Chapter 25
Cab Suspension Vibration Isolation Analysis Based on Vibration Decoupling Theory

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Abstract A certain engineering vehicle is taken as an example to build up the three-dimensional and full-floating cab 3D model and establish the vibration response model in the Adams platform. The vibration decoupling theory is applied to optimize the vibration stiffness of suspension and arrangement angle. The basic vibration isolation theory is used in the frequency domain for the analysis and calculation of transmission. Optimization results of the suspension system are verified, and the cab suspension system dynamics model of vehicle is created on the RecurDyn platform. The cab suspension system is placed in the vehicle for the comfort simulation to improve the comfort of the driver. NVH performance is improved with satisfactory results.

Keywords Cab suspension · Vibration decoupling · Multiple rigidity system dynamics · Vibration isolation analysis

25.1 Introduction

To guarantee the large load, high-speed, and high-power performance and low self-load, the engineering vehicles are designed with thinner body and lighter components. However, it results in larger vibration and higher noise. According to the limit of the market and related laws and regulations and requirements of customers, the vehicle designers can not ignore how to reduce the vibration and noise in the technical innovation to improve the competence [1]. According to the vibration measurement on vehicles in working at present, the vibration of a certain vehicle reaches as much as 2.5 G, several times higher than that of the present

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vehicles. It shows the extremely strong vibration in the cab [2]. The driver seat is directly connected with the vehicle base plate with bolts. The vehicle vibration passes to the driver through the seat supporter. It is hard to meet the driving comfort only with the cushion. It requires exploring new vibration reduction method according to the vehicle features to improve the driving comfort [3].

In the thesis, the whole built-in cab is designed within the vehicle according to the thought of modularization design. The vibration reduction optimization design is made on the basis of the vibration reduction optimization of the action system. The decoupling is made in various vibration directions on the vibration decoupling theory, obtaining optimized parameters of the suspension system in the cab.

25.2 Modeling

25.2.1 Structure Model

The 3D structure diagram is made on a certain engineering vehicle after the simplification, as indicated in Fig. 25.1.

In the figure, a capsule is added inside the cab shell to make overall vibration reduction, as indicated by the dotted line. On the basis of the action system, the capsule of composite materials is made with the casting molding technique inside the cab shell to reduce the vibration from the passing access and reducing the components' coupling vibration. The improvement is also made on the design of the operation parts in the cab according to the new can structure of the vehicle. The suspension unit is added between the inner capsule and the base plate, to reduce the vibration and noise and to improve the driving comfort in the whole cab.

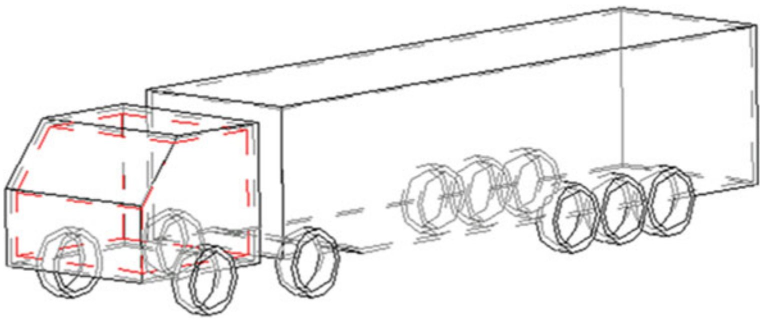

Fig. 25.1 The structure model of some engineering vehicle

25.2.2 Dynamic Modeling

The capsule inside the vehicle cab (the capsule) is suspended on the chassis with the suspension system of four groups of spring damping components in the direction of front and back, and left and right. Suppose the capsule only moving vertically, take the balance position of the capsule as the origin and also consider the spring force in the initial deformation stage in balance with the gravity, then [4]:

$$m\ddot{x} = -k(x - y) - c(\dot{x} - \dot{y}) \tag{25.1}$$

If the cabin movement is harmonic vibration in the vertical direction, can be expressed as:

$$y = Y \cos\omega t \tag{25.2}$$

To solve it, then

$$\frac{X}{Y} = \sqrt{\frac{k^2 + c^2\omega^2}{(k - m\omega^2)^2 + c^2\omega^2}} \tag{25.3}$$

If let the frequency ratio $\gamma = \omega/\sqrt{k/m}$, damping ratio $\xi = c/(2\sqrt{mk})$, then the transfer ratio X/Ywill be

$$\frac{X}{Y} = \sqrt{\frac{1 + (2\xi\gamma)^2}{(1 - \gamma^2)^2 + (2\xi\gamma)^2}} \tag{25.4}$$

According Formula (25.4), we can obtain the changing curve of the transfer ratio with frequency domain, as indicated in Fig. 25.2. When the frequency ratio is larger, the elastic support will make the vibration isolation. The smaller the damping ratio is, the better the vibration isolation is. However, too small damping ratio will lead to larger resonance peak value.

The capsule structure finite element simulation is made with the inertial parameters and coordinate positions of the equipment within the cab. The suspension system is installed in positions with high strength. The shell finite element simulation shows that the four corners of the base plate of the cab are ideal [5]. The cab vibration analysis model is created on Adams platform. The cab model is directly introduced with 3D modeling software. The vehicle head base plate is replaced with a rectangle vibration platform. The suspension system between the cab and vibration platform is connected with bushing force, with the position, rigidity, and damp of the electric support applied according to design values. The input stimulus is set up as the sinusoidal acceleration on the vibration platform, and the output channel is set up as the vertical vibration acceleration, vertical displacement at the center of mass and vertical vibration acceleration and vertical displacement at the driver's seat.

A multiple rigid body dynamic model of the whole vehicle is set up at RecurDyn platform, including the position of the center of mass and inertial

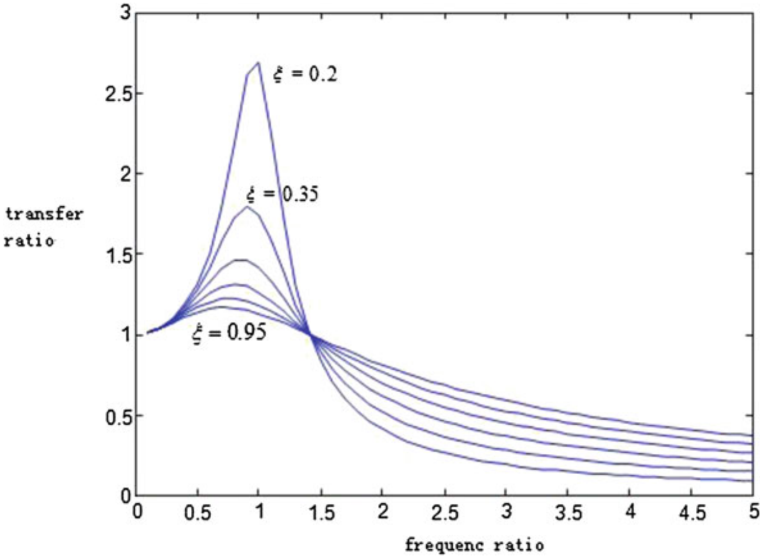

Fig. 25.2 The curve of the transfer ratio with frequency domain

parameters of the front axle, rear axle, chassis, cab, driver's seat, and tank. The comfort analog simulation is made [6].

25.3 Dynamic Analysis and Optimization Calculation

To obtain the optimized results, it requires modeling and optimization on independent cab at first. The sweep frequency mode is used to determine the vibration response curve to decide the point of resonance and effective vibration isolation frequency. For the whole vehicle simulation, generally there is no actual sample vehicle for experiment in the design process. The primary analog simulation can be made with the force curve calculated in dynamics.Optimized design process shown in (Fig. 25.3).

25.3.1 Calculation of Vibration Isolation of Suspension Unit in Cab

According to the simulation model, as shown in Table 25.1, set front suspension and rear suspension stiffness and damping ratio. The acceleration stimulus sweep frequency simulation of 1 g is made within 0.1–100 Hz. The inherent frequency, vibration type, and frequency domain response of the suspension system of the cab can be figured out.

Fig. 25.3 The flow of the optimization design

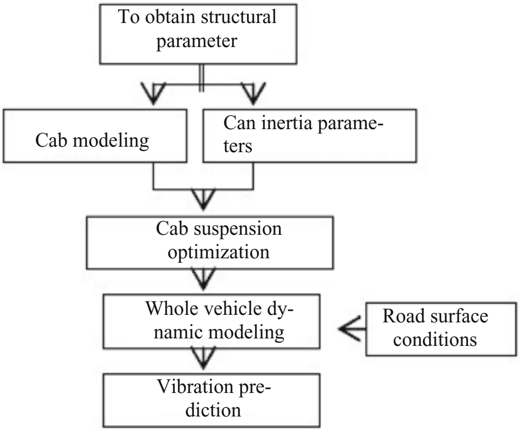

Table 25.1 Suspension system parameter of cabin

Parameter	Value
Front suspension rigidity	285 N/mm
Rear suspension rigidity	335 N/mm
Suspension damping ratio	0.12

The frequency domain response of the acceleration of the center of mass of the cab is calculated, of which the result shows that the vertical acceleration of the center of mass reaches the peak at 8.75 Hz approximately. The human sensitive frequency 4–8 Hz has larger effect on the driving comfort. The vibration isolation effect of the suspension system displays the frequency range of 18 Hz. In the whole frequency range, the vertical vibration contributes most. The inherent frequency and vibration type can be obtained by the data analysis.

25.3.2 Modality Decoupling Calculation and Optimization

The rigidity matrix [7] can be obtained from the model parameters:

$$[K_{\text{com}}] = \sum_{i=1}^{n} [T_i]^T [B_i]^T [K_i][B_i][T_i] \tag{25.5}$$

where $[K_{\text{com}}]$is rigidity matrix, $[T_i]$ displacement matrix, and $[B_i]$ angle matrix.

Then, the maximum kinetic energy of main vibration in stage i of the system is

$$T_{\max}^{(i)} = \frac{1}{2}\omega_i^2 \{\psi_i\}^T [M]\{\psi_i\} \tag{25.6}$$

where ω_i is the modality frequency of the system and $\{\psi_i\}$ modality matrix vector. The percentage of the kinetic energy distributed to general coordinate k out of the total kinetic energy of the system will be

$$T_p = \frac{T_k^{(i)}}{T_{\max}^{(i)}} = \frac{\frac{1}{2}\omega_i^2 \sum_{l=1}^{6} \{\psi_i\}_l \{\psi_i\}_k m_{kl}}{\frac{1}{2}\omega_i^2 \sum_{l=1}^{6} \sum_{k=1}^{6} \{\psi_i\}_l \{\psi_i\}_k m_{kl}} \tag{25.7}$$

The energy distribution diagram is obtained.

The energy distribution diagram is obtained after determining the vibration decoupling by adjusting suspension rigidity, angle, and position, as indicated in Fig. 25.4:

To make simulation on vibration transfer ratio, we can obtain (Fig. 25.5).

Table 25.2 shows the comparison of the inherent frequency of the vibration isolation system before and after the optimization. The inherent frequency at the highest stage reduces from 13.76 to 12.60 after the optimization, and the inherent frequency of the first stage increases from 3.34 to 3.51. It means that the modality frequency at the highest stage reduces, while the modality frequency at the lowest stage increases, thus reducing the width of the inherent frequency of the system resonance.

To make further investigation on the vibration isolation effect of the suspension unit in the cab after the optimization, the whole vehicle simulation model including the independent cab is created. The straight running simulation is made

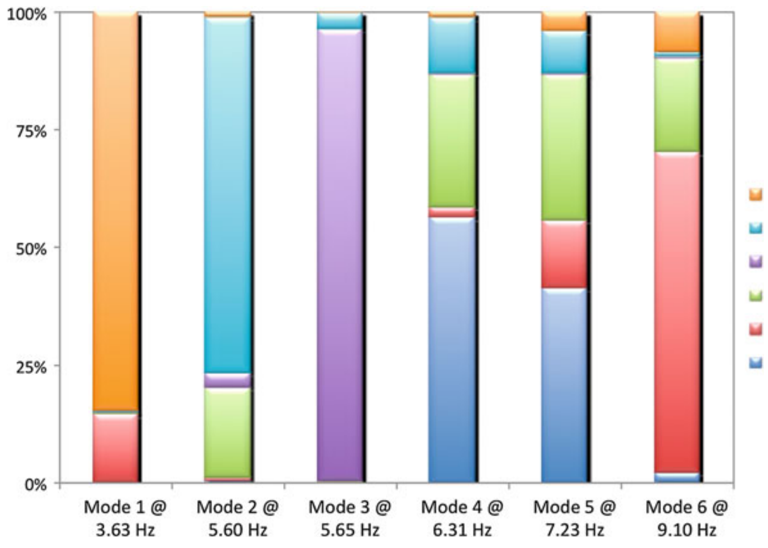

Fig. 25.4 The chart of energy distribution

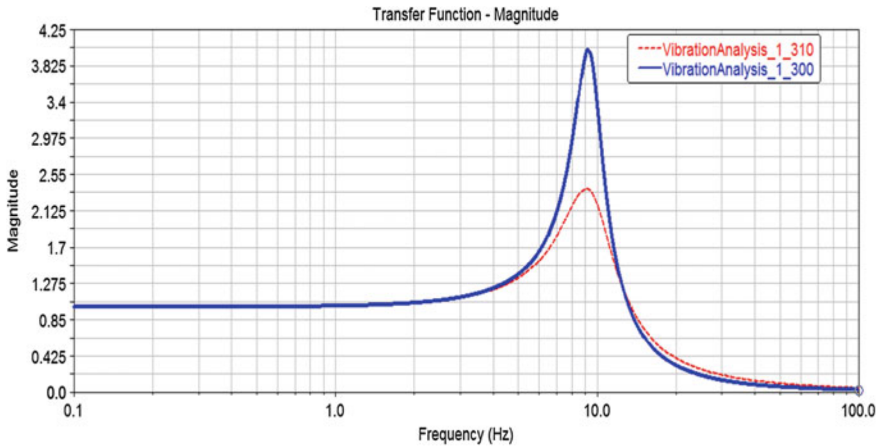

Fig. 25.5 The contrast of transfer ratio curve before and after the optimization

Table 25.2 The contrast of inherence frequency before and after the optimization

Stage	1	2	3	4	5	6
Before optimization	6.2	10.3	10.6	11.0	16.0	17.3
After optimization	6.3	10.5	10.8	11.1	15.7	17.0

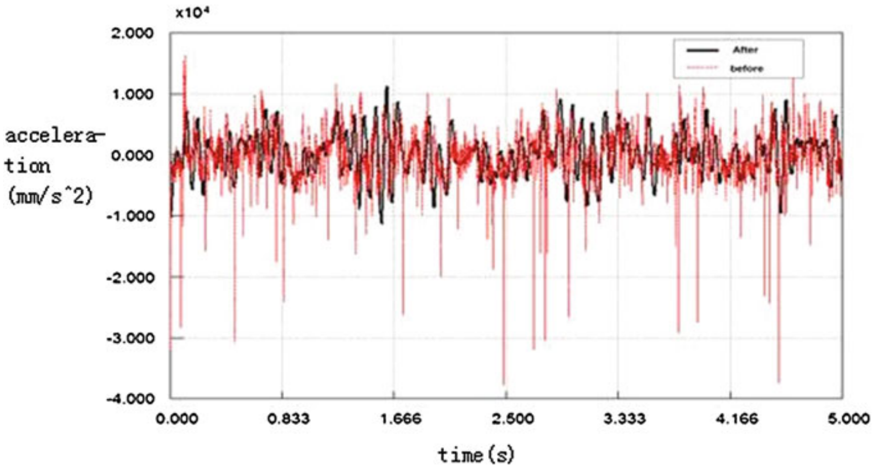

Fig. 25.6 The acceleration curve of upright direction in driver place

in the time domain at the speed of 50 km/h on the road of level E. The simulation calculation results are shown in Fig. 25.6. The acceleration RMS value reduces from 3.8 m/s^2 to 3.2 m/s^2.

25.4 Conclusion

In the thesis, a certain engineering vehicle is taken as an example to create independent cab structure model, cab dynamic model, and whole vehicle dynamic model. The simulation optimization is made on the basis of the vibration decoupling theory. The best state of the suspension rigidity of the cab is decided after the optimization calculation of the suspension rigidity and damping ratio matching. The comparison and simulation experiment show that it improves the comfort of the operator of special vehicles and also improves NVH performance of the whole cab.

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