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Modelling and Test Verification of Suspension Optimal Damping Ratio for Electric Vehicles Considering Occupant-cushion and In-wheel Motor Effects

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ABSTRACT

The damping ratio of chassis suspension is a key parameter for damping matching of in-wheel motor vehicles (IWMVs). Because the motor is attached to the driving wheel, the initial design method of the damping ratio for traditional cars is not entirely suitable for IWMVs. This paper proposes an innovative initial design method of the damping ratio for IWMVs. Firstly, a traveling vibration model of occupant-vehicle-road (OVR) for IWMVs is established. The model involves the occupant, cushion, suspension, in-wheel motor, road, and running speed. Secondly, on the basis of the model, using a special form of infinite integral, a mathematical expression of the occupant root-mean-square (RMS) acceleration is derived. Thirdly, based on the RMS optimization criterion for ride comfort, an 8 order polynomial equation about the suspension optimal damping ratio is deduced. Subsequently, through factors analysis, the change principles of the optimal damping ratio versus vehicle parameters are unveiled. Finally, the reliability of the optimal damping ratio is validated by test. The relative deviation of the calculated optimal damping ratio and the tested damping ratio is 5.4%. The results show that the proposed optimal damping ratio can effectively guide the suspension damping matching for IWMVs.

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1. INTRODUCTION

For in-wheel motor vehicles (IWMVs), the motor is integrated into the driving wheel and the traditional transmission system is omitted, such as the transmission shaft and the differential mechanism [1, 2]. This kind of high integration makes them have obvious technical advantages in the aspects of the components space layout, energy saving, and environmental protection, etc [3, 4]. However, due to the increase of unsprung mass, the ride comfort of IWMVs is often degraded [5].

The damping matching of chassis suspension system is critical to the improvement of ride comfort for IWMVs [6-8]. For example, Shao et al. researched about the dynamic damping of an active suspension of an IWMV [9]. The damping ratio is a key parameter of suspension system [10]. The design of the damping ratio has an important influence on ride comfort [11]. At present, for IWMVs, the damping matching of suspension system mainly refers to the damping matching theory for traditional cars [12]. The traditional method is as follows: firstly, according to engineering experience and subjective judgment, using the feasible design range of the damping ratio (0.2~0.4) for traditional cars, an initial value of the damping ratio is selected; then, based on the initial value, a damper considered good is determined by trial and error or optimization algorithm. A good initial value as the starting point of the exact optimal matching can reduce the workload and the design difficulty, so that it can greatly shorten the whole design cycle of suspension system.

The shortcomings of the above traditional method are as follows: (1) the initial value selection of the damping ratio relies on the design experiences to a great degree, which has a great blindness and subjectivity, therefore, it is not easy to guarantee the whole design cycle of suspension system; (2) because the motor is attached to the driving wheel, the feasible design range of the damping ratio for traditional cars is not entirely suitable for IWMVs; (3) the feasible design range does not involve the effects of occupant-cushion system,

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although this system has an important influence on the suspension damping matching [12, 13]. Therefore, it is difficult to provide a good initial design value of the damping ratio for IWMVs by the traditional method, which makes the suspension design cycle longer and the cost higher. In addition, due to the vehicle model complexity and the convergence ability constraints of the optimization algorithm [14, 15], the accurate design of suspension damping is also dependent on the initial design value and the feasible design range of the damping ratio. To effectively guide the suspension damping matching for IWMVs, a reliable initial design method of the damping ratio for suspension system should be created.

The main aim of this paper is to provide a practical and reliable initial design method of the suspension damping ratio for IWMVs. In Section 2, a traveling vibration model of occupant-vehicle-road (OVR) for IWMVs is established. In Section 3, using a special form of infinite integral, a mathematical expression of the occupant RMS acceleration is derived. Moreover, based on the RMS optimization criterion, an 8 order polynomial equation about the optimal damping ratio of suspension system is deduced. In Section 4, through factors analysis, the change principles of the optimal damping ratio versus vehicle parameters are unveiled. In Section 5, the reliability of the optimal damping ratio is validated by test. In Section 6, some useful conclusions are given.

2. TRAVELLING VIBRATION MODEL OF OVR FOR IWMVS

The vibration model for IWMVs is the base of suspension damping matching and comfort analysis. It is difficult to carry out dynamic analysis based on complex models in the initial design stage , because of a large number of unknown vehicle parameters [16]. Although the simplified model is different from the actual situation, its analysis results are close to the actual situation to a great extent. The simplified model has important reference value for the solution of some actual problems [17, 18]. Therefore, simplified vehicle models can be used to provide a simple and feasible method for the initial design of the suspension damping ratio for IWMVs. Considering the occupant-cushion system and motor system, a simplified traveling vibration model of OVR for IWMVs is established, as shown in Figure 1.

In Figure 1, the cushion is characterized as a shock absorber with damping C_3 and a spring with stiffness K_3; the chassis suspension is simplified as a shock absorber with damping C_2 and a spring with stiffness K_2; the tire is characterized as a spring with stiffness K_1; m_i, i=0,1,2,3, represents the in-wheel motor mass, the

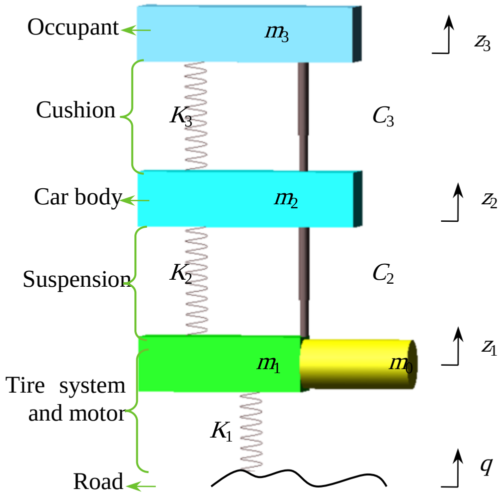

Figure 1. The travelling vibration model of OVR for IWMVs

tire system mass, the car body mass, and the occupant mass, respectively; z_i, i=1,2,3, represents the vertical displacement of the corresponding mass and q represents the road displacement input.

The vibration equation of the model can be expressed as:

$$\boldsymbol{M}\ddot{\boldsymbol{z}} + \boldsymbol{C}\dot{\boldsymbol{z}} + \boldsymbol{K}\boldsymbol{z} = K_1\boldsymbol{q} \tag{1}$$

where,

$$\boldsymbol{M} = \begin{bmatrix} m_3 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_1 \end{bmatrix}, \quad \boldsymbol{C} = \begin{bmatrix} C_3 & -C_3 & 0 \\ -C_3 & C_3+C_2 & -C_2 \\ 0 & -C_2 & C_2 \end{bmatrix},$$

$$\boldsymbol{K} = \begin{bmatrix} K_3 & -K_3 & 0 \\ -K_3 & K_3+K_2 & -K_2 \\ 0 & -K_2 & K_2+K_1 \end{bmatrix}, \quad \boldsymbol{z} = \begin{bmatrix} z_3 \\ z_2 \\ z_1 \end{bmatrix}, \quad \boldsymbol{q} = \begin{bmatrix} 0 \\ 0 \\ q \end{bmatrix}.$$

The power spectral density (PSD) of the road surface displacement input q can be expressed as [19]:

$$G_q(\omega) = 4\pi^2 G_q(n_0) n_0^2 \frac{v}{\omega^2} \tag{2}$$

where, n_0=0.1m^{-1}, v is the running speed of IWMVs, $G_q(n_0)$ is the road irregularity coefficient and ω is the excitation frequency.

The road surface velocity spectrum can be expressed as:

$$G_{\dot{q}}(\omega) = 4\pi^2 G_q(n_0) n_0^2 v \tag{3}$$

3. SUSPENSION OPTIMAL DAMPING RATIO FOR IWMVS

3. 1. Mathematical Expression of the Occupant RMS Acceleration

The occupant RMS acceleration

is often used for comfort evaluation. In this section, based on the traveling vibration model of OVR for IWMVs, a mathematical expression of the occupant RMS acceleration is derived.

From Equation (1), the transfer matrix $\boldsymbol{H}(\mathrm{j}\omega)$ of the displacement vector $\boldsymbol{z}$ to the road surface input q can be expressed as:

$$\boldsymbol{H}(\mathrm{j}\omega)=\left[\boldsymbol{K}-\omega^2\boldsymbol{M}+\mathrm{j}\omega\boldsymbol{C}\right]^{-1}K_1 \tag{4}$$

For the convenience in analyzing the vibration characteristics of IWMVs, the following variables are introduced:

$$r_2=\frac{m_3}{m_2},\quad r_1=\frac{m_2}{m_1+m_0},\quad \omega_3=\sqrt{\frac{K_3}{m_3}},\ \omega_2=\sqrt{\frac{K_2}{m_2}},$$
$$\omega_1=\sqrt{\frac{K_1}{m_1+m_0}};\ \xi_3=\frac{C_3}{2\sqrt{K_3m_3}},\ \xi_2=\frac{C_2}{2\sqrt{K_2m_2}}.$$

where, ξ_2 and ξ_3 are the damping ratios of the suspension system and the occupant-cushion system, respectively.

For the sake of convenience, the following notations are given:

$$\begin{cases}\beta_1=2\left[\xi_2\omega_2(1+r_1)+\xi_3\omega_3(1+r_2)\right]\\ \beta_2=\omega_1^2+\omega_2^2\left(1+r_2\right)+\omega_3^2\left(1+r_1\right)+\\ 4\xi_2\xi_3\omega_2\omega_3\left(1+r_1+r_1r_2\right)\\ \beta_3=2[\xi_2\omega_1^2\omega_2+\xi_3\omega_1^2\omega_3\left(1+r_2\right)+\\ \xi_2\omega_2\omega_3^2\left(1+r_1+r_1r_2\right)+\xi_3\omega_2^2\omega_3\left(1+r_1+r_1r_2\right)]\\ \beta_4=\omega_1^2\omega_2^2+\omega_1^2\omega_3^2\left(1+r_2\right)+\\ \omega_2^2\omega_3^2\left(1+r_1+r_1r_2\right)+4\xi_2\xi_3\omega_1^2\omega_2\omega_3\end{cases} \tag{5}$$

The transfer function of the occupant acceleration to the road excitation velocity can be expressed as:

$$H(\mathrm{j}\omega)=\frac{\omega\omega_1^2\omega_2\omega_3[\left(\omega_2\omega_3-4\xi_2\xi_3\omega^2\right)\mathrm{j}-2(\omega_2\xi_3+\omega_3\xi_2)\omega]}{-\omega^6+\beta_1\omega^5\mathrm{j}+\beta_2\omega^4-\beta_3\omega^3\mathrm{j}-\beta_4\omega^2+2\omega_1^2\omega_2\omega_3(\omega_3\xi_2+\omega_2\xi_3)\omega\mathrm{j}+\omega_1^2\omega_2^2\omega_3^2} \tag{6}$$

According to reference [20], the occupant mean square acceleration can be expressed as:

$$\sigma_a^2=\frac{1}{2\pi}\int_0^{+\infty}\left|H(\mathrm{j}\omega)\right|^2G_{\dot q}(\omega)\mathrm{d}\omega \tag{7}$$

Substituting Equation (3) into Equation (7), one obtains the following equation:

$$\sigma_a^2=2\pi G_q(n_0)n_0^2v\int_0^{+\infty}\left|H(\mathrm{j}\omega)\right|^2\mathrm{d}\omega \tag{8}$$

Substituting Equation (6) into Equation (8), then, Equation (8) can be further expressed as [21]:

$$\sigma_a=\pi n_0\sqrt{\frac{G_q(n_0)v\lambda_2}{\lambda_1}} \tag{9}$$

where, $\lambda_2=h_2(\beta_1\beta_3\beta_6-\beta_1\beta_4\beta_5+\beta_5^2)+h_3(\beta_1^2\beta_6+\beta_3\beta_5-\beta_1\beta_2\beta_5)+h_4(\beta_1^2\beta_4-\beta_1\beta_2\beta_3-\beta_1\beta_5+\beta_3^2)$

$$\lambda_1=\beta_5^3+3\beta_1\beta_3\beta_5\beta_6-2\beta_1\beta_4\beta_5^2-\beta_2\beta_3\beta_5^2-\beta_3^3\beta_6+\beta_3^2\beta_4\beta_5+\beta_1\beta_2\beta_3^2\beta_6+\beta_1^3\beta_6^2-\beta_1^2\beta_3\beta_4\beta_6+\beta_1^2\beta_4^2\beta_5-\beta_1\beta_2\beta_3\beta_4\beta_5+\beta_1\beta_2^2\beta_5^2-2\beta_1^2\beta_2\beta_5\beta_6$$

$\beta_5=2\omega_1^2\omega_2\omega_3(\omega_3\xi_2+\omega_2\xi_3)$, $h_3=4(\omega_2^2\xi_3^2+\omega_3^2\xi_2^2)\omega_1^4\omega_2^2\omega_3^2$, $\beta_6=\omega_1^2\omega_2^2\omega_3^2$, $h_2=16\xi_2^2\xi_3^2\omega_1^4\omega_2^2\omega_3^2$, $h_4=\omega_1^4\omega_2^4\omega_3^4$.

3. 2. Solution of suspension optimal damping ratio for IWMVs

Due to a large number of unknown parameters in the initial phase of vehicle design, designers often only consider the comfort index — the occupant RMS acceleration. Thus, in the following, based on Equation (9), the initial design method of the damping ratio of suspension system for IWMVs is discussed.

Taking the partial derivative of Equation (9) with respect to the damping ratio ξ_2, setting the derivative to zero:

$$\frac{\partial\sigma_a}{\partial\xi_2}=0 \tag{10}$$

Substituting Equation (9) into Equation (10), one obtains the following equation:

$$\frac{\partial}{\partial\xi_2}\left(\pi n_0\sqrt{\frac{G_q(n_0)v\lambda_2}{\lambda_1}}\right)=0 \tag{11}$$

Through the identity transformation, Equation (11) can be expressed as:

$$\frac{\partial}{\partial\xi_2}\left(\frac{\lambda_2}{\lambda_1}\right)=0 \tag{12}$$

According to Equation (12), one obtains the following equation about ξ_2:

$$P_8\xi_2^8+P_7\xi_2^7+P_6\xi_2^6+P_5\xi_2^5+P_4\xi_2^4+P_3\xi_2^3+P_2\xi_2^2+P_1\xi_2+P_0=0 \tag{13}$$

where, P_i is determined by vehicle parameters K_1, K_2, K_3, m_0, m_1, m_2, C_3, and the occupant mass m_3.
Equation (13) can be abbreviated as:

$$F(\xi_2)=0 \tag{14}$$

The real root of Equation (14) is the optimal damping ratio of the suspension system. The optimal damping ratio $\xi_{2\mathrm{op}}$ can be expressed as:

$$\xi_2=\xi_{2\mathrm{op}}(K_1,K_2,K_3,m_0,m_1,m_2,m_3,C_3) \tag{15}$$

The optimal damping ratio $\xi_{2\mathrm{op}}$ of suspension system for IWMVs is determined by vehicle parameters K_1, K_2, K_3, m_0, m_1, m_2, C_3, and the occupant mass m_3.

For instance, the basic parameters of an IWMV are as follow: m_1=15.5 kg, m_2=202.0 kg, K_1=200.0 kN/m, K_2=22.4 kN/m, K_3=24.0 kN/m, C_3=1000 Ns/m; m_3=65.0 kg. The most commonly running road condition of the vehicle is as follows: v=60.0 km/h, B-class road. Under different values of the in-wheel motor mass m_0=0, 12, 24, 36, and 48 kg, the damping ratio ξ_{2op} and the corresponding occupant RMS acceleration σ_a are shown in Table 1. Where, m_0=0 represents the condition of the traditional car. Under different values of the in-wheel motor mass m_0=0, 12, 24, 36, and 48 kg, the relationship between the occupant RMS acceleration σ_a and the damping ratio ξ_2 is shown in Figure 2. From Figure 2, it can be seen that with the increase of the damping ratio ξ_2, the occupant RMS acceleration σ_a decreases firstly and then increases. Moreover, when the damping ratio ξ_2 equals ξ_{2op}, σ_a reaches its minimum.

4. FACTORS ANALYSIS OF SUSPENSION OPTIMAL DAMPING RATIO FOR IWMVS

The thorough factors analysis of ξ_{2op} was conducted. The basic parameters and the possible values range of an IWMV are shown in Table 2 [6].

4. 1. Effect of Cushion Damping The curve of the optimal damping ratio ξ_{2op} versus cushion damping C_3, is shown in Figure 3.

TABLE 1. The optimal damping ratio ξ_{2op} and the corresponding acceleration σ_a

Motor mass m_0 (kg)	Optimal ratio ξ_{2op}	Acceleration σ_a (m/s^2)
0	0.2895	0.6261
12	0.2834	0.6382
24	0.2772	0.6511
36	0.2712	0.6649
48	0.2653	0.6797

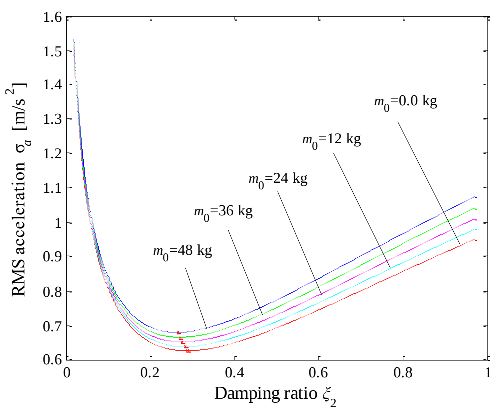

Figure 2. The occupant RMS acceleration σ_a versus the damping ratio ξ_2

TABLE 2. The basic parameters and the possible value ranges of an IWMV

Vehicle parameter	Nominal value	Value range
m_0 (kg)	24.0	[0, 48]
m_1 (kg)	15.5	[10, 30]
m_2 (kg)	65.0	[50, 100]
m_3 (kg)	202.0	[150, 300]
K_1 (kN/m)	200.0	[100, 300]
K_2 (kN/m)	22.4	[10, 40]
K_3 (kN/m)	24.0	[10, 100]
C_3 (Ns/m)	1000	[0, 2000]

From Figure 3, it can be seen that with the increase of the cushion damping C_3, the optimal damping ratio ξ_{2op} decreases firstly, then increases, finally trends to gentle. Moreover, the larger the value of the in-wheel motor mass m_0 is, the smaller the design value of the optimal damping ratio ξ_{2op} needs.

4. 2. Effects of stiffness parameters The stiffness parameters of the OVR model for IWMVs include the cushion stiffness K_3, the suspension stiffness K_2, and the tire stiffness K_1. Figures 4 to 6 depict the relationship of the optimal ratio ξ_{2op} and the stiffness parameters.

From Figure 4, it can be seen that the optimal damping ratio ξ_{2op} decreases with the increase of cushion stiffness K_3. When the cushion stiffness K_3 belongs to the range [10, 100] kN/m, the range of the optimal damping ratio ξ_{2op} is about [0.15, 0.45], rather than the damping ratio range [0.2, 0.4] for traditional cars.

Figure 5 illustrates that the optimal damping ratio ξ_{2op} increases with the increase of suspension stiffness K_2. Moreover, when the suspension stiffness K_2 is less than 6.5 kN/m and the in-wheel motor mass m_0 is larger than 12 kg, the optimal damping ratio ξ_{2op} is less than 0.2.

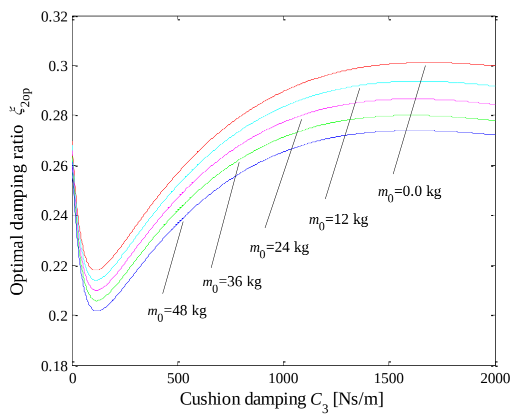

Figure 3. The optimal ratio ξ_{2op} versus the cushion damping C_3

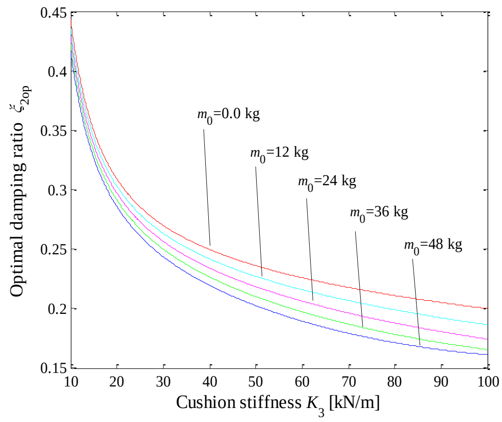

Figure 4. The optimal damping ratio ξ_{2op} versus cushion stiffness K_3

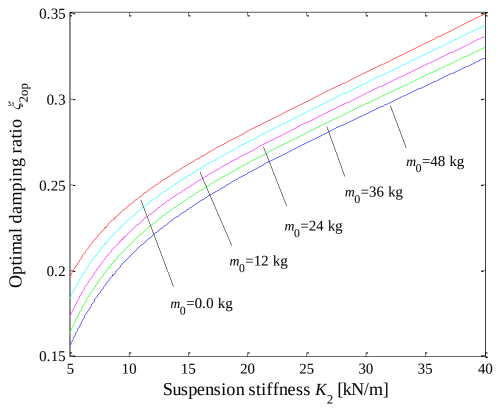

Figure 5. The optimal damping ratio ξ_{2op} versus suspension stiffness K_2

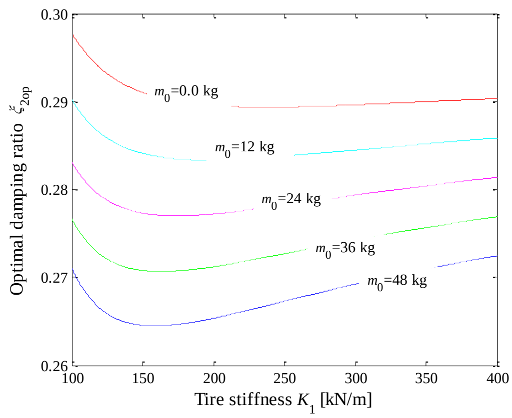

Figure 6. The optimal damping ratio ξ_{2op} versus tire stiffness K_1

That is to say, the value of the optimal damping ratio ξ_{2op} is out of [0.2, 0.4]. In such conditions, if the damping ratio range [0.2, 0.4] for the traditional cars is referred for IWMVs, it will be hard to realize optimal damping matching of suspension system. Figure 6 shows that the optimal damping ratio ξ_{2op} decreases initially and then increases with the increase of tire stiffness K_1. The smaller the in-wheel motor mass m_0 is, the more gentle the change trend becomes.

4. 3. Effects of Mass Parameters The mass parameters of the OVR model for IWMVs include the occupant mass m_3, the car body mass m_2, the tire system mass m_1, and the in-wheel motor mass m_0. Figures 7 to 9 depict the relationship of the optimal ratio ξ_{2op} and the mass parameters.

Figure 7 depicts that the optimal damping ratio ξ_{2op} increases approximately linearly as the occupant mass m_3 increases. Figure 8 illustrates that the optimal damping ratio ξ_{2op} is nonlinearly related to the car body mass m_2. The optimal damping ratio ξ_{2op} decreases with the increase of the car body mass m_2. Figure 9 shows that there is almost a linearly descending relationship between the optimal damping ratio ξ_{2op} and the tire system mass m_1. Figures 7 to 9 all prove that the larger the value of the in-wheel motor mass m_0 is, the smaller the design value of the optimal damping ratio ξ_{2op} needs.

Totally, the occupant-cushion and the in-wheel motor system have important effects. The suspension optimal damping ratio ξ_{2op} is irrelevant to driving road conditions. Moreover, the suspension optimal damping ratio ξ_{2op} range [0.15, 0.45] for IWMVs is recommended, rather than the damping ratio range [0.2, 0.4] for traditional cars. The variation laws of the suspension optimal damping ratio ξ_{2op} provide a theoretical basis for the initial design of suspension damping ratio ξ_2 for IWMVs.

5. TEST VERIFICATION OF SUSPENSION OPTIMAL DAMPING RATIO FOR IWMVS

To verify the reliability of the suspension optimal damping ratio ξ_{2op}, a comparison of the calculated ξ_{2op} and the tested ξ_2 should be conducted.

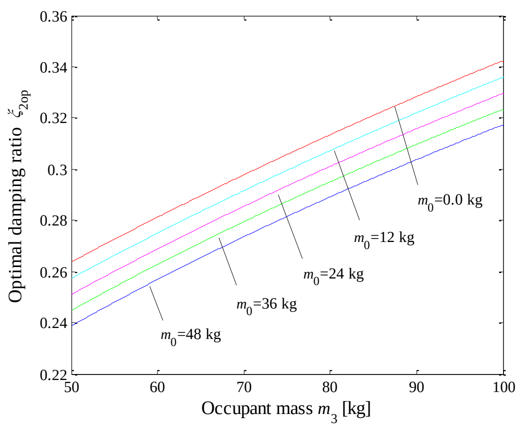

Figure 7. The optimal damping ratio ξ_{2op} versus occupant mass m_3

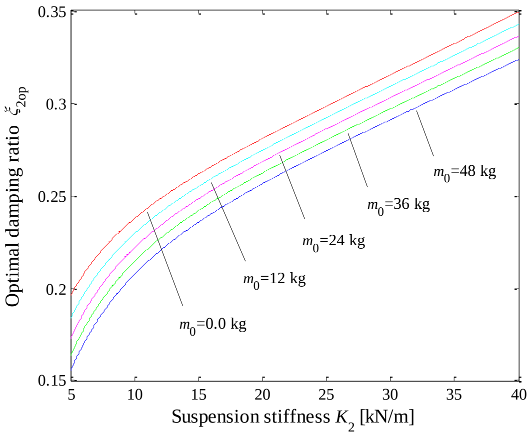

Figure 8. The optimal damping ratio ξ_{2op} versus car body mass m_2

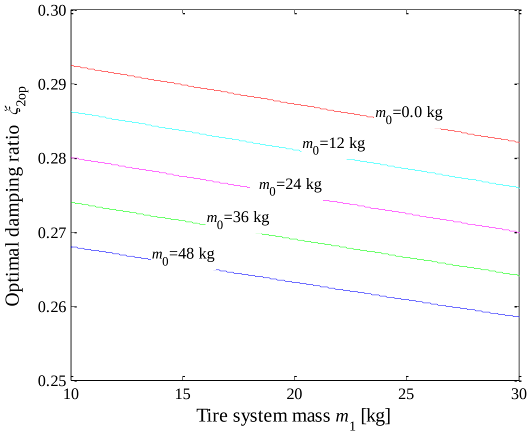

Figure 9. The optimal damping ratio ξ_{2op} versus tire system mass m_1

The calculated ξ_{2op} is regarded as the initial design value, while the tested ξ_2 is regarded as the final precise design value of suspension damping ratio ξ_2.

5. 1. Tested IWMV and Suspension Optimal Damping Ratio The driving mode of the tested vehicle is rear-wheel-drive using two in-wheel motors. The configuration and parameters of the tested IWMV are as follows:

Length×Width×Height (4501 mm×1704 mm×1469 mm), wheelbase 2604 mm, curb-weight 1115 kg; MacPherson front independent suspension, torsion beam rear suspension, two seats per row (two rows, no seat suspension), rigid polyurethane foam cushion with equivalent damping 1400 Ns/m and stiffness 28 kN/m. Under the full load condition, the front axle load 540 kg, the rear axle load 575 kg, ① for one front suspension system: the suspension vertical stiffness 12.3 kN/m, the tire vertical stiffness 189 kN/m, the tire system mass 24.0 kg; ② for one rear suspension system: the suspension vertical stiffness 19.3 kN/m, the tire vertical stiffness 210 kN/m, the tire system mass 20.5 kg, the in-wheel motor mass 17.0 kg.

Because the motor is attached to the rear driving wheel for the tested IWMV, we took the rear suspension system as an example to calculate the suspension optimal damping ξ_{2op}. According to the tested IWMV parameters, solving Equation (14), the calculated optimal damping ξ_{2op} is 0.25. The value 0.25 is regarded as the initial design value of the rear suspension damping ratio.

5. 2. Measurement of Suspension Damping Ratio
According to the Chinese standard GB/T 4783-1984, the measurement of the rear suspension damping ratio for the tested IWMV is carried out. The used equipment is German ICM data acquisition system.

According to the standard GB/T 4783-1984, the test is carried out under the full load condition. The specific practice is as follows: firstly, dismantled the seat cushion and two dampers in two front suspension systems; then, installed one DH105E acceleration sensor on the rear chassis frame; thirdly, put each rear wheel on a semi trapezoidal bump with height 60 mm, respectively; subsequently, pushed the vehicle off the bump and ensured that the two wheels were able to fall to the ground at the same time as much as possible; meanwhile, used the ICM data acquisition system to collect the free attenuation time history of the frame vertical vibration acceleration for 7 s. In order to ensure the reliability of the collected data, the test was repeated three times. Given space limitations and the similarity of each time history, the collected data for the first test is provided, as shown in Figure 10. According to the time histories of three tests, the adjacent three peak values and the corresponding time points for each test were extracted as shown in Table 3.

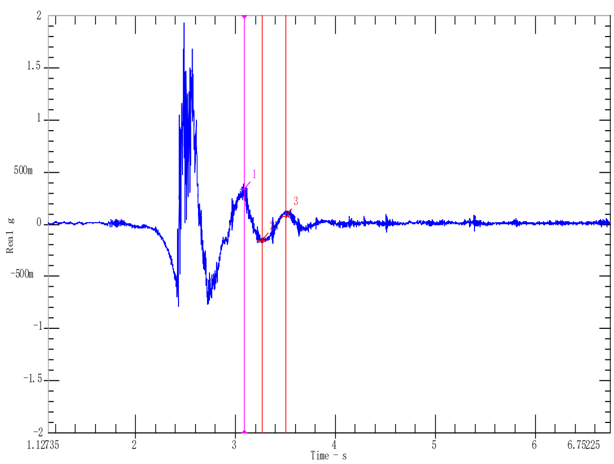

Figure 10. The free attenuation time history of the frame vertical acceleration for the first test

TABLE 3. The adjacent three peak values and the corresponding time points

Test		Point 1	Point 2	Point 3
First	Time (s)	t_1=3.0970	t_2=3.2760	t_3=3.5120
	Peak (10^{-3}g)	A_1=337.27	A_2=-149.89	A_3=75.608
Second	Time (s)	t_1=3.0280	t_2=3.2070	t_3=3.4430
	Peak (10^{-3}g)	A_1=335.89	A_2=-148.72	A_3=75.218
Third	Time (s)	t_1=3.0460	t_2=3.2280	t_3=3.4640
	Peak (10^{-3}g)	A_1=338.02	A_2=-150.63	A_3=76.112

From Table 3, using the time-history method, it can be obtained that the tested values of the rear suspension damping ratio for three tests are 0.238, 0.239, and 0.237, respectively. The mean value of the tested damping ratio is 0.24 which can be regarded as the final precise design value of suspension damping ratio ξ_2, where, the formula of the damping ratio for the time-history method for our calculation is as follows:

$$\xi = \frac{1}{\sqrt{1+\frac{\pi^2}{\ln^2\left[(A_1-A_2)/(A_3-A_2)\right]}}} \tag{16}$$

5. 3. Comparison of the Initial Value and the Precise Value for ξ_2

A comparison of the damping ratio between initial design value (the calculated ξ_{2op}) and the final precise design value (the tested ξ_2) is shown in Table 4.

From Table 4, it can be seen that the calculated optimal damping ratio ξ_{2op} is 0.25, the final precise design value is 0.24 and their absolute deviation and relative deviation are 0.01 and 5.4%, respectively.

The results show that the optimal damping ratio ξ_{2op} is very close to the final precise design value. Thus, the optimal damping ratio ξ_{2op} can provide a good initial design value for the accurate design stage of IWMVs. Compared with the traditional method, the proposed method can effectively avoid the blindness and subjectivity of the damping ratio selection among 0.2~0.4, so that it can reduce the workload and the design difficulty.

TABLE 4. A comparison of the calculated ξ_{2op} and the final precise design value

Final precise design value	Initial design value	
0.24	Optimal damping ratio ξ_{2op}	Traditional method
	0.25	0.2~0.4

6. CONCLUSION

A traveling vibration model of OVR for IWMVs is created, which considers the effects of the occupant-cushion system and the in-wheel motor system. Based on the model, using a special form of infinite integral, a mathematical expression of the occupant RMS acceleration is derived. Using the RMS optimization criterion for ride comfort, an 8 order polynomial equation about the suspension optimal damping ratio is deduced. Finally, the reliability of the optimal damping ratio is validated. Some instructive conclusions and proposals are as follow:

(1) The optimal damping ratio ξ_{2op} of suspension system is determined by vehicle parameters K_1, K_2, K_3, m_0, m_1, m_2, C_3, and occupant mass m_3.

(2) The optimal damping ratio ξ_{2op} increases approximately linearly as the occupant mass m_3 increases. The larger the value of the in-wheel motor mass m_0 is, the smaller the design value of the optimal damping ratio ξ_{2op} needs.

(3) The optimal damping ratio ξ_{2op} decreases with the increase of cushion stiffness K_3. With the increase of the cushion damping C_3, the optimal damping ratio ξ_{2op} decreases firstly, then increases, finally trends to gentle.

(4) The optimal damping ratio ξ_{2op} range [0.15, 0.45] for IWMVs is recommended, rather than the damping ratio range [0.2, 0.4] for traditional cars. That is to say, the value of the optimal damping ratio ξ_{2op} for IWMVs may be out of the range [0.2, 0.4].

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Modelling and Test Verification of Suspension Optimal Damping Ratio for Electric Vehicles Considering Occupant-cushion and In-wheel Motor Effects

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چکیده

نسبت میرایی تعلیق شاسی یک پارامتر کلیدی برای تطبیق میرایی موتورهای چرخ دنده ای (IWMV) می باشد. از آنجا که موتور به چرخ متصل است، روش طراحی اولیه نسبت میرایی اتومبیل های سنتی به طور کامل برای IWMV مناسب نیست. این مقاله روشی برای طراحی اولیه ابتکاری نسبت میرایی برای IWMV ها ارائه می دهد. در ابتدا، یک مدل ارتعاشات جابجایی وسیله نقلیه جاده ای (OVR) برای IWMV ایجاد شده است. مدل شامل ساکنان، کوسن، تعلیق، موتور درایو، جاده و سرعت حرکت است. ثانیا، بر اساس مدل، با استفاده از یک فرم خاص انتگرال بی نهایت، عبارت ریاضی متوسط مربعات ریشه (RMS) شتاب حاصل می شود. سوم، بر اساس معیار بهینه سازی RMS برای راحتی سواری، یک معادله چند جمله ای مرتبه ۸ در مورد نسبت خمش مطلوب تعلیق محاسبه می شود. پس از آن، از طریق تجزیه و تحلیل عوامل، اصول تغییر نسبت میرایی بهینه نسبت به پارامترهای خودرو نمایش داده می شود. در نهایت، قابلیت اطمینان نسبت میرایی مطلوب با آزمون تایید می شود. انحراف نسبی نسبت به محدوده مطلوب محاسبه شده و نسبت میرایی آزمایشی ۵/۴ درصد است. نتایج نشان می دهد که نسبت میرایی بهینه مطلوب می تواند راهنمای خوبی برای تطبیق میرایی تعلیق برای IWMV ها باشد.

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