Frequency Domain Control Simulation of In-Wheel Motor Electric Vehicle

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Abstract. Aiming at the optimization of the suspension structure and control strategy of electric vehicle driven by hub motor, a magnetorheological suspension fuzzy control form based on the idea of ceiling was proposed to optimize the suspension system. Taking a hub motor driven electric vehicle as the research object, a vehicle simulation model was established in Matlab/Simulink. Taking the vertical acceleration, pitching Angle acceleration and roll Angle acceleration of the vehicle as the optimization target, the smoothness of the vehicle was simulated and analyzed and compared with the initial hub motor drive form. The simulation results show that the proposed optimization method is effective and provides a new way to further study the optimization method of hub motor driven electric vehicle suspension parameters.

1. Introduction
Hub motor drive, as a kind of pure electric vehicle that has been put into practical use, has great development potential due to its strong performance and many driving combinations. The hub motor drive mode does not adopt the traditional transmission system arrangement, but integrates the power, braking and transmission structures into the vehicle hub, thus greatly simplifying the mechanical structure of the vehicle. Moreover, the simplified structure has the advantages of high transmission efficiency, simple structure and high space utilization. It is especially suitable for the modular production of new energy vehicles and reduces the manufacturing cost. Although hub motor drive has many advantages, its structure also brings many vehicle dynamics problems. Due to the presence of the hub motor, the unsprung weight of the automobile is greatly increased, resulting in a corresponding increase in the moment of inertia of the hub, and consequently the vibration of the wheels will increase, resulting in a significant increase in the root mean square value of the body acceleration. The passengers feel that the vibration of the road surface increases, and the increase of unsprung weight will also worsen the tire grounding performance, increase the dynamic load of the tire, affect the handling stability, and even damage the motor in serious cases. At present, these problems are mainly solved by using lightweight materials to manufacture unsprung components, designing integrated hub motors, transferring suspension quality, and adopting semi-active and active suspension control [1].

In the literature [2], the ANSYS motor model and the Maxwell motor model were established. The stator stator electromagnetic force wave obtained by Maxwell analysis was applied as an excitation to the ANSYS motor stator. The variation of the stator displacement with time was obtained, and the motor rotation was analyzed in detail. Moment pulsation and electromagnetic vibration have proposed several methods to reduce electromagnetic vibration, but no external control methods such as suspension are used to reduce the amplitude and vibration of vibration. Document [3] simplifies the flat annular channel model according to the working characteristics of the damper, summarizes the calculation method of the damping force of the magnetorheological damper, analyzes the structure, principle and control method...
of the magnetorheological damper, establishes the suspension model through experiments and simulations, discusses the optimization effect of the magnetorheological damper under different control methods, but does not consider the negative effect brought by the electromagnetic vibration of the hub motor. In the literature [4], two adaptive neuro-fuzzy inference systems are designed. The positive and negative models of the MR damper are simulated and all achieve good results. Based on the mixed mode operation, the magneto-rheological damper is designed and manufactured in literature [5], and combined with the suspension model, two adaptive neuro-fuzzy inference systems are designed. The forward and inverse models of MR dampers are simulated. After establishing the dynamic model of MR suspension system for half-car, a human simulation intelligent control (HSIC) scheme for suppressing unexpected vibration such as pitch acceleration is proposed, which achieves good results.

At present, there are relatively few literatures on the matching of electromagnetic vibration and suspension. Firstly, the situation of permanent magnet synchronous motor (PMSM) is not considered, and several cases of electromagnetic vibration are not analyzed and discussed one by one. Secondly, the calculation of electromagnetic vibration is still in the analytical method, which cannot take into account the radial electromagnetic force on the structure of the motor. Influences; At last, the modeling and simulation of suspension is not perfect enough, the whole vehicle model is not established, and there is no relevant dynamic simulation analysis, which leads to inaccurate and incomplete final simulation results. The research on the matching of hub motor vibration and suspension mainly focuses on the suspension structure, electromagnetic vibration and control methods of hub motor. In this paper, a permanent magnet synchronous motor (PMSM) model is established to make the suspension system more realistic by modeling and simulating the suspension system driven by hub motor. The structure and control method of MR suspension are studied. The RMS of suspension performance evaluation parameters is greatly reduced, and the ride comfort of electric vehicle driven by hub motor is effectively improved.

2. Establishment of Pavement Motivation Model

Running bumpy road surface is the main excitation source of suspension system. The simulation of road excitation generally takes into account the various working conditions of automobiles. Class B random pavement simulation continuous pavement is a common pavement input model, and power spectral density function is commonly used to express random pavement excitation [6].

\[ G_q(n) = G_q(n_0) \left( \frac{n}{n_0} \right)^n \]

In the formula: \( n_0 \) is the reference control frequency, usually \( n_0=0.1 \) m\(^{-1} \); \( n \) is the frequency of space, the unit is m\(^{-1} \); \( n \) is the frequency coefficient, Usually \( n=2m3 \); \( G_q(n_0) \) is the road roughness coefficient, defined as the power spectral density at \( n_0 \), and its unit is m\(^3 \).

Substituting the time frequency \( f=un \) into equation (1) can be converted into:

\[ G_q(f) = G_q(n_0) \left( \frac{u}{n_0} \right)^2 \frac{\omega}{f^2} \]

When the vehicle speed \( u \) is expressed by the angular frequency \( \omega \) (rad/s), the equation (2) is converted into:

\[ G_q(\omega) = (2\pi)^2 G_q(n_0) n_0^2 \frac{\omega}{\omega^2} \frac{\omega_0^2}{\omega^2 + \omega_0^2} \]

It can be seen that when \( \omega = 0 \), \( G_q(\omega) \), does not meet the actual situation. The road surface spectrum model of the low frequency region is corrected by using the lower cutoff frequency \( \omega_0 = 2\pi n_c \), wherein the lower cutoff frequency \( n_c = 0.1 \) m\(^{-1} \), then the equation (3) can be converted into:

\[ G_q(\omega) = (2\pi)^2 G_q(n_0) n_0^2 \frac{\omega}{\omega^2 + \omega_0^2} \]

According to the classical literature on random vibration, the following relations are obtained:

\[ G_s(\omega) = |H(\omega)|^2 S_n \]

Spectrum parameters of random vibration are not discussed.
Where: $H(\omega)$ is the frequency response function of the vibration system; $S$ is the ideal white noise power spectral density, which is generally taken as 1 [7].

If $U$ is the driving speed of the car, $S$ is the horizontal moving distance, and $Zr(t)$ is the random roughness coefficient of the road surface, the spatial frequency response function can be obtained according to equations (4) and (5) as follows:

$$H(\omega)=\frac{2\pi n_s \sqrt{G_x(n_x)u}}{w_o + j\omega}$$

(6)

From equation (6), the differential equation of road roughness in the spatial domain can be further obtained as follows

$$\dot{z}_s(s)+w_o z_s(s)=2\pi n_s u(s)\sqrt{G_x(n_x)u}$$

(7)

Because the differential relationship between horizontal displacement and vehicle speed is as follows:

$$dv=ds=d\left(\frac{ds}{dt}\right)$$

(8)

The differential equation of pavement roughness in time domain obtained by substitution (7) is as follows:

$$H(\omega)=\frac{2\pi n_s \sqrt{G_x(n_x)u}}{w_o + j\omega}$$

(9)

According to the differential relation between the horizontal displacement and the vehicle speed, the differential equation of road roughness in the time domain can be obtained as follows:

$$\dot{Z}_r(t)=-2m_1 u Z_r(t)+\sqrt{G_x(n_x)u}\omega(t)$$

(10)

Where: $Zr(t)$ is the road surface irregularity displacement of the suspension system; $\omega(t)$ is Gaussian white noise with a mean value of zero.

### 3. Optimization of Suspension Structure and Control Strategy

#### 3.1. Suspension structure optimization

Since the four wheel hub motor wheels need to be separately added with independent driving and speed reducing devices and the quality is large, the vibration system of the whole vehicle will have an important influence, which directly affects the maneuverability of the car and the ride comfort. The dynamic vibration absorber is installed on the vibrating object and absorbs the vibration energy through the reaction force when the mass spring resonance system generates resonance [8]. It has obvious effect on single-frequency or low-frequency excitation force, and can play its role to the greatest extent by adding corresponding dynamic vibration absorbers according to specific frequencies. Based on the principle of dynamic vibration absorber, the outer shell of the hub motor stator and the wheel shaft are connected through a passive suspension system to make the hub motor suspend in the wheel, thus forming a structure similar to the dynamic vibration absorber to counteract the vibration, thus improving the maneuverability and ride comfort of the automobile.

| Parameter                  | Symbol | Numerical |
|----------------------------|--------|-----------|
| Spring mass /kg            | $m_s$  | 315       |
| Wheel assembly mass /kg    | $m_l$  | 29        |
| Wheel hub motor mass /kg   | $m_d$  | 31        |
| Stiffness of passive suspension /N/m | $K$     | 29509     |
| Passive suspension damping/n.s/m | $C$     | 1767      |
| Tire stiffness /N/m        | $K_t$  | 217751    |
The 1/4 hub motor-driven suspension system model of the wheel power absorber is shown in Figure 1.

![Matlab/Simulink model of dynamic vibration absorber suspension](image)

Fig.1. Matlab/Simulink model of dynamic vibration absorber suspension

The dynamic differential equation of the suspension system model is:

\[ m \ddot{Z}_s + K(Z_s - Z_d) + C \dot{Z}_s = 0 \]  \hspace{1cm} (11)

\[ m \ddot{Z}_d + K_d(Z_d - Z_s) + C_d \dot{Z}_d = F(t) \]  \hspace{1cm} (12)

When the passive suspension of the wheel hub part works, the distance between the rim and the motor housing needs to be limited, i.e. the maximum deformation of the passive suspension spring needs to be limited to prevent the motor from hitting the hub. The maximum displacement of suspension vertical motion is set to be 30 mm, that is: \(|Z_s - Z_d| \leq 30\text{mm}\). The three performance parameters of vertical acceleration, suspension dynamic deflection and wheel dynamic load are optimized. The rms value of the three indicators is reduced and the performance balance is achieved. The optimal parameters of the passive suspension are obtained. That is, the optimal damping coefficient \(C_d\) and the optimal stiffness coefficient \(K_d\) are sought. Table 2 shows the changes of the root mean square values of the three performance parameters of the vehicle body vertical acceleration, suspension dynamic deflection and wheel dynamic load with \(C_d\) and \(K_d\) under normal driving at 85 km/h.

| \(C_d\) | \(K_d\) | Vertical acceleration | Dynamic deflection | Dynamic displacement |
|--------|--------|-----------------------|--------------------|---------------------|
| 4000   | 40000  | 1.2281                | 0.0074             | 0.0034              |
| 5000   | 40000  | 1.2446                | 0.0074             | 0.0034              |
| 4000   | 50000  | 1.2268                | 0.0074             | 0.0034              |
| 5000   | 50000  | 1.2437                | 0.0074             | 0.0034              |

From the table above, it can be seen that stiffness has no obvious influence on the effect of vibration reduction; when the damping increases, the vertical vibration acceleration increases, but at the same time the dynamic deflection of the motor suspension decreases accordingly. When the damping is \(C_d = 4000\) n s/m and the rigidity is \(K_d = 40000\) N/m, the suspension arrangement meets the requirements and the maximum displacement of suspension vertical motion is less than 30 mm.

3.2. **Magnetorheological damper**

Rheological liquid is a suspended liquid composed of fine magnetic particles, dispersed in carrier liquid and stabilizer, which exhibits Newtonian fluid characteristics when no magnetic field is applied and Bingham fluid characteristics when magnetic field is applied [9]. The rheological liquid viscosity is not affected by the magnetic field, and the greater the applied magnetic field strength, the greater the yield
stress. When the magnetic field strength increases to a certain limit, the fluidity of the liquid disappears. At this time, the damping coefficient is infinite, and the fluidity is restored after the magnetic field is removed. This characteristic is called the magnetorheological effect [10].

Magnetorheological damper works based on the controllable characteristics of magnetorheological fluid flow. The signals of the body and wheel motion sensors are used as input to judge the road condition and driving environment. Then the damping force is changed in real time by using the characteristics of magnetorheological fluid to filter and reduce the vibration. At present, automobile magneto-rheological dampers usually work in a mixed mode of flow mode and shear mode. The internal structure of the damper is shown in Fig. 2. The magnitude of coil current is changed to generate magnetic fields with different intensities to control the rheological characteristics of magneto-rheological fluid in the damping channel. The piston moves in a reciprocating straight line in the working cylinder, and the pressure of the left cavity and the right cavity are different, thus changing the magnitude of damping force.

\[
F = \left( \frac{12\eta A_v^2}{bh^3} + \frac{\eta lA_v}{h} \right) v + \left( \frac{c lA_v^2}{h} + bt \right) \tau, \text{sgn}(v) \quad (14)
\]

Where: \( \tau_y = 78\mu_0NI/(2h) \); \( A_p = \pi/4(D^2-d^2) \); \( b = \pi D \); \( N \) is the number of coil turns; \( D \) is the inner diameter of the cylinder barrel; \( d \) is the piston diameter; \( l \) is the length of damping channel; \( h \) is the damping channel gap; \( \eta \) is zero field viscosity; \( \mu_0 \) is the permeability of magnetorheological fluid; \( v \) is the rotational speed; \( c \) is the correction factor.

The finishing formula (14) can be converted into:

\[
F = \left( \frac{3\pi\eta(D^2-d^2)l}{4Dh^2} + \frac{\pi\eta Dl}{h} \right) v + \left( \frac{2\pi D(D^2-d^2)}{4h} + \pi Dl \right) \tau, \text{sgn}(v) \quad (15)
\]

The sum of viscous damping force and Coulomb damping force is the damping force of MR damper.

\[
C_e = \left( \frac{3\pi\eta(D^2-d^2)l}{4Dh^2} + \frac{\pi\eta Dl}{h} \right)
\]

Let the viscous damping force coefficient be \( S = \left( \frac{2\pi D(D^2-d^2)}{4h} + \pi Dl \right) \), and the Coulomb damping force coefficient is \( C_e \). Equation (14) can be reduced to:
The coefficient of viscous damping force $C_v$ has been determined at the design stage, and the damping value is fixed. The Coulomb damping force coefficient $S$ is determined by the control current and can be controlled in real time.

3.3. Fuzzy controller of magnetorheological suspension based on sky-hook control

Sky-hook control is a classical active suspension control strategy based on optimal control theory, because of its good control performance and simple structure, it is widely used in semi-active suspension control research [12]. The essence of sky-hook control is based on the negative feedback control of suspension speed of a virtual fixed fulcrum, but the false fixed fulcrum does not exist in fact, so sky-hook control can not be realized for a moving vehicle. In the actual research, the ceiling damping control strategy is often used as a reference for the improved control strategy.

According to the idea of sky-hook control strategy, in order to make magnetorheological suspension control close to its effect, the function of current control is taken as follows:

$$I = \text{abs} \left( \dot{Z}_s \cdot X \right)$$

(17)

In the formula: $I$ is the current in the input MR damper; $X$ is the optimization coefficient of the current function.

According to relevant documents, the evaluation of optimization results is mostly in the form of comprehensive performance evaluation function of formula:

$$J = \lim_{\rho \rightarrow \infty} \left[ \rho_1 \frac{\frac{1}{T} \int_{0}^{T} (\ddot{Z}_s)^2 \, dt - \frac{1}{T} \int_{0}^{T} (\dot{Z}_s)^2 \, dt}{\sqrt{\frac{1}{T} \int_{0}^{T} (\dot{Z}_s)^2 \, dt}} \right]$$

$$+ \rho_2 \frac{\frac{1}{T} \int_{0}^{T} (\dot{Z}_s - \dot{Z}_u)^2 \, dt - \frac{1}{T} \int_{0}^{T} (Z_s - Z_u)^2 \, dt}{\sqrt{\frac{1}{T} \int_{0}^{T} (Z_s - Z_u)^2 \, dt}}$$

$$+ \rho_3 \frac{\frac{1}{T} \int_{0}^{T} (\dot{Z}_r)^2 \, dt - \frac{1}{T} \int_{0}^{T} (Z_r)^2 \, dt}{\sqrt{\frac{1}{T} \int_{0}^{T} (Z_r)^2 \, dt}}$$

(18)

Where $\rho_1$ is the weight coefficient of body acceleration; $\rho_2$ is the suspension dynamic deflection weighting coefficient; $\rho_3$ is the weighting coefficient of tire relative dynamic load; $Z_s$ is vertical displacement of mass above spring, $Z_u$ is vertical displacement of mass below spring and $Z_r$ is random road excitation of B grade.

The weighting coefficients are fixed under different driving conditions and have been defined in the design process. Therefore, it is necessary to determine the relative importance of each performance index in the comprehensive performance evaluation function. The optimization objective of this paper is to optimize the body suspension system as much as possible under the condition of ensuring the normal operation of the motor, so the weighted coefficient value is taken as follows: $\rho_1=0.4$, $\rho_2=0.15$, $\rho_3=0.3$.

The weighted calculation of the optimum percentage of the evaluation parameters in formula (18) can eliminate the problem that the order of magnitude of the initial values is too different, and it is closer to the actual situation, so as not to make some evaluation parameters "one-size-fits-all". At the same time, the evaluation method is changed from the expectation to the optimization percentage of the root mean square value, which can reflect the degree that the evaluation parameter deviates from the origin of the ideal position, more intuitively reflects the goal of suspension optimization, and facilitates the evaluation of the optimization result.

According to the comprehensive performance evaluation function, the optimization coefficient $x$ of the current function is optimized through the optimization toolbox pattern search function of Matlab.
The curves of the comprehensive performance evaluation parameters varying with X value based on sky-hook control MR suspension model at different speeds are obtained as shown in Fig. 3.

Fig. 3. Magnetorheological suspension model optimization curve based on skylight control

From the above three figures, it can be seen that the maximum comprehensive performance optimization coefficient corresponds to: X=7 at 10km/h speed; X=9 at 85km/h speed; X=10 at 120km/h speed, considering X=9 as the optimization coefficient.

The vertical velocity v and acceleration a of the sprung mass of the car are taken as the inputs of the fuzzy controller, and the control force f is taken as the output of the fuzzy controller. the magneto-
rheological suspension fuzzy controller designed to suppress the vertical acceleration of the sprung mass is shown in Fig. 4.

4. Optimization Simulation and Analysis

4.1. Establishment of vehicle model

Due to the large number of structural components and parameters of the vehicle model, a reasonable simplification of the vehicle vibration system is conducive to the analysis of the core issues, based on the following assumptions: 1) The body, wheel and hub motor systems are rigid mass blocks. 2) The stiffness of passive suspension and tire is linear, and the tire damping is neglected. 3) The front and rear suspensions are regarded as independent suspensions. 4) The pitch angle and roll angle of the car body caused by the pitch and roll motion of the sprung mass are smaller. 5) The vehicle suspension model is established without considering the change of the moment of inertia and the yaw motion of the vehicle during the pitch and roll of the vehicle.

When the body pitch angle and roll angle are small, the vertical displacement of the connection between the spring mass and the suspension system satisfies the following formula:
\[
\begin{align*}
Z_{i1} &= Z - L_i \dot{\theta}_t + B_i \dot{\theta}_t \\
Z_{i2} &= Z - L_i \dot{\theta}_t - B_i \dot{\theta}_t \\
Z_{i3} &= Z + L_i \dot{\theta}_t + B_i \dot{\theta}_t \\
Z_{i4} &= Z + L_i \dot{\theta}_t - B_i \dot{\theta}_t 
\end{align*}
\]

By deriving formula (19), the vertical displacement velocity at the connection point between spring mass and suspension system is obtained as follows:
\[
\begin{align*}
\dot{Z}_{i1} &= \dot{Z} - L_i \dot{\theta}_t + B_i \dot{\theta}_t \\
\dot{Z}_{i2} &= \dot{Z} - L_i \dot{\theta}_t - B_i \dot{\theta}_t \\
\dot{Z}_{i3} &= \dot{Z} + L_i \dot{\theta}_t + B_i \dot{\theta}_t \\
\dot{Z}_{i4} &= \dot{Z} + L_i \dot{\theta}_t - B_i \dot{\theta}_t 
\end{align*}
\]

The unsprung mass dynamics differential equations for four wheels that do not include the hub motor suspension are:
\[
\begin{align*}
&\left[ m_i \ddot{Z}_{i1} + K_i(Z_{i1} - Z_{i3}) + C_i(\dot{Z}_{i1} - \dot{Z}_{i3}) + F_{i1} \right] + F_{i1} = 0 \\
&\left[ m_i \ddot{Z}_{i2} + K_i(Z_{i2} - Z_{i3}) + C_i(\dot{Z}_{i2} - \dot{Z}_{i3}) + F_{i2} \right] + F_{i2} = 0 \\
&\left[ m_i \ddot{Z}_{i3} + K_i(Z_{i3} - Z_{i3}) + C_i(\dot{Z}_{i3} - \dot{Z}_{i3}) + F_{i3} \right] + F_{i3} = 0 \\
&\left[ m_i \ddot{Z}_{i4} + K_i(Z_{i4} - Z_{i3}) + C_i(\dot{Z}_{i4} - \dot{Z}_{i3}) + F_{i4} \right] + F_{i4} = 0 
\end{align*}
\]

The dynamic differential equation of hub motor suspension is:
\[
\begin{align*}
F_i(t) &= m_j \ddot{Z}_{i1} + K_j(Z_{i1} - Z_{i3}) + C_j(\dot{Z}_{i1} - \dot{Z}_{i3}) \\
F_i(t) &= m_j \ddot{Z}_{i2} + K_j(Z_{i2} - Z_{i3}) + C_j(\dot{Z}_{i2} - \dot{Z}_{i3}) \\
F_i(t) &= m_j \ddot{Z}_{i3} + K_j(Z_{i3} - Z_{i3}) + C_j(\dot{Z}_{i3} - \dot{Z}_{i3}) \\
F_i(t) &= m_j \ddot{Z}_{i4} + K_j(Z_{i4} - Z_{i3}) + C_j(\dot{Z}_{i4} - \dot{Z}_{i3}) 
\end{align*}
\]

The dynamic differential equations of vertical vibration acceleration, pitch motion and roll motion of the vehicle model are as follows:
\[
\begin{align*}
M_z &= -K_i(Z_{i1} - Z_{i3}) - C_i(\dot{Z}_{i1} - \dot{Z}_{i3}) + F_{i1w} - K_i(Z_{i2} - Z_{i3}) - C_i(\dot{Z}_{i2} - \dot{Z}_{i3}) + F_{i2w} \\
&- K_i(Z_{i3} - Z_{i3}) - C_i(\dot{Z}_{i3} - \dot{Z}_{i3}) + F_{i3w} - K_i(Z_{i4} - Z_{i3}) - C_i(\dot{Z}_{i4} - \dot{Z}_{i3}) + F_{i4w} \\
I_{\phi} \ddot{\phi}_t &= -L_i \left[ K_i(Z_{i1} - Z_{i3}) + C_i(\dot{Z}_{i1} - \dot{Z}_{i3}) - F_{i1w} - K_i(Z_{i2} - Z_{i3}) - C_i(\dot{Z}_{i2} - \dot{Z}_{i3}) - F_{i2w} \right] \\
&- L_i \left[ K_i(Z_{i3} - Z_{i3}) + C_i(\dot{Z}_{i3} - \dot{Z}_{i3}) - F_{i3w} - K_i(Z_{i4} - Z_{i3}) - C_i(\dot{Z}_{i4} - \dot{Z}_{i3}) - F_{i4w} \right] \\
I_{\psi} \ddot{\psi}_t &= B \left[ K_i(Z_{i1} - Z_{i3}) + C_i(\dot{Z}_{i1} - \dot{Z}_{i3}) - F_{i1w} + K_i(Z_{i2} - Z_{i3}) + C_i(\dot{Z}_{i2} - \dot{Z}_{i3}) - F_{i2w} \right] \\
&- B \left[ K_i(Z_{i3} - Z_{i3}) + C_i(\dot{Z}_{i3} - \dot{Z}_{i3}) - F_{i3w} + K_i(Z_{i4} - Z_{i3}) + C_i(\dot{Z}_{i4} - \dot{Z}_{i3}) - F_{i4w} \right] 
\end{align*}
\]

4.2. Simulation and analysis of vehicle model
According to the above dynamic differential equation, a vehicle suspension model is established. The simulation model is shown in Fig. 6.
Fig. 6. Eleven degree of freedom model of vehicle suspension model

The Simulink simulation time is set to 8s, and the simulation results show that when the hub motor-driven electric vehicle runs normally at different speeds, the vertical acceleration amplitude-frequency characteristics of the vehicle model under the initial hub motor driving form under the class B random road surface and the fuzzy control form of the magnetorheological suspension based on the ceiling idea are compared as shown in Fig. 7:

Fig (a) Amplitude-frequency characteristics of 10km/h traditional form and fuzzy form

Fig (b) Amplitude-frequency characteristics of 85km/h traditional form and fuzzy form
From the comparison of the three amplitude-frequency characteristics, it can be seen that the vibration frequency peaks of the vertical acceleration of the whole vehicle model at the three speeds are greatly attenuated after the fuzzy control mode is adopted. The vibration distribution in the high frequency area is uniform, and there is no abnormal mutation. It shows that the ride comfort of the electric vehicle driven by hub motor has been greatly improved by using the fuzzy control method.

The comparison results of root mean square values of vertical acceleration, pitch angle acceleration and roll angle acceleration of the initial hub motor drive form under Class B random road surface and the magneto-rheological suspension fuzzy control form based on ceiling idea are shown in Table 3-5.

| Suspension form  | Vertical acceleration | Pitch angular acceleration | Roll angular acceleration |
|------------------|-----------------------|---------------------------|--------------------------|
| Traditional form | 0.2424                | 0.0835                    | 0.3360                   |
| Fuzzy form       | 0.1617                | 0.0611                    | 0.2462                   |
| Optimized percentage | 33.29%             | 26.83%                    | 26.73%                   |

Table 3 Comparison of root mean square values under 10km/h

| Suspension form  | Vertical acceleration | Pitch angular acceleration | Roll angular acceleration |
|------------------|-----------------------|---------------------------|--------------------------|
| Traditional form | 0.5394                | 0.2359                    | 0.9230                   |
| Fuzzy form       | 0.4120                | 0.1748                    | 0.6830                   |
| Optimized percentage | 23.62%             | 25.90%                    | 26.00%                   |

Table 4 Comparison of root mean square values under 85km/h

| Suspension form  | Vertical acceleration | Pitch angular acceleration | Roll angular acceleration |
|------------------|-----------------------|---------------------------|--------------------------|
| Traditional form | 0.6146                | 0.2734                    | 1.0574                   |
| Fuzzy form       | 0.4647                | 0.2007                    | 0.7731                   |
| Optimized percentage | 24.39%             | 26.59%                    | 26.89%                   |

Table 5 Comparison of root mean square values under 120km/h
From the above three tables, it can be seen that after the magnetorheological suspension is controlled in fuzzy form, when the hub motor drives the electric vehicle to run normally at different speeds, the vertical acceleration, pitch angle acceleration and roll angle acceleration of the suspension model of the whole vehicle are greatly optimized under B-level random road excitation, and the optimization percentage exceeds 20%, thus greatly enhancing the ride comfort and handling stability.

5. Conclusion
In this paper, a dynamic simulation model of a hub motor-driven electric vehicle is established in Matlab, and the suspension configuration is optimized by using the principle of dynamic vibration absorber, so that the hub motor becomes a dynamic vibration absorber to reduce the root-mean-square values of body vertical vibration acceleration, suspension dynamic deflection and wheel dynamic load. For the optimized suspension structure, a magnetorheological damper is added between the body and the wheel, and the suspension damping force is adjusted in real time by using the fuzzy control method based on the skylight idea, so as to further optimize the ride comfort and handling stability of the vehicle. The comparison between pre-optimization and post-optimization shows that the vertical acceleration, pitch angle acceleration and roll angle acceleration of the hub motor-driven electric vehicle have been greatly optimized after adopting the magnetorheological suspension fuzzy control form based on the ceiling idea, and the ride comfort and handling stability of the whole vehicle have been improved, which shows that the proposed optimization method is effective. It provides a new idea for further research on the optimization method of the suspension motor parameters of the electric motor vehicle.

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