Research on Energy Reclaiming Active Suspension Control Strategy Based on Linear Motor and Hydraulic Hybrid System

Zhijun Deng¹, Bo Shi², Dongping Wei³ and Shengchang Wang⁴

¹School of Automotive and Transportation Engineering, Shenzhen Polytechnic, Shenzhen, China
²BYD Auto Industry Co., Ltd, Shenzhen, China
³Department of Math and Physics, Shenzhen Polytechnic, Shenzhen, China
⁴School of Automobile, Chang’an University, Xi’an, China
Email: 821126299@qq.com

Abstract. Based on linear motor and hydraulic hybrid energy reclaiming suspension, control strategy for multi-mode switched active suspension is designed. By establishing a quarter vehicle suspension system model, a fuzzy sliding mode controller is designed to control the active suspension system, and design active suspension mode switching strategy to select and switch suspension modes of vehicles under different conditions, it improves the performance of active suspension as well as the feed-ability of the suspension. The accuracy of the model and control strategy is verified by simulation, at the same time, a quarter suspension bench is built for test. The test results show the effectiveness of the simulation. The energy-feed test shows that the energy-feed efficiency of the designed energy reclaiming suspension can reach 17.54%.

Keywords. Energy reclaiming, Active suspension, Fuzzy sliding mode control, Multi mode conversion.

1. Introduction

As a new type of suspension system, energy reclaiming suspension can be divided into liquid-point type, piezoelectric type and electromagnetic type according to the way of vibration energy recovery. Among them, electromagnetic suspension has become a hot spot in the research of energy reclaiming suspension due to its fast response, good controllability and wide application range of energy recovery [1]. Suitable suspension control strategies can provide good performance for driving. Common control strategies include sky-hook control, ground-hook control, hybrid control strategy of sky-hook control and ground-hook control [2-4] and LQG control [5]. With the development of modern control theory, many control algorithms, such as \( H_\infty \) control, adaptive control, sliding mode control [6-7], are widely used in suspension system.

The main component of active suspension with electromagnetic energy feed is electromagnetic actuator. In this paper, the switching and selection of different suspension modes are used to realize the switching of several suspension actuator modes, such as electromagnetic active, electromagnetic energy feed, electro-hydraulic mixing, so as to realize the control of active suspension with energy feed and improve the performance of the active suspension system. Finally, the suspension system model is established for simulation and experimental verification.
2. Quarter Vehicle Active Suspension Model
A quarter vehicle active suspension model based on a hybrid electro-magnetic and hydraulic actuator can be simplified as shown in Figure 1.

![Figure 1](image)

Figure 1. 1/4 Vehicle feed suspension model. Note: $m_s$ is the weight mass of the body (spring mass); $m_t$ is the tire mass (unspring mass); $k_s$ is the spring stiffness; $k_t$ is the tire stiffness; $x_s$, $x_t$, $q$ are the vertical displacement coordinates of the vehicle body, wheel and ground relative to the coordinate origin, where the coordinate origin is their respective equilibrium position; $c_0$ is the basic damping coefficient of the throttle valve adjustable shock absorber.

The equation of motion for the energy reclaiming active suspension system is:

$$
\begin{align*}
& m_s \ddot{x}_s + c_0 (\dot{x}_s - \dot{x}_t) + k_s (x_s - x_t) + F_t + F_e = 0 \\
& m_t \ddot{x}_t - c_0 (\dot{x}_s - \dot{x}_t) - k_t (x_s - x_t) + k_t (x_t - q) - F_t - F_e = 0
\end{align*}
$$

where $F_t$ is the regulating damping force of the throttle valve adjustable damper; $F_e$ is the electromagnetic force generated by a linear motor.

Select the state vector $X = [x_t \ x_s \ \dot{x}_t \ \dot{x}_s]^T$, convert equation (1) to equation of state:

$$
\dot{X} = AX + BU + Fq \tag{2}
$$

where

$$
A = \begin{bmatrix}
0 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 \\
-(k_s / m_s) & k_t / m_t & -c_0 / m_t & c_0 / m_t \\
k_s / m_s & -k_t / m_t & c_0 / m_s & -c_0 / m_s
\end{bmatrix}, \quad
B = \begin{bmatrix}
0 \\
0 \\
1 / m_t \\
-1 / m_s
\end{bmatrix}, \quad
F = \begin{bmatrix}
k_t / m_t \\
0 \\
0 \\
0
\end{bmatrix}, \quad
U = F_t + F_e, \quad q
$$

is road input.

3. Design of Control Strategy for Energy Reclaiming Suspension

3.1. Design of Sliding Mode Controller
Sliding mode control is a kind of special nonlinear control in essence. The control has discontinuity, strong robustness and adaptability to external interference. It is suitable for vehicle suspension system. In recent years, the application of sliding mode control in suspension system has been widely studied. Sliding mode control essentially makes the system a hypersurface in the state space $s(x) = 0$. In other words, the "sliding mode area" moves with high frequency and small amplitude along the prescribed state track, which is called "sliding mode motion" [8]. When the moving point in the sliding mode area reaches the switching surface $s = 0$, the following conditions should be satisfied:

$$
\lim_{s \to 0} ss \leq 0 \tag{3}
$$
According to the sliding mode control rules, the simultaneous equation (2) and the sliding mode switching equation can be obtained as follows:

\[
\begin{align*}
\dot{X} &= AX + BU + Fq \\
\dot{s} &= CX \\
\end{align*}
\]

(4)

In equation (4): \( s \) is the sliding mode switching surface function, \( C = [c_1 \ c_2 \ c_3 \ c_4] \) is the coefficient matrix of the switching function. In order to simplify the calculation, we usually take \( c_4 = 1 \).

For the above controlled system, since the input coefficient matrix \( \text{rank}(B) = 1 \) is full column rank, then \( B \) can be decomposed into \( B = [B_1 \ B_2]^T, B_1 = [0 \ 0 \ 1/m_1] \), \( B_2 = [-1/m] \). Because \( \det(B_2) \neq 0 \), there exists a nonsingular linear transformation matrix \( Q \), which makes \( X' = QX \) hold. Because it is necessary to ignore the external disturbance in the state equation when designing the sliding mode switching function, then equation (4) becomes:

\[
\begin{align*}
\dot{X}' &= A'X' + B'U \\
\dot{s}' &= C'X' \\
\end{align*}
\]

(5)

where \( A' = QAQ^{-1} = \begin{bmatrix} A_{11} & A_{12} \\ A_{21} & A_{22} \end{bmatrix}, Q = \begin{bmatrix} I_3 & -B_1B_2^{-1} \\ 0 & 1 \end{bmatrix}, B' = QB = [0 B_2]^T, C' = CQ^{-1} = [C'_1 \ C'_2] \).

After the above transformation, the original system state equation and sliding mode switching equation are changed into:

\[
\begin{align*}
\dot{X}_1 &= A_{11}'X_1' + A_{12}'X_2', \\
\dot{X}_2 &= A_{21}'X_1' + A_{22}'X_2' + B_2'U \\
\dot{s}' &= C_1'X_1' + C_2'X_2' \\
\end{align*}
\]

(6)

when the system enters sliding mode, the equation should be satisfied: \( s' = 0, \dot{s}' = 0 \). By simultaneous equations (5) and (6), it can be obtained that:

\[
\dot{X}_1' = (A_{11}' - A_{12}'C_1'C_2^{-1})X_1' \\
\]

(7)

Suppose \( K = C_1'C_2^{-1} \), \( K \) is the state feedback control gain matrix, \( A_{11}' \) and \( A_{12}' \) are controllable matrices. Thus, the characteristic polynomial of equation (7) can be calculated as follows:

\[
D(\lambda) = -\lambda^3 + \frac{m_1c_2 - m_1c_4}{m_1 - c_1m_1}\lambda^2 + \frac{k_1}{m_1 - c_1m_1}\lambda + \frac{k_2}{m_1 - c_1m_1} \\
\]

(8)

In order to satisfy Hurwitz condition and make sliding mode stable [9], the following conditions should be satisfied:

\[
\begin{align*}
\frac{m_1c_2 - m_1c_4}{m_1 - c_1m_1} &> 0, \\
\frac{k_1}{m_1 - c_1m_1} &> 0, \\
\frac{k_2}{m_1 - c_1m_1} &> 0, \\
\frac{k_2(m_1c_2 - m_1c_4)}{(m_1 - c_1m_1)^2} &> 0 \\
\end{align*}
\]

(9)

Constant velocity reaching law is selected for the sliding mode control rate:

\[
\dot{s} = -\varepsilon \text{sgn}(s) \\
\]

(10)

where \( \varepsilon \) is the rate at which the moving point approaches the sliding mode switching surface, and the value of \( \varepsilon \) will affect the stability of the system; \( \text{sgn}(s) \) is the symbol function.
In order to reduce the control error, the saturation function \( \text{sat}(s) \) is used to replace the symbolic function \( \text{sgn}(s) \). Substituting equation (10) into equation (4), the damping force required by suspension system can be obtained as follows:

\[
U = -(CB)^{-1}[CAX + CFq + \varepsilon \text{sat}(s)]
\]  

(11)

3.2. Design of Fuzzy Controller

In order to eliminate the instability and chattering caused by sliding mode control in the control process, combining sliding mode control with fuzzy control, it is fuzzy sliding mode variable structure control [10], it is used to realize the optimization of the two control methods in the active suspension control problem. In equation (10), the value of coefficient \( \varepsilon \) affects the stability of the system, so fuzzy controller is selected to optimize the value of coefficient.

In this paper, the sliding mode switching surface function \( s \) and \( \dot{s} \) are used as the input of the fuzzy controller, and the coefficient \( \varepsilon \) is the output value. The input variables \( s \), \( \dot{s} \) and output \( \varepsilon \) of the system are divided into seven different variables. The process of fuzzification adopts triangular membership function, and the fuzzy subdomain is \([-3 -2 -1 0 1 2 3]\). The fuzzy control rule table based on barycenter method is established as shown in table 1.

### Table 1. Table of fuzzy control rules based on center of gravity method.

| \( \varepsilon \) | NB | NB | NB | NB | NB | NB | NB |
|------------------|----|----|----|----|----|----|----|
| \( s \)           | NB | NB | NB | NM | NM | NS | NS |
|                  | NM | NB | NB | NM | NS | NS | NS |
|                  | NS | NM | NM | NS | ZE | ZE | ZE |
| \( \dot{s} \)    | ZE | NM | NM | NS | ZE | ZE | PS |
|                  | PS | NS | NS | ZE | ZE | PS | PS |
|                  | PM | ZE | PS | PS | PS | PM | PB |
|                  | PB | PS | PS | PM | PB | PB | PB |

3.3. Control Strategy Design of Suspension Mode Switching

Due to the influence of vehicle driving conditions, environment and other factors, the damping force required for active suspension is constantly changing during driving. In order to improve the ride comfort and stability of the vehicle, as well as the energy feedback of the electromagnetic suspension, it is necessary to switch between different modes of energy-feed suspension when working.

Since the root mean square value of vehicle body acceleration is proportional to the road roughness coefficient and vehicle speed [11], the changes of road surface and vehicle speed have a direct impact on vehicle body acceleration. In case of poor road conditions and high vehicle speed, the energy feedback effect of energy reclaiming suspension is significant [12]. Generally, the energy feedback efficiency is higher on the road surface above grade C. Therefore, the simulated speed is selected as 20m·s\(^{-1}\). The average root mean square value of vehicle body acceleration \( \bar{x}_c \), \( \bar{x}_D \) and the average value of root mean square value of tire deflection \( \bar{x}_c - q_c \), \( \bar{x}_D - q_D \) on the simulated road surface of grade C and D are selected as the threshold value.

In the subsequent simulation, Simulink/Stateflow is used for modeling, the input signal needs to be changed into logical variables. If the suspension performance evaluation index is greater than the threshold value, it means that the index needs to be adjusted by mode switching, and the output is 0; otherwise the output is 1. Through the switching strategy, the working mode of energy reclaiming suspension can be divided into four modes: standard mode, comfort mode, stability mode and energy feedback mode. The mode control switching strategy is shown in table 2.
Table 2. Automatic mode switching control strategy for energy reclaiming active suspension.

| Logic value of body acceleration | Logical value of tire deflection | Suspension working mode          |
|----------------------------------|----------------------------------|----------------------------------|
| 0                                | 0                                | Standard mode                    |
| 0                                | 1                                | Comfort mode                     |
| 1                                | 0                                | Stability mode                   |
| 1                                | 1                                | Energy feedback mode             |

The standard mode is the default mode of the energy reclaiming suspension mode, and the effect of this mode is between the other three modes. The standard mode conversion is shown in Table 3.

Table 3. Standard mode conversion strategy for energy reclaiming active suspension.

| Suspension status | Relative displacement | Relative speed | Dynamic mode                                      |
|-------------------|-----------------------|----------------|---------------------------------------------------|
| Stretching        | $x_s - x_t > 0$       | $\dot{x}_s - \dot{x}_t \geq 0$ | Electromagnetic active and hydraulic semi-active |
|                    |                       | $\dot{x}_s - \dot{x}_t \leq 0$ | Electromagnetic energy feedback and hydraulic semi-active |
| Compress          | $x_s - x_t < 0$       | $\dot{x}_s - \dot{x}_t \leq 0$ | Electromagnetic active and hydraulic semi-active |
| Static            | $x_s - x_t = 0$       | $\dot{x}_s - \dot{x}_t = 0$   | -                                                 |

The design idea is: when the suspension system starts to work and the vehicle suspension state changes. If $x_s - x_t > 0$, the suspension is in the stretching state. At this time, if $\dot{x}_s - \dot{x}_t \geq 0$, the suspension is stretched to a large extent, which requires the motor to generate electromagnetic force to hinder its deformation; if $\dot{x}_s - \dot{x}_t < 0$, the suspension has a small degree of tension, and the elongation is reduced, the suspension has a compression trend, and then it is converted to electromagnetic energy feedback mode. If $x_s - x_t < 0$, the suspension is in the state of compression. At this time, if $\dot{x}_s - \dot{x}_t \geq 0$, the degree of suspension compression is small, and the amount of compression is reduced, the suspension tends to stretch, and the electromagnetic energy feedback circuit starts to work; if $\dot{x}_s - \dot{x}_t < 0$, the suspension compression degree is large, and electromagnetic force is required to work on the suspension to maintain the stability of the vehicle body.

When the suspension works in the energy feedback mode, the linear motor turns into a generator and starts to recover the vibration energy caused by road excitation and store it in the energy storage element. At this time, the energy feedback control circuit is connected and starts to work. In order to prevent the phenomenon of "dead zone" during charging [13], this study uses a boost converter to boost the rectified electric energy, so that the voltage reaches the threshold voltage of charging at both ends of the super capacitor as an energy storage element, and the super capacitor can be charged. The motor energy feedback circuit includes four parts: linear motor, full wave bridge rectifier, boost converter and super capacitor. The topology of feed energy control circuit is shown in Figure 2.

Figure 2. Topology of feed energy control circuit. Note: $E_a$, $E_b$ and $E_c$ represent the induced electromotive force of each phase of the linear motor; $L_a$, $L_b$, $L_c$ denote the inductance of each winding; $L_d$ represents the energy storage inductance; VT represents the MOS transistor controlling the circuit on and off; SC represents the super capacitor.
In order to prevent the stability of the system from being affected by too frequent mode switching in a short period of time during the active suspension action, the system stability module should be designed. Considering the response time of linear motor and adjustable orifice hydraulic shock absorber, the reference time of mode switching is determined as 1s. If the input duration of logic signal is greater than 1s, the logic signal is output for the system judges whether to switch mode. The stateflow model of suspension mode switching control strategy is shown in figure 3. By judging the switching function coefficient matrix $C$ of output fuzzy sliding mode control, the output signals of $C$ are 1, 2, 3, and 4, and the corresponding suspension working modes are standard mode, stable mode, comfort mode and energy feedback mode. The coefficient matrix $C$ corresponding to different modes is also different. The damping force of suspension can be controlled by changing the output of coefficient matrix $C$, so as to achieve the purpose of suspension adapting to the influence of different working conditions on its performance; “a” and “x” in figure 3 represent the logic signals of body acceleration and tire deflection, and “cf” and “d” correspond to the gears of hydraulic shock absorber and linear motor with adjustable throttle aperture respectively.

**Figure 3.** Mode switching control stateflow model.

4. Active Suspension Control Strategy Simulation and Experiment

4.1. Simulation of Suspension Control Strategy

The simulation model of suspension control strategy is established in Matlab / Simulink software. The road input model selects a filtered white noise as the road input, the road input expression is:

$$
\dot{q}(t) = -2\pi n_0 u q(t) + 2\pi n_0 \sqrt{Gq} w(t)
$$

where $n_0$ is the lower cut-off space frequency, $u$ is vehicle speed, $n_0$ is the reference space frequency, $Gq$ is road roughness coefficient, $w(t)$ is Gaussian white noise with mean value of zero.

In this simulation, the C-grade pavement ($Gq=256 \times 10^{-6}$m$^3$) is used, the vehicle speed $u$ is taken as 20m/s$^{-1}$. The basic parameters of vehicle suspension are selected as follows: spring mass $m_s$ is 289 kg, unspring mass $m_t$ is 45 kg, spring stiffness $k_s$ is 18182N/m, and tire stiffness $k_t$ is 190000N/m, the basic damping coefficient $c_0$ is 750N·s·m$^{-1}$.

Through the established suspension control strategy model and disconnecting the mode switching module, the comparative simulation results of celestial control semi-active suspension, sliding mode control active suspension and fuzzy sliding mode control active suspension on vehicle body acceleration, suspension deflection and tire dynamic travel are shown in figures 4-6 and table 4.
Figure 4. Comparison of vehicle body acceleration response curves.

Figure 5. Comparison of suspension deflection response curves.

Figure 6. Comparison of tire deflection response curves.

Table 4. Simulation data results.

| Root mean square value of suspension performance index | Skyhook Control | Sliding mode control | Fuzzy sliding mode control |
|-------------------------------------------------------|----------------|----------------------|---------------------------|
| Vehicle body acceleration /m·s⁻²                      | 1.5230         | 1.2320               | 1.0680                    |
| Suspension deflection /m                              | 0.0130         | 0.0113               | 0.0094                    |
| Tire deflection/m                                     | 0.0088         | 0.0072               | 0.0069                    |

Figures 4-6 and table 4 show that the performance of the suspension system under the sliding mode control and fuzzy sliding mode control strategy is better than that of the skyhook control. Specifically, compared with skyhook control, sliding mode control has decreased 19.11%, 13.08% and 18.18% respectively in vehicle body acceleration, suspension deflection and tire deflection, which represent suspension performance. Compared with the single sliding mode control strategy, on the basis of sliding mode control, fuzzy sliding mode control has decreased by 13.31%, 16.81% and 4.17% respectively. Meanwhile, the fuzzy sliding mode control strategy can reduce the curve amplitude, improve the chattering problem of sliding mode control to a certain extent, and improve the performance of active suspension.

4.2. Control Strategy Experiment of Suspension

In order to verify the actual effect of the designed suspension control strategy, a quarter suspension bench is built for test, as shown in figure 7.
There are two kinds of road excitation used in the test: sinusoidal road surface and swept frequency road surface. Firstly, sinusoidal road surface is taken as road input, and the test and simulation results of root mean square value of suspension performance are shown in Table 5.

Table 5. Comparison of suspension performance simulation and experiment results.

| Root mean square value of performance evaluation index | Simulation result | Experiment result | Error |
|------------------------------------------------------|-------------------|------------------|-------|
| Vehicle body acceleration /m·s$^{-2}$                | 0.47831           | 0.49120          | 2.62% |
| Suspension deflection /m                              | 0.00763           | 0.00813          | 6.17% |
| Tire deflection /m                                    | 0.00108           | 0.00112          | 7.14% |

Table 5 shows that there are certain errors in the test results compared with the simulation results, among which the errors of suspension deflection and tire deflection are larger, while the error of body acceleration is small. This is because the acceleration sensor can directly collect the vehicle body acceleration signal, and the error is small after ignoring the clutter interference; while the suspension deflection and the tire deflection need to be sensed to the relative displacement. The displacement information is processed by a series of differential and integral processing, so the displacement signal error is large. In conclusion, the simulation results are consistent with the experimental results after ignoring the influence of errors, which verifies the effectiveness of the simulation results.

After adding the mode switching control model, according to the verification design road conditions of the model: when $0 < t < 10s$, the frequency of sinusoidal road is 2Hz, the amplitude is 0.01M; when $10 \leq t < 20$, the frequency of sinusoidal road is 3Hz, the amplitude is 0.01M; when $20 \leq t < 30$, the frequency of sinusoidal road is 3Hz, the amplitude is 0.02M; when $30 \leq t < 40$, the frequency of sinusoidal road is 2Hz, and the amplitude is 0.02M. The simulation results of suspension mode switching logic signal output are shown in Figure 8.

Figure 8 shows that the suspension mode at the beginning is the standard mode. After judging the driving conditions of the vehicle, it is switched to the comfort mode to improve the ride comfort of the
vehicle. At the same time, the first section of the road is relatively gentle, so after satisfying the ride comfort performance of the vehicle, it switches to the energy feedback mode to recover the energy of vehicle body vibration caused by road excitation; the road surface of the second and third section fluctuates greatly, so the mode switching controller switches the comfort mode and stability mode of suspension mode according to the driving conditions, so as to improve the ride comfort and stability of the vehicle qualitative; in the last section of the road surface undulation is reduced, so when the driving performance of the vehicle meets the requirements, it switches to the energy feedback mode and continues to recover and store the vibration energy.

In order to verify the effect of mode switching on suspension performance, the evaluation indexes before and after the mode switching of suspension are simulated. As shown in figure 9 and table 6.

Figure 9. Comparison of suspension performance before and after mode switching.

Table 6. Comparison of results before and after mode switching.

| Root mean square value of suspension performance index | Before switching | After switching | Optimization percentage |
|--------------------------------------------------------|-----------------|----------------|------------------------|
| Vehicle body acceleration /m·s^{-2}                    | 0.2989          | 0.2606         | 12.81%                 |
| Tire deflection /m                                      | 0.0006          | 0.0005         | 16.67%                 |

Figure 9 and table 6 show that when the frequency and amplitude of sinusoidal road are small, the optimization of suspension performance after mode switching is small, and the road excitation energy is mainly recovered; when the sinusoidal road surface fluctuates greatly or violently, the performance of suspension is mainly improved after mode switching. And the performance of active suspension can be improved by switching the suspension mode in the whole experimental process compared with that without mode switching, it can improve the performance of active suspension, and improve the energy feedability of the suspension system.

The road excitation is changed to 0 Hz to 15 Hz Sinusoidal sweep road input, and the set parameters are as follows: the conventional sweep frequency is set to 1Oct/Min, the conventional compression rate is set to 0.5dB/s, the sweep frequency is set to 1, and the sweep direction is upward sweep frequency. Set the suspension mode to the energy feedback mode, the initial terminal voltage of energy storage element is set as 8V, the results are shown in figure 10.

Figure 10. Comparison of road excitation input energy and regenerative energy.
Figure 10 shows that under the sinusoidal sweep road excitation, the vibration energy recovered by the suspension system changes with the change of excitation. Specifically, with the change of time, the frequency of sinusoidal sweep excitation increases and the amplitude decreases. At the initial moment, although the amplitude of sinusoidal excitation is large, the vibration frequency is small, so the energy recovery is less at this stage, which is almost zero at the beginning; with the increase of input excitation vibration frequency, the linear motor motion intensifies, and the vibration energy recovery of suspension system also begins to increase. In the last stage, because the vibration amplitude of input excitation decreases continuously, although the vibration frequency is large, the motion of linear motor is relatively gentle, so the energy recovery is less. In the whole process, the input energy of sinusoidal sweep road excitation is 794J, the energy recovered from energy feedback is 139.3J, and the energy feedback efficiency reaches 17.54%.

5. Conclusion
By establishing a quarter vehicle energy reclaiming active suspension system model based on electromagnetic force and hydraulic hybrid actuator, a fuzzy sliding mode control strategy is designed to control the active suspension, and a suspension mode switching strategy is designed to judge the performance status of the active suspension by comparing the set threshold value, so as to realize the multi-mode switching of active suspension in the process of vehicle driving. The effectiveness of the fuzzy sliding mode control strategy and suspension mode switching strategy for active suspension control is verified by simulation and experiment. The ride comfort and stability of the vehicle are effectively improved, and the energy feedback efficiency of the suspension reaches 17.54%, which improves the energy feedback efficiency of the suspension.

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