The Design and Analysis of New Hydraulic Energy-Regenerative Active Suspension based on Adaptive Backstepping Sliding Mode Variable Structure Control

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***Abstract* — In this paper we design a new type of hydraulic Energy-Regenerative vibration active suspension system. The system has continuous change of damping force from zero to the maximum value and high energy conversion efficiency. Also we designed and simulated an adaptive back-stepping sliding mode variable structure controller. The simulation results show the designed controller has good robustness. The system has good riding comfort and operational stability.**

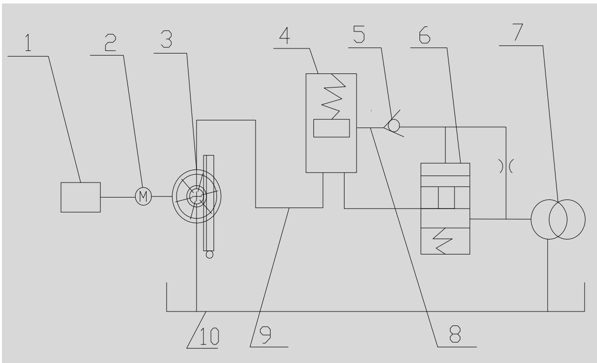
Keywords - Energy- regenerative, Active Suspension, Adaptive, Backstepping, Sliding Mode

I. INTRODUCTION

Suspension is an important part of the car. It electrically connects the car body and the axle, which bear the force between the wheel and the body, and buffers the impact load from rough pavement. In addition, it attenuates the vehicle body vibration caused by various dynamic loads[1]. Vehicle vibration is the main factor affecting the riding comfort [2-4]. In order to overcome current suspension disadvantages, we designed a new type of hydraulic feedback vibration active suspension system with the eccentric vane pump, which not only can automatically recovery vehicle vibration energy, but also can produce the required damping force. At the same time, for the nonlinear, unpredictable state and external disturbance factors of vehicle suspension system, we put forward the back-stepping adaptive sliding mode variable structure controller to control uncertainty nonlinear system of vehicle suspension by the theoretical derivation and simulation results prove that the designed controller can improve the riding comfort and handling stability of vehicles.

II. STRUCTURE AND WORKING PRINCIPLE OF NEW HYDRAULIC ENERGY-REGENERATIVE VIBRATION ACTIVE SUSPENSION SYSTEM

Eccentric vane pump of active new feed suspension system is consisting of the electronic control unit, control electric motor, actuator, accumulator, control valve, hydraulic motor, generator, one-way valve, oil channel, etc. As shown in Figure 1. Eccentric vane pump is consisting of the fixed pump body, movable pump body, impeller, blade. Movable pump body can move around to change the cavity between the pump body and impeller. High pressure oil hole is interlinked with the upper part of vane pump movable shell. The low pressure oil hole is interlinked with the lower part of vane pump movable shell. Vane pump high pressure oil holes are connected to the accumulator through the high pressure oil pipe. The low pressure oil hole connected to the tank through the low pressure oil pipe. Gear connected to the eccentric pump impeller. The lower end of the rack is connected to the axle, the vane pump fixed shell connected with the body. The control rod of control electric motor connected to the vane pump movable shell, which is used to control the size and direction of the force generated by the shock absorber.



1.Electronic control unit 2.Control electric motor 3.Vane pump 4.Hydraulic accumulator 5.One-way valve 6.Control valve 7.Hydraulic motor and generator 8.Control oil pipe 9.High pressures tubing 10.Tank

Figure 1. New hydraulic Energy-Regenerative vibration suspension system structure.

III. THE CHOICE OF CONTROL METHOD AND CONTROLLER DESIGN

Current methods of control includes: sky-hook control [5], ground hook control[6], optimal control, adaptive control, variable structure control[7] and intelligent control[8-9], etc. In this paper, the adaptive back-stepping sliding mode variable structure controller is designed based on the active control strategy, which is combined with the back-stepping design theory, and the adaptive algorithm is used to estimate the disturbance factor.

Figure 2 presents the model of 1/4 vehicle suspension with single-degree-of-freedom, which neglects the flexibility and quality of the tires.

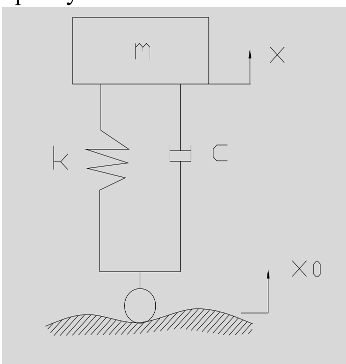


Figure 2. 1/4 Simplified model of vehicle suspension.

In Figure 2, m is the mass of the vehicle body, k is the nonlinear spring of the suspension, c is the nonlinear damping of the suspension; x is the displacement of the vehicle body; x_0 is the excitation of road surface.

According to the Newton's second law, we can set up the suspension system's motion differential equations as shown in Figure 2.

$$m\ddot{x} - f_k + f_d = 0 \tag{1}$$

Where, f_k 、 f_d for spring force and damping force generated by nonlinear spring and nonlinear damper respectively.

To fit the measured data of the measuring device of the suspension system parameters, we can get [10]

$$f_k = k_0 + k_1\Delta x + k_2\Delta x^2 + k_3\Delta x^3 \tag{2}$$

$$f_d = c_0\Delta\dot{x} + c_1\Delta\dot{x}^2 \tag{3}$$

Where, $\Delta x = x - x_0$, $k_0 = -2216.5N$, $k_1 = 12294N/m$, $k_2 = -729696N/m$, $k_3 = 3090400N/m^3$, $c_0 = 1379.9N\cdot s/m$, $c_1 = 519.37N\bullet s^2/m^2$

f_k *and* f_d substituted the equation (1), and by making $y = \Delta x$:

$$\ddot{y} = -\ddot{x}_0 - B_0 - B_1 y - B_2 y^2 - B_3 y^3 - C_0\dot{y} - C_1\dot{y}^2 \tag{4}$$

Where, $B_0 = k_0/m$, $B_1 = k_1/m$, $B_2 = k_2/m$, $B_3 = k_3/m$, $C_0 = c_0/m$, $C_1 = c_1/m$

Order $y_1 = y$, $y_2 = \dot{y}$, introducing control signal u(t) into the model:

$$\begin{cases} \dot{y}_1 = y_2 \\ \dot{y}_2 = -\ddot{x}_0 - B_0 - B_1 y_1 - B_2 {y_1}^2 - B_3 {y_1}^3 \\ \quad - C_0 y_2 - C_1 {y_2}^2 + u(t) \end{cases} \tag{5}$$

The equation (5) is corrected:

$$\begin{cases} \dot{y}_1 = y_2 \\ \dot{y}_2 = -\ddot{x}_0 - B_0 - (B + \Delta B) y_1 - (C_0 + \Delta C) y_2 \\ \quad + f(y_1, y_2, t) + d(t) + u(t) \end{cases} \tag{6}$$

Where, ΔB and ΔC are time-varying part of B and C_0; $f(y_1,y_2,t)$ is unknown nonlinear part of system model; d (t) is the excitation of random road.

q(t) is uncertainties inside the system, that is:

$$q(t) = -\Delta B y_1 - \Delta C y_2 + f(y_1, y_2, t) \tag{7}$$

The equation (6) can be further finished as follows:

$$\begin{cases} \dot{y}_1 = y_2 \\ \dot{y}_2 = -\ddot{x}_0 - B_0 - B y_1 - C_0 y_2 \\ \quad + F(x) + u(t) \end{cases} \tag{8}$$

Where, F (t) is the sum of uncertainty factors, its expression:

$$F(t) = q(t) + d(t)$$

(9)

Assuming F(T) is bounded and satisfied $|F(t)| \le F_0$

Assuming that the tracking control target signal of the vehicle suspension system equation (5) is y_d, and the tracking error is defined:

$$e = y_1 - y_d \tag{10}$$

The change rate of error:

$$\dot{e} = \dot{y}_1 - \dot{y}_d = y_2 - \dot{y}_d \tag{11}$$

Introduce virtual control law:

$$\alpha = re \tag{12}$$

Where, the constant $r \in R^+$, introducing variable z:

$$z = \dot{e} + \alpha \tag{13}$$

According to the equation (8):

$$\begin{aligned} \dot{z} &= \ddot{e} + \dot{\alpha} = \dot{y}_2 - \ddot{y}_d + \dot{\alpha} = -x_0 - B_0 - By_1 \\ &- C_0 y_2 + F(t) + u(t) - \ddot{y}_d + \dot{\alpha} = -\ddot{x}_0 - B_0 \\ &- B(e + y_d) - C_0(z - \alpha + \dot{y}_d) - \ddot{y}_d + \dot{\alpha} \\ &+ u(t) + F(t) \end{aligned} \tag{14}$$

$$S = \lambda e + z \tag{15}$$

Because the unknown F (t) will have bad influence on control effect, the estimation value $\hat{F}(t)$ of F (t) is produced by the adaptive algorithm.

The adaptive law of $\hat{F}(t)$ is taken as:

$$\hat{F}(t) = \varepsilon S \tag{16}$$

Where, ε is the proportionality constant.

Defining estimation error:

$$\tilde{F}(t) = F(t) - \hat{F}(t) \tag{17}$$

$$\begin{aligned} u(t) &= -\lambda(z-\alpha) + \ddot{x}_0 + B_0 + B(e + y_d) + C_0(z - \alpha + \dot{y}_d) \\ &+ \ddot{y}_d - \dot{\alpha} - \hat{F}(t) - \gamma(S + \beta \operatorname{sgn}(S)) \end{aligned} \tag{18}$$

Where, the sgn(•) is the symbol switching function, γ and β are constants of greater than zero .

Selecting the appropriate r 、 λ and γ to meet $\gamma(r+\lambda)-\frac{1}{4}\rangle 0$, so that the controlled system (8) is gradually stable, that is, which meet $t \to \infty$, at that time $e(t) \to 0, \dot{e}(t) \to 0$.

IV. SIMULATION MODEL

According to the established system of differential equations (8) and controller equation (18), we build a simulation model of the suspension system in Matlab as shown in Figure 3.

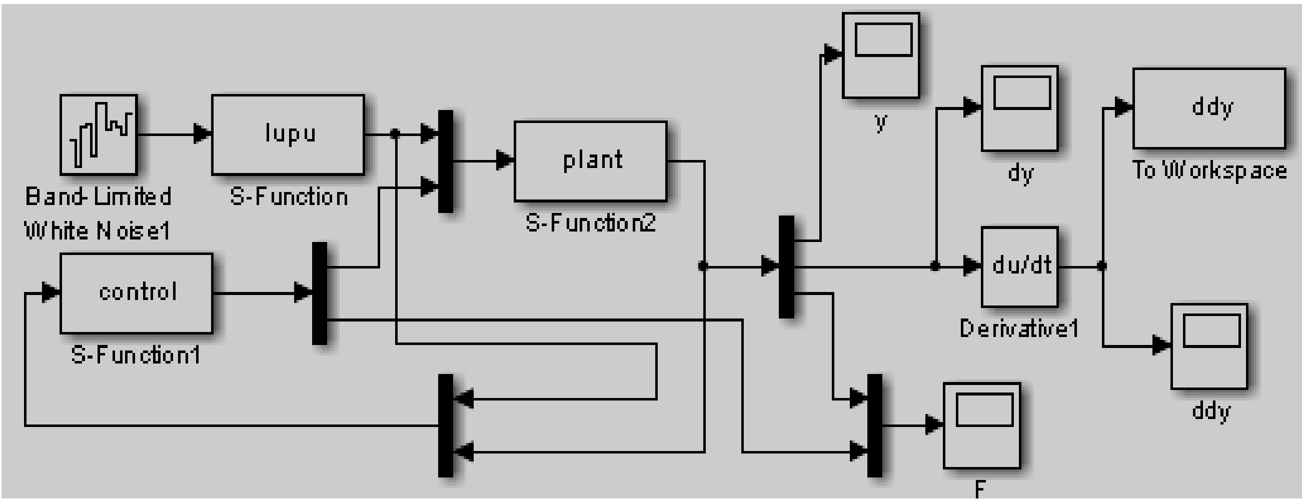


Figure 3. Matlab/Simulink simulation model of suspension system.

V. SIMULATION RESULTS ANALYSIS

We get $y_d = 0$ as the reference target, and assume that the initial estimate value of the uncertain influence is 0, and selection of related parameters r=10, γ=5, β=0.08,

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ε=1500.We use the hyperbolic function tanh(•) to replace the switching function sign（•）in the controller [11]. To verify the robustness of the control method, we introduce the slowly varying signal q(t)=10sin(t) to describe uncertainty of system parameters time-varying and mode nonlinear term in simulation.

A. *q(t)=10sin(t),d(t)=0*

We only considered time-varying and the uncertainty of the nonlinear term in the system. The vehicle is only affected by the road excitation, so F (t) = 10 sin (t). In y_d=0 the vehicle is traveling at a speed of 20 m/s in B level road, and vertical vibration displacement and velocity diagram of the controlled suspension system are as shown in Figure 4 and 5 respectively. It shows that vertical vibration displacement and velocity has periodic stable change in small scope around zero besides a very short time fluctuation at the beginning, which shows the suspension system can still obtain expected steady state even in time-varying interference of sine law.

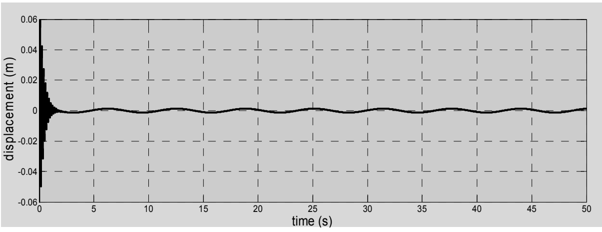

Figure 4. Vertical vibration displacement diagram.

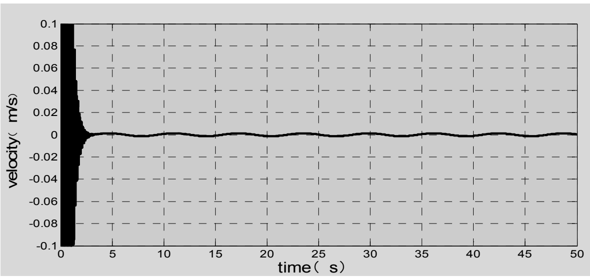

Figure 5. Velocity diagram.

The time domain diagram of the vertical vibration acceleration of the suspension system with control and without control are shown in Figure 6—8 . Compared with Figure 6 and 7 without control the vertical acceleration highly change irregular, and the change is bigger, which reflects the poor vehicle ride. With control the acceleration tends to be stable and single cyclical change, and change greatly decreases, which shows significant improvement of vehicle comfort. Compared with no interference estimation control ε=0, the back-stepping sliding mode control with adaptive interference estimation is better (Figure 8).

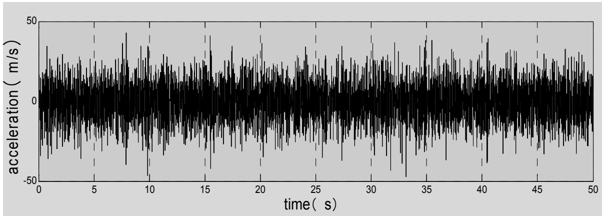

Figure 6. Vibration acceleration diagram witrh control.

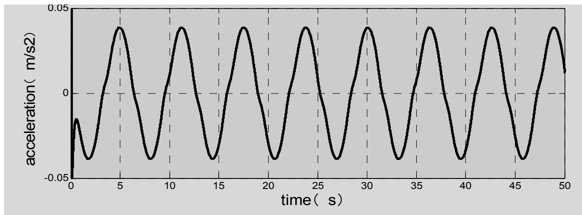

Figure 7. Vibration acceleration diagram with control ε =0.

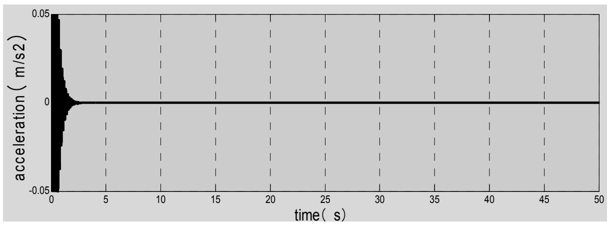

Figure 8. Vibration acceleration diagram with control ε =1500.

B. *q(t)=10sin(t)，d(t)=-3sin（0.1t）*

At that moment, we consider the uncertainties of the suspension system parameters and the nonlinear of the model, random road surface disturbance and other factors. The vehicle is going at a constant speed of 20 km/h on the B level road, when the vehicle suspension system is in control, the time domain response curve of vibration displacement, velocity and acceleration is shown in Figure 9-11. we can see from the figure that even in time-varying of system parameters, model's nonlinear and uncertain factors such as random interference of road conditions, the displacement, velocity and acceleration are stable periodic vibration near zero, and the numerical also greatly reduced after the implementation of adaptive back-stepping sliding mode control.

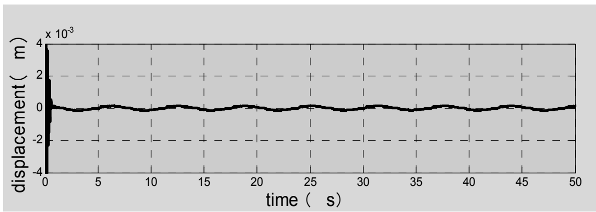

Figure 9. Displacement diagram.

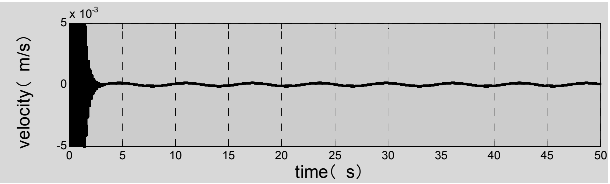

Figure 10. Velocity diagram.

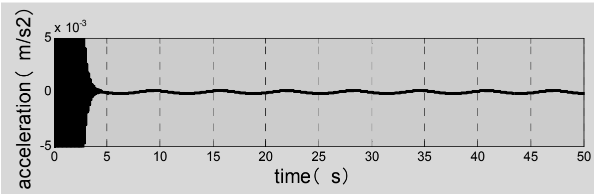

Figure 11. Acceleration diagram.

VI. CONCLUSION

In the complicated circumstances of parameter time-varying, outside the road disturbance incentive and the function of the system strongly nonlinear uncertain factors, with sliding mode variable structure controller with adaptive inversion of automobile active new hydraulic feedback damping suspension system, the vertical vibration of suspension system is still a good inhibition in different grades and different speed of the state of the random road conditions, and makes human sensitive frequency range of vertical vibration acceleration decrease obviously, which is very good to improve the vehicle ride comfort and comfort.

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