Improving the Sound Pressure Level for a Simplified Passenger Cabin by using Modal Participation and Size Optimization

Adrian COROIAN
Technical University of Cluj-Napoca, Department of Mechanical Systems Engineering, 103-105 Muncii Blvd., 400641 Cluj-Napoca, Romania, e-mail: adycoroian@yahoo.com

Iulian LUPEA
Technical University of Cluj-Napoca, Department of Mechanical Systems Engineering, 103-105 Muncii Blvd., 400641 Cluj-Napoca, Romania, e-mail: iulian.lupea@mep.utcluj.ro

Abstract: - The interior acoustics is an important part in the design stage for automakers in order to reduce the sound pressure level in the passenger compartment by using Computer Aided Engineering (CAE). The sound field in the vehicle interior is affected by the vibration of the body in white (BIW) and by the acoustic modal characteristics of the enclosed volume of air. The Finite Element Method (FEM) was used for a vibro-acoustic analysis of a simplified passenger compartment. In order to simulate the sound pressure level (SPL) at the hearing point of the driver while the firewall panel is excited, the modal frequency response analysis has been used. The SPL versus frequency curve shows that at the excitation frequency of 50 Hz the highest value of 87.09 dB(A) is recorded. The modal participation analysis is carried out to find the structural modes which contribute most to the sound pressure level inside the vehicle cab. Based on the analysis results, a size optimization was performed in order to lower the SPL at the driver’s ear location at the excitation frequency of 50 Hz applied to the firewall panel. The thickness parameters of the structural panels have been altered in order to reduce the interior noise level at the frequency of interest.

Keywords: - vibro-acoustic analysis, modal frequency response analysis, modal participation factors, vehicle interior noise, size optimization

1. INTRODUCTION

“Noise, vibration and harshness” (NVH) is an engineering discipline that deals with the structural dynamic and acoustics aspects of the automobile design. This NVH study represents the acoustic analysis of the passenger compartment, sound playing an important role in the development of motor vehicle. The NVH study in the passenger compartment can be divided in two categories of interest:

- Noise: audible sound in the frequency range 30 – 4000 Hz;
- Vibration: tactile vibration in the frequency range 30 – 200 Hz.

The structure-borne noise contains the noise that comes from the excitation forces transmitted from the vibrating sources to the whole vehicle body through the structural attachment points [3]. In this study, we investigated the structure borne noise, which is caused by vibrating panels enclosing the vehicle. One excitation for the structure-borne noise comes from the engine, which causes the cabin to vibrate at the resonant frequencies. The vibrating panels will produce a variation in the sound pressure level within the passenger cabin, and consequently will generate an undesirable booming noise. In Figure 1 can be seen the engine, the firewall and the front floor panels, which have been extracted from a Ford Taurus FE model.

Figure 1. Structural FE model showing the firewall panel, the front floor panel and the engine

The finite element analysis (FEA) can be used to predict the sound pressure level inside the passenger cabin [5], [6], [9], [10], [12].
In this study we utilized the modal frequency response analysis performed with Radioss solver, a module of HyperWorks [14].

In order to reduce the interior noise, it is important to understand the dynamics of the vehicle panels and how they interact with the air inside the vehicle cabin. Therefore, the next step, after the predicting of sound pressure level, is to determine the participation of the structural modes to the acoustic response. In this scope, we utilized the modal/panel participation utility of Radioss. HyperGraph allows us to plot the complex contribution of the structural modes to the sound pressure at the driver’s ear location for a given excitation frequency [14]. We can establish which structural panel is found as the highest contributor to the sound pressure level at the driver’s ear at each excitation frequency. These panels can participate in a size optimization in order to find the optimum configuration by using OptiStruct solver capabilities.

The main purposes of this study are as follow: modeling a coupled acoustic-structure system, which represents a simplified passenger compartment, by means of finite element method; performing an acoustic analysis to calculate the sound pressure level at the driver’s ear position; finding the modal participation factors to know which structural panels contribute more to the SPL at driver’s ear at a given frequency; optimizing the thickness of those panels using a size optimization in order to reduce the SPL at a given frequency.

2. FINITE ELEMENT MODEL CREATION

At the pre-processing stage of the finite element analysis (FEA), the following basic steps were completed in HyperMesh:

- Design the 3D geometry of the structural panels that looks like the interior cabin of the vehicle. The overall dimensions of the model are: 1200 mm x 700 mm x 782 mm. The thickness values of the panels are presented in Table 1.

- Starting from this 3D geometry, the middle surfaces of the panels and the air cavity were discretized by using proper finite elements. For modeling the structural panels, CQUAD4 finite elements were used. These elements are 2D quadrilateral elements which have four corner nodes with six degrees of freedom (DOFs) per node: three translational displacements in X, Y and Z directions and three rotations about the X, Y, and Z axes. For the acoustic cavity, CHEXA finite elements were used. These are 3D elements bounded by six faces, having eight nodes with one degree of freedom per node: the acoustic pressure at that location.

Table 1. Panel thicknesses

<table>
<thead>
<tr>
<th>Panel name</th>
<th>Thickness [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>floor</td>
<td>0.9</td>
</tr>
<tr>
<td>front &amp; back doors</td>
<td>1</td>
</tr>
<tr>
<td>firewall</td>
<td>0.8</td>
</tr>
<tr>
<td>windshield</td>
<td>5</td>
</tr>
<tr>
<td>roof</td>
<td>0.7</td>
</tr>
<tr>
<td>bulkhead</td>
<td>0.8</td>
</tr>
</tbody>
</table>

Table 2. Material properties

<table>
<thead>
<tr>
<th>Structure</th>
<th>Young's modulus</th>
<th>Poisson’s ratio</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>2.1e11 Pa</td>
<td>0.312</td>
<td>7850 kg/m^3</td>
</tr>
<tr>
<td>glass</td>
<td>6.2e10 Pa</td>
<td>0.24</td>
<td>2300 kg/m^3</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fluid</th>
<th>Speed of sound</th>
<th>Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>air</td>
<td>343 m/s</td>
<td>1.21 kg/m^3</td>
</tr>
</tbody>
</table>

3. ACOUSTIC ANALYSIS

At the interface between structure part and fluid part, the accelerations ($\ddot{u}$) on the structural grid points excite the air particles on the fluid part, and the pressures ($p$) on the fluid grids excite the structural part. This is a coupled problem between the vibration of the panels and the pressure inside the passenger cabin.

The acoustic analysis of the coupled system – the structure and the air - can be done either by direct
integration or by modal frequency response analysis. The outputs of the analysis are:

- the responses for the structure part - the displacements and rotations at structural grid points
- the responses for the fluid part – the pressures at the fluid grid points.

After finite element discretization, the equations of motion for the fluid part and for the structure part are as follows [3], [5], [9], [12]:

\[
M_f \ddot{u} + B_f \dot{u} + K_f u = f_s + f_{sf}
\]

\[
M_f \ddot{p} + B_f \dot{p} + K_f p = f_f + f_{fs}
\]

where

- \(M_f\) and \(M_s\) represent the mass matrix for the fluid part and for the structure part, respectively.
- \(B_f\) and \(B_s\) represent the damping matrix for the fluid part and for the structure part, respectively.
- \(K_f\) and \(K_s\) represent the stiffness matrix for the fluid part and for the structure part, respectively.
- \(p\) and \(f_f\) are the acoustic nodal pressures vector and the external source vector acting on the fluid part.
- \(u\) and \(f_s\) are the structural nodal displacements vector and the external source vector acting on the structure part.
- \(f_{fs} = A^T \ddot{u}\) is the vector of forces from the structure part that are acting on the fluid part.
- \(f_{sf} = -Ap\) is the vector of forces from the fluid part that are acting on the structure part.
- \(A\) represents the interface matrix that describes the geometric relationship between the faces of the fluid elements on the surface of the air mesh and the structural nodes on the wetted surface of the structural mesh.

The differential equations of motion for the coupled fluid - structure system, can be written in the following matrix form:

\[
\begin{bmatrix}
M_f & O \\
-A^T & M_f
\end{bmatrix}
\begin{bmatrix}
\ddot{u} \\
\dot{p}
\end{bmatrix}
+ \begin{bmatrix}
B_f & O \\
O & B_f
\end{bmatrix}
\begin{bmatrix}
\dot{u} \\
p
\end{bmatrix}
+ \begin{bmatrix}
K_f & A \\
O & K_f
\end{bmatrix}
\begin{bmatrix}
u \\
p
\end{bmatrix}
= \begin{bmatrix}
f_s \\
f_f
\end{bmatrix}
\]

(3)

The vibration of the firewall panel was considered to be the most dominant source of noise in the cabin of the car, when the engine is running. Engine forces are transmitted to the vehicle panels through the engine mount locations. The amplitude variations of the force data can be obtained experimentally at each mount location while the engine is running at different speeds.

In this study, for the frequency response analysis, we have chosen a unit load along the direction of the length of the car, that is applied on a grid point of the firewall panel which is located near the geometric center of the firewall. The unit load is a sinusoidal harmonic force in the frequency range of 0-300 Hz. The response is the sound pressure level variation as a function of a frequency-dependent force. During the post-processing with HyperGraph (module of HyperWorks), the pressure was converted to ‘A’ weighted sound pressure by using the relation (4), [3], [9].

\[
SPL_{dB(A)} = 20\log\left(\frac{p/\sqrt{2}}{p_{ref}}\right) + A_{weighting}
\]

(4)

where \(p\) is predicted by the Radioss solver and \(p_{ref} = 20*10^{-6}\) Pa is the reference sound pressure in air.

The location of the response is a specific grid point inside the cabin, which represent the driver’s hearing point. As can be seen in Figure 3, the highest sound pressure level is 87 dB(A) which is achieved at a frequency of 50 Hz.

![Figure 3. Sound pressure level at driver’s ear location as a function of excitation frequency](image)

4. MODAL PARTICIPATION

Modal participation is a measure of how much each structural mode participates at the sound pressure level inside the passenger cabin at a given excitation frequency [14]. Modal participation is also a CAE driven vehicle NVH development [5].

In our study, to output the modal participation factors for all frequencies, the PFMODE output request card was used from the modal frequency response analysis.

The participation factors are obtained from equation (3) [13].
where the matrix $\{\phi_i\}$ contains the uncoupled, undamped structural modes, and the matrix $\{\phi_j\}$ contains the uncoupled, undamped, rigid–walls acoustic modes. The vectors $\{\xi_s\}$ and $\{\xi_f\}$ are the modal amplitudes for the structure and the fluid. Substituting these relations into equation (3) and pre-multiplying by the modal matrices $\phi^T$ and $\phi^T$, we get the following equation:

\[
\left[ \begin{array}{c}
\phi^T M \phi_s & O \\
-\phi^T A \phi_s & \phi^T M \phi_f
\end{array} \right] \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} + \begin{bmatrix} \phi^T B_s \phi_s & O \\
O & \phi^T B_f \phi_f
\end{bmatrix} \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} + \\
-k_s a_s \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} = \begin{bmatrix} O_s \\ Q_f \end{bmatrix}
\]

or by considering the orthogonal properties of the structural and acoustic modes results:

\[
\left[ \begin{array}{cc}
m_s & O \\
-a_f m_f & O
\end{array} \right] \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} + \begin{bmatrix} b_s & O \\
O & b_f
\end{bmatrix} \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} + \\
k_s a_s \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} = \begin{bmatrix} O_s \\ Q_f \end{bmatrix}
\]

The following harmonic solution at the forcing frequency $\omega$ are assumed:

\[
\begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} = e^{i\omega t} \begin{bmatrix} \xi_s^0 \\ \xi_f^0 \end{bmatrix}
\]

The first two derivatives are:

\[
\begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} = i\omega \begin{bmatrix} \xi_s^0 \\ \xi_f^0 \end{bmatrix} e^{i\omega t} = i\omega \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix}
\]

\[
\begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix} = -\omega^2 \begin{bmatrix} \xi_s^0 \\ \xi_f^0 \end{bmatrix} e^{i\omega t} = -\omega^2 \begin{bmatrix} \xi_s \\ \xi_f \end{bmatrix}
\]

For the second (fluid) equation of the relation (8), we get:

\[
\omega^2 [a_f]^T \{\xi_s\} + \\
-\omega^2 [m_f] + i\omega [b_f] + [k_f] \{\xi_f\} = \{Q_f\}
\]

Defining $\{z_2\}$ as:

\[
\{z_2\} = -\omega^2 [m_f] + i\omega [b_f] + [k_f]^{-1}
\]

from equation (12) results: 

\[
\{\xi_f\} = -\omega^2 [z_2]^T \{\xi_s\} + [z_2] \{Q_f\}
\]

The structure mode participation is defined as:

\[
\{p_s\} = -\omega^2 [\phi_s]^T [z_2]^T \{\xi_s\} \]

where $[\{\xi_s\}]$ is the diagonalized vector containing the structural modal amplitudes.

In a similar manner the fluid mode participation is determined [13].

Figure 4 shows the structural modes participation at the driver’s ear location at 50 Hz.

Figure 5 depicts the sound pressure level curves corresponding to the structural modes. In figure 6 are presented the first 10 highest contributors to the sound pressure level at the driver’s ear location in the frequency range from 0 to 300 Hz.
Figures 6 to 8 show the alternative views of the most important structural modes that contribute to the sound pressure level at the driver’s ear position at the excitation frequency of interest. Analyzing these structural modes, we have decided the structural panels that can participate in the optimization problem using finite element method. The structural panels are the floor panel, the roof panel and the bulkhead panel.

5. SIZE OPTIMIZATION PROBLEM

The optimization with an optimization solver is an automatic process that makes a system as good as possible based on the objective function and subjected to certain design constraints. Mathematically the applied optimization problem can be seen as [14]:

$$\min(f(x) = f(x_1, x_2, \ldots, x_n))$$

subjected to the following constraints:

$$g_j(x) \leq 0, \quad j = 1, \ldots, m$$

$$x_i^l \leq x_i \leq x_i^u, \quad i = 1, \ldots, n$$
where: $f(x)$ represents the objective function, in this case the mass of the system; $x$ is the vector of the design variables $x_1, x_2, ..., x_n$; $g(x)$ is the constrain function; $x_l^i$ and $x_u^i$ are the lower and upper bounds for the $i^{th}$ design variables.

Design variables that belong to the vector $x$ are system parameters that can vary to optimize the system performance. For the size optimization, DVs are scalar parameters such as shell thickness, spring stiffness or material properties that affect the system responses. To define the DVs we used DESVAR cards which are related to size properties in the model by using a DVPREL1 or DVPREL2 bulk data entry.

The responses are defined using DRESP1 bulk data entries and represent any value or function that is dependent of the DVs and is evaluated during the solution.

The objective function $f(x)$ and the functions $g(x)$ are structural responses obtained from the finite element analysis. Changing the values of DVs should change the value of the objective function. Constrain functions are defined as a lower bound or upper bound on any response that is dependent of the design variable.

Table 3. Panel thicknesses

<table>
<thead>
<tr>
<th>Panel name</th>
<th>Baseline thickness (mm)</th>
<th>Lower bound (mm)</th>
<th>Upper bound (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>floor</td>
<td>0.9</td>
<td>0.81</td>
<td>0.99</td>
</tr>
<tr>
<td>bulkhead</td>
<td>0.8</td>
<td>0.72</td>
<td>0.88</td>
</tr>
<tr>
<td>roof</td>
<td>0.7</td>
<td>0.63</td>
<td>0.77</td>
</tr>
</tbody>
</table>

For our optimization problem we chose the thicknesses of the panels that vibrate in the $11^{th}$ structural mode as design variables. Table 3 contains the panel name, the baseline thickness and the lower and the higher values respectively.

The objective is to minimize the global structural mass while keeping the magnitude of the sound level at the driver’s ear below 1.3055E-05 MPa at the excitation frequency of 50 Hz. The initial model has a mass of 33.84 kg and the maximum sound level is at 87.09 dB(A).

The optimization process converges in 12 iterations with the sound level at the driver’s ear being reduced from a maximum of 87.09 dB(A) to 66.75 dB(A) at the excitation frequency of interest.

The structural mass is reduced from 33.84 kg to 33 kg. The objective function history is shown in Figure 10. In Figure 11 it can be seen the history of the DVs at each iteration. The sound pressure level at the driver’s ear location for the initial and optimized designs are shown in the Figure 12.

**6. CONCLUSIONS**

In this study, we considered a vibro-acoustic FE model of a simplified passenger compartment. This FE model which represents a coupled fluid–structure system was utilized to simulate the sound pressure level at the driver hearing point due to the harmonic force applied to the geometric center of the firewall panel for the frequency interval of interest. The simulation was performed by using the modal frequency response analysis implemented in Radioss solver, a HyperWorks module. The highest sound pressure level of 87.09 dB(A) was recorded for the excitation frequency of 50 Hz. For this frequency, the modal participation analysis was performed to identify the most influential structural modes to the sound pressure level at the driver’s ear location. The highest contributor was the $11^{th}$ structural mode in which only three structural panels, the floor, the bulkhead and the roof, have a significant motion.

In order to reduce locally the peak of the sound pressure level at the drive’s ear a size optimization problem was formulated and solved by using the...
OptiStruct module. The thicknesses of the floor, bulkhead and roof panels were chosen as design variables. The sound pressure level for the baseline FE model was compared with the sound level for the optimized FE model.

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