A SIMULATION AND EXPERIMENTAL ANALYSIS OF INTERIOR NOISE IN AN AMBULANCE AND NOISE REDUCTION WITH DAMPING MATERIALS

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Based on the theory of multi-body system dynamics and the method of finite element and boundary element, the finite element model of vehicle body and the rigid-flexible coupled model of vehicle and the acoustic boundary element model of carriage were established by applying a variety of simulation software. Road roughness was simulated and the SPL of field point in carriage under the excitation was computed at the frequency range of 20-300Hz. Considering the size and the positive/negative nature of panels acoustic contribution at SPL peak of field point, damping treatments were taken on different combinations of panels to decrease multiple field point’s SPL peak and the optimal treatment was finally confirmed. Finally, the structural modification with damping materials is performed to the most contributing panels and the experimental results indicate that maximum SPL peak is decreased by 11dB and OSPL is decreased by 3dB.

1. Introduction

Ambulance carriage is the coexistence place for the operations of the medical staff, medical instruments work and the sick and wounded. In the process of driving and parking, interior noise not only interferences the crews’ auditory for the normal work, but also harm to the crews’ health when it is severe[1,2].

Due to the auto body panels are susceptible to vibrations, they mainly radiate structure-borne noise below 300 Hz low frequency into the passenger compartment which produce the main components of interior noise[3]. In view of low frequency noise, different panels usually have different acoustic contribution. Indentifying the panels with greater acoustic contribution and reducing theirs vibration are the most directly method to reduce noise. Currently, the research of panels’ acoustic contribution mostly adopt value simulation which employs the panel acoustic contribution analyti-
ical method to find the panels with greater acoustic contribution in simulation model[4-10]. However, the simulation method need construct accurate models of structure and sound field, and whether the model is accurate or not will influence the accuracy and reliability of the results.

In this paper, a vehicle model was simulated by applying CAE technology to study the internal acoustic characteristics of carriage when it was on the rough road and provided guidance for improving the structure of vehicles. Based on the theory of multi-body system dynamics, the analytical method of finite element and boundary element, the structural noise from panel vibration was studied in the frequency range of 20 to 300Hz when the vehicle was on the rough roads, combining with ADAMS, ANSYS and SYSNOISE software. At last, the structural modification with damping materials is performed to the most contributing panels and the experiment result show that maximum SPL peak is decreased by 11dB and OSPL is decreased by 3dB. The interior noise reduction was realized by acoustic contribution analysis and damping treatments.

2. Model analysis

2.1 The finite element model of carriage

The finite element model of carriage was constructed in ANSYS software (Figure.1). The mesh of carriage panels were divided by shell63/181 element and the pillars and beams were divided by three-dimensional elastic beam4/188 element. The rigid joints of panels were simulated by coupling nodes’ displacement and mechanics. The interior and seats were connected to the vehicle body using mass21 element. The structural modalities of carriage and cab were calculated by Block Lanczos method. There were 209 order modal of carriage in the frequency range of 20 to 300Hz and partial modal frequencies were shown in Table 1.

![Figure.1. The finite element model of carriage](image)

<table>
<thead>
<tr>
<th>range</th>
<th>12</th>
<th>47</th>
<th>81</th>
<th>102</th>
<th>146</th>
<th>178</th>
</tr>
</thead>
<tbody>
<tr>
<td>frequency [Hz]</td>
<td>48.38</td>
<td>84.17</td>
<td>110.82</td>
<td>179.84</td>
<td>230.06</td>
<td>295.42</td>
</tr>
</tbody>
</table>
2.2 The rigid-flexible coupled model

The rigid-flexible coupled model of vehicle was constructed by combining the structural flexible coupled model and the rigid coupled model by inputting the modal neutral files of carriage structure which were generated in ANSYS into ADAMS software (Figure 2).

![Figure 2. The rigid-flexible coupled model of vehicle](image)

For the convenience of study, the chassis was appropriately simplified. The brake system, steering system and driving system which has little effect on study were Omitted. The suspension was approximate seen as linear system and adopted of the equivalent stiffness and equivalent damping at E-Class road to describe the physical characteristics of suspension (Table 2) [11].

<table>
<thead>
<tr>
<th></th>
<th>equivalent stiffness [N/m]</th>
<th>equivalent damping [N•s/m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Front suspension</td>
<td>45565.91</td>
<td>14513.25</td>
</tr>
<tr>
<td>Rear suspension</td>
<td>25782.90</td>
<td>12822.93</td>
</tr>
</tbody>
</table>

2.3 The acoustic boundary element model

Appropriate modifications to the finite element model of the carriage were made to form closed cavity model and extract shell element as the mesh of boundary element model. To ensure that the vibration response results can accurately input to the boundary element model, numbers of elements and nodes were not modified in them. The model had at least six elements in a wavelength range and met the precision requirement. The acoustic boundary element model of carriage was constructed by inputting shell elements into SYSNOISE software as shown in Figure 3.
3. Road roughness simulation

The road roughness was the random process of each steady states when the vehicle was driving at a constant velocity\cite{12,13}. In order to improve the precision of calculation, the correlation of inputting the front and rear wheels was considered and adopted filter-white noise method to construct the road roughness model.

The formula of time-domain model for the roughness of steady road which related to the front and rear wheels was expressed as\cite{14,15}:

\[
\dot{I}(t) = -2\pi n_0 v F_0 I(t) + 2\pi \sqrt{G_0 v} B_{xi} x(t) + 2\pi \sqrt{G_0 v} B_{yi} W(t) \\
\dot{x}(t) = A_x x(t) + B_x W(t)
\]  

In the formula, \(I(t)\) was the road roughness vector of the front and rear wheels; \(x(t)\) was the state transition vector; \(F_0\) was the unit coefficient matrix; \(W(t)\) was the zero mean white noise time series; \(G_0\) was the coefficient of road roughness, denoting the road levels; \(n_0\) was the spatial frequency cutoff; \(v\) was the speed; \(B_{xi}, B_{yi}, A_x, B_x\) were the coefficient matrix which describing the characteristics of the front and rear wheels. The formula was expressed as follows:

\[
B_{xi} = \begin{bmatrix} 0 & 0 \\ 1 - t_d/2 & 0 \end{bmatrix} \quad B_{yi} = \begin{bmatrix} 1 \\ 1 \end{bmatrix} \quad A_x = \begin{bmatrix} -6/t_d & 12/t_d^2 \\ 1 & 0 \end{bmatrix} \quad B_x = \begin{bmatrix} 12/t_d^2 \\ 0 \end{bmatrix}
\]

In the formula, \(t_d = L/v\) was the time difference of inputting the front and rear wheels; \(L\) was the tread. Inputting the zero-mean white noise signals which was formed in Matlab into the Eq. (1) and Eq. (2), the road roughness was calculated under the condition of the E-level road and the speed of 60 km/h using Runge-Kutta method (Figure.4).
4. Noise simulation and result analysis

The road roughness was loaded in the simulation model, the vibration response of carriage was obtained. Then, the vibration response as boundary condition was loaded into the compartment acoustic boundary element model. Set the left supine wounded ear for the field points. In the SYSNOISE software, the point sound pressure level was calculated using the direct boundary element method at the frequency of 20 to 300Hz, step 1Hz and the reference sound pressure $2 \times 10^{-5}$Pa. Fig.5 showed A-weighted sound pressure level curve of the left supine wounded ear.

![Figure 5. The SPL curve at left prone wounded ear](image)

As shown in Fig.5, the SPL peak was mainly distributed in the 86Hz (69dB), 182Hz (54dB), 228Hz(48dB), 296Hz (55dB). Referring to Tab 1, the sound pressure level peak was basically corresponding to the structural modal of carriage. As the sound pressure level maximum peak 69dB appeared in the vicinity of 86Hz, corresponding to the forty-seventh order modal frequency 84.17Hz.

5. Panel acoustic contribution analysis

Reducing the panel vibration was the most direct way to reduce low frequency noise in vehicle. The damping material was adopted to reduce the noise of panels in order to reduce the noise of the field. In order to reduce noise targetedly, panel acoustic contribution was analyzed for the acoustic boundary element model in SYSNOISE software, the acoustical contributions of the compartment of each point in different frequency were obtained (Fig. 6). The corresponding relationships between walls and numbers were showed as follows: 1-front roof; 2-rear roof; 3-front panel; 4-the left...
anterior wall; 5- the left rear panel; 6-the right front panel; 7-theright rear panel; 8- front plate; 9- rear plate; 10- rear panel.

As shown in Fig. 6, the sound pressure level peak of the left supine wounded ear, the front and rear roofs showed the large positive contribution and became the main factor affecting the noise. Front panel had the different positive contribution and noise can be reduced by reducing the vibration. The rear panel was in the neutral area and showed the relatively small acoustic contribution and little effect on noise. Other panels showed the different acoustic contribution to different sound pressure level peaks.

Figure 6. The histogram of panel acoustic contribution of left prone wounded ear’s SPL peak

Through the analysis of the panel acoustic contribution, we found that the size and positive and negative contribution of panels to the sound pressure lever were quite different in the different frequency. Although the sound pressure level of a frequency can be reduced when reducing a panel vibration, sound pressure level of another frequency position may increase. This paper pointed to reduce peak sound pressure levels of multiple fields, generally considering distribution of peak value of the sound pressure level and acoustic contribution of panels. Noise-reducing measures were taken to reach the maximum effect of the noise reduction.

6. Damping experiment

By tried with combination of different panels for noise reduction, the optimal damping noise-reduction protocol was determined ultimately. On the basis of the results above, the roof panel, left and right panel that have greater sound field contributions are chosen for placement of the 3mm damping materials layers. Considering the damping properties of damping materials, the sound absorption performance will not be considered. The damping materials is made of specially synthetic polymer, which are used for reducing vibration of sideboards and floor board (Table 3). Figure 7 shows the panels covered with damping materials.
Tab 3. Damping materials parameters

<table>
<thead>
<tr>
<th>parameter</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>areal density [kg/m²]</td>
<td>1.2</td>
</tr>
<tr>
<td>tensile-strength [MPa]</td>
<td>≥0.5</td>
</tr>
<tr>
<td>tear strength [N/mm]</td>
<td>≥6</td>
</tr>
<tr>
<td>critical damping factor</td>
<td>≥0.5</td>
</tr>
<tr>
<td>acoustic attenuation coefficient</td>
<td>≥30</td>
</tr>
</tbody>
</table>

Fig. 7. Panels’ surface structure after covering damping materials layer

Fig. 8 showed the sound pressure level curve of the left supine wounded ear before and after the implementation of damping treatment. As shown in Fig. 8, the maximum sound pressure level peak was reduced from 69dB to 58dB and overall sound pressure level was reduced from 80.7dB to 77.1dB by implementing of the noise reduction protocol. Sound pressure level peak frequency also changed, which means that the damping treatment on partial panels changed the original structure of carriage. When reducing the panel vibration and sound pressure level peak, the distribution of the frequencies can also be changed at the same time. We will continue to focus on the phenomena.

Additionally, comparing with the condition of covering damping materials layer at all panels, the added mass of the condition of covering damping materials layers at partial panels is 38.4kg, saving 24 kg. The damping treatment at partial panels places the damping layers in the optimum locations and enhances the efficiency of the damping layers so that the dual requirements of sound field refinement and light auto body weight can be met simultaneously.
7. **Summary**

The results showed that, the peak sound pressure level and frequency distribution of the peaks were related to structure modal frequency under the road roughness excitation condition. The size and properties of panels’ acoustic contribution varied with the frequency which should be considered in reducing noise. The damping materials will also change the frequency distribution of the sound pressure level when reducing noise.

According to the results of simulations, we forecasted the low-frequency structural noise inside carriage and found the optimal damping noise reduction protocol. The maximum SPL peak is decreased by 11dB and OSPL is decreased by 3dB. These results could provide guidance for the next vehicle modification.

**References**


