q \), then introduce \( q \) into the suspension equation of state to obtain the acceleration of the suspension body. [5]

Because the suspension system is a complex multi-variable and non-linear system, simplifying modeling by assuming conditions cannot obtain a very accurate mathematical model. Therefore, the use of traditional control methods (such as PID control strategy) to control the suspension system often does not achieve a good vibration reduction effect. The intelligent control theory based on fuzzy control theory can realize inexact control of complex systems. Taking the two DOF 1/2 suspension system as the research object, Considering the nonlinear and damping characteristics of suspension stiffness, controlling suspension with fuzzy PID method. Real-time control of the duty ratio of the solenoid valve of the damper according to the road surface information, so that the acceleration of the suspension body is always kept to a minimum.

4. Simulation optimization analysis

The Simulink module of MATLAB software is used to build the suspension state equation into a simulation model of the suspension system. According to different control methods, passive suspension simulation model and fuzzy PID control suspension simulation model are respectively established. As shown in figure 6 and 7.

![Figure 6. Passive suspension simulation.](image-url)
The variables in the simulation model are:

\[ m_1 = 20; \quad m_2 = 320; \quad k_1 = 15000; \quad k_2 = 12000; \quad C = 1422; \]

Because it is necessary to simulate the performance of the car on the actual road surface, the input source of this simulation module is to select a Band-Limited White Noise and obtain the simulated road surface after integration. Among them, the power spectrum of the limited bandwidth white noise is:

\[ G_q(f) = 4\pi^2 G_n(n_0) n_0^2 u. \]

On the actual road surface, the amplitude of the road speed power spectrum can be regarded as a constant in the entire frequency range, that is, "white noise".

\[ u = 20 \text{m/s}, \quad n_0 = 0.1 \text{m}^{-1}, \quad G_q(n_0) = 256*10^{-6} \text{m}^2 / \text{m}^{-1} \]

As we all know, the performance of automobile suspension system is most directly related to the ride, comfort and handling stability of the vehicle. The reflected physical quantities are vehicle body acceleration, tire dynamic load, and suspension dynamic disturbance. And the scientific and reasonable suspension system evaluation index is especially important to match the parameters of the vehicle suspension system.

In terms of body acceleration, body acceleration is related to the smoothness and ride comfort of the car. Therefore, for passenger cars, this indicator can be regarded as a key criterion for evaluating the strength of the suspension. In terms of the dynamic load of suspension, due to the fact that the rigid tires of the integrated hub motor and power loss brake are used, factors affecting tire pressure that affect dynamic load characteristics cannot be considered; in terms of the dynamic disturbance of suspension, due to the fact that the torsion bar spring and vane damper are used, the dynamic disturbance of suspension is so little that is not suitable for observation and comparison.

With the same road excitation, the values of the above data are substituted into the simulation model, and the simulation results of the body acceleration obtained are shown in figure 8 and 9.
Figure 8. The left: passive suspension simulation diagram; the right: PID control simulation diagram.

Figure 9. The left: Fuzzy PID control simulation diagram; the right: Comparison of body acceleration.

From the simulation results, it can be seen that the uncontrolled passive suspension has the largest body acceleration fluctuation range, and the value jumps back and forth between $-5.14 \sim 4.60 \text{ m/s}^2$; after PID control, the suspension body acceleration fluctuation becomes smaller and the curve becomes gentler. The value jumps between $-3.65 \sim 3.42 \text{ m/s}^2$; after the fuzzy PID control, the acceleration of the suspension's body becomes further relaxed, and the fluctuation range becomes $-3.22 \sim 2.85 \text{ m/s}^2$. Comparing the simulation results, the fuzzy PID control is superior to the traditional PID control, and the feasibility of the control strategy of the vane damper based on the road surface is verified, which can improve the ride comfort of the suspension.

5. Summary
This paper introduces a damping force optimization control algorithm based on active arm-type torsion suspension with a vane damper that incorporates road information. The road surface information is identified based on the characteristics of the suspension structure, and the fuzzy PID control method is used to match the damper damping force of the corresponding road condition in real time to improve the ride comfort of the suspension. Then, the Simulink module of MATLAB was used to build the
simulation model. From the simulation results, it can be concluded that fuzzy PID control is better than PID control, PID control is better than no control. On top of PID control, the body acceleration of fuzzy PID control get 14.14% more optimized; on top of passive suspension, the body acceleration of fuzzy PID control get 37.68% more optimized which verified the correctness of the control strategy of the vane damper based on the road surface information.

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