Research on Vehicle Damping Control Strategy Based on In-Arm Torsional Electromagnetic Active Suspension

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Abstract. This paper proposes a new control strategy based on the In-Arm Torsional Electromagnetic Active Suspension (ITEAS), which can improve the vehicle ride comfort through adjusting the damper in ITEAS to optimize the kinetic characteristics of the vehicle composed by two ITEAS systems. Firstly, the structure and principle of the new type system is introduced followed by the description of the vehicle and the damper in ITEAS. Secondly, according to theoretical mechanics and fluid mechanics, the dynamic model of ITEAS system and the hydraulic model of shock absorber are established under forced vibration. Then, the PID control simulation model of the front suspension is established in AMESIM with a passive model as reference, due to the special assemble features of the vehicle. The rear suspension adopts predictive control method and the simulation model is built in MATLAB/SIMULINK, whose theory can be summarized as adjusting the duty ratio of the valve in damper to improve the stability of the vehicle under the condition that the road surface information is already known. Lastly, this paper tests the suspension characteristics on the MTS bench, which verifies the proposed control method improves the ride comfort performance and the feasibility of predictive system.

1. Introduction
The main function of the vehicle suspension system is to reduce the vibration and impact caused by gravamen road and improve ride quality and handling stability. As a mainstream research direction, the semi-active suspension can timely adjust the damping force of the controllable damping according to the road conditions to achieve the best damping effect. This article researches the structural design of the new suspension system [1] and the verification of the damping adjustable vane damper [2] based on the "In-Arm Torsional Electromagnetic Active Suspension (ITEAS)”. It is verified that ITEAS, as a semi-active suspension system, can realize damping adjustment through the vane damper after adding the suspension. In the field of suspension control, Ming Xu in China has conducted research on different intelligent control methods of vehicle semi-active suspension [3], Qiu Liu implement semi-active suspension control based on convolutional neural network [4], Ling Zheng proposed a sliding mode variable structure control strategy to achieve semi-active control of the suspension [5], and Long Chen proposed damping multi-mode switching control for semi-active suspension [6]. To sum up, most of the research on semi-active suspension control is achieved through continuous feedback adjustment of the
unknown road surface, but not related to the vehicle control analysis through the road surface information as a predictive input. Therefore, this article proposes an optimal control strategy for vehicle damping adjustment for wheeled vehicles equipped with ITEAS system, as shown in Figure 1, to improve ride quality and handling stability.

This paper studies the driving state of wheeled vehicles. Because the front suspension receives random road excitation information first and needs to respond instantaneously, the front suspension adopts a simple structure, fast response speed, and strong adaptability. The PID controller is the most suitable for controlling damping. At the same time, the road condition information received by the front suspension is quickly transmitted to the rear suspension controller, and the rear suspension controls the current duty cycle by predicting the input state of the road excitation and a more accurate mathematical model. Because predictive control has obvious advantages such as low model accuracy requirements, large model information redundancy, and good system robustness and stability, the rear suspension can be used in complex and changeable road conditions through predictive control strategies. Respond accurately to make the rear suspension more stable. The PID adjustment of the front suspension and the predictive control of the rear suspension adjust the damping, and the control strategy can adjust the damping in real time, which can achieve the stability of the whole vehicle and improve the ride comfort of the whole vehicle.

Figure 1. Structure diagram of wheeled vehicle

This paper studies the driving state of wheeled vehicles. Because the front suspension receives random road excitation information first and needs to respond instantaneously, it is the most suitable for damping control of the front suspension to adopt the PID controller which has simple structure, fast response speed and strong adaptability. At the same time, the road condition information received by the front suspension is quickly transmitted to the rear suspension controller, and the rear suspension controls the current duty cycle by predicting the input state of the road excitation and a relatively accurate mathematical model. Because predictive control has obvious advantages such as low model accuracy requirements, large model information redundancy, and good system robustness and stability, through predictive control strategy, the rear suspension can accurately respond to complex and changeable road conditions, and make the rear suspension more stable. Through the front suspension PID adjustment and rear suspension predictive control to adjust the damping, the method of adjusting damping in real time by control strategy, it can realize the vehicle approaching stability and improve the vehicle ride comfort effect.

2. Structural Design and Control Principle

2.1. Suspension Structure Introduction
The front and back of the wheeled vehicle are connected to the body with two back-to-back ITEAS systems. The suspension structure is shown in Figure 2. The semi-active suspension system is composed
of torsion bar springs, vane dampers, trailing arms and connecting parts. The torsion bar spring passes through the center of the vane damper, and is fixed in parallel with the vane damper rotor and trailing arm through splines. When the suspension is excited by the road, the vane damper sleeve is driven to rotate around the torsion bar spring through the rotation of the trailing arm, and the blade on the vane damper sleeve rotates relative to the cylinder block. Then, the hydraulic oil flows between the internal partitions of the cylinder block to produce damping force, and the torsion of the torsion bar spring produces elastic force, so as to produce the corresponding elastic moment and damping force to slow down the suspension vibration. Realize the basic functions of suspension system to relieve shock and absorb vibration.

Figure 2. Structure diagram of semi-active suspension

2.2. Introduction of Damper Structure
The structure diagram of the vane damper is shown in Figure 3. The main components of the vane damper include the shell, sleeve, blade, diaphragm, hydraulic pipe, solenoid valve and trailing arm. In order to facilitate processing and sealing, the interior of the damper is divided into two chambers by blades and partitions. The outer end of the sleeve is connected with the trailing arm through a spline to transmit road excitation. In order to adjust the damping through an external solenoid valve, two oil holes are designed on the shell to connect the hydraulic pipe and the solenoid valve. When the solenoid valve is fully open, the fluid flow in it can be regarded as the nozzle flow. By adjusting the motor input PWM ratio, the internal flow and velocity of the hydraulic oil are changed, and the pressure difference between the two ends of the solenoid valve is changed. The working principle of the vane damper is: when the trailing arm is rotated by the impact of the road, the rotation of the blade drives the oil to flow to another chamber through the external solenoid valve or the gap. In this process, severe friction occurs between the oil and the inner wall of the cylinder and the molecules inside the oil, which generates the damping force of the damper.

Figure 3. Model diagram of vane damper

The designed trailing arm length of ITEAS suspension is 0.2m, the angle between static balance and road surface is 45 °, the sprung mass is 450 kg, the torsion bar spring stiffness is 12000 n / m, the
unsprung mass is 20 kg, and the unsprung stiffness is 120000 N/m. The design parameters of shock absorber and the selection of relevant solenoid valve options are shown in Table 1. The damping coefficient of the damper is determined by AMESIM kinematics analysis and MATLAB simulation.

| Name                                | Value       | Name                          | Value   |
|-------------------------------------|-------------|-------------------------------|---------|
| Inner diameter of damper housing    | 0.075 m     | Hydraulic oil density         | 860 kg\cdot m^{-2} |
| Inner diameter of damper shell      | 0.015 m     | Gap width                     | 0.003 m |
| Blade and partition thickness       | 0.008 m     | Gap depth                     | 0.003 m |
| Length of blade and partition       | 0.08 m      | Cross-sectional area of solenoid valve orifice | 0.0004 m² |
| Hydraulic pipe length              | 0.25 m      | Natural frequency of solenoid valve | 5 Hz |
| Inner diameter of hydraulic pipe    | 0.005 m     | Solenoid valve damping ratio  | 1.8     |
| Hydraulic oil dynamic viscosity     | 0.00894 Pa·s | Flow characteristics of full open solenoid valve | 22 L/min |

2.3. The Principle of Vehicle Damping Control
For the suspension system, when it is impacted by the road, the vehicle body is displaced relative to the ground. The spring and shock absorber generate elastic force and damping force under the action of the relative displacement to resist the impact of the ground on the vehicle body. In order to improve the performance of the suspension, a semi-active suspension composed of a spring with adjustable stiffness or a shock absorber with adjustable damping is proposed. According to the acceleration response of the sprung mass and other feedback signals, the suspension can adjust the spring stiffness and damper damping according to a certain control law to meet the requirements of suspension ride quality. When the wheeled vehicle composed of ITEAS system passes through different working conditions, different methods of damping adjustment are adopted for the front and rear suspension to achieve vehicle control, which can improve ride quality and handling stability.

According to the characteristics of quick self-adaptation of suspension, a PID controller is designed for front suspension. When the front wheels pass through the road surface, the height change is detected according to the displacement sensor installed at the front end of the vehicle body, and the PID controller is used to adjust duty cycle of the solenoid valve to realize active adjustment of damping force of the blade shock absorber. The front end of the vehicle body collects the road information and transmits it as an electrical signal to the rear suspension. The rear suspension performs predictive control according to the collected road surface excitation changes, and takes the optimal effect of suspension dynamic deflection and vehicle acceleration as the expected value, based on the connection between suspension vibration model and shock absorber hydraulic model and the relationship between hydraulic flow and pressure difference in shock absorber. The duty cycle of the solenoid valve to be adjusted is determined with the fluid dynamics theory. The electromagnetic valve controller converts the required damping force into PWM with different frequencies to adjust duty cycle of the electromagnetic valve. When rear wheels pass through the detected road surface, the shock absorber is adjusted in real time according to the changes to the predicted road surface information.
3. **Theoretical model**

3.1. **Suspension mechanics vibration model**

The semi-active suspension adjusts the duty cycle of the solenoid valve of the vane damper according to the road surface information, thereby achieving adjustable damping. Build a two-degree-of-freedom 1/4 suspension model based on its working principle, as shown in Figure 5.

![Semi-active suspension dynamics model](image)

**Figure 5.** Semi-active suspension dynamics model

In the above figure: \( m_1 \) is the tire mass; \( m_2 \) is the body mass; \( k_1 \) is the tire stiffness; \( z_1 \) is the tire vertical displacement; \( z_2 \) is the body vertical displacement; \( q \) is the road surface excitation; \( k_2 \) is the body elasticity coefficient; \( C_t \) is the suspension adjustable damping coefficient.

Establish the state equation of the mechanical vibration model of the suspension:
3.2. Hydraulic Model of Vane Damper

According to the two degree of freedom vibration model equation, the damping force $F_c$ of shock absorber and the vertical speed $z_i$ of wheel are determined. When the wheel is excited by the road, based on the structural characteristics of the system, the damping moment $M$ can be calculated by equation.

\[ M = F_c \cdot l \cdot \cos \theta \] (3)

In the above formula: $\theta$ is the angle between the longitudinal arm and the road surface.

When the vane damper is working, according to the characteristics of the double-chamber vane damper, assuming that the sleeve and the blade are uniformly loaded by the hydraulic oil, according to the reference, the damping torque of the shock absorber [2] can be expressed as:

\[ M = L \Delta P \frac{D_1 - D_2}{2} + \frac{D_1 + D_2}{4} = L \Delta P \frac{D_1^2 - D_2^2}{8} \] (4)

In the above formula: $M$ is the damping moment of the vane damper; $D_1$ is the inner diameter of the damper cylinder; $D_2$ is the outer diameter of the damper sleeve blade; $L$ is the length of the blade and the diaphragm; $\Delta P$ is the pressure difference between the chambers. Combining (3) (4), the pressure difference $\Delta P$ of the shock absorber chamber can be obtained as:

\[ \Delta P = \frac{8F_c \cdot l \cdot \cos \theta}{L(D_1^2 - D_2^2)} \] (5)

As shown in Figure. 6, when the hydraulic oil flow $Q$ flows from the high-pressure chamber to the low-pressure chamber, the main channel includes the solenoid valve and hydraulic pipe in series, which is equivalent to the blade clearance and diaphragm clearance of a constant through hole. Set the flow through the hydraulic pipe and solenoid valve as $Q_1$, and the flow through the blade clearance and diaphragm clearance as $Q_2$. Let the pressure difference between the two chambers be $\Delta P$, the pressure difference between the two ends of the hydraulic pipe in the high pressure chamber be $\Delta P_1$, the pressure difference between the two ends of the solenoid valve be $\Delta P_2$, the pressure difference between the two ends of the hydraulic pipe in the low pressure chamber be $\Delta P_3$, and

\[ \Delta P = \Delta P_1 + \Delta P_2 + \Delta P_3 \] (6)

![Figure 6. Fluid structure diagram of vane damper](image-url)
According to the literature [1], the relationship among the total flow \( Q \) of the oil inside the damper, the flow \( Q_2 \) flowing through the gap between the blades and the diaphragm, and the hydraulic difference between the two chambers \( \Delta P \):

\[
Q = \int_{D_1}^{D_2} \frac{\rho}{2} \pi l Q \, dr = \frac{L(D_1^2 - D_2^2)}{8l} \xi_1 \cos \theta
\]

\[
Q_2 = C_q \sqrt{\frac{2\Delta P \pi d_1^2}{\rho}}
\]

In the above formula: \( \rho \) is the density of hydraulic oil; \( C_q \) is the flow coefficient; \( d_1 \) is the diameter of the constant through hole that replaces the gap flow.

The flow through the hydraulic pipe and solenoid valve is \( Q_1 \):

\[
Q_1 = L(D_1^2 - D_2^2) \xi_1 \cos \theta - C_q \sqrt{\frac{2\Delta P \pi d_1^2}{\rho}}
\]

According to the pressure drop between the solenoid valve and the pipeline, the pressure difference can be determined as \( \Delta P_1 \), \( \Delta P_2 \), and \( \Delta P_3 \) respectively:

\[
\Delta p_2 = \left( \frac{Q_1}{C_q A_0} \right)^2 \frac{\rho}{2}
\]

\[
\Delta p_1 = \Delta p_3 = \frac{128 \mu l Q_1}{\pi d_2^4}
\]

In the above formula: \( A_0 \) is the cross-sectional area of the orifice of the solenoid valve, \( L_1 \) is the flow length of the hydraulic pipe, and \( D_1 \) is the inner diameter of the hydraulic pipe.

According to the working principle of the solenoid valve, when the solenoid valve is fully open, the flow of fluid in it can be regarded as nozzle flow. By adjusting the PWM ratio of motor input and changing the flow rate and flow rate of hydraulic oil, the pressure difference between the two ends of solenoid valve \( \Delta P_2 \) can be changed. The relationship between the duty ratio \( r \) and the hydraulic pressure difference between the two chambers \( \Delta P \) can be obtained:

\[
\gamma = \left( \frac{\Delta p - 64 \mu l Q_1}{\pi d_2^4} \right) \frac{2 C_q A_0^2}{\rho Q_1^2}
\]

By combining (5) (9) (12), the relationship among duty cycle \( \gamma \), damping force \( F_c \) and wheel vertical velocity \( \ddot{z}_w \) can be obtained: \( \gamma = f(P_c, \ddot{z}_w) \).

3.3. PID Control Model of Front Suspension

In order to achieve fast and adaptive motion control, in the research of front suspension control, a PID controller with good stability and fast corresponding speed is adopted, as shown in Figure 7, to reduce the body’s dynamic deflection as the control purpose, through the PID controller improve the suspension performance. The ratio (P), integral (I), and derivative (D) of the deviation between the given value \( r(t) \) and the actual output value \( c(t) \) are combined to form a control variable to control the control object. The principle is shown in Figure 7, the control algorithm differential equation is:
Figure 7. PID control principle diagram

\[ u(t) = k_p e(t) + k_i \int_0^t e(t) dt + k_d \frac{de(t)}{dt} \]  

(13)

In the above formula: \( e(t) = r(t) - c(t) \), \( u(t) \) is the output of the controller; \( e(t) \) is the difference between the preset value \( r(t) \) of the controlled object and the true value \( c(t) \); \( k_p \) is the proportional coefficient; \( k_i \) is the integral coefficient; \( k_d \) is the derivative coefficient. The PID controller calculates the difference between the real value and the preset value of the body dynamic deflection by proportion, integration and differentiation. And use trial and error method to determine PID control parameters \( k_p, k_i, k_d \). At first, adjust the proportional coefficient, then adjust the integral coefficient, and finally adjust the differential coefficient. In this way, the PID control parameters of the front suspension are obtained after repeated debugging.

3.4. Predictive Control Model of Rear Suspension

Model predictive control (MPC) is a special kind of control. Its current control action is obtained by solving an open-loop optimal control problem in finite time domain at each sampling moment. The current state of the process is regarded as the initial state of the optimal control problem, and the optimal control sequence only performs the first control function. There are dozens of predictive control algorithms, among which the representative ones are model algorithm control (MAC), dynamic matrix control (DMC) and generalized predictive control (GPC). Because the rear suspension can establish a more accurate mathematical model, the model algorithm control is adopted. The basic idea of this algorithm is to first predict the future input state of the object, and then determine the current control action, that is, first predict and then control.

According to the designed vehicle motion characteristics, the front end of the vehicle body uses the perceived road condition information as the input state of the predicted object in the future, and let the expected value of the suspension dynamic deflection and the body acceleration be equal to zero to determine the control action at the current moment. The relationship between the damping force and the vertical wheel displacement under the optimal solution determined by the degree vibration model is:

\[ \frac{m}{k_2} \dot{z}_i + z_i = q \]  

(14)

\[ F_c = k_2 z_i \]  

(15)

Determine the damping force and the vertical displacement of the wheels by inputting the road excitation \( q \), and use the damper hydraulic model to solve the solenoid valve duty ratio \( r \) value at the moment.
In the above formula:

\[
y = \frac{8F \cdot l \cdot \cos \theta}{L(D_1^2 - D_2^2)} - \frac{64 \mu_l l Q_i}{\pi d_1^2} + 2C_i^2 A_i^2 \rho \Omega_1^2
\]  

In the above formula:

\[
Q_i = \frac{L(D_1^2 - D_2^2)l \cos \theta}{8l} - \frac{C_i \left( \frac{16F \cdot l \cdot \cos \theta}{\rho L(D_1^2 - D_2^2)} \right) \pi d_1^2}{4}
\]  

4. Suspension Characteristics Simulation and Analysis

4.1. Front Suspension Simulation and Comparison

In order to reflect the characteristics of the ITEAS suspension, this paper uses the hydraulic simulation software AMESIM to build a hydraulic simulation model. As shown in Figure 8, the mass and stiffness of the model tire are represented by the mass and the coil spring respectively; On the structure, the torsion bar spring and the blade shock absorber are in parallel. Therefore, the trailing arm lever is used to input the signal of the cam blade of the damper. Since the output of the trailing arm end is inconsistent with the input end interface data of the cam blade, the torque and angle conversion module is added; and the scale factor is calculated based on the basic principle of theoretical mechanics; there are gap flow inside the shock absorber and the external solenoid valve pipeline flow, two parallel fluid models, so in the model, the cam blade is used to represent the two-chamber vane damper, and the normal through hole instead of gap flow, the solenoid valve adopts the normally closed switch valve model used in the prototype; the car body is represented by another mass block. And according to the theoretical model of the suspension and the shock absorber, the actual prototype parameters are matched and input into the hydraulic simulation model.

Figure 8. Front suspension hydraulic simulation model

The simulation model aims to reduce the dynamic deflection of the vehicle body. The difference between the actual value and the preset value is input to the PID controller for proportional, integral,
and differential calculations, as follows: \( kp=1300 \), \( ki=100 \), \( kd=1.7 \), and it is converted into PWM duty cycle signal and input to the solenoid valve. The front suspension is input with random road excitation, and the duty cycle of the solenoid valve is 0 as a passive control, which is compared with PID control, to evaluate the acceleration of the body, the dynamic displacement of the tire and the dynamic deflection of the suspension, and analyze the ride comfort effect of the suspension.

![Comparison of PID control and passive control performance indicators of the front suspension](image)

**Figure 9.** Comparison of PID control and passive control performance indicators of the front suspension

By comparing PID control and passive control, as shown in Figure. 9(a)(b)(c), under random road excitation, the semi-active suspension with PID controller can significantly reduce the vibration of the vehicle. Compared with the passive suspension, the semi-active suspension reduces the RMS value of the body acceleration from 1.1230m/s\(^2\) to 0.9099m/s\(^2\), and the index reduction is 18.98%; the suspension dynamic stroke RMS value is reduced from 13.4013mm dropped to 8.6084mm, the index reduction is 35.82%; the RMS value of tire dynamic displacement is reduced from 6.4582mm to 4.7664mm, the index reduction is 26.16%. The simulation results show that: PID control of the front suspension can
effectively improve the ride comfort of the vehicle without affecting the suspension's motion characteristics and tire grounding performance.

4.2. Simulation and Comparison of Rear Suspension

By inputting the road excitation collected by the front wheels, the rear suspension takes the non-fluctuation of suspension dynamic deflection as the expected value. According to the suspension vibration model, the appropriate damping force is obtained to achieve better suspension ride comfort. According to the damper hydraulic model, the PWM duty cycle of the solenoid valve is determined, as shown in Figure 10. Based on the Matlab/SIMULINK environment, build a simulation solution model according to formula (12).

As shown in Figure 11(a), the random road surface excitation is used as the road condition information that the front end of the car body will perceive. According to the natural frequency of the solenoid valve, the road excitation fluctuation frequency is set to 5HZ, and duty cycle r value of solenoid valve in Figure 10(b) is obtained by inputting the road surface excitation. The duty cycle fluctuates in the range of 0.0445 ~ 0.4253m, indicating that there is no unsolvable situation through the equation solving. And comparing the road excitation and the duty cycle ladder diagram can verify the real-time corresponding relationship between them. According to the duty cycle value r, it can be converted into the solenoid valve PWM signal as shown in Figure 10(c).

Figure 10. Simulation diagram of rear suspension predictive control

Figure 11. Stochastic road excitation fluctuation diagram
In order to verify the predictive control effect of the rear suspension, the random road information and the corresponding duty cycle value are input into the hydraulic simulation model built by AMESIM, and compared with the passive control, the suspension characteristic curve as shown in Figure 14 (a) (b) (c) is obtained. The simulation results show that: according to the ideal hydraulic theory model established by AMESIM simulation (normally through hole model instead of gap flow [5]), by taking the body dynamic deflection as zero as the expected value to calculate the duty cycle of the solenoid valve, the suspension dynamic deflection and body acceleration have both zero fluctuation, the maximum range of tire dynamic displacement fluctuation is -1.3 – 1.3mm, and the RMS value of tire dynamic displacement is extremely small and close to 0. Therefore, it is proved that the predictive control has high precision and can achieve the optimal effect of each performance index of suspension characteristics, which indicates that the predictive control of rear suspension is feasible.
5. Verification on Test Bench
In order to further verify the control feasibility of ITEAS semi-active suspension system, small and medium-sized vehicles are selected as the application objects for small-scale suspension prototype machining, and a semi-active suspension system test bench is built for active control testing. The bench test layout is shown in the figure. 15. In the experiment, two sets of springs are used to simulate the tire stiffness, and the existing MTS vibration actuator is used to simulate the road surface excitation and drive the vane damper to work through the trailing arm. The laser displacement sensor is fixed on the lifting lugs, and the body acceleration and suspension dynamic deflection are measured by the
acceleration sensor and the displacement sensor respectively. The tire dynamic displacement can be obtained by the displacement sensor on the vibration platform, and the data is collected through the NI card communicating with LabVIEW. The measured displacement sensor information is obtained through the designed combined filter circuit to obtain the displacement signal required by the suspension controller. After the displacement signal is converted, the ideal damping force is obtained, and the STM32 single-chip microcomputer is used to control the duty ratio of the solenoid valve.

Through the designed combined filter circuit, the displacement signal needed by the suspension controller is obtained by using the measured displacement sensor information. The ideal damping force is obtained by converting the displacement signal. And the duty cycle of the solenoid valve is controlled by STM32 single chip microcomputer.

As shown in Figure 16(a)(b)(c), through the test, it can be obtained that the semi-active suspension reduces the RMS value of the body acceleration from 1.1230m/s² to 0.9424m/s², the index reduction is 16.08%; The RMS value of suspension dynamic travel decreases from 13.4013mm to 9.0839mm, and the index reduction is 32.22%; the RMS value of tire dynamic displacement is reduced from 6.4582mm to 4.8316mm, and the index reduction is 25.19%. The simulation results show that: PID control of the front suspension can effectively improve the ride comfort of the vehicle without affecting the suspension's motion characteristics and tire grounding performance.
Table 2 shows the comparison between the simulation results and the test results of the ITEAS suspension. From Table 2, it can be seen that the simulation and test results are relatively consistent, but there are still certain errors. During the test, due to the deformation of the bearing plate and the connecting bearing and the slight jitter of the device, the acceleration of the vehicle body, the dynamic displacement of the tire and the dynamic deflection of the suspension were slightly increased.

Figure 16. Comparison of PID control and passive control performance indicators of the front suspension
Table 2. Comparison of rms values of each performance between simulation and experiment

| project                 | RMS of body acceleration / (m/s²) | RMS of suspension dynamic deflection / mm | RMS of tire dynamic displacement / mm |
|-------------------------|-----------------------------------|------------------------------------------|---------------------------------------|
| simulation              | 0.9099                            | 8.6084                                   | 4.7664                                |
| test                    | 0.9424                            | 9.0839                                   | 4.8316                                |
| error%                  | 3.57                              | 5.52                                     | 1.37                                  |

6. Conclusions
According to the characteristics of ITEAS semi-active suspension system, this paper proposes a vehicle control strategy which combines the damping force adjustment of suspension system and real-time adjustment by collecting information. According to the forced vibration model of ITEAS system, the damping control is verified by simulation in AMESIM and MATLAB / SIMULINK environment and by a bench test. Based on this, the dynamic deflection, body acceleration and other parameters are evaluated. And the following conclusions are obtained:

1. According to the characteristics of the vane damper, the front suspension adopts a fast adaptive PID controller to adjust the damping force, which has good performance and a good effect of optimizing ride comfort.

2. Utilizing the characteristics of wheeled vehicles, the rear suspension can be predicted based on the information of the road through which the vehicle passes, and converted into a solenoid valve duty cycle signal through the solution model, so as the damping force can be adjusted in real time on the corresponding road surface to achieve the optimal effect of ride comfort.

3. Through the damping adjustment and control of the front and rear suspension of ITEAS under road conditions in turn, not only the damping performance of the suspension system is optimized, but the ride comfort and handling stability of the vehicle are guaranteed.

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