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Research Article

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Simulation and Analysis of Dual-mass Vibration for a Quarter Vehicle Model

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Abstract In this paper, the dual-mass vibration system of 1/4 vehicle model is selected as the research object, and a simplified two-degree-of-freedom dual-mass vibration system model is established. The software named MATLAB/Simulink is used for simulation analysis, and the suspension model with symmetric damping is compared with that with asymmetric damping under the ground step input. The results show that the asymmetric damping suspension has better damping effect. Moreover, the effects of damper diastolic travel damping coefficient, suspension stiffness, suspended mass, non-suspended mass and tire stiffness on the vertical displacement and vertical acceleration of the vehicle body are studied respectively. The results show that reducing the suspension stiffness, increasing the damper diastolic travel damping coefficient or decreasing tire stiffness can improve the ride comfort of the vehicle

Keywords Dual-mass vibration system, Asymmetric damping, Suspension stiffness, Suspended mass

1. Introduction

Suspension is an indispensable part of the car, which transmits various forces and torques to the vehicle body, and can slow down the vibration and impact caused by the road surface, make the driver more comfortable, and ensure the car has good handling stability and ride comfort[Error! Reference source not found.]. Therefore, reasonable selection of the damping coefficient and stiffness of the elastic element in the suspension system will significantly promote the optimization of vehicle ride comfort and handling.

In this paper, the passive suspension is taken as the research object. And the vehicle vibration is simulated by MATLAB/Simulink to analyze the influence of the stiffness and damping coefficient of the suspension system, suspension mass, non-suspension mass and tire stiffness for the performance of the suspension.

2. Dual-mass Vibration System Model

2.1 Model Simplification

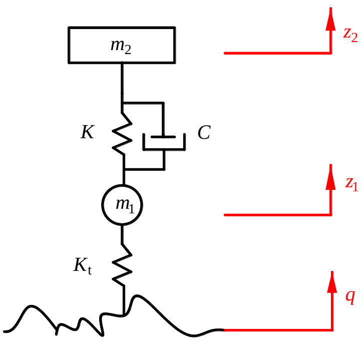

Figure 1: Two-degree-of-freedom vibration system model

Vehicle can actually be regarded as a vibration system, and it is very difficult to describe the vibration system accurately with a definite model. However, if you ignore some unimportant factors and then simplify the model, it will be much easier **Error! Reference source not found.**. Assuming that the left and right sides of the vehicle are symmetrical along its vertical axis, the road roughness received by the left and right wheels of the vehicle is approximately the same, and when the suspension mass distribution coefficient of the vehicle is close to 1, the vertical vibration of the front and rear suspension system is almost independent, so it can be simplified into a two-degree-of-freedom vibration system **Error! Reference source not found.**, as shown in *Figure 1*.

2.2 Mathematical Model

The coordinate origin of vehicle body and the wheel is selected after their respective equilibrium positions, and the dynamics equation of the two-degree-of-freedom system is established according to Newton's second law:

$$m_2\ddot{z}_2 + C\left(\dot{z}_2 - \dot{z}_1\right) + K\left(z_2 - z_1\right) = 0 \qquad (1)$$

$$m_1\ddot{z}_1 + C\left(\dot{z}_1 - \dot{z}_2\right) + K\left(z_1 - z_2\right) + K_t\left(z_1 - q\right) = 0 \qquad (2)$$

Where, m_1 is the non-hanging mass (kg); m_2 is the suspension mass (kg); C is the damping coefficient of the shock absorber ($\mathrm{N \cdot s/m}$); K is the suspension stiffness ($\mathrm{N/m}$); K_t is the tire stiffness ($\mathrm{N/m}$); z_1 is the vertical displacement of the wheel (m); z_2 is the vertical displacement of the vehicle body (m); q is the road roughness (m).

2.3 Modeling

After establishing the dynamic equation of the two-degree-of-freedom system, a model can be built in Simulink, the core idea of which is to divide m_1 and m_2 to the right side of the equation, and only put $\ddot{z}_1$ and $\ddot{z}_2$ on the left side of the equation. Such as:

$$\ddot{z}_2 = \frac{1}{m_2}\left[\left(C\dot{z}_1 - C\dot{z}_2\right) + \left(Kz_1 - Kz_2\right)\right] \qquad (3)$$

$$\ddot{z}_1 = \frac{1}{m_1}\left[\left(C\dot{z}_2 - C\dot{z}_1\right) + \left(Kz_2 - Kz_1\right) + \left(K_t q - K_t z_1\right)\right] \qquad (4)$$

To integrate a variable twice, the displacement obtained by the last integral and the velocity obtained by its derivative are both multiplied by the coefficient and then added, equal to the variable on the left side of the differential equation; The road excitation is represented by a step signal with a size of 0.1m. Asymmetric damping is realized by switch module in Simulink. When $\dot{z}_2 - \dot{z}_1 > 0$, it is stretch stroke and output C_2. When $\dot{z}_1 - \dot{z}_2 > 0$, it is compression travel and output C_1, so that you can easily build a model. The Simulink model of dual-mass vibration system is shown in *Figure 2*.

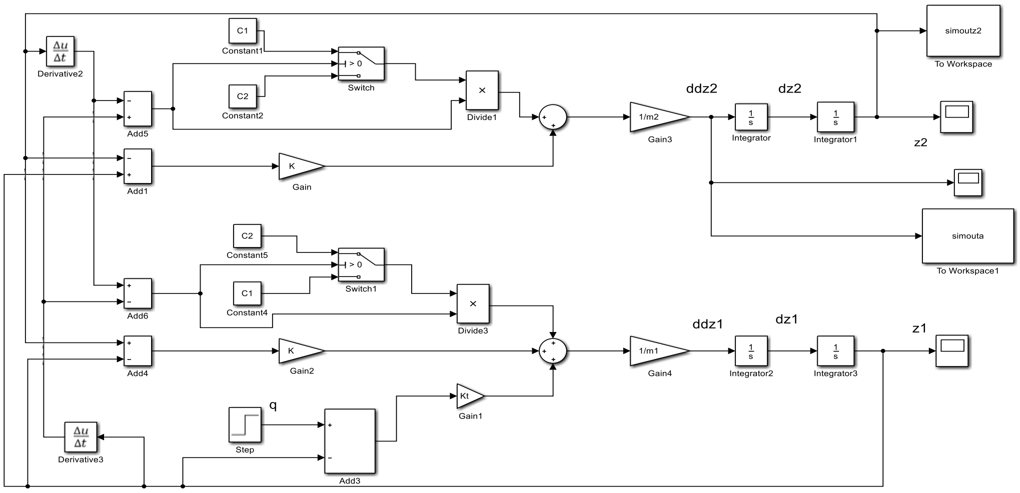

Figure 2: Simulink model of dual -mass vibration system

3. Simulation and Analysis

In this chapter, using the MATLAB/Simulink R2016b was to conduct simulation. In this simulation experiment, the Normal mode was selected, the simulation step was set as Fixed-step, and the sampling time was set as 0.01s. Firstly, the simulation of dual-mass vibration system with symmetric damping suspension is carried out, and then dual-mass vibration system of asymmetrically damped suspension and symmetrically damped suspension is simulated and compared. Finally, on the basis of keeping the parameters of the dual-mass vibration system of the symmetrical damping suspension unchanged, the suspension stiffness, the damping coefficient of the shock absorber's relaxation stroke, the suspension mass, the non-suspension mass and the tire stiffness were changed respectively to analyze the influence on the ride comfort of the vehicle.

Simulation one:

1/4 vehicle dual-mass vibration system (symmetric damping) model parameters: non-suspension mass $m_1 = 100\text{kg}$, suspension mass $m_2 = 1000\text{kg}$, suspension stiffness $K = 30000\,\text{N/m}$, tire stiffness $K_t = 100000\,\text{N/m}$, damping coefficient of compression and relaxation stroke of shock absorber $C_1 = C_2 = 2500\,\text{N}\cdot\text{s/m}$, ground input q is step input at 1s. After inputting the above data, the vertical displacement and vertical acceleration of vehicle body under symmetrical damping are obtained, as shown in *Figure 3* and *Figure 4* respectively.

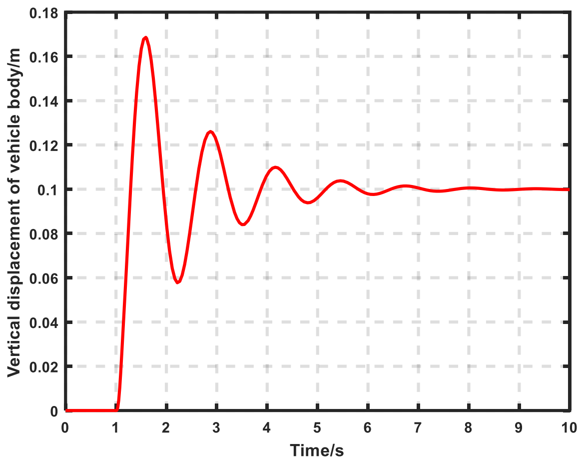

Figure 3: Vertical displacement of vehicle body

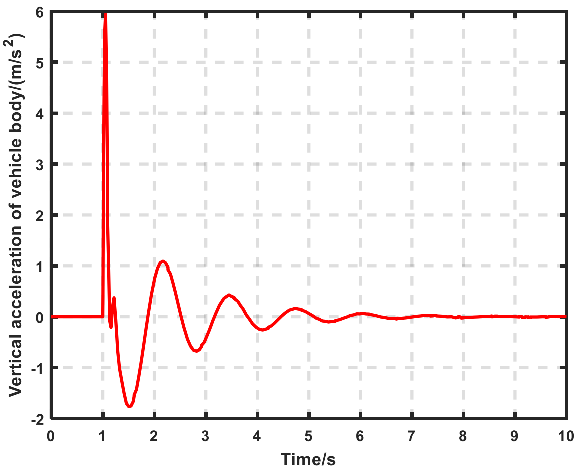

Figure 4: Vertical acceleration of vehicle body

According to *Figure 3*, *Figure 4*, it can be seen that under the above parameters, the peak displacement of the vehicle body is about 0.168 m, the peak acceleration of vibration is about 5.9m/s^2, and the time for vehicle body vibration to reach steady state is about 7.9s.

Simulation two:

The model parameters of 1/4 vehicle dual-mass vibration system (asymmetric damping) are: non-suspension mass $m_1 = 100\text{kg}$, suspension mass $m_2 = 1000\text{kg}$, spring stiffness $K = 30000\,\text{N/m}$, tire stiffness $K_t = 100000\,\text{N/m}$, damping coefficient of shock absorber compression stroke $C_1 = 2500\,\text{N}\cdot\text{s/m}$, damping coefficient of relaxation stroke $C_2 = 5000\,\text{N}\cdot\text{s/m}$, ground input q is step input at 1s. *Figure 5*. shows the comparison of vertical displacement between symmetric and asymmetrically damped suspension, and *Figure 6*. shows the comparison of vertical acceleration between symmetric and asymmetrically damped suspension.

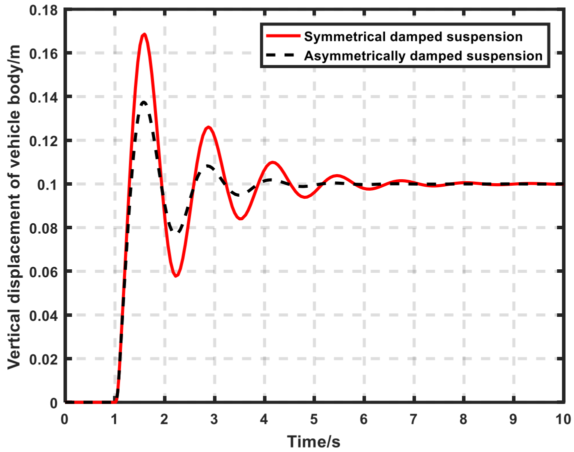

Figure 5: Comparison of vertical displacement between symmetric and asymmetrically damped

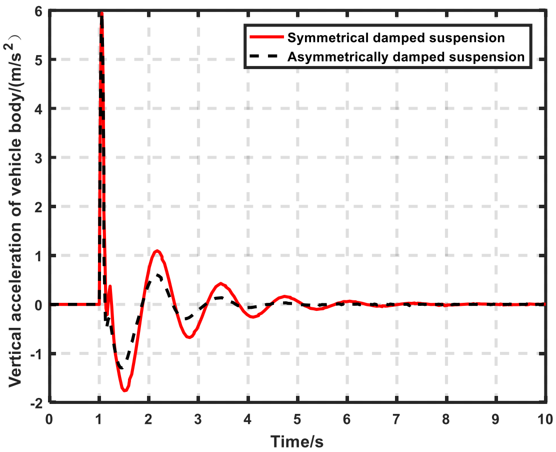

Figure 6: Comparison of vertical acceleration between symmetric and asymmetrically damped

It can be seen from *Figure 5* and *Figure 6* that the dual-mass vibration model with asymmetric damping damper has better damping effect than that of the dual-mass vibration model with symmetrical damping damper. Under the same step input, the peak value of vertical displacement is lower for the vibration model with asymmetric damping, and the vertical acceleration is also lower than that of the vibration model with symmetrical damping, and the time to reach steady state is shorter.

Simulation three:

The parameters of vehicles in simulation one are kept unchanged, and only changed the damping coefficients of the shock absorber during the relaxation stroke The three damping coefficients of the shock absorber, $C_2 = 300\,\mathrm{N \cdot s/m}$, $C_2 = 2500\,\mathrm{N \cdot s/m}$, $C_2 = 5000\,\mathrm{N \cdot s/m}$, during the relaxation stroke are respectively used for simulation. *Figure 7* shows the vertical displacement with different damping coefficients of diastolic stroke, and *Figure 8* shows the vertical acceleration with different damping coefficients of diastolic stroke.

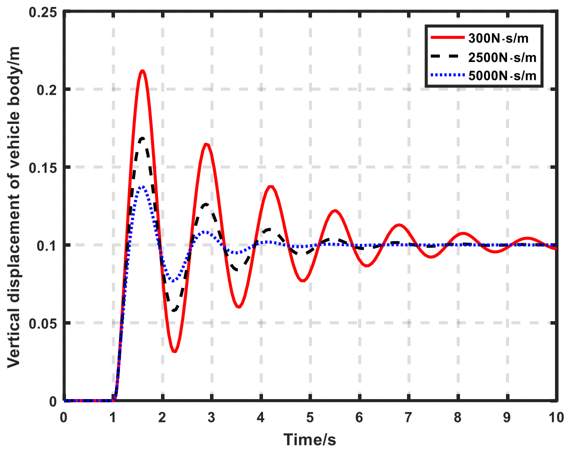

Figure 7: Vertical displacement with different damping coefficients of diastolic stroke

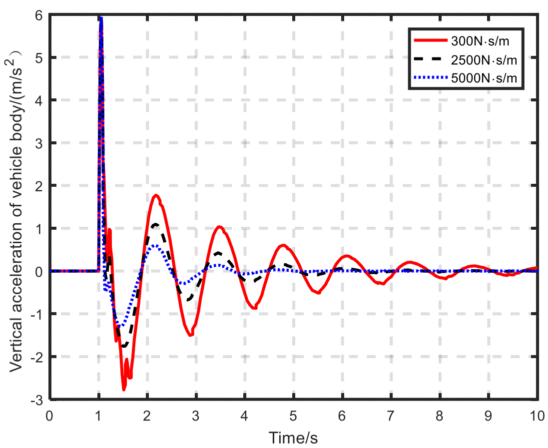

Figure 8: Vertical acceleration with different damping coefficients of diastolic stroke

It can be seen from *Figure 7* and *Figure 8* that with the gradual increase of the damping coefficient, the peak value of the vertical displacement decreases and the time to reach steady state is shortened. With the increase of the damping coefficient, the peak vertical acceleration does not change, but the time to reach the steady state is shortened. Therefore, in a certain range, the larger the damping coefficient of the relaxation stroke of the shock absorber, the better the damping effect of the suspension.

Simulation four:

While keeping other vehicle parameters unchanged in simulation one, only changed the suspension stiffness, and three different suspension stiffness, $K = 10000\,\mathrm{N/m}$, $K = 30000\,\mathrm{N/m}$, $K = 50000\,\mathrm{N/m}$, were simulated respectively. *Figure 9* is the vertical displacement of different suspension stiffness, and *Figure 10* is the vertical acceleration for different suspension stiffness

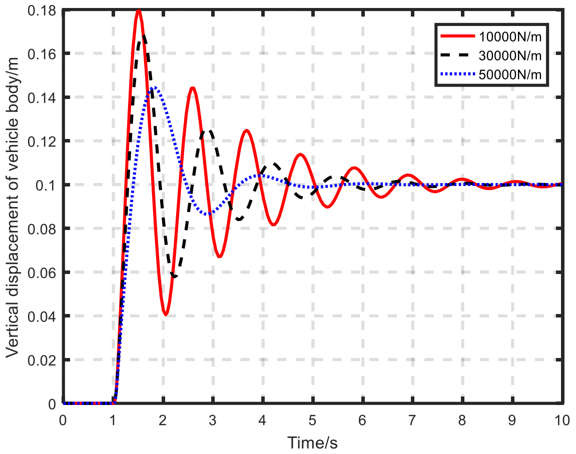

Figure 9: Vertical displacement for different suspension stiffness

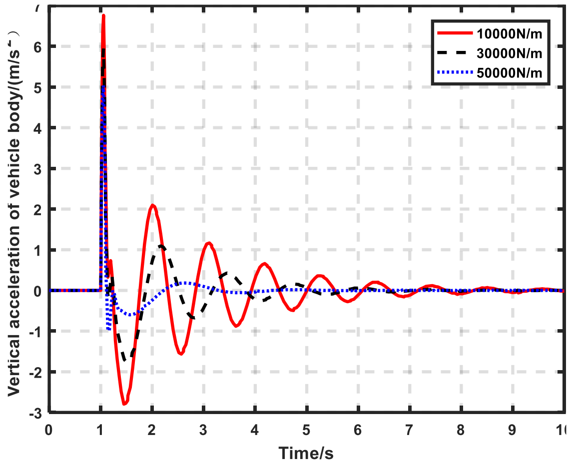

Figure 10: Vertical acceleration for different suspension stiffness

It can be seen from *Figure 9* and *Figure 10* that as the suspension stiffness gradually decreases, the peak value of the vertical vibration displacement increases, the time to reach the peak value becomes, but the time to enter the steady state becomes longer. With the gradual increase of suspension stiffness, the peak of vertical acceleration of the vehicle body becomes smaller, and the time to reach steady state becomes shorter.

Simulation five:

While keeping the other vehicle parameters unchanged in simulation one, only the suspended mass was changed, and three different suspension masses, $m_2 = 500\text{kg}$, $m_2 = 1000\text{kg}$, $m_2 = 2000\text{kg}$, were simulated respectively. *Figure 11* is the vertical displacement for different suspension masses, and *Fig.12.* is the vertical acceleration for different suspension masses.

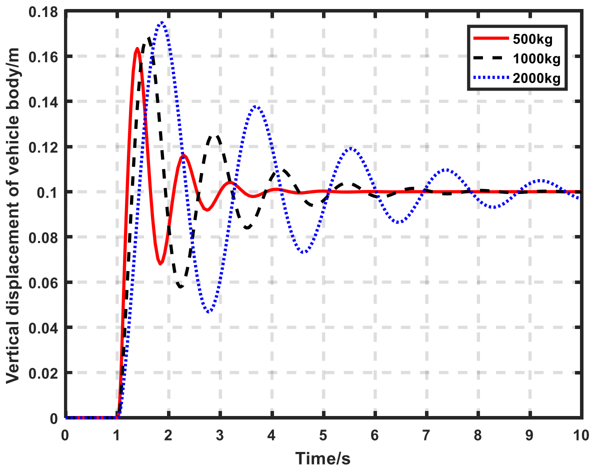

Figure 11: Vertical displacement for different suspension masses

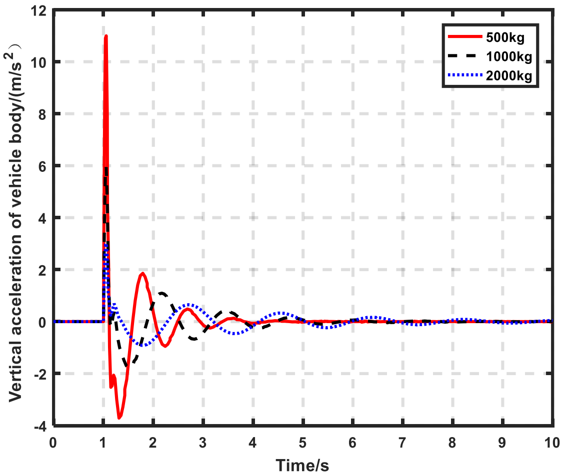

Figure 12: Vertical acceleration for different suspension masses

As can be seen from *Figure 11* and *Figure 12.*, with the increase of suspended mass, the peak value of vertical displacement increases slightly, the time to reach steady state increases significantly, and the peak value of vertical acceleration decreases significantly.

Simulation six:

While keeping other vehicle parameters unchanged in simulation one only changed the non- suspension mass, and three different non-suspension masses, m=50kg m= 100kg m= 200kg, were used for simulation respectively. Figure 13 shows the vertical displacement of the vehicle body for different non-suspended masses, and Figure 14 shows the vertical acceleration for different non-suspended masses.

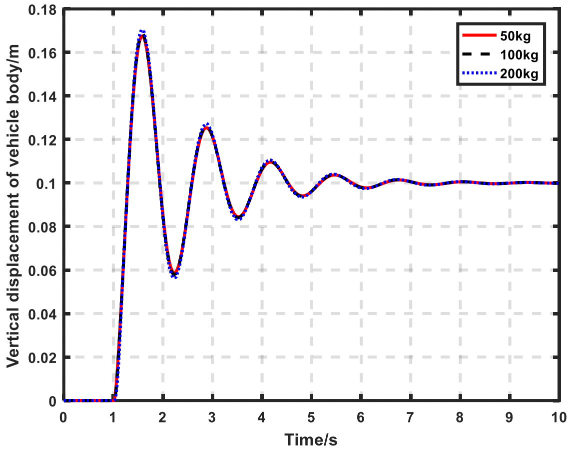

Figure 13: Vertical displacement for different non-suspended masses

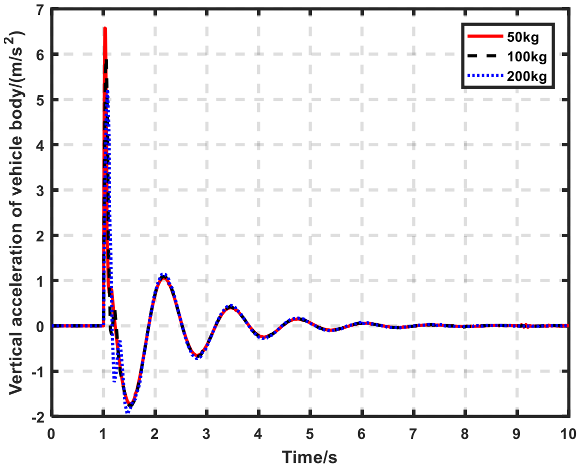

Figure 14: Vertical acceleration for different non-suspended masses

The non-suspended mass of this model only includes the tire mass, which has little influence on ride comfort because the tire mass is small. It can be seen from *Figure 13* and *Figure 14* that under different non-suspended masses, the changes of vertical displacement and acceleration are not obvious.

Simulation seven:

While keeping the parameters of other vehicles unchanged in simulation one, only changed the tire stiffness, and the three different tire stiffness, $K_t = 50000\,\text{N/m}$, $K_t = 100000\,\text{N/m}$, $K_t = 200000\,\text{N/m}$, were used for simulation respectively. *Figure 15* is the vertical displacement for different tire stiffness, and *Figure 16.* is the vertical acceleration for different tire stiffness.

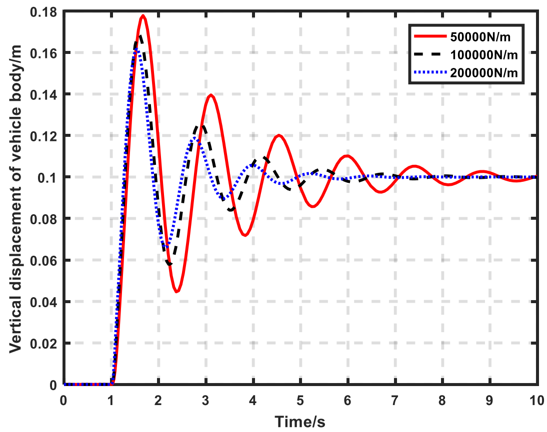

Figure 15: Vertical displacement for different tire stiffness

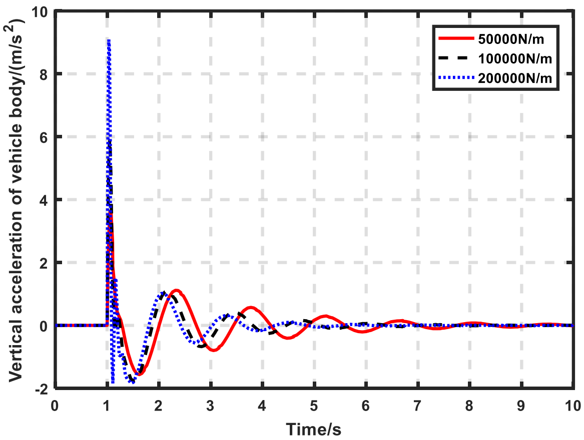

Figure 16: Vertical acceleration for different tire stiffness

As can be seen from *Figure 15* and *Figure 16*, with the decrease of tire stiffness, the peak value of vertical displacement slightly increases, and the time to reach steady state becomes longer. However, when 1s is stimulated by the road surface, the peak value of vertical acceleration significantly decreases, that is, the impact transmitted to the human body decreases. Therefore, appropriate reduction of tire stiffness is conducive to improving vehicle ride comfort.

4. Conclusion

In this paper, the dual-mass vibration system of 1/4 vehicle model is analyzed, and the suspension system with symmetric damping damper and asymmetric damping damper are compared. On the basis of maintaining the parameters of the symmetric damping system, the damping coefficient of the damper's diastolic stroke, suspension stiffness, suspended mass, non-suspended mass and tire stiffness were changed to analyze the influence on vehicle ride comfort. The following conclusions can be drawn from the simulation results.

1) The suspension system with asymmetric damping damper has better damping effect and ride comfort than that with symmetrical damping suspension system;
2) Increasing the damping coefficient of the shock absorber's diastolic stroke and reducing the suspension stiffness within a certain range can reduce the peak value of the vertical displacement during vehicle vibration, accelerate the attenuation speed of vehicle vibration, and make the system reach the steady state faster;
3) With the increase of suspended mass, the peak of the vertical acceleration of the vehicle body decreases obviously, and the time to reach the steady state increases obviously;
4) The change of non-suspended mass has little effect on the displacement and acceleration of the body.
5) With the decrease of tire stiffness, the peak value of vertical acceleration decreases obviously.

Therefore, using asymmetrically damped suspension, increasing the damping coefficient of the shock absorber's diastolic stroke or reducing the spring stiffness or reducing the tire stiffness can help improve the ride comfort of the car, but it is not always possible to pursue the ride comfort of the vehicle by increasing the damping coefficient of the shock absorber's diastolic stroke and reducing the spring stiffness. These parameters also affect the performance of the car's handling stability, so comprehensive consideration is needed.

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