Research Article

Active Fault-Tolerant Control Based on the Fault of Electromagnetic Hybrid Active Suspension

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In order to improve the ride comfort and handling stability of the vehicle and realize the recovery of vibration energy, an electromagnetic linear hybrid suspension actuator composed of linear motor and solenoid valve shock absorber is proposed. At the same time, a fault diagnosis and fault-tolerant control strategy is designed to solve the system instability caused by the fault of electromagnetic hybrid active suspension. The 1/4 vehicle two-degree-of-freedom suspension model, the linear motor mathematical model, and the solenoid valve shock absorber test model are established. In this paper, the fuzzy sliding mode controller is used as the controller and the unknown input observer is used to estimate the state of the suspension. According to the residual obtained from the unknown input observer and compared with the residual threshold, the suspension fault is determined. In the case of fault, the fuzzy sliding mode controller is used to compensate the force and realize the suspension fault-tolerant control. The performance of the suspension is simulated on random road and bumped road, respectively. The simulation results show that the fault-tolerant control effect of the three performance indexes of the suspension is good, and the ride comfort and safety of the suspension are improved.

1. Introduction

The active suspension can adjust the stiffness and damping and improve the comfort and stability of the vehicle [1–3]. At present, the research on active suspension is mostly carried out under the condition of all components in good condition [4, 5]. However, when the actuator or sensor of suspension shows fault, the active suspension cannot provide ideal force according to the designed requirement, which influences the safety of suspension [6–8].

Fault-tolerant control can provide real-time compensation force for the suspension based on the detection result through fault diagnosis and isolation [9–11]. In recent years, many scholars have researched fault-tolerant control of vehicle suspension system from different perspectives, including passive fault-tolerant control method and active fault-tolerant control method. The fault-tolerant control objects mainly include the suspension actuator and sensor.

Fault diagnosis technology is a comprehensive technology, which involves modern control theory, fuzzy set theory, reliability theory, and signal processing [12]. As for the fault diagnosis of the suspension system, it is mainly realized by the state observer. The state observation value obtained by the state observer is compared with the residual threshold, the suspension fault is determined. In [13], Chen et al. used the extended Kalman filter bank to diagnose the fault of the electronically controlled air suspension system due to frequent sensor faults during the vehicle height adjustment process. In [14], Commault et al. used the observer group to estimate the state of the system and detect system faults. In [15], Gao et al. proposed an active fault-tolerant control method for electric power steering, aiming at the influence of sensor and actuator faults on EPS performance. Besides, ABBAS [16] used a group of sliding mode observers to detect and isolate the
sensor fault, and estimated the sensor fault based on the residual value information. Then, according to the results of fault detection and isolation, the sensor fault value was used to replace the failed sensor signal to achieve fault-tolerant control. In [17], Du et al. used an unknown input observer to detect the fault of MR damper; at the same time, the correlation coefficient method based on system residual was used to isolate the fault of MR damper.

Active fault-tolerant control is to redesign the controller based on the results of fault diagnosis after a fault occurs, so that the control system is stable, and at the same time, the performance of the control system is close to the performance level when it is in a fault-free state. [18–20]. In [21], Sun et al. designed a reliable fault-tolerant controller based on adaptive compensation control method considering actuator fault, parameter uncertainty and external disturbance of semi-active suspension. Yang and Chen [22] proposed an active fault-tolerant control strategy based on sensor signal reconstruction in order to improve the control effect of vehicle active suspension with sensor fault. Morato et al. [23] designed a fault-tolerant dynamic output-feedback controller for Semi-Active Suspension Systems, and the performance of the proposed control structure is demonstrated through simulation. In [24], Liu et al. proposed an observer-based active fault-tolerant controller for a half-vehicle active suspension system subjected to the actuator fault as the nonzero offset fault. Tudon-Martinez et al. [25] proposed a novel fault-tolerant strategy to compensate multiplicative actuator faults in a semi-active suspension system. Simulation results show that the proposed fault-tolerant semi-active suspension improves the vehicle comfort up to 60% with respect to a controlled suspension without fault-tolerant strategy and 82% with respect to a passive suspension.

In this paper, a fault diagnosis and fault-tolerant control method is designed based on electromagnetic linear hybrid actuator suspension (EMLHAS). In Section 2, the structure and working principle of EMLHAS are introduced, which can work in three different modes according to different driving conditions. In Section 3, the 1/4 vehicle two degree of freedom model, linear motor mathematical model, and solenoid valve shock absorber test model are established. In Section 4, the models of fuzzy sliding mode controller and unknown input observer (UIO) are established, and based on these two models, the fault diagnosis and fault-tolerant control methods are designed. In Section 5, the simulation analysis is completed, including the performance analysis of unknown input observer and fault-tolerant control under different road inputs. Finally, in Section 6, the bench test of the fault-tolerant control is done.

2. Structure and Principle of the EMLHAS System

The structural of EMLHAS system is shown in Figure 1 [26], which includes EMLHAS actuator (composed of linear motor and solenoid valve shock absorber in series), hydraulic damper, coil spring, controller, power module circuit, super capacitor, acceleration sensor etc.

The working modes of the EMLHAS can be divided into three modes, which are energy regenerative state, semi-active state and active state. The suspension controller can control the two components of EMLHAS to work in different modes according to different requirements.

The structure of EMLHAS actuator is shown in Figure 2 [26], which mainly includes upper ear, permanent magnet (PM), throttle, solenoid valve shock absorber, piston, hydraulic oil, etc. EMLHAS’s linear motor and solenoid valve shock absorber work in different states according to different working modes. When EMLHAS works in energy regenerative state: the body vibration drives the linear motor to cut the magnetic induction line, and the generated electric energy is stored in the supercapacitor through the energy recovery circuit. Meanwhile, EMLHAS does not supply power to the solenoid valve shock absorber, which provides the base damping force; when EMLHAS works in the semi-active state, the suspension adjusts the current through the power module circuit to control the solenoid valve shock absorber and the linear motor is still recovering energy. When EMLHAS works in the active state, it can control the linear motor by adjusting the current through the power module circuit. The linear motor can output the corresponding active force, and the solenoid valve shock absorber provides the base value damping force.

3. Modeling of the EMLHAS Dynamic Model

3.1. Dynamic Model of EMLHAS. The two degree of freedom model can clearly reflect the dynamic characteristics of suspension. In this paper, the two degree of freedom model of 1/4 vehicle suspension is established, which is shown in Figure 3.

Based on Newton’s laws of motion, the dynamic motion equations for the quarter vehicle suspension can be expressed as

\[
\begin{align*}
\begin{cases}
\dot{m}_s \ddot{x}_s + k_s (x_s - x_u) + c_s (\dot{x}_s - \dot{x}_u) &= u, \\
\dot{m}_u \ddot{x}_u - k_s (x_s - x_u) - c_s (\dot{x}_s - \dot{x}_u) + k_t (x_u - z) &= -u,
\end{cases}
\end{align*}
\]

(1)

where \(x_s\) is the displacement of sprung mass, \(\dot{x}_s\) is the speed of sprung mass, \(\ddot{x}_s\) is the acceleration of sprung mass, \(x_u\) is the displacement of unsprung mass, \(\dot{x}_u\) is the speed of unsprung mass, \(\ddot{x}_u\) is the acceleration of unsprung mass, \(u\) is the control force of suspension, \(m_s\) is the sprung mass, \(m_u\) is the unsprung mass, \(z\) is the displacement of road input, \(k_t\) is the tire stiffness coefficient, \(k_s\) is the damping coefficient.

The state variable vector, output vector, control input vector, and disturbance input vector of the suspension system are selected as

\[
\begin{align*}
X &= [x_s - x_u \dot{x}_s x_u - z \dot{x}_u]^T, \\
Y &= [\dot{x}_s x_s - x_u k_t (x_u - z) \dot{x}_u]^T, \\
U &= [u], \\
W &= [\dot{z}].
\end{align*}
\]

(2)
In this way, the state space equations of suspension can be expressed as follows:

\[
\begin{cases}
\dot{X} = AX + BU + EW, \\
Y = CX + DU,
\end{cases}
\]

where \( A \) is the state matrix, \( B \) is the control matrix, \( C \) is the output matrix, \( D \) is the direct connection matrix, and \( E \) is the disturbance input matrix.

\[
A = \begin{bmatrix}
0 & 1 & 0 & -1 \\
\frac{k_s}{m_s} & \frac{c_s}{m_s} & 0 & \frac{c_s}{m_s} \\
0 & 0 & 0 & 1 \\
\frac{k_{x}}{m_{u}} & \frac{c_{x}}{m_{u}} & \frac{k_{t}}{m_{u}} & \frac{c_{x}}{m_{u}}
\end{bmatrix}
\]
\[
B = \begin{bmatrix}
0 & 1 & 0 \\
0 & 0 & 1 \\
\frac{1}{m_x} & \frac{c_x}{m_x} & 0 \\
\frac{1}{m_y} & 0 & \frac{c_y}{m_y} \\
\frac{1}{m_z} & \frac{c_z}{m_z} & 0
\end{bmatrix},
\]
\[
C = \begin{bmatrix}
1 & 0 & 0 & 0 \\
0 & 0 & 1 & 0 \\
0 & 0 & 0 & 0
\end{bmatrix},
\]
\[
D = \begin{bmatrix}
\frac{1}{m_x} \\
\frac{1}{m_y} \\
\frac{1}{m_z}
\end{bmatrix},
\]
\[
E = \begin{bmatrix}
0 & 0 & -1 & 0
\end{bmatrix}^T,
\]

3.2. Mathematical Model of the Linear Motor. When the linear motor works in the active mode, the linear motor outputs electromagnetic thrust. The mathematical model of linear motor is complicated to derive under the ABC coordinate system. Without considering the harmonics and magnetic saturation effects, the voltage equation of the linear motor in the ABC phase coordinates is expressed as

\[
\begin{align*}
\psi_A &= R_Ai_A + \frac{d\psi_A}{dt}, \\
\psi_B &= R_Bi_B + \frac{d\psi_B}{dt}, \\
\psi_C &= R_CI_C + \frac{d\psi_C}{dt},
\end{align*}
\]

where \(u_A, u_B, u_C\) are the phase voltages, \(i_A, i_B, i_C\) are the phase currents, \(R_A\) is the winding resistance, and \(\psi_A, \psi_B, \psi_C\) are the three-phase winding fluxes.

The active force generated by the linear motor is expressed as

\[
F_m = \frac{\pi}{\tau} \varphi_f \left[ i_A \sin \frac{\pi}{\tau} x + i_B \sin \left( \frac{\pi}{\tau} x - \frac{2\pi}{3} \right) + i_C \sin \left( -\frac{\pi}{\tau} x + \frac{2\pi}{3} \right) \right],
\]

where \(\varphi_f\) is the permanent magnet flux.

The motion equation of linear motor is expressed as

\[
m \frac{dv}{dt} = F_m - f - B\dot{x}_s - u,
\]

where \(m\) is the mover mass, \(f\) is the linear motor load, and \(B\) is the viscous damping coefficient.

In order to simplify the mathematical model and achieve the best control of it, then the transformation of the two-phase rotating d-q coordinate system is carried out successively.

It is difficult to deduce the mathematical model of linear motor in abc three-phase coordinate system. In order to simplify the mathematical model and realize the optimal control, abc is the natural coordinate system, \(\alpha-\beta\) is the stationary coordinate system, and d-q is the coordinate system rotating at a certain angular speed. Two-phase static coordinate changes and two-phase rotation coordinate changes are carried out successively. The space vector relationship of the three coordinate systems is shown in Figure 4 [26].

In the d-q coordinate system, the state equation of the linear motor is expressed as [26]

\[
\begin{align*}
\frac{d}{dt}(id) &= \frac{R_S}{L_d}id + \frac{L_d}{L_d} \omega i_d + \frac{u_d}{L_d}, \\
\frac{d}{dt}(iq) &= \frac{R_S}{L_q}iq - \frac{L_d}{L_q} \omega i_d - \frac{\varphi_f}{L_q} \omega + \frac{u_d}{L_q} + \frac{u_q}{L_q}.
\end{align*}
\]

where \(i_d\) is the direct axis current, \(i_q\) is the quadrature axis current, \(L_d\) is the direct axis inductance, \(L_q\) is the quadrature axis inductance, \(\omega\) is the electrical angular velocity, \(u_d\) is the direct axis voltage, and \(u_q\) is the quadrature axis voltage.

In the d-q coordinate system, the active force generated by the linear motor is expressed as

\[
F_L = \frac{1.5P_n\pi \varphi_f i_q}{\tau},
\]

where \(P_n\) is the pole logarithm and \(F_L\) is the active force generated using the linear motor.

3.3. Test Modeling of the Solenoid Valve Shock Absorber. There is a nonlinear relationship between the output damping force of solenoid valve shock absorber and the relative velocity of suspension and the input current of solenoid valve shock absorber. In order to obtain the mathematical model of the solenoid valve shock absorber, the test modeling is carried out in Figure 5. The frequency is at 2 Hz and the amplitude is at 5 mm of the sinusoidal excitation from vibration table, which is taken as road input. The damping force signals are measured by a force sensor. Also, suspension displacement signals generated by the solenoid valve shock absorber are measured by using a displacement sensor. The different input current of the solenoid valve shock absorber is changed by the regulated DC power supply. Under different input current, multiple groups of force signals and displacement signals are obtained. Finally, the regression fitting curves of velocity characteristics for the solenoid valve shock absorber are obtained under various currents, as shown in Figure 6.

After analysis, there is a nonlinear relationship between the damping force, the suspension speed, and the input current of solenoid valve damper. According to the least square method, the correlation expression between the damping force and suspension speed is determined as follows [26]:

\[
F_L = \frac{1.5P_n\pi \varphi_f i_q}{\tau},
\]

where \(P_n\) is the pole logarithm and \(F_L\) is the active force generated using the linear motor.
Figure 4: Space vector relation of three coordinate systems.

Figure 5: Velocity characteristic test of the solenoid valve shock absorber.

Figure 6: Regression fitting curves of the velocity characteristic test.
\[ F_s = \sum_{k=0}^{3} (b_k I^k + c_k I + d_k)x_s^{\Delta} \]  

(10)

where \( b_k, c_k, \) and \( d_k \) are polynomial coefficients and \( I \) is the control current of the solenoid valve shock absorber.

Equation (10) is a fitting expression. The more times \( k \) is taken, the more accurate the fitting effect is. In this model, \( k \) from 0 to 3 can meet the fitting requirements of the least square method.

The parameters of the polynomial model for the solenoid valve shock absorber are obtained by regression analysis method, and the polynomial coefficients are shown in Table 1 [27]. Then the polynomial model of the solenoid valve shock absorber is obtained by taking the identified parameter results into equation (10).

4. Fault-Tolerant Control of EMLHAS

4.1. Design of the Fuzzy Sliding Mode Controller. Sliding mode control is a kind of nonlinear control, which has strong robustness and anti-interference ability. However, due to the influence of delay, system inertia, and other factors, the actual sliding mode control will have chattering problem, which will affect the stability of the control system. In order to optimize the sliding mode controller, based on the model reference sliding mode controller, the sliding mode controller is designed. The structure block diagram of fuzzy sliding mode control for EMLHAS is shown in Figure 7.

Where \( x_{sr} \) is the sprung mass displacement of reference model, \( x_{sr} \) is the sprung mass acceleration of reference model, \( \Delta K \) is the switching gain which is fuzzy controller output, \( K \) is the sliding mode switching gain, \( u_d \) is the actual output active force, \( u \) is the unconstrained input force, and \( S \) is the sliding mode switching function.

The sliding mode switching function \( S \) and its derivative \( \dot{S} \) are selected as the input variable of the fuzzy controller, and the switching gain \( \Delta K \) is selected as the output variable of the fuzzy controller. Five language sets are used to describe the fuzzy state of input variables and output variables. The five specific language sets are named [negative big, negative medium, zero, positive medium, positive big], and the short of them is [NB, NM, ZO, PM, PB], respectively.

The basic fuzzy domain of sliding mode switching function \( S \) and its derivative \( \dot{S} \) are set as \([-3, 3]\), and the basic fuzzy domain of switching gain \( \Delta K \) is set as \([-3, 3]\). When the basic fuzzy domain is determined, the membership function of fuzzy language variables, which is called the assignment of fuzzy variable, should be determined according to actual situation. In this paper, the triangle membership function is selected to fuzzify the input and output variables of fuzzy system, the membership function of input and output variables is expressed as

\[
\begin{align*}
 f(x, a, b, c) = \begin{cases} 
 0, & x \leq a, \\
 x - a, & a \leq x \leq b, \\
 \frac{c - x}{c - b}, & b \leq x \leq c, \\
 0, & x \geq c.
\end{cases} 
\end{align*}
\]  

(11)

The rules of fuzzy control are formulated according to the empirical method, and the fuzzy control rules are shown in Table 2.

According to membership functions of input and output variable and rules of fuzzy control, the input and output surfaces of the fuzzy control system are shown in Figure 8.

The upper bound \( \tilde{K}(t) \) is estimated by integral method, which is expressed as

\[ \tilde{K}(t) = G \int_0^t \Delta K dt, \]  

(12)

where \( G \) is scale factor, \( G > 0 \).

Considering the uncertainty of the system, the sprung mass acceleration of the suspension is expressed as

\[ \ddot{x}_s = -\frac{k_s}{m_s} \left(x_s - x_{u}\right) - \frac{e_k}{m_s} \left(x_s - x_{u}\right)^3 + \frac{1}{m_s} [-u + E(t)], \]  

(13)

where \( E(t) \) is the unknown disturbance.

In order to make the actual control of the EMLHAS better track the reference model, take the tracking error:

\[ e = [\dot{x}_s - \dot{x}_{sr} - \dot{\lambda} \lambda' x_s - x_{sr} - \lambda]\]  

(14)

The sliding mode function is designed by pole placement method, which is expressed as

\[ s = [1 \ c] e, \]  

(15)

where \( c \) is the sliding mode parameters, \( c > 0 \).

In order to ensure the dynamic quality of the sliding mode, the designed switching control is expressed as

\[ u_{dsw} = K(t) \sgn(s), \]  

(16)

where \( K(t) = \max|E(t)| + \eta, \eta > 0 \).

The designed sliding mode controller of EMLHAS is expressed as

\[
\begin{align*}
 & x_{sr} = x_{sr} + \frac{\Delta K}{K} \left(\dot{x}_{sr} - \dot{x}_s - \dot{\lambda} \lambda' x_s - x_{sr} - \lambda\right) + \Delta K \left(\dot{x}_{sr} - \dot{x}_s - \dot{\lambda} \lambda' x_s - x_{sr} - \lambda\right), \\
 & u_d = K(t) \sgn(s),
\end{align*}
\]  

(17)
Figure 7: The structure block diagram of fuzzy sliding mode control for EMLHAS.

Table 2: Rules of fuzzy control.

| s     | NB | NM | ZO | PM | PB |
|-------|----|----|----|----|----|
| NB    | NB | NB | NM | NM | ZO |
| NM    | NB | NB | ZO | NM | ZO |
| ZO    | NM | ZO | ZO | PM | PM |
| PM    | ZO | ZO | PM | PM | PB |
| PB    | PM | ZO | PM | PB | PB |

Figure 8: Input and output surface of the fuzzy inference system.

\[ U = m_s \left[ c_2 \ddot{e}_2 - c_1 \dddot{x}_u + c_i \left( -\frac{k_i}{m_s} (x_s - x_u) - \frac{e_k}{m_s} (x_s - x_u)^3 + a_1 (-a_1 \lambda_1 + \lambda_2) + a_2 \lambda_2 \right) \right] + K(t) \text{sgn}(s). \] (17)
4.2. Design of the Unknown Input Observer. Because the road disturbance is random, the full input observer is not suitable for suspension system. The unknown input observer can make use of the redundant output of the system to realize the progressive observation of the system state and can completely decouple the external disturbance [28]. Therefore, in this paper, the unknown input observer is used to observe the suspension state. The structure of UIO is shown in Figure 9.

The structure of the unknown input observer is expressed as [29]
\[
\begin{align*}
\dot{Z} &= GZ + TBU + KY, \\
\dot{X} &= Z + HY,
\end{align*}
\] (18)

where \( \dot{X} \) is the state estimation vector, \( Z \) is the observation vector of UIO, and \( G, T, K, H \) are matrices for unknown input decoupling.

The residual of output estimation is expressed as
\[
\rho = Y - \hat{Y}. \tag{19}
\]

\[
\begin{align*}
\dot{X} &= \hat{X} - \dot{\hat{X}} \\
&= AX + BU + EW - (\dot{Z} + HY) \\
&= (A - HCA - K_1C)\dot{X} + (A - HCA - K_1C - G)Z + [(A - HCA - K_1CH - K_2)]Y \\
&+ (B - TB - HCB - K_1D)U + (I - HC)EW - HDU,
\end{align*}
\] (20)

where \( I \) is the suitable dimensional matrix and \( K = K_1 + K_2 \).

When equation (20) satisfies equation (21), then the state estimation error is \( G\overline{X} \).

\[
\begin{align*}
HD &= 0 \\
(I - HC)E &= 0 \\
A - HCA - K_1C &= G \\
GH &= K_2 \\
B - TB - HCB - K_1D &= 0
\end{align*}
\] (21)

If \( G \) has all stable eigenvalues, \( \overline{X} \) will approach zero, and the system output residual \( r \) will also approach zero, so UIO can effectively estimate the state of the suspension system.

4.3. Fault Diagnosis and Fault-Tolerance Control. The structure diagram of fault diagnosis and fault-tolerant control is shown in Figure 10. When EMLHAS occurs fault, the fault must be diagnosed in real time. If it is judged that the EMLHAS has a fault, then the controller can control the EMLHAS to provide compensation force. The output and disturbance input of the EMLHAS are sent to the UIO, and the state observation value and the residual of sprung mass acceleration \( r_s \) are obtained through the calculation of the UIO, then the residual of sprung mass acceleration is input to the residual module. If the residual of sprung mass acceleration is bigger than the set residual threshold value, then the output signal of the residual module is transmitted to the stability module, and the stability module judges it. If the residual of sprung mass acceleration is bigger than the set residual threshold value, it is judged that the suspension occurs fault. Finally, the EMLHAS's force is compensated by the fuzzy sliding mode controller.

There are three common actuator faults: stuck fault, gain fault, and deviation fault [30]. In order to make the research general, this paper only considers the actuator gain fault. When the EMLHAS occurs fault, the system state equation is expressed as

\[
\begin{align*}
\dot{X} &= AX + B\delta U + EW, \\
Y &= CX + DU,
\end{align*}
\] (22)

where \( \delta \) is gain factor, \( \delta \in [0, 1] \), \( \delta = 1 \) indicates that the EMLHAS occurs no fault, \( \delta = 0 \) indicates that the EMLHAS occurs complete gain fault, and \( 0 < \delta < 1 \) indicates that the EMLHAS occurs partial gain fault.

When EMLHAS occurs fault, the residual of output estimation is expressed as
\[
\begin{align*}
r &= Y - \hat{Y} = CX + DU - C\overline{X} - D\delta U \\
&= C(X - \overline{X}) + D(U - \delta U). \tag{23}
\end{align*}
\]

The output force of the fault EMLHAS is expressed as
where $u_n$ is the output force of the EMLHAS when there is no fault.

The compensation force is expressed as

$$F(t) = (1 - \delta)u.$$  \hfill (25)

### 4.4. Design of Stability Module.

Random road shock may lead to instantaneous suspension fault, so the active force will be compensated by controller. In order to avoid the frequent force compensation for EMLHAS when the instantaneous fault occurs, which leads to the instability of the system, a stability module is designed to improve the stability of the suspension system. In this paper, the root mean square (RMS) value of the sprung mass acceleration and the total energy of the system are judgment basis to select reasonable sampling frequency. When the judgment times are 4, the root mean square value of the sprung mass acceleration of the suspension system is the minimum. When the judgment times are 6, the total energy of suspension system is the minimum. Considering the dynamic performance and energy consumption characteristics of suspension, a compromise method is adopted, and the judgment times are selected as 5. The flow chart of the stability module is shown in Figure 11. The influence of judgment times on the sprung mass acceleration and system total energy is shown in Figure 12.

### 5. Simulation Analysis

In order to verify the effectiveness of the fault-tolerant control for the EMLHAS, a fault-tolerant control model is established by using MATLAB/Simulink software. The simulation analysis of the UIO performance and the effect of fault-tolerant control under random road and bumped road are, respectively, carried out.

The suspension system parameters are shown in Table 3.

#### 5.1. Simulation Analysis of the Unknown Input Observer.

The performance of the unknown input observer will affect the accuracy of fault diagnosis. In this paper, the actual value of suspension and the state estimation by UIO value are compared. The simulation vehicle speed is set to 60 km/h, the road level is selected as level B, and the simulation time is 5 s. The comparison chart of state estimation value and actual value is shown in Figure 12. The RMS of three suspension performance index is shown in Table 4.

Figure 13 and Table 4 show that there is little difference between the actual value and the state estimation value of the suspension performance index. The error of sprung mass acceleration is 5.66%, the error of suspension working space is 5.86%, and the error of dynamic tire load is 2.91%. The results show that the errors of the three performance indexes are small, and within a reasonable range, so it shows that the unknown input observer can accurately estimate the suspension state.

#### 5.2. Simulation Analysis of Fault-Tolerant Control under the Random Road Surface.

A filtered white noise is adopted as the random road surface input model, which is expressed as

$$z(t) = -2\pi f_0 z(t) + 2\pi \sqrt{G_0} \omega(t),$$  \hfill (26)

where $G_0$ is the road irregularity coefficient, $\omega(t)$ is the unit white noise, and $f_0$ is the lower cutoff frequency.

In order to verify the performance of the designed fault-tolerant control strategy, a fault-tolerant control model is established by using MATLAB/Simulink software. The 0.6 times gain fault is set at the initial time. The vehicle speed is set to 60 km/h, the road level is selected as level B, and the simulation time is 10 s. The time-domain and frequency-domain response of sprung mass acceleration, suspension working space, and dynamic tire load under different state are obtained, as shown in Figure 14. The RMS values of dynamic performance and control effect in time domain are
shown in Table 5. The highest peak values of dynamic performance and control effect in frequency domain are shown in Table 6.

Table 6 shows that, in time domain, compared with the fault state, the dynamic performance parameters of the suspension in the normal state are greatly reduced, and there is little difference with the normal state. The fault-tolerance effect (compared with fault state) of sprung mass acceleration is 38.93%, the fault-tolerance effect of suspension working space is 54.03%, and the fault-tolerance effect of dynamic tire load is 42.61%.

The frequency-domain resonance range of car body is 1–1.5 Hz [32], and the most sensitive frequency-domain resonance range of human body is 0.5–80 Hz [33]. The performance indexes in time domain are transformed by fast Fourier transform to get the performance index in

| Parameters                                              | Values  |
|---------------------------------------------------------|---------|
| Sprung mass (kg)                                        | 300     |
| Unsprung mass (kg)                                      | 40      |
| Base value damping of solenoid valve shock absorber (V·s·m⁻¹) | 522     |
| Tire stiffness coefficient (N·m⁻¹)                      | 150000  |
| Spring stiffness coefficient (N·m⁻¹)                    | 12000   |
| Thrust coefficient of the linear motor (N·A⁻¹)          | 78.54   |
| CEMF coefficient of the linear motor (V·s·m⁻¹)          | 68.42   |
| Winding resistance of the linear motor (Ω)               | 10.1    |

CEMF is the short form of the counter electromotive force.
Table 4: RMS of the suspension performance index.

| Index       | Actual value | State estimation value | Error (%) |
|-------------|--------------|------------------------|-----------|
| SMA (m·s²)  | 1.149        | 1.084                  | 5.66      |
| SWS (m)     | 0.0273       | 0.0257                 | 5.86      |
| DTL (N)     | 475.0        | 461.2                  | 2.91      |

SMA is the short form of sprung mass acceleration, SWS is the short form of suspension working space, and DTL is the short form of dynamic tire load.

Figure 13: Comparison chart of state estimation value and actual value. (a) Comparison of sprung mass acceleration. (b) Comparison of suspension working space. (c) Comparison of dynamic tire load.

Figure 14: Continued.
frequency domain. In this part, the highest peak value is used as the quantitative standard of suspension performance index. Table 7 shows that, in frequency domain, especially in low frequency, the highest peak values of the suspension dynamic performance are greatly reduced. The fault-tolerance effect of sprung mass acceleration is 34.34%, the fault-tolerance effect of suspension working space is 51.56%, and the fault-tolerance effect of dynamic tire load is 38.79%.

According to the simulation results, under the fault-tolerant control strategy, the performance indexes of the suspension are reduced, and the ride comfort and safety of

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Table 5: The output RMS values in time domain under random road.

| Indexes            | N-S | F-S  | F-T-C-S | Fault-tolerance effect (%) |
|--------------------|-----|------|---------|----------------------------|
| SMA (m s⁻²)        | 0.385 | 0.619 | 0.378   | 38.93                      |
| SWS (m)            | 0.0098 | 0.0211 | 0.0097 | 54.03                      |
| DTL (N)            | 158.7 | 272.0 | 156.1   | 42.61                      |

Table 6: The highest peak values in frequency domain under random road.

| Indexes            | N-S              | F-S              | F-T-C-S          | Fault-tolerance effect (%) |
|--------------------|------------------|------------------|------------------|----------------------------|
| SMA (m² s⁻⁴ HZ⁻¹)  | 0.169            | 0.265            | 0.174            | 34.34                      |
| SWS (m² HZ⁻¹)      | 0.0067           | 0.0128           | 0.0062           | 51.56                      |
| DTL (N² HZ⁻¹)      | 49.87            | 82.16            | 50.92            | 38.79                      |

N-S is the short form of normal state, F-S is the short form of fault state, and F-T-C-S is the short form of fault-tolerant control state.
the vehicle are improved. Meanwhile, the damage caused by road shock to human body and car body is reduced. It shows that the fault-tolerant control strategy is reasonable.

5.3. Simulation Analysis of Fault-Tolerant Control under the Bumped Road Surface. The input expression of the bumped road surface is

\[ y_0 = 0.01 \left( 1 - \cos 8\pi t \right), \quad 0.5 \leq t \leq 0.75 \]  

In order to verify the performance of the designed fault-tolerant control strategy on another road surface, a fault-tolerant control model is established by using MATLAB/ Simulink software. The 0.6 times gain fault is set at the initial time. The vehicle speed is set to 60 km/h, the road level is selected as level B, and the simulation time is 5 s. The time-domain response of sprung mass acceleration, suspension working space, and dynamic tire load under different state are obtained as shown in Figure 15. The RMS values of dynamic performance and control effect in time domain are shown in Table 7.

Table 7 shows that, in time domain, compared with the fault state, the dynamic performance parameters of the suspension in the normal state are greatly reduced, and there is little difference with the normal state. For example, the fault-tolerance effect of suspension working space is 35.71%, and the convergence speed is faster; therefore, it takes less time for the EMLHAS to reach a stable state. It is verified that the fault-tolerant control method has a better ability to control the influence of disturbance.

| Indexes         | N-S | F-S | F-T-C-S | Fault-tolerance effect (%) |
|-----------------|-----|-----|---------|-------------------|
| SMA (m·s²)      | 0.295 | 0.382 | 0.288 | 24.61 |
| SWS (m)         | 0.0028 | 0.0042 | 0.0027 | 35.71 |
| DTL (N)         | 97.2 | 125.8 | 91.6 | 27.19 |

![Figure 15](image-url)
6. Bench Test

In order to verify the actual control effect of fault-tolerant control, the bench test is carried out. The EMLHAS structure in this paper is composed of linear motor and solenoid valve in series, and the cost of the whole mechanical structure is high. If the structural destructive test is carried out by structural destructive test, the cost is high. In this paper, combined with the actual situation, the suspension is simulated in the fault state by changing the current. The schematic diagram of test system is shown in Figure 16. The test equipment mainly includes DSPACE, acceleration sensor, DSPACE PC, data acquisition system, driver module, hydraulic vibration table, etc.

First, the model built in MATLAB/Simulink software is transmitted to DSPACE. At the initial time, the input current is controlled to 10 A through DSPACE, and the EMLHAS works in normal state. At the 5th second, the input current of the EMLHAS is suddenly changed to 5 A, and the active force provided by the EMLHAS is insufficient. At this time, the EMLHAS is in a fault state. From the 5th second, the controller starts fault-tolerant control of EMLHAS. Because of the limitation of test condition, only the sprung mass acceleration dynamic response of the EMLHAS is measured in this test. The time-domain response of the sprung mass acceleration test is shown in Figure 17. The frequency-domain response of sprung mass acceleration test is shown in Figure 18. The output RMS values of sprung mass acceleration in time domain are shown in Table 8. The highest peak values of sprung mass acceleration in frequency domain are shown in Table 9.

From Table 8, the sprung mass acceleration of EMLHAS in fault-tolerant control state is 1.193 m·s⁻², and the fault-
tolerance effect of sprung mass acceleration is 31.69%. From Table 9, in frequency domain, the highest peak values of sprung mass acceleration in fault-tolerant control state are 0.138 (m² · s⁻⁴ · Hz⁻¹), which is reduced by 46.92% compared with fault state. Especially in the 1–1.5 Hz, which is the frequency-domain resonance range of car body, the improvement effect of power spectral density is more obvious. Test results show that the effect of fault-tolerant control is good and reasonable.

7. Conclusions

(1) A new kind of EMLHAS is put forward that is composed of a linear motor and a solenoid valve shock absorber in series. The EMLHAS can work in three different modes, namely energy regenerative state, semiactive state, and active state. The mathematical model of the linear motor and the test model of the solenoid valve shock absorber are established.

(2) Aiming at the suspension system instability caused by the fault of the EMLHAS, a fault diagnose and fault-tolerant control method is proposed. Using fuzzy sliding mode controller as the controller and unknown input observer as the state observer, fault diagnose and fault-tolerant control method is designed.

(3) A fault diagnose and fault-tolerant control model is built in MATLAB/Simulink software. The simulation analysis is carried out on random road and bumped road, respectively. The simulation analysis results show that in the time domain and frequency domain, the fault-tolerant control can effectively improve the dynamic characteristics of the EMLHAS compared with the fault state. Finally, a hydraulic vibration table was used to carry out a bench test. The test results showed that the sprung mass acceleration of the EMLHAS was reduced by 31.69% under fault-tolerant control, which verified the rationality of the fault-tolerant control strategy.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

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