Analysis and Improvement of Acoustic Contribution of Tractor Cab

Kai Zhu[a], Xintan Ma, Weiwei Xu

Henan University of Science and Technology Vehicle & Transportation Engineering Institute of Hust Luoyang, China

[a]694140648@qq.com

Keywords: Cab, Boundary Element, Plate Acoustic Contribution, Modal Contribution

Abstract: In order to study the influence of structural low-frequency vibration on the driver's ear noise, the combination of plate acoustic contribution and modal acoustic contribution was used to overcome the shortcomings of the plate acoustic contribution method which is easy to generate new sound pressure peak. By establishing the tractor cab boundary element model, the acoustic transfer vector and modal acoustic transfer vector analysis were combined with the panel and modal shape analysis with the largest contribution of the driver's ear position sound pressure peak to the right door and front block of the cab. The wind is thickened, and the maximum position of the right baffle vibration is reinforced, which effectively reduces the total sound pressure at the driver's ear and does not generate a new sound pressure peak, thereby improving the cab noise level.

1. Introduction

The tractor travels slowly during the working process. (1) The main source of noise at the driver's ear is the low-frequency vibration noise of the 20-200Hz plate. For the vibration and noise characteristics of the plate, (2-3) domestic and foreign scholars use the theory and test methods to find out the plate that contributes to the sound pressure of the field point, and correlate the vibration of the plate with the sound pressure level of the field; (4-5) Use the plate contribution and test method to find out the plate that contributes the most to the acoustical point of the field and improve it; (6-7) modal contribution was used for a reasonable layout method for improving the structure of the engine and the reduction gearbox by using. (8) Research shows that the box radiation noise can be effectively reduced by finding the plate that contributes the most to the sound pressure of the field and reducing the local vibration mode. However, the acoustic contribution of the plate is a vector. The contribution and phase of a plate are different at different frequencies. (9) If the plate is simply modified, other peak noise peaks may be generated while reducing the peak noise.

In this paper, the acoustic contribution of the plate is combined with the modal contribution, and the acoustical-solid coupling model vibration mode is used as the boundary condition to find the plate and mode shape that contributes the most to the cab noise peak. Reinforced ribs are processed by corresponding vibration modes to overcome the shortcomings of plate acoustic contribution method.

2. Basic method

2.1 Acoustic transfer vector and plate contribution method

When the air disturbance pressure is small, the acoustic transfer equation can be considered linear, and the field sound pressure is:

$$P=\{ATV(\omega)^T\{V_n(\omega)\} \tag{1}$$

In the formula, ATV(ω) is an acoustic transfer vector, and the vibration of the structure is related to the sound pressure of the field point by the above formula. ATV is mainly related to structural characteristics, analysis frequency and position of field points, and is an inherent characteristic of structural surface vibration and field point sound pressure. The plate acoustic contribution can be seen as the contribution of the set of n elements to the sound pressure of the field. Different units have different acoustic contributions to the same field. The acoustic contribution of the plate is the vector sum of the acoustic contributions of all the elements of each plate to the field. Therefore, the acoustic contribution amplitude and phase of the different plates for the field are different.

$$P_c = \sum_{i=1}^{n} \{ATV(\omega)^T \{V_n(\omega)\} \tag{2}$$

Acoustic contribution coefficient is:

$$D_c = \mathrm{Re}(\frac{pp*}{|p|}) \tag{3}$$

Where P is the sound pressure at the field point, $p*$ is the conjugate of the sound pressure at the field point. When the D_c is positive, the sound pressure phase generated by the plate is the same as the total sound pressure, and the total sound pressure increases as the vibration of the plate increases; When D_c is negative, the phase is opposite, and the total sound pressure decreases as the vibration of the plate increases. Therefore, the control of the noise is to suppress the vibration of the plate member whose positive contribution coefficient is positive, and to enhance the vibration of the plate member having a negative contribution coefficient.

2.2 Modal Acoustic Transfer Vector and Modal Contribution Method

The linear superposition of modal modes can obtain the structural vibration displacement response:

$$\{x(\omega)\} = \sum_{i=1}^{n} (\varphi)_i \{MRSPS(\omega)_i\} \tag{4}$$

$\{MRSP(\omega)_i\}$ is the i-th modal participation factor vector.

Normal vibration velocity:

$$\{V_n(\omega)\} = j\omega \sum_{i=1}^{n} (\varphi)_i \{MRSPS(\omega)_i\} \tag{5}$$

$(\varphi)_{ni}$ is the component of each mode shape in the normal direction.

The sound pressure at the field is:

$$P(\omega) = \{ATV(\omega)\}^T \sum_{i=1}^{n} (\varphi)_{ni}\, j\omega \{MRSP(\omega)\} = \{MATV(\omega)\}^T \{MRSP(\omega)\} = \sum_{i=1}^{n} P_{si} \tag{6}$$

MATV is a modal acoustic transfer vector.

P_{si} represents the sound pressure generated by the ith-order structural mode, the percentage of P_{si} in the total sound pressure $P(\omega)$ is called the modal contribution, so it's a vector with size and orientation. The positive contribution is consistent with the total sound pressure projection direction, and the negative contribution is opposite to the total sound pressure projection direction. It associates structural modal vibration with field point sound pressure. The modal contribution order method is used to obtain the structural modal order that contributes the most to the peak voltage of the sound pressure. The analysis process is to introduce the structural vibration of the coupled model into the acoustic boundary element model as the boundary condition, and calculate the modal acoustic transfer vector MATV, and then obtain the contribution of each modal order in the acoustic response.

3. Cab acoustic-solid coupling model

The cab is composed of sheet metal parts and doors and windows, and its vibration is the main source of noise in the cab. Considering the vibration of the structure and air, the structure interacts with the acoustic cavity. By establishing the acoustic-solid coupling model, the vibration mode of the structure can be truly reflected, and the boundary conditions of the acoustic response and contribution of the vehicle are analyzed below.

The structural finite element model uses the SHELL63 shell element, the grid size is set to 30mm, the material is mainly steel and glass, and the connection between the door and window and the structure is simplified to a rigid connection. (9) The acoustic cavity finite element model mesh size is set to 60mm. The coupling plane between the structural finite element and the acoustic cavity finite element model is defined, and the acoustic-solid coupling model of the cab is established as showed in the following figure.

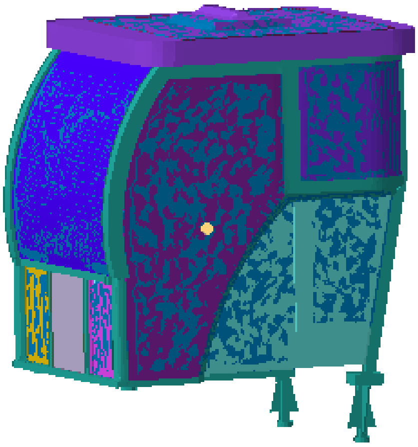


Figure 1. Cab acoustic-solid coupling model

The acoustic-solid coupling model was established as the research object to make the model more accurate considering the influence of sound cavity on structural vibration. The normal vibration velocity of cab plate was obtained, which provided boundary conditions for the study of acoustic frequency response and contribution analysis in the cockpit.

4. Cab acoustic response analysis

In the virtual lab grid division module, the finite element surface mesh of the structure is used to establish the boundary element model, as shown in figure 2.Since the boundary element grid is directly generated by the surface grid of structural finite element, the shell element and node number are not changed, so the consistency of boundary element grid and finite element grid is maintained, and no conflict will occur in the data mapping. In order to reduce the calculation amount, only the position of the driver's right ear is set at the field point with coordinates (-750, 1250, 1200).

Figure 2. Cab boundary element model

The vibration response analysis of the model is carried out, and the Z-direction unit excitation is placed on the four suspension points of the coupled model. The frequency calculation range is 20-200 Hz, and the step size is 2 Hz [1]. The vibration response of the coupling model of the cab is calculated as the boundary condition, and the frequency response analysis is performed on the boundary element grid. The sound pressure curve of the driver's right ear is showed in Figure. 3.

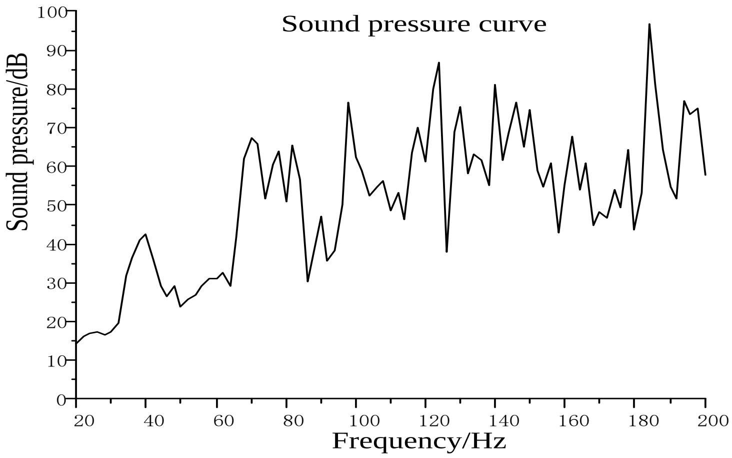
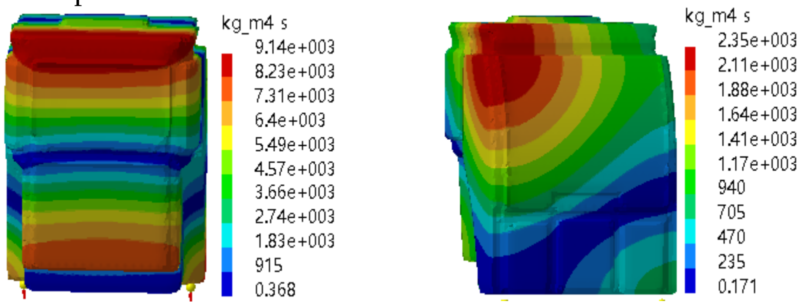


Figure 3. Driver's right ear sound pressure frequency response curve

As can be seen from the above figure, there are multiple sound pressure level peaks in the driver's right ear noise in the low frequency range, where the maximum position reaches 96.64 dB(A) at 184 Hz and the second peak reaches 86.71 dB at 124 Hz (A). It is analyzed that at these two frequencies, the cab structure and the natural frequency of the acoustic cavity are close, which is easy to generate resonance, resulting in the generation of peak sound pressure level. Therefore, 184 Hz and 124 Hz are used as the contribution analysis objects.

Calculate the driver's right ear peak frequency acoustic transmission vector ATV, 184 Hz and 124 Hz right ear ATV cloud image as showed in Figure 4. From the cloud image, the driver's right ear is shown in a higher sound pressure environment.



(a)184Hz ATV cloud imag (b)124Hz ATVcloud image

Figure 4. Driver right ear ATV cloud map

5. Cab acoustic contribution analysis

5.1 Plate acoustic contribution

The boundary element model of Figure 2 is divided into panels, and the entire cab is divided into left door-1, left baffle-2, ceiling-3, left window-4, rear window-5, tailgate-6, right window- 7. Right baffle-8, right door-9, windshield-10, front right window-11, front left window-12, instrument panel-13, front floor-14, rear floor-15, total 15 plates.

According to the above ATV calculation, the acoustic contribution of the plate at the two peak frequencies of the driver's right ear is shown in Figure 5 and Figure 6.

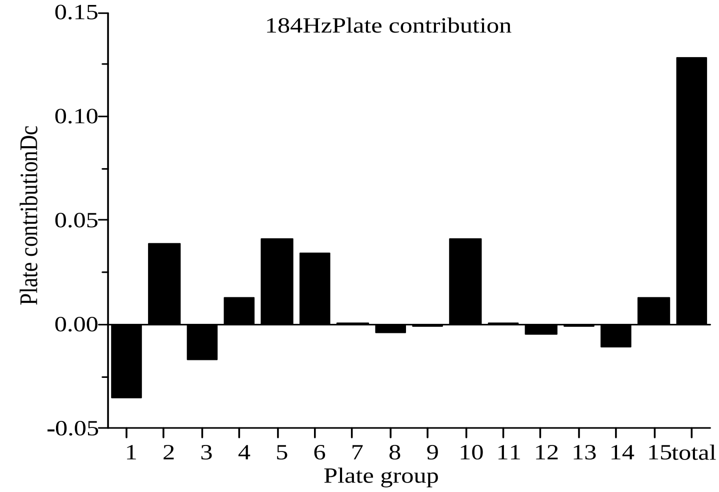


Figure 5. 184 Hz board contribution

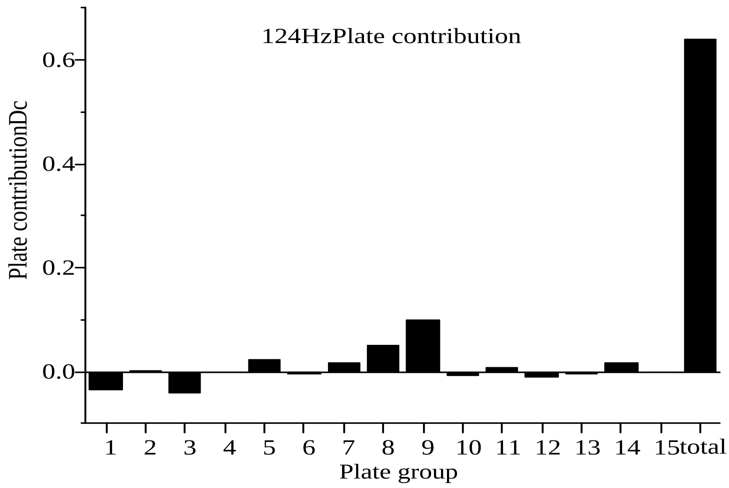


Figure 6. 124 Hz board contribution

From the results of the plate contribution, it can be seen that the plates that contribute the most to the driver's ear sound pressure at the frequency of 184 Hz are the windshield and the rear window respectively; the 124 Hz frequency is the right door and the right baffle. The contribution of the four plates is the same as the total contribution and is positive, indicating that the driver's ear sound pressure increases with the vibration of the four plates at the peak frequency, which is the object of research.

5.2 Modal acoustic contribution

The modal transfer vector MATV calculation is performed on the cab boundary element model, and the two peak frequency modal contributions in the sound pressure frequency response curve are obtained. As shown in Figure 7 and Figure 8.

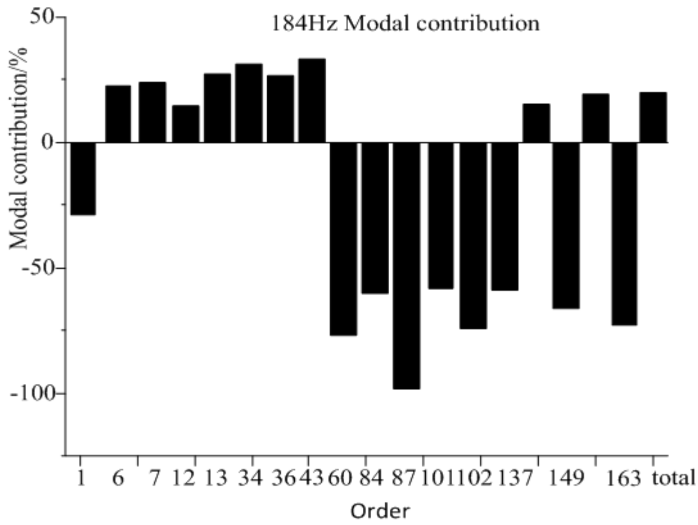


Figure 7. 184 Hz modal contribution

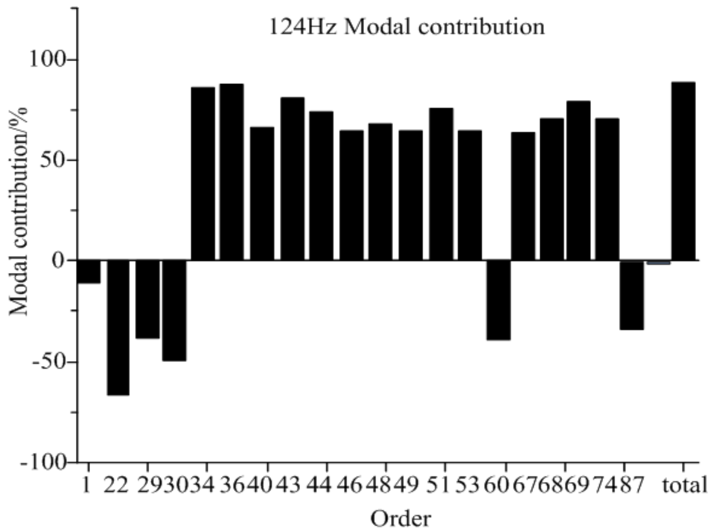


Figure 8. 124 Hz modal contribution

From the modal contribution graph, the modal order that contributes a lot to the acoustic response can be obtained. The order with the largest contribution at the frequency of 184 Hz is 43 orders, 34 orders, 13 steps; the order with the largest contribution at 184 Hz is 36th order, 34th order, 43th order, 69th order. The mode shape can directly reflect the structural vibration, and the mode shape indicated by the order of large modal contribution is the object of subsequent improvement. Table 1 show these order vibrating plates. It can be clearly seen from the following figure that in the order with the largest contribution, the vibration modes are mainly concentrated on the right bezel, the right door, the front windshield and the rear window, and the suppression of these modes can effectively reduce the sound of the field Pressure.

Table.1. Modal shape

Modal Order	**Mode Shape**
34	Right baffle-8
36	Right baffle-8
42	Left doors-1, right doors-9
43	right doors-9
69	Right baffle-8, front windshield-10

6. Improved sound pressure in the cab

Comprehensive analysis of plate contribution and modal contribution results, combined with the largest contribution of the plate and mode shape, the cab right door, right baffle and front windshield is the object of structural improvement. The low-frequency structure noise is mainly caused by the vibration of the plate member, so the vibration of the plate member is reduced by increasing the stiffness and the natural frequency of the plate member, thereby reducing its contribution to the interior noise. Consider that the right door and the front windshield belong to the glass material and are thickened by 1 mm. In the area where the right baffle mode vibration mode is large, the rib reinforcement is added, as shown in Figure 9.

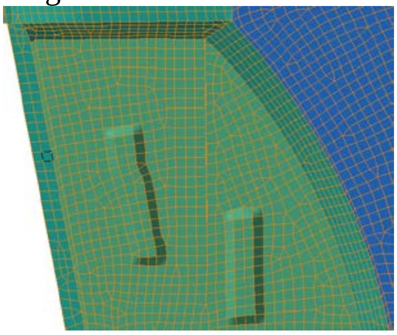


Figure 9. Right baffle rib treatment

For the improved cab, due to structural changes, the above acoustic transfer vector ATV is not applicable, recalculating the ATV and performing a sonic-induced coupling frequency response calculation. The noise response curve of the driver's ear before and after the structural improvement is shown in Fig.ure 10.

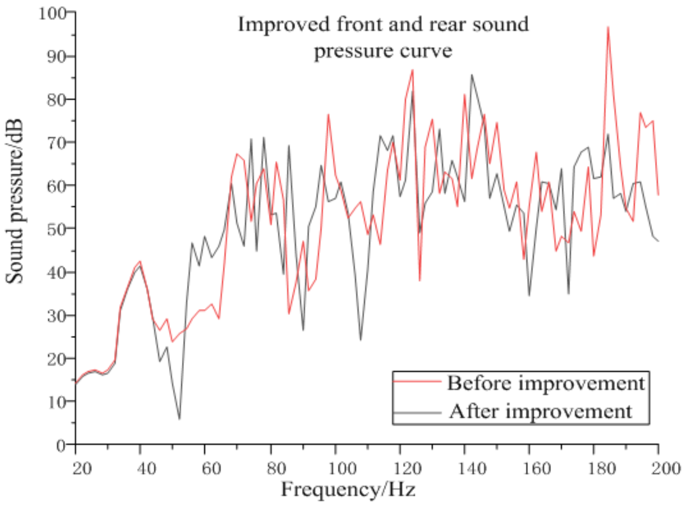


Figure 10. Improved sound pressure response curve before and after

It can be seen from the above figure that the driver's right ear sound pressure response curve is significantly reduced after the structural improvement, from 96.6 dB(A) to 72.0 dB(A) at the peak frequency of 184 Hz; From 86.7 dB(A) drops to 82.0 dB(A) at 124 Hz. and does not produce a new apparent peak sound pressure. The overall sound pressure level is reduced by 5.4 dB(A) from 91.1 dB(A) to 85.7 dB(A), and the noise reduction effect is obvious.

7. Conclusion

(1) Using the acoustic transmission vector to simulate the low-frequency structure noise of the tractor cab, calculate the frequency response of the driver's right ear field point, obtain the driver's ear noise and obtain the plate with large contribution to the peak frequency noise. The modal mode

contribution is used to obtain the structural mode shape with large contribution to the peak frequency noise, which provides a basis for subsequent structural improvement.

(2) Combining the acoustic contribution of the plate and the analysis result of the modal contribution, the improvement of the applied rib is performed. This method can effectively overcome the shortcomings of the single board contribution improvement and easy to generate new peak noise. After the improvement, the noise at the peak frequency of the sound pressure is significantly reduced and no new sound pressure peak is generated, and the overall sound pressure level is also improved to improve the noise level of the cab.

References

[1] Bai Song, Xu Xinxi, et al. Analysis of interior noise in a cab and noise reduction with damping [J]. Noise and Vibration Control, 2012, 03: 139-142.

[2] Fouladi M H, Nor M J M, Ariffin A K. Spectral analysis methods for vehicle interior vibro-acoustics identification [J]. Mechanical Systems & Signal Processing, 2009, 23(2): 489-500.

[3] Wu S F, Natarajan L K. Panel acoustic contribution analysis [J], The Journal of the Acoustical Society ofAmerica, 2013, 133(2): 799-809.

[4] Chen Jian, Jiang Fengxin, Xiao Yue. Low frequency noise analysis and control of excavator cab [J]. China Mechanical Engineering, 2014, 25(15): 2124-2130.

[5] He Li, Zhang Liming. Modal contribution analysis of vehicle’s body based on static bench test [J]. Noise and Vibration Control, 2018, 38(4): 97-99.

[6] Song Wen, Liang Yue, Gao Aijun, et al. Vibration control of engine support structure based on modal contribution method [J]. Torpedo Technology, 2015, 23(6): 439-443.

[7] Wang Jinpeng, Chang Shan, Liu Geng, et al. Gearbox noise reduction based on model acoustic contribution [J]. Journal of Vibration and Shock, 2015, 34(17): 50-56.

[8] Zheng Xiaowen, Zhang Jinmei. Simulation for gearbox structure on vibration and noise of reducer [J]. Science Technology and Engineering, 2014, 14(12): 253-256.

[9] Zao Yuwei, ZhouHong. Bus body panel acoustic contribution analysis [J]. Automobile Technology, 2012(12): 30-33.