Design and Simulation of Vehicle Cab Noise Reduction based on Symplectic Geometry Method

Ziyu Zheng[1, a]
[1]Shenzhen Senior High School, China

Keywords: Symplectic Geometry; Automobile Cab; Acoustics; Noise Reduction.

Abstract: The noise caused by the vibration of the car body will reduce the comfort of the car. In order to reduce the indoor noise, through the experimental modal analysis of the cab, the change of natural frequency and basic mode of vibration is obtained. It is found that when the frequency is 120.44 Hz, the cab noise has the greatest impact. Through symplectic geometric analysis and panel acoustic contribution analysis, it is found that the cab panel has the greatest influence on the sound pressure near the ear of the roof driver. In order to solve the above problems, the noise reduction method of the cab roof using stiffened panels and the sound radiation theory is used to simulate the noise reduction effect of the improved cab. It is found that the sound pressure of the improved cab is significantly reduced, the low-frequency noise is effectively suppressed, and the comfort of the car is improved.

1 INTRODUCTION

With the increasing demand for car ride comfort, more and more attention has been paid to the impact of body panel vibration on the human body. The research on the vibration level of cab panel has penetrated into the whole vehicle development process, so it is important to analyze and control cab vibration and noise. From 1966 GLADWELL et al. (G ML Gladwell, G Zimmermann, 1966) for the first time, the finite element technology has been applied to the field of acoustics. Today, the finite element technology has been more and more applied to the prediction and analysis of low-frequency structural vibration and noise. In recent years, scholars at home and abroad have made a lot of efforts in perfecting and applying the finite element boundary element technology in vehicle vibration and noise. So far, the basic research method of reducing vibration and noise of automobile cab is to use three-dimensional software to model and then use finite element software to calculate theoretical modal and compare with experimental modal. Finally, it is imported into the acoustic software for acoustic analysis. This method is more complex, and the software interface is a difficult problem. In this paper, the natural frequencies and modes of cab vibration are obtained directly through modal test analysis. Based on the thin plate theory, the symplectic geometry method of Hamilton system is used to calculate the natural frequencies of cab vibration accurately. Compared with the frequencies obtained from the test analysis, the more influential panels are analyzed, and the acoustic analysis is carried out directly in the acoustic software, and the noise reduction measures are put forward, so as to improve the cab structure design.

2 MODAL TEST ANALYSIS OF AUTOMOBILE CAB

The modal test is an effective method for identifying the dynamic characteristics of engineering structures by experimental methods (Shilei, 2012). Its main task is to measure the input and output signals of the system at the same time and to process them digitally, so as to estimate the frequency response function or impulse response function of the system under test, so as to provide an accurate and reliable basis for modal analysis. The single-point excitation analysis method is mainly used in the cab modal test of this automobile. The single-point excitation modal vibration testing system is mainly composed of three parts: excitation part (including signal generator, power amplifier and exciter), sensing part

Zheng, Z.
Design and Simulation of Vehicle Cab Noise Reduction based on Symplectic Geometry Method.
DOI: 10.5220/0008870401020107
In *Proceedings of 5th International Conference on Vehicle, Mechanical and Electrical Engineering (ICVMEE 2019)*, pages 102-107
ISBN: 978-989-758-412-1

(including sensor and adjustable amplifier and related connections) and analysis part (including analysis system and peripheral equipment such as plotter and printer) (Fu Zhifang, 1990). The block diagram of the test system is shown in Figure 1.

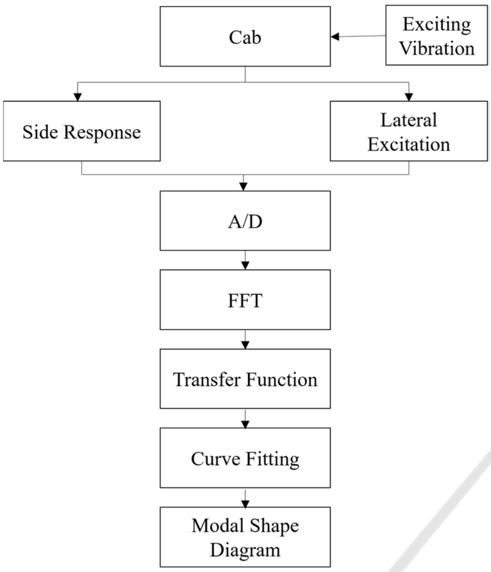


Fig 1. Block diagram of test system.

When the modal test is carried out, the exciting force is generated by the exciting equipment, which causes cab vibration. Then the force signal and the response signal measured by the acceleration sensor mounted on the cab are amplified and transmitted to the structural dynamic analyzer for FFT analysis, and the frequency response function is obtained. The average frequency response function is obtained by fitting the whole frequency response function, and then the modal parameters are identified, from which the experimental modal parameters are obtained. The first ten vibration modes and mode characteristics of the cab are calculated by the software, as shown in Table 1.

From the test data measured in Table 1, it can be seen that the frequency of the cab roof vibration is relatively frequent, and the left and right panels and the rear panels also have local vibration. Therefore, it is preliminarily determined that roof vibration is the main cause of the noise. From the modal analysis, it can be seen that the roof vibration is the largest when the modal frequency is the seventh order of 120.44Hz. The mode shapes of the cab are shown in Fig. 2, which show the obvious vertical vibration of the cab roof, accompanied by the overall rotation of the cab around the Z axis. If we want to improve the interior noise of the cab, the most effective way is to reduce the vibration response of the roof.

3 SYMPLECTIC GEOMETRY METHOD FOR ANALYZING ROOF NATURAL FREQUENCY

The model of the cab is shown in Figure 3. The cab is considered to consist of 10 thin-walled panels of different sizes. Cab roof vibration is the largest, so it is taken as the research object. Because the cab is fixed on the surrounding panel by welding and screw connection, the panel is set as a rectangular thin plate, the boundary condition of the four sides is adopted, and the structure is set as full restraint.

Table 1. The first ten modes and modal characteristics of cab structure measured.

Modal order	Natural Frequency(Hz)	Modal mode characteristics
1	21.186	Slight vibration of roof
2	29.325	Local Vibration of Rear Wall Plate
3	37.508	Roof Vibration, Transverse Local Vibration of Left Wall
4	66.085	Local Vibration of Rear and Right Wallboard
5	75.341	Roof Vibration, Right Wall Transverse Local Vibration
6	113.534	Roof Vibration, Local Vibration of Left and Right Wallboard
7	120.44	Vertical vibration of roof
8	167.055	Vertical vibration of roof and torsional vibration of back wall
9	183.892	Vibration of rear panel and left and right panel
10	195.45	Left and right lateral vibration

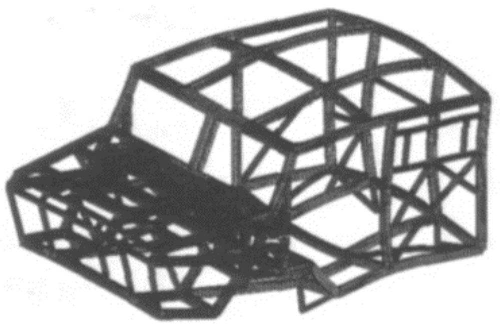


Fig 2. The seventh mode shapes of the cab.

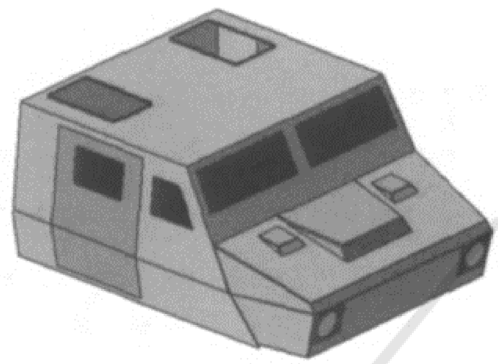


Figure 3. Cab structure.

The basic equation of free vibration for the cab roof is established:

$$D^4W-\rho\omega^2W=0 \qquad (1)$$

Where W is the mode function of the roof; D is the flexural strength; ρ is the mass of the unit area of the roof; ω is the natural frequency of the roof.

The relationship between roof bending moment and torsion is as follows:

$$\begin{cases}\frac{\partial M_x}{\partial x}+\frac{\partial M_{xy}}{\partial y}-Q_x=0\\ \frac{\partial M_y}{\partial y}+\frac{\partial M_{xy}}{\partial x}-Q_y=0\\ \frac{\partial Q_x}{\partial x}+\frac{\partial Q_y}{\partial y}+\rho\omega^2W=0\end{cases} \qquad (2)$$

Where, M_x, M_y and M_{xy} are the bending moment of the roof, Q_x and Q_y are the torsion of the roof, and V_x and V_x are the shear force and total shear force. After analysis and transformation, the relationship among the parameters can be expressed by Hamilton equation (Zhong Yang, Li Rui, Tian Bin, 2011), which can be written in the form of matrix:

$$\frac{\partial Z}{\partial y}=\begin{bmatrix}F & G\\ Q & -F^T\end{bmatrix}Z \qquad (3)$$

Where, F= $\begin{bmatrix}0 & 1\\ -\nu\partial^2/\partial x^2 & 0\end{bmatrix}$, G= $\begin{bmatrix}0 & 1\\ 0 & -1/D\end{bmatrix}$, Q= $\begin{bmatrix}-\frac{(\nu^2-1)\,\partial^4}{\partial x^4}+\rho\omega^2 & 0\\ 0 & 2D(1-\nu)\,\partial^2/\partial x^2\end{bmatrix}$, Z is the state vector of a thin plate.

The symplectic geometry method is used to analyse the vibration mode function. The formula (3) is solved by the method of separating variables. The solutions of the vibration mode function in the X and Y directions are as follows:

$$\begin{cases}W(x)=A_1\cos\left(\sqrt{R-\beta^2x}\right)+B_1\sin\left(\sqrt{R-\beta^2x}\right)+\\ \quad C_1\cosh\left(\sqrt{R+\beta^2x}\right)+F_1\sinh\left(\sqrt{R+\beta^2x}\right)\\ W(x)=A_2\cos(\beta\gamma)+B_1\sin(\beta\gamma)+\\ \quad C_2\cosh\left(\sqrt{2R-\beta^2y}\right)+F_2\sinh\left(\sqrt{2R-\beta^2x}\right)\end{cases} \qquad (4)$$

Where, A_1, A_2, B_1, B_2, C_1, C_2, F_1 and F_2 are undetermined constants.

The boundary conditions in the X and Y directions of the clamped thin plates are as follows:

$$\begin{cases}W(x)\,|_{x=\pm a}=\frac{\partial W_x}{\partial x}\,|_{x=\pm a}=0\\ W(y)\,|_{x=\pm b}=\frac{\partial W_y}{\partial y}\,|_{x=\pm b}=0\end{cases} \qquad (5)$$

The partial derivative substitution formula (5) of equation (4) is obtained, and the equation about coefficient is obtained. The determinant of coefficient matrix is zero. The eigenvalue transcendental equation about X and Y axisymmetry is obtained as follows:

$$\begin{cases}\sqrt{R-\beta^2}\tan\left(a\sqrt{R-\beta^2}\right)+\sqrt{R+\beta^2}\tanh\left(a\sqrt{R+\beta^2}\right)=0\\ \beta\tan(b\beta)+\sqrt{2R-\beta^2}\tanh\left(b\sqrt{2R-\beta^2}\right)=0\end{cases} \qquad (6)$$

The values of R and β can be obtained from equation (6). From R=$\omega\sqrt{\rho/D}$, the frequency of rectangular thin plate with four edges fixed can be further obtained.

The sound pressure on the roof surface can be expressed by Helmholtz integral equation (Yang Xiaowei, et al, 2009):

$$p(R)=-\frac{1}{2\pi}\int p(R_0)\frac{e^{ikr}}{r}\left(\frac{1}{r}-ik\right)\cos\beta\, dS(R_0)-\frac{i\omega\rho_0}{2\pi}\int v(R_0)\frac{e^{ikr}}{r}dS(R_0) \qquad (7)$$

Among them, p is the sound pressure of the structure surface, ν is the normal velocity of the structure surface, r=$|R-R_0|$ is the distance between two points on the structure surface, and S is the surface of the vibration structure.

The elastic modulus of the cab panel is 206 GPa, Poisson's ratio is 0.3, the strength limit is 450 MPa, the thickness of the thin panel is 7850 kg/m3, the thickness is 5 mm, the length and width of the roof are 2510 mm and 1263 mm, respectively. By substituting the parameters of cab panel (6), the natural frequency of the roof can be calculated to be 121.42Hz, which is basically consistent with the model test value. Thus, it can be judged that the analytical solution of the symplectic geometry method is correct and applicable.

4 CAB ACOUSTIC CHARACTERISTIC CALCULATION

The acoustic characteristics of the cab are analyzed in the acoustic software, and the sound pressure level curve near the driver's ear is obtained, as shown in Figure 4. It can be seen from the curve that the peak sound pressure appears at frequencies of 18 Hz, 40 Hz, 64 Hz, 70 Hz, 114 Hz, 120 Hz, 129 Hz, 170 Hz, 180 Hz and 195 Hz. The results are basically consistent with the frequencies of cab modal test analysis. Further modal participation factor analysis is carried out for the cab (Li Zenggang, 2005). The so-called modal participation factor refers to the degree to which each order of modal participates in the dynamic response of the structure when the modal superposition method is used to calculate the dynamic response. From the analysis of the partially modal participation factor curve shown in Fig. 5, it can be seen. At frequencies of 47 Hz, 120 Hz and 195Hz, the effects of the last five frequencies on the sound pressure near the driver's ear are greater than those of the first five frequencies. The modal participation factor corresponding to the seventh order is always at the maximum, and its corresponding frequency is 120 Hz, which coincides with the natural frequency value of the roof vibration analyzed above. Therefore, it can be accurately judged that roof vibration is the main source of cab noise.

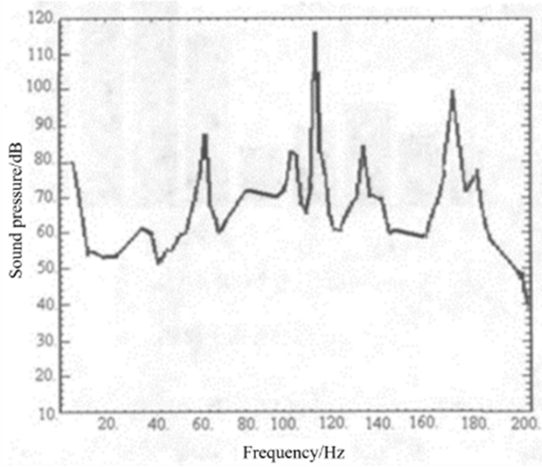
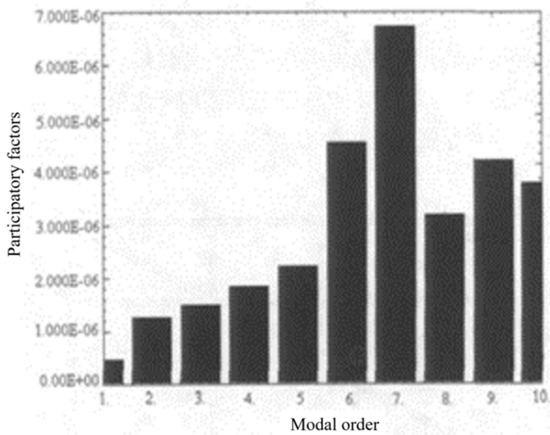


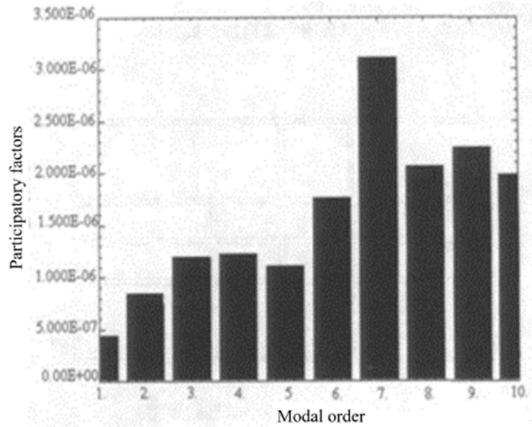
Fig 4. The curve of sound pressure level near the driver's ear.

5 CAB SIMULATION ANALYSIS AND IMPROVEMENT

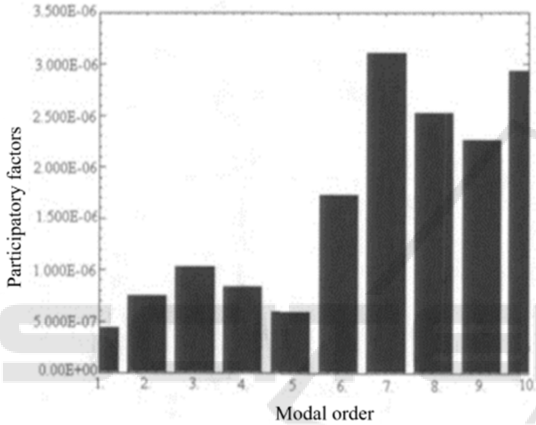
According to the above analysis, the cab structure is improved and designed. Because the vibration of roof is the main cause of the noise, the structural improvement of roof is mainly carried out in the redesign, mainly through the following two ways: first, the surface of cab panel is damped to reduce the indoor noise; second, the structure improvement of cab panel is carried out. To improve its stiffness and reduce the interior noise. Because the vehicle belongs to the engineering field, the strength of the cab is required to be high (Chen Shuming, et al, 2012). Considering the economic point of view and practical aspects, this paper adopts the method of improving the stiffness of the cab roof to reduce vibration and noise.



(a) Modal Participation Factor Curve at 47Hz



(b) Modal Participation Factor Curve at 120Hz



(c) Modal Participation Factor Curve at 195Hz

Fig 5. Modal Participation Factor Curve.

There are two ways to improve the stiffness of the cab panel. The first is to increase the thickness of the wallboard; the second is to reinforce the wallboard (Wang Xianyi, et al, 2008). Thickening of cab roof will increase the dead weight of cab, and the material is not fully utilized. Reinforcement of roof under certain conditions can increase the dead weight of cab roofless and improve the performance. In this paper, two kinds of reinforcement methods are compared. One is the four corner reinforcement of the roof, as shown in Figure 6, and the other is the reinforcement through the center of the roof, as shown in Figure 7.

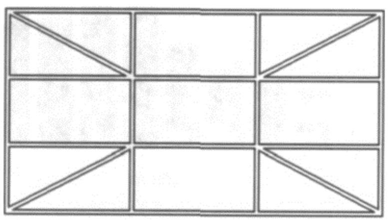


Fig 6. Cross-sectional diagrams of four corner stiffeners on the roof.

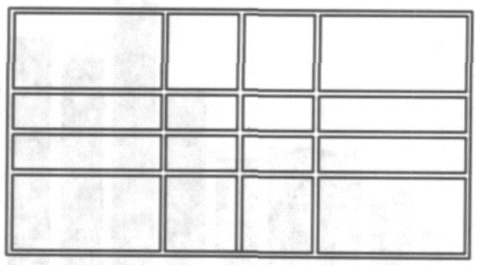


Fig 7. Cross-sectional diagrams of stiffened roof through the center of roof.

Acoustic analysis is carried out for the two kinds of reinforcement arrangement. The sound pressure level curve is shown in Fig. 8. The calculation results of sound pressure near the driver's ear are shown in Table 2.

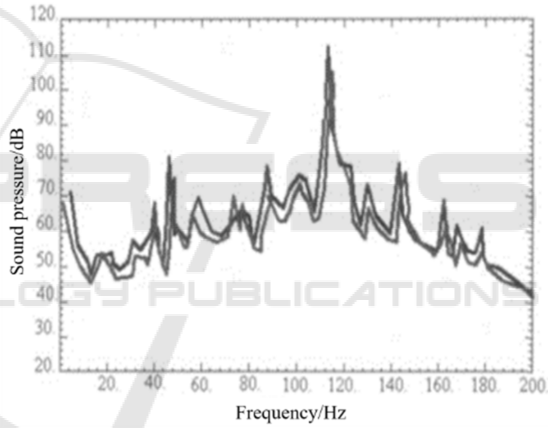


Figure 8. Sound pressure curves using stiffened panels for noise reduction.

Table 2. Comparison of driver's ear sound pressure before and after reinforcement.

Modal order	1	2	3	4	5
No reinforcement /dB	53.2	54.1	58.9	72.3	61.8
Corner stiffening /dB	51.3	52.7	55.5	68.6	58.0
Central reinforcement /dB	49.8	51.4	53.2	60.2	53.9
Modal order	6	7	8	9	10
No reinforcement /dB	75.3	118.7	55.6	59.2	48.9
Corner stiffening /dB	70.4	114.8	52.7	55.1	48.2

Central reinforcement /dB	73.1	108.4	51.9	50.4	49.1

The black line and the gray line in Figure 8 show the sound pressure curves of the cab roof reinforced by four corner stiffeners and through the center stiffeners, respectively. It can be seen from the curve that the sound pressure of cab roof decreases after stiffening, and the noise reduction effect of stiffeners applied through the center of the roof is better than that of four corner stiffeners. For the measurement of sound pressure, 387 nodes near the center of the cab roof are selected as the test points. The peak value of the sound pressure curve is basically consistent with the fixed frequency value measured in front. The maximum value appears at 120 Hz, which is consistent with the maximum vibration of the roof analyzed in front when the natural frequency is 120.44 Hz. The sound pressure value decreases from 118.7 dB to 114.8 dB by applying ribs at the edge corners, while the sound pressure value decreases to 108.4 dB by applying ribs through the center of the roof, and the decrease of sound pressure in the latter arrangement is greater than that in the former arrangement at other peak points. This is because the sound pressure is gradually divergent from the central part of the roof. Therefore, the cab roof is eventually laid out through the center of the roof with reinforcement bars. By improving the cab roof design, the cab's internal noise has been significantly reduced, which not only improves the comfort of passengers but also improves the driver's response ability, so that the car can be better applied.

6 CONCLUSIONS

In this paper, the cab of an engineering vehicle is taken as the research object, and the natural frequency and mode of cab vibration are preliminarily determined by means of the experimental modal analysis method. It is concluded that the vibration of the cab roof is the main cause of indoor noise. The symplectic geometry method of Hamilton system is used to analyze the natural frequency of cab noise, which is consistent with the experimental data. It provides a theoretical basis for the future study of vehicle interior vibration. In the application process, the frequency value can be resolved by directly substituting the relevant data of the research object into the formula, without modeling and calculation, which saves time and improves work efficiency. The acoustic characteristics of the cab are analyzed, and the method of panel contribution is used to further prove that the roof vibration contributes the most to the indoor noise. The sound pressure near the driver's ear is calculated, and the seventh mode has the greatest influence. Two methods are put forward to improve the cab structure. The noise level near the driver's ear is obviously reduced by stiffening through the center of the roof, which effectively reduces the noise in the cab as a whole.

REFERENCES

G ML Gladwell, G Zimmermann. On energy and complementary energy formulations of acoustic and structural vibration problems [J].Journal of Sound and Vibration, 1966, (3): 233-241.

Shilei. Application of modal test analysis to solving vehicle vibration problems [J].Automotive Science and Technology, 2012, (5): 22-26.

Fu Zhifang. Vibration modal analysis and parameter identification [M]. Beijing: Machinery Industry Press, 1990.

Zhong Yang, Li Rui, Tian Bin. Hamiltonian analytical solution for free vibration of rectangular thin plates with fixed edges [J]. Journal of Applied Mechanics, 2011, 28 (4): 323-327.

Yang Xiaowei, et al. [J].Journal of Mechanical Engineering, 2009, 45 (8): 221-227.

Li Zenggang.SYSNOESE Rev5.6 [M]. Beijing: National Defense Industry Press, 2005.

Chen Shuming, et al. [J].Computer simulation, 2012 (4): 287-291.

Wang Xianyi, et al. [J].Journal of Vibration Engineering, 2008, (2): 13-17.