The Analysis of the Influence of Cab Structure to Noise Under Random Excitation

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ABSTRACT

The vibration and acoustic coupling FEA of light truck cab is established using the combining way of experiment and finite element simulation in this paper. The influence of cab structure to noise under random excitation is studied by acoustic plates contribution analysis. Extract a sound pressure level of the points beside driver’s right ear. Finally, the plates that contribution a lot to acoustic are respectively front windshield, the left door, left side window, the top cover, rear floor and the front floor which provides a reference for the improvement of the cab acoustic environment in the future.

KEYWORDS

 component; Light Truck Cab; Random excitation; Acoustic-structure Coupling; Finite Element Simulation

INTRODUCTION

One of the most important indicators to measure design and manufacture standards of modern vehicles is NVH(Noise、Vibration、Harshness).[1]

In recent years, the research of NVH characteristics for modern vehicles is mostly accomplished by computer simulation technology and experiment data accumulation.[2]

The main source of vibration and noise of light truck cab are respectively through by structure and air.[3] However, when the structure spread vibration, it will interact with acoustic cavity and then may enlarge structural distortion of coupling model. The noise spread by structural also will increase due to vibrate plate.

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Therefore, the research of light truck cab in this article uses vibration and acoustic coupling finite element model. We studied the influence of interior noise caused by vibration of cab structure under random excitation at low frequency range (200Hz below). For further study, we found out the plates that hold the greater impact on the structure noise of cab using acoustic plate contribution analysis, which providing reference for following structure optimization. All above has a guiding significance to design reasonable cab structures and improve the NVH characteristics for light truck cab.[4]

THE ACOUSTIC-STRUCTURE COUPLING FINITE ELEMENT MODEL OF CAB

The Establishment of Vibration and Acoustic Coupling Finite Element Model

In this paper, the cab finite element model was established basing on modular modeling idea. We built up the geometric model of each component in CATIA software at first and then imported them respectively into Hypermesh. After geometry cleaning-up, meshing and unit quality inspection, we assembled every subassembly structure finite element model. In the end, assembled the whole cab structure finite element model by solder joint and bolt.[5]

Pretreatment process was accomplished by the Altair’s Hyperworks 10.0 pretreatment software and then we used MSC. Nastran to solve the structural modal as well as compared with experimental modal. It can concluded that calculating modal have the same corresponding inherent frequency with experimental modal and the relative errors are within 10%. So the finite element model of cab we established is of high accuracy.

Removed plates that unnecessary or do less influence on acoustic analysis from cab structure finite element model, then filled the hole. The airtight internal cab space was formed.

Modeling from top to bottom again on the modified model, it could form acoustic finite element model.[6] Finally using LMS Virtual.Lab/Acoustic software to couple the structure finite element model and acoustic finite element model, regardless of the seat, the vibration and acoustic coupling finite element model of cab was set up as shown in figure 1. The envelope surface of acoustic space was the coupling surface of two finite element model.

![Figure 1. The vibration and acoustic coupling finite element model of cab (door removed).](image)

The Modal Analysis of the Vibration and Acoustic Coupling FEM

To solved the coupled system, modal superposition method was taken to superimpose structure modal and acoustic modal by LMS Virtual.Lab software.

The former five orders of coupling modal and the corresponding structure modal for cab is shown in table 1. It is also respectively listing the two modal inherent frequencies and the main deformation plates.
Table 1. The coupling modal of cab and the corresponding structure modal.

| Order | The coupling modal of cab | The structure modal of cab |
|-------|---------------------------|---------------------------|
|       | Frequency (Hz) | Main deformation plate | Frequency (Hz) | Main deformation plate |
| 1     | 21.41          | A column of both sides  | 21.46          | A column of both sides  |
| 2     | 30.85          | The central of top cover| 29.46          | The central of top cover|
| 3     | 33.66          | Top cover/rear wall     | 33.02          | Top cover/rear wall     |
| 4     | 35.26          | Gusset of both sides   | 35.32          | Gusset of both sides   |
| 5     | 36.31          | Rear wall/front beam of top cover | 36.71 | Rear wall/front beam of top cover |

The table shows that the inherent frequencies of other orders for coupling modal are lower than it corresponding structure modal expect the second order and the third order. This is because the air in acoustic space of cab has damping effect, making the coupling modal frequency decreased. The variation is within 0.4 Hz. But the mode shape of coupling modal and structure modal is consistent, it can conclude that the structure deformation dominant the coupling modal which means that the change of sound pressure distribution is caused by the vibration of structure.

**THE ANALYSIS OF COUPLED ACOUSTIC FIELD**

For light truck cab, the direct excitation source of vibration is caused by suspension vibration in the upper end of the cab. Using LMS Virtual. Lab/System Analysis software force signals of direction X, Y, Z to the cab’s four suspension. These signals are from the random excitation force of grade B level road which obtained by road test in the cab suspension.

In LMS.Virtual lab, it uses acoustic finite element module and adopts the coupling method to predict the sound field in the car. Analysis frequency is set from 0 to 100 Hz and analysis step is set 0.25 Hz. The results are represented by A weight sound pressure levels. The sound pressure is collected from the driver's right ear in 60km/h uniform driving conditions of grade B road. The results are shown in Fig.2:
According to Fig.2, it can calculate the peak frequency in the 0~100Hz of the driver's right ear and the frequency is respectively in 36 Hz, 44 Hz, 57 Hz, 72 Hz, 82 Hz, 92 Hz. The corresponding A weighting sound pressure levels are 52.3 dB, 49.0 dB, 64.8 dB, 61.0 dB, 61.9 dB, 62.3 dB. Among them, the maximum sound pressure level of the driver's right ear appears at 57 Hz. Extracting 57 Hz sound pressure response cloud chart and corresponding to analysis with the comparison of coupled modes.

![57Hz sound pressure response cloud chart](a)

(*Figure 3. Vibration type of 57 Hz sound pressure response cloud chart and the cab’s fifteenth order modal coupling.)*

Comparative analysis can be seen in Fig.3 that the maximum sound pressure of sound pressure response cloud chart appears in top position chart. The cab’s fifteenth order coupled modes were excited. The vibration at the roof is larger and the roof becomes the main noise source. Therefore, it should take measures at the top to make the noise reduced. Many domestic and foreign scholars are studying the optimization method such as optimization design of damping reducing vibration [7] and noise, topology optimization of damping material [8] and others. The article will not elaborate.

The sound pressure response cloud chart can be reflected in the given excitation, the distribution of coupled acoustic field. The main noise source can be found by comparing the coupling mode of the cab. But the influence of the cab’s plate vibration on the driver's right ear pressure cannot be quantified. The contributions of acoustic plate can solve the problem.[9] So the contributions of acoustic plate will be analyzed in subsequent articles.

**ACOUSTIC PLATE CONTRIBUTION ANALYSIS OF CAB**

Cab wall is divided into 10 panels by LMS Virtual.lab software before acoustic contribution analysis is processed. reference numbers are as follows: 1 front wind plate, 2 dash panel, 3 the left side of the door, 4 the left side of the window, 5 the top cover, 6 right door, 7 windows on the right, 8 rear around panel, 9 rear floor, 10 front floor.

Acoustic plate contribution below peak frequencies has been paid more attention. Acoustic plate contribution coefficient on driver’s right ear is shown in Fig.4.
Figure 4. the plate acoustic contribution coefficient under the peak frequency.

Figure 4 shows that at different frequencies, sound positive pressure contribution factor of each of the plates to the driver’s right ear is different. Dash plate is the main positive contribution between 44Hz and 57Hz for drive’s right ear. It is the main negative contribution plate between 82Hz and 92Hz. When the main measuring point positive contribution plate is found, plate contribution about measuring point acoustic pressure at different frequencies should be taken into account synthetically. The size of contribution coefficients and positive or negative of contribution coefficients must take into account. In order to reduce the vibration of these plates and achieve the purpose of improving the acoustic environment of the cab, these panels can be used as optimization objects to optimize its structure thickness.[10]

CONCLUSIONS

This paper analyzes the Vibration and Noise of a light truck cab at low frequencies by establishing sound-structure coupling finite element model and processing simulation analysis. Conclusions are as follows.

1) Coupled Modal and structure Modal are consistent. coupled modes dominated by structural deformation.
2) Peak SPL of the right ear of the driver appears at 44Hz, 57Hz, 82Hz, 92 Hz.
3) The driver's right ear main positive sound pressure contributions under peak frequency are 1 front wind plate, 3 the left side of the door, 4 the left side of the window, 5 the top cover, 8 rear around panel, 9 rear floor and 10 front floor.

Through these analyzes, we can propose the following measures to improve the acoustic environment.

1) The vibration of the roof has great influence on cab field. Therefore it can increase the roof plates thickness and restrain the vibration of the roof, but it is not conducive to the structure of the lightweight. The strengthening member can be disposed on the vibration peak of the roof and it can effectively improve the vibration.
2) Through statistical analysis, the sound pressure measuring point can find this major positive contribution to the plate: front windshield, the left door, left side
window, the top cover, rear floor and the front floor. The optimization design for these structure can increase its local stiffness and reduce the vibration and noise.

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