Analysis of Urban Bus Body Vibration and Coupling Interior Noise

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Abstract. Bus body vibration and interior noise caused by vibration directly affect the ride comfort, cause driving fatigue, discomfort, and bring to the urban environmental noise pollution at the same time. Based on urban bus as the research object in the paper, the modal analyses of body structure, interior spoke and solid coupling model have been carried out by LMS. Virtual Lab software, and compared with the results of modal experiment results on body. The interior sound pressure response has been calculated under the engine incentive by using the boundary element method, and the sound pressure response curves for low frequency have been gotten. Finally, the local structure of body has been improved, and this method reaches the purpose of noise reduction by the simulation calculation. At the same time, references have been provided for improving vehicle vibration and noise.

1. Introduction

With the improvement of people's attention for comfortableness of the car, the level of control car body vibration and noise has gradually become one of the important indicators of quality. Bus interior noise is mainly caused by engine, transmission system, tires, etc, and other structural vibration, road machinery [1,2]. Modern cars increasingly tend to be lightweight and high speed, research shows that quality of light body and reduce car like quality measures increase the vibration of the body structure, leads to enhanced interior noise, especially the low frequency noise [3].

2. Body Structure Modeling and Modal Analysis

2.1 Establish the Finite Element Model of Body Structure

An urban bus body is the paper modeling object, it's a semi-load body, consists of frame, floor, ceiling, dash, scarf, and left and right side, the parts are connected by welding. There are two kinds of materials used in the body: Q235 steel and 16Mn steel, 16Mn steel is used in the frame section, Q235 steel is used in other parts. Material properties are shown in table 1.

<table>
<thead>
<tr>
<th>materials</th>
<th>Tensile limit (MPa)</th>
<th>Yield limit (MPa)</th>
<th>Modulus of elasticity (GPa)</th>
<th>Shear modulus (GPa)</th>
<th>density (Kg/mm³)</th>
<th>Poisson's ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>16Mn</td>
<td>480</td>
<td>280</td>
<td>200</td>
<td>79.4</td>
<td>7.85e.9</td>
<td>0.3</td>
</tr>
<tr>
<td>Q235</td>
<td>375</td>
<td>235</td>
<td>203</td>
<td>79.4</td>
<td>7.81e.9</td>
<td>0.3</td>
</tr>
</tbody>
</table>

On the basis of establishing a body structure model, the grid has meshed in LMS Virtual Lab software, the size of tetrahedron grid cell is 60 mm, through the model simplification, specifying the unit type and meshing, finally, the finite element model of car body structure is generated, showing in figure 1. The total number of nodes is 56475, and the total number of units is 28669.
2.2 Modal Analysis of Body Structure

In this paper, the modal calculation is conducted of the finite element model, compared the first 4 order modal frequency and test modal frequency. Table 2 shows that the maximum error is 4.81% and the error is small, which proves that the finite element model has high precision and can meet the requirements of subsequent analysis. The modal mode is shown in figure 2.

Table 2. Comparison Between Vehicle Body Modal Simulation Results and Test Results.

<table>
<thead>
<tr>
<th>Order number</th>
<th>Natural frequency calculation results /Hz</th>
<th>test results /Hz</th>
<th>modal vibration mode</th>
<th>e/%</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>28.47</td>
<td>29.91</td>
<td>top swing</td>
<td>4.81</td>
</tr>
<tr>
<td>2</td>
<td>29.28</td>
<td>30.15</td>
<td>top swing</td>
<td>2.89</td>
</tr>
<tr>
<td>3</td>
<td>35.48</td>
<td>36.98</td>
<td>first-order bending torsional</td>
<td>4.06</td>
</tr>
<tr>
<td>4</td>
<td>39.39</td>
<td>41.35</td>
<td>first-order bending</td>
<td>4.74</td>
</tr>
</tbody>
</table>

(a) $f_1=23.1434\text{Hz}$  (b) $f_2=28.4448\text{Hz}$  (c) $f_3=0.9575\text{Hz}$  (d) $f_4=55.8006\text{Hz}$

Figure 2. The First 4 Order Modal Vibration Mode of the Body Structure.

It can be seen from the vibration model diagram that the vibration form of the body structure is mainly manifested as the overall torsion and bending, and local vibration of the body ceiling, side circumference and front circumference. Overall, the roof deformation is more severe.

In the body design, the natural frequency of structural mode should be avoided when the engine idle. Body and suspension resonance frequency of this urban bus is between 2Hz and 3.4Hz, frequency of engine idle speed is about 32Hz, therefore, structure natural frequency and frequency of engine idle speed is not the same, and it’s not in the body and suspension frequency range, at the same time, road incentive is usually less than 20Hz, it can avoid resonance theoretically.

3. Sound Cavity Modeling and Modal Analysis

3.1 Establish Finite Element Model of Coustic Cavity

A coustic cavity model with seat of the passenger car is established in LMS Virtual.Lab. The size of grid unit is 70mm, as shown in figure 3.
3.2 Simulation Analysis of Acoustic Modal

The modal calculation of acoustic cavity finite element model is carried out, and the modal frequency is shown in table 3 and the vibration shape is shown in figure 4.

<table>
<thead>
<tr>
<th>Order number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>frequency /(Hz)</td>
<td>31.3</td>
<td>60.9</td>
<td>79.3</td>
<td>85.1</td>
<td>88.3</td>
<td>90.2</td>
<td>98.7</td>
<td>100</td>
</tr>
</tbody>
</table>

(a) \( f_1 = 31.287 \text{Hz} \)  \hspace{1cm}  (b) \( f_2 = 60.924 \text{Hz} \)
\hspace{1cm}  (c) \( f_3 = 79.263 \text{Hz} \)  \hspace{1cm}  (d) \( f_4 = 85.122 \text{Hz} \)

Figure 4. First 4 Modes of Sound Cavity Structure.

It can be seen from the vibration mode that the first two vibration modes are generally distributed symmetrically from left to right. In the later stage, due to the existence of the seat model, the sound pressure is no longer uniformly distributed and there are no rules to follow.

4. Establishment of Sound-Solid Coupling Model and Modal Analysis

4.1 Establish Sound-Solid Coupling Model

Based on the finite element model of bus body and sound cavity, a sound and solid coupling model is established, which requires that the nodes of the sound cavity model and the body structure model must correspond to each other, that is, the nodes have the same coordinate, so that they can move together and have the same degree of freedom when analyzing. The coupling model established in LMS Virtual. Lab is shown in figure 5.

Figure 5. Fem Model of Sound-Solid Coupling.

4.2 Modal Analysis of Coupled Model

The body structure is an elastic system, the body structure and sound cavity interact and influence
each other. In order to describe the interaction between structure and sound cavity accurately, it is necessary to analyze the dynamic characteristics of the coupling system composed of structure and sound cavity.

If the interior space of the bus is regarded as a closed system and the sound damping is ignored, the finite element model of the sound cavity can be obtained by discrediting the interior air domain. The acoustic finite element equation can be expressed as:

\[
\left( \begin{bmatrix} K_p \end{bmatrix} - \omega^2 \begin{bmatrix} M_p \end{bmatrix} \right) \{P\} = \omega^2 \begin{bmatrix} R \end{bmatrix} \{U_s\} \tag{1}
\]

In the type: \([M_p]\) is the acoustic mass matrix, \([K_p]\) is the acoustic stiffness matrix, \([P]\) is the sound pressure vector, \([R]\) is the fluid-solid coupling matrix, \([U_s]\) is the node displacement amplitude vector.

The finite element equation of the structural system can be expressed as:

\[
\left( \begin{bmatrix} K_s \end{bmatrix} - \omega^2 \begin{bmatrix} M_s \end{bmatrix} \right) \{U_s\} = \begin{bmatrix} F_p \end{bmatrix} + \begin{bmatrix} F_s \end{bmatrix} \tag{2}
\]

In the type: \([M_s]\) is the structure mass matrix, \([K_s]\) is the structural stiffness matrix, \([F_s]\) is the force applied to the car, \([F_p] = [R]^T \{P\}\)

The steady state frequency domain equation of the sound and solid coupling system can be obtained by combining equations 1 and 2:

\[
\left( \begin{bmatrix} K_s \end{bmatrix} - \omega^2 \begin{bmatrix} M_s \end{bmatrix} \right) - \omega^2 \begin{bmatrix} R \end{bmatrix}^T \begin{bmatrix} K_p \end{bmatrix} - \omega^2 \begin{bmatrix} M_p \end{bmatrix} \right) \begin{bmatrix} U_s \end{bmatrix} = \begin{bmatrix} F_p \end{bmatrix} + \begin{bmatrix} F_s \end{bmatrix} \tag{3}
\]

\[
\begin{bmatrix} K_s \end{bmatrix} - \omega^2 \begin{bmatrix} M_s \end{bmatrix} \begin{bmatrix} R \end{bmatrix} - \begin{bmatrix} K_p \end{bmatrix} - \omega^2 \begin{bmatrix} M_p \end{bmatrix} \right) \right. \left. \begin{bmatrix} U_s \end{bmatrix} = \begin{bmatrix} F_p \end{bmatrix} + \begin{bmatrix} F_s \end{bmatrix} \right. \left. \begin{bmatrix} 0 \end{bmatrix} \right. \right.

is the coupling stiffness matrix, \([U_s] = [F_s] = \begin{bmatrix} 0 \end{bmatrix}\) is the system response vector, \([F_p] = [R]^T \{P\}\) is the external motivation vector.

Modal calculation of coupling model is accomplished by modal superposition method in virtual. Lab Acoustic module. The mode frequency and mode are shown in table 4 and figure 6.

<table>
<thead>
<tr>
<th>Order time</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
</tr>
</thead>
<tbody>
<tr>
<td>frequency/(Hz)</td>
<td>23.1</td>
<td>28.4</td>
<td>31.8</td>
<td>41.0</td>
<td>55.8</td>
<td>61.4</td>
<td>61.9</td>
<td>62.3</td>
</tr>
</tbody>
</table>

![Figure 6](image)

(a) \(f_1 = 23.13\)Hz  (b) \(f_2 = 28.40\)Hz  (c) \(f_3 = 31.78\)Hz  (d) \(f_4 = 40.95\)Hz

As can be seen from the figure, the first 4 order non-zero coupling modes are mainly the deformation of bus structure, and there is little difference between the mode modes of coupling structure and the modal modes of body structure. The coupling mode mainly based on structural deformation is relatively dense, the coupling mode at most frequencies is close to the corresponding structural mode, and the influence of air on structural vibration mode is small. The vibration deformation of the roof, side and front of the body is prominent and should be paid attention to.
5. Analysis of Sound Field and Structural Improvement of the Bus

5.1 Calculation and Analysis of Sound Field in the Bus

The indoor noise of the bus is mainly low-frequency noise, while the engine is the main source of low-frequency noise. Due to the complexity of the actual engine operating conditions, it is difficult to obtain the real excitation. Therefore, this paper mainly investigates the acoustical characteristics when the body structure is driven by the engine harmonic [4]. Full constraint is applied to the positions where the suspension is connected with the bus body, that is, the rotation and movement of x, y and z are restricted, 1 unit vertical load is applied to the engine mount, the calculated frequency is set to 5-150Hz, the step length is 5Hz, the modal frequency is set to 5-150Hz, and the modal damping is 0.03. The vibration response of the body structure under the simple harmonic excitation of the engine is calculated and the body vibration velocity is obtained in the Virtual Lab. Then, the acoustic boundary element environment is introduced, and the body structure grid, acoustic boundary element grid, field point grid and vibration velocity are introduced, the sound pressure distribution of the field point is calculated by data mapping transfer, defining the boundary conditions and the sound absorption properties of the seat, etc. The sound pressure response curve of the driver and a passenger by the ear is shown in figure 7.

![Figure 7. Response Curve of Sound Pressure near the Ear of Driver and Passenger.](image)

It can be seen from the response curve of sound pressure level that the sound pressure level of drivers and passengers is basically the same, but not exactly the same. The peak of sound pressure reached 85dB at 100Hz. Sound pressure also peaked near 90Hz and 115Hz. The maximum sound pressure peak significantly exceeded the noise limit of 82dB stipulated in GB/t25982-2010 of the national standard "noise limit value and measurement method in passenger cars". The peak of sound pressure mainly occurs at the low-frequency stage, i.e. 5~150Hz, indicating that the interior space noise caused by engine excitation is mainly low-frequency noise, which is consistent with the actual situation. If the structural modal frequency is similar to or the same as these peak frequencies, the peak of sound pressure may be caused by resonance phenomenon. Therefore, measures such as improving the body structure or changing the structural materials can be taken to avoid resonance frequency to reduce noise sound pressure.

5.2 Analysis of Structural Improvement

According to the above analysis results, the roof position with large modal deformation of the body was strengthened to improve the rigidity. Considering the lightweight design of modern bus, the roof was not thickened, but three short beams were added to the roof frame, as shown in fig.8.

![Ceiling Skeleton (before Improvement) and The Improved Ceiling Skeleton](image)

Figure 8. Comparison of Roof Skeleton Before and After Improvement.
After the structural improvement, the boundary element method was used again to calculate the sound pressure at the field point, and the result was shown in figure 9.

![Acoustic pressure curve of driver's ear and passenger's ear](image)

**Figure 9.** Response Curves of Driver and Passenger Ear Pressure After Structural Improvement.

Compared with the results of sound pressure response before and after the improvement, it can be seen that the sound pressure peak still mainly occurs near the frequencies of 90Hz, 100Hz and 115Hz. After the improvement, the maximum sound pressure of the passenger ear was around 100Hz, about 82dB, which was reduced by about 3dB. Meanwhile, the peak of sound pressure at other frequencies also decreased. The structural improvement scheme is proved correct by comparison, which provides an effective method for the bus design of vibration and noise reduction.

6. Conclusion

(1) The modal results of the body structure and the sound and solid coupling structure show that the local vibration deformation of the roof, side circumference and front of the body is large within the range of low frequencies, especially the ceiling, and its stiffness is poor.

(2) The calculation of the sound pressure at the field point indicates that the maximum sound pressure of the driver and passenger ear exceeds the noise sound pressure 1~3dB as specified by the national standard. The peak of sound pressure corresponding to low frequency may be caused by structural resonance, which needs to be paid attention to.

(3) Through the structure improvement of the roof of the bus body and simulation analysis, the peak of sound pressure of the driver and passenger ear is reduced, and the interior noise of the bus is greatly improved, providing an effective method for its reasonable noise reduction. In addition, noise can be reduced and ride comfort can be improved by adding vibration isolation material, sound absorbing material, and adopting active control etc.

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References


