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FINITE ELEMENT ANALYSIS FOR INTERIOR BOOMING NOISE REDUCTION IN A TRACTOR CABIN

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ABSTRACT

This paper presents a finite element approach to analyze the "boom" noise for a compact tractor cabin. The tractor cabin is initially designed to have a structure made up of steel beams and aluminum panels, as well as PMAA panels in windshield, backlight and windows. Cavity acoustic modes of the cab are evaluated and the acoustic resonant frequencies are identified. The study on the structural-borne noise from the cabin structural vibration generated by the engine of the vehicle is performed. A coupled-field finite element model, counting the interactions between the air fluid inside the cabin compartment and the cabin exterior structure, is presented for investigating the structural-borne noise in a low frequency range of 20 Hz to 80 Hz. This range has shown strong boom effects. The interior noise level at driver's right ear position is investigated. The peak noise levels at the position are determined. The effects of additional stiffeners and damping layers on the boom noise are also investigated.

INTRODUCTION

Interior noise levels in vehicle passenger compartment have been a long-time concern in automobile industry. Noise plays a significant role in automobile industry because of the market competition in satisfying customer's requirement for high comfortableness in vehicle driving practice. The tractor studied in this paper is a typical light-weighted design, which is managed to be constructed with aluminum panels, PMAA (polymethyl methacrylate) panels and steel-tube columns and beams. The reason for the light-weighted design is not for fuel saving, but for expedient manufacture and easy propulsion. The tractor designed targets at the market of all-terrain vehicles (ATV), and not for large-scale commercial vehicle market. Usually, the major sources of automobile vehicle interior noise are categorized as 1) engine and transmission operation, 2) road excitation and 3) aerodynamic excitation [1]. For the tractor considered, engine excitation of the vehicle is the main vibration source. The tractor features low speed driving in fieldwork, therefore, road excitation and aerodynamic excitation do not contribute overwhelming noise as the engine does.

Boom is defined as the sonic noise due to medium's objectionable response in the frequency ranged from 20 Hz to 80Hz [2]. The main factors affecting the boom noise in a tractor cabin are acoustic cavity resonance and the cabin panel's vibration of various modes. The noise propagating in the vehicle cabin is a complex phenomenon and the contributions of the cabin cavity and the cabin panel have to be jointly considered. For comprehensively understanding the characteristics of the noise, a theoretical approach is almost impossible. In investigating low-frequency noise of vehicle cabins, finite element method is widely accepted as a powerful tool and it has been verified as an effectively accurate technique in noise assessments [3, 4]. It is therefore utilized in the present research. The acoustic finite element analysis for interior noise in the vehicle cabin considered consists of two parts in this paper: modal analysis of cavity acoustic resonance and harmonic analysis of coupled structural-acoustic model of vehicle cabin. The former aims at finding the natural frequencies of the air fluid filling up the interior compartment of the cabin. In the cases that the natural frequencies coincide with the external excitation frequencies or have strong coupling property with structure vibration, intense noise problems can be generated. The investigation of the harmonic analysis demonstrates the interactions between the vehicle cabin structure and the interior air cavity, which couples the structure elements and the acoustic fluid elements. This investigation is to directly show how the structure vibrations affect the sound pressure inside the cabin.

FINITE ELEMENT FORMATION

The governing equation for the sound pressure p in an enclosed cavity can be described as

$$\nabla^2 p - \frac{\partial p^2}{\partial^2 t}\frac{1}{c^2} = 0\,, \tag{1}$$

where, c is the sound speed in fluid medium, t the time, and ∇^2 is the Laplacian operator [3, 5]. The air fluid is assumed to be compressible and inviscid. The air fluid has no mean flow and the density and pressure of the medium are uniform throughout the fluid. With these assumptions, p can be expressed in the following equation for harmonic pressure varying.

$$p = p_0 e^{i\omega t} \tag{2}$$

where p_0 is the amplitude of sound pressure, and ω is the radian frequency. Equation (1) thus reduces to

$$\nabla^2 p_0 + \left(\frac{\omega}{c}\right)^2 p_0 = 0\,. \tag{3}$$

The boundary condition for a boundary surface S_1, which is in small amplitude motion, can be written as

$$\frac{\partial p}{\partial n} = -\rho \ddot{u}\,, \tag{4}$$

where ρ is the fluid density, $\ddot{u}$ stands for the normal acceleration component of the boundary surface and n denotes the normal direction of the surface. Considering harmonic vibration of the cabin structure, u is in the form of

$$u = u_0 e^{i\omega t} \tag{5}$$

For a rigid/hard surface,

$$\partial p / \partial n = 0\,. \tag{6}$$

For a free surface opening to the ambient, the pressure on the boundary surface is

$$p = 0\,. \tag{7}$$

If the cavity volume is discretized and represented by the elements of proper meshing, Equation (1) can be written in a matrix form as

$$\left[M_f\right]\{\ddot{p}\} + \left[K_f\right]\{p\} = [F_f] - (\rho c)^2 [R]^T \{\ddot{u}\} \tag{8}$$

where $\{p\}$ is the sound pressure vector, $\left[M_f\right]$ and $\left[K_f\right]$ are acoustic fluid mass and stiffness matrices, $[F_f]$ is the vector of pressure force applied to the fluid, $\{\ddot{u}\}$ represents the structural acceleration vector, and $[R]^T$ is the transposed matrix that represents the effective surface area associated with each node on the fluid-structure interface. Equation (8) is the basic governing equation for acoustic analysis with finite element approach. In solving this equation for eigenfrequencies of air cavity with rigid wall, the right hand side of Equation (8) becomes 0. Solving the acoustic problem is then reduced to solving a Helmholtz equation in finite element form.

Taking the structural-acoustic interaction into consideration, at the interface of the fluid and the structure, the acoustic pressures exert a force on the structure surfaces whereas the structural motions apply excitations to the air fluid. The structural finite element equations in this case can be expressed in the following form.

$$\left[M_s\right]\{\ddot{u}\} + \left[K_s\right]\{u\} = [R]\{p\} + [F_s]\,, \tag{9}$$

where $\left[M_s\right]$ and $\left[K_s\right]$ are structural mass and stiffness matrices, and $[F_s]$ is the external force vector representing exertions applied on the structure.

In Equations (8) and (9), $\{u\}$ and $\{p\}$ are the unknown vectors to be solved. The equations (8) and (9) can be combined and rewritten as

$$\begin{bmatrix} [M_s] & [0] \\ (\rho c)^2 [R]^T & [M_f] \end{bmatrix} \begin{bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{bmatrix} + \begin{bmatrix} [K_s] & -[R] \\ [0] & [K_f] \end{bmatrix} \begin{bmatrix} \{u\} \\ \{p\} \end{bmatrix} = \begin{bmatrix} [F_s] \\ [F_f] \end{bmatrix} \tag{10}$$

Several finite element software packages available in the market provide the capability of analyzing the numerical models expressed mathematically in the form of Equation 10 [6, 7]. NASTRAN and ANSYS are two of them.

CAVITY ACOUSTIC RESONANCE

For modeling the tractor cabin, a three-dimensional finite element model is built with utilization of Element Fluid30 of ANSYS. The elements used have eight corner nodes with four degrees of freedom per node: translations in the nodal x, y and z directions and pressure [5]. The translations, however, are applicable only at nodes that are on the interface. For cavity acoustic resonance analysis, the "structure absence" option of Fluid30 is chosen in application. The option of structure absence to Element Fluid30 makes the finite element matrices symmetric as only one degree of freedom, pressure, is included in the model. This arrangement provides the advantage of reducing the computation time.

The acoustic resonance concerned in the present research is to the cavity featuring rigid boundaries. With this consideration, the right hand side of Equation (8) is set to be zero, and then the only data necessary to compute for the resonance of the vehicle compartment are the geometric data of the compartment [3]. As such, Equation (8) reduces to the form of

$$\left[M_f\right]\{\ddot{p}\} + \left[K_f\right]\{p\} = [0]\,. \tag{11}$$

Due to the irregular shape of the cavity contained in the vehicle cabin, the model is meshed with tetrahedral fluid elements of free mesh. The finite element model thus established is as shown in Figure 1. One may note the finer meshing at the sharp corners of the cabin.

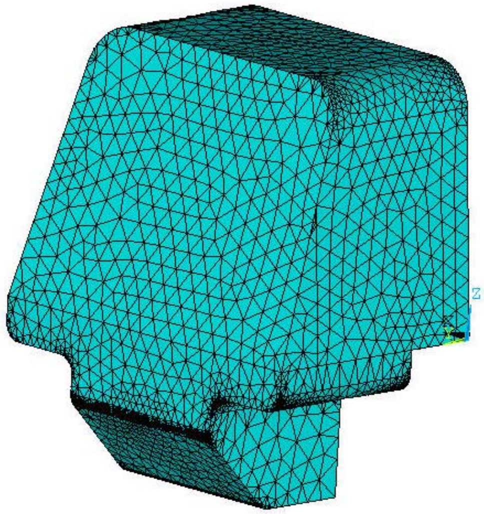

Figure 1: Finite Element Model of Air Cavity.

For evaluating the medium response of an irregular shaped air volume as the one considered by the present research, numerical simulation is almost the unavoidable approach, as the resonant frequencies and mode shapes of the air volume can hardly be predicted by pure analytical approaches. The modal analysis for finding the natural frequencies of the acoustic cavity is conducted with different mesh densities. Totally five differently meshed models (Model 1 to 5 as listed in Table 1) are established and computed. The number of elements for the five models is 3348, 7884, 12536, 30093 and 51811 respectively. The lowest ten resonant frequencies of the cavity and the corresponding total number of elements of the model are tabulated in Table 1.

Item	Model 1	Model 2	Model 3	Model 4	Model 5
1st	0.39E-05	0.19E-05	0.24E-05	0.23E-05	0.24E-05
2nd	109.57	108.66	108.35	107.92	107.79
3rd	122.96	121.64	121.27	120.86	120.76
4th	177.16	173.08	172.22	171.04	170.72
5th	190.03	184.66	183.57	182.18	181.83
6th	195.68	192.34	191.25	189.92	189.50
7th	238.20	230.12	228.56	225.88	225.18
8th	247.79	238.22	236.22	233.52	232.79
9th	256.99	248.41	246.69	244.05	243.36
10th	267.19	259.94	257.78	255.74	255.21
No. of Elements	3348	7884	12536	30093	51811

Table 1: The First Ten Resonant Frequencies (Hz) of the Cabin Air Cavity in Different Mesh Densities.

The results in Table 1 reveal the fact that the finer the meshing, the more accurate in obtaining higher resonant frequencies. In performing the numerical calculations, it is found that at least 4 elements within a wavelength are needed. Accurate and stable results are obtained if 6 elements in the finite element model are employed. This agrees with that reported by [8, 9]. Therefore, as the observing frequency increases, the accuracy of the results depends more on how fine the model can be meshed. In Table 1, the second modal frequency obtained by the model with total elements of 3348 and 51811 are 109.57Hz and 107.79 Hz respectively. The difference between the two is only 1.78 Hz. But to the tenth modal frequency, 267.19 Hz and 255.21 Hz are obtained, the variation increases to 11.98 Hz. While there is no conceptual and theoretical difficulty in applying finer mesh to the finite element model, the drawback is the cost of the computation time [3].

The first acoustic mode generated is always a zero frequency, i.e., the mode corresponding to the static compressibility of the air according to an ideal gas [2]. As shown in Table 1, the results of first acoustic mode for the same air cavity in different mesh densities are all approximately equal to zero as they should be. With a mathematical interpretation, this implies a trivial solution for the equations. The first non-zero acoustic modal frequency is 107.79 Hz. Figure 2 shows the corresponding mode shape. In the figure, MX and MN stand for maximum and minimum values respectively.

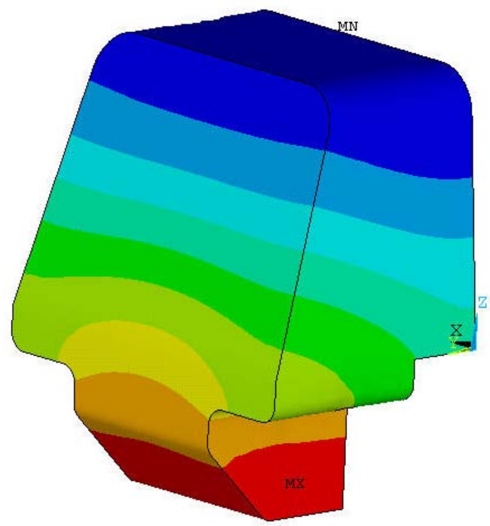

Figure 2: Acoustic Mode Shape of the Air Cavity at 107.79 Hz.

According to the abovementioned results, the first non-zero resonant frequency of the air cavity is 107.79 Hz. The small size of the compact tractor cabin brings this high first non-zero modal frequency, comparing to some published results of vehicle compartments in different shapes, the corresponding first non-zero modal frequency is normally a frequency around 50 Hz to 80 Hz. Obviously, the above-mentioned acoustic modes have no significant effects to the boom noise with frequencies in between 20 Hz to 80 Hz.

STRUCTURAL-ACOUSTIC SYSTEM MODEL

The structure components of the cabin are shown in Figure 3 and Figure 4. In order to quantify the sound pressure at driver's right ear position when the cabin is excited by harmonic force of different frequencies, the coupled-field structural-acoustic model of the tractor cabin is established. The model is as shown in Figure 5. Shell elements and beam elements are introduced to mesh the structure components. The interior air cavity is also meshed with Fluid30. The fluid layer that has interface with the structure components are chosen to have structure present option featuring the degree of freedom of both the displacements in x, y, z directions and pressure. This makes the coupling between the fluid and the structure applicable. The air volume inside of this structure-present fluid layer is meshed with structure-absent elements without employing the displacement [5]. This helps to reduce the size of the total matrix for computation. Nevertheless, the computation is still a very time-consuming task. Since the structure has to be meshed sufficiently fine to provide accurate response of the structure, the interior air cavity is meshed with large quantity of elements accordingly. It should be noted, in ANSYS, the air cavity is meshed based on the exterior structure elements to assure the structure elements and the outermost fluid elements have the same nodes to realize the coupling. Thus, the fluid elements are much more than enough to satisfy the requirement mentioned in last section that 6 elements in a wavelength provide sufficiently accurate results.

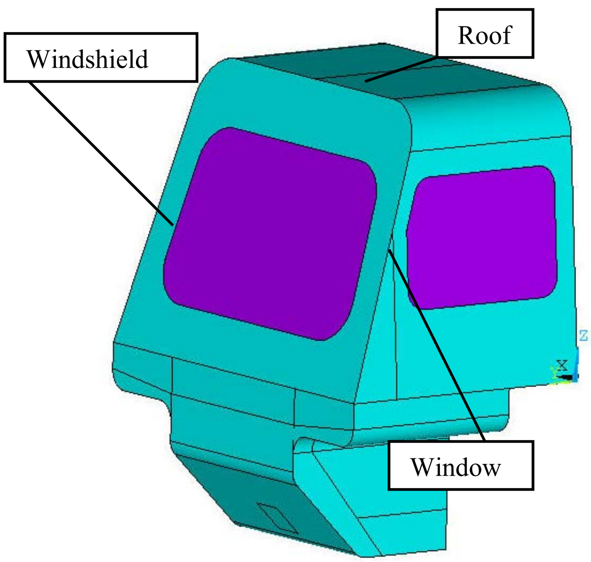

Figure 3: Cabin Viewed from Left-front.

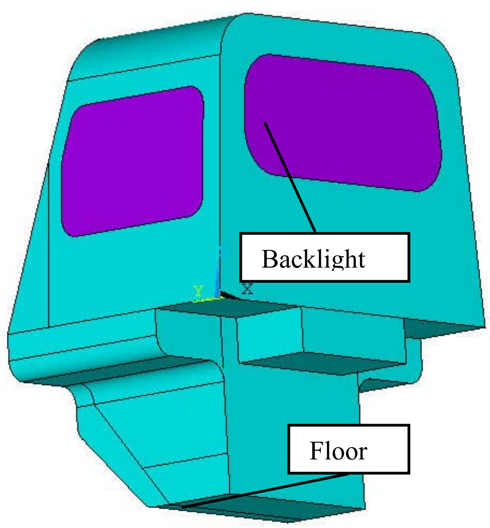

Figure 4: Cabin Viewed from Left-back.

In the harmonic analysis for obtaining the sound pressure inside the cabin, the excitation of unit amplitude is acted at the location near the left-bottom corner of the cabin structure, where two beams jointed together (as indicated in Figure 5). The excitation acted is a sinusoidal loading. The vibration excitation sweeps in the boom frequency with the range from 20 Hz to 80 Hz.

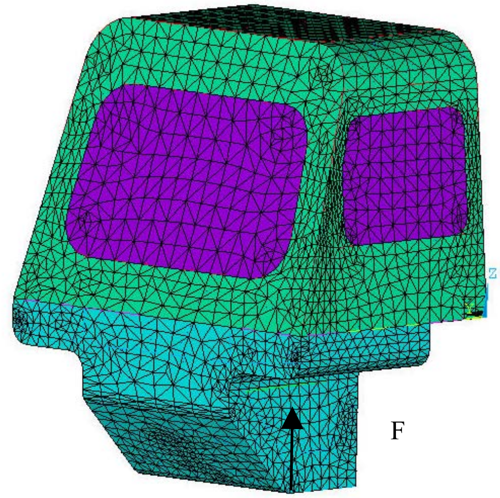

Figure 5: Excitation Force on the Meshed Structure Model.

In the model, only structural excitation force is introduced, therefore, $\left[F_f\right]$in Equation (9) is equal to $[0]$, as no external sound pressure excitation forces. Equation (9) is then reduced to the following form

$$\begin{bmatrix} [M_s] & [0] \\ (\rho c)^2[R]^T & [M_f] \end{bmatrix} \begin{bmatrix} \{\ddot{u}\} \\ \{\ddot{p}\} \end{bmatrix} + \begin{bmatrix} [K_s] & -[R] \\ [0] & [K_f] \end{bmatrix} \begin{bmatrix} \{u\} \\ \{p\} \end{bmatrix} = \begin{bmatrix} [F_s] \\ [0] \end{bmatrix}. \quad (11)$$

In the present research, the sound pressure values at the driver's right ear position are transferred to sound pressure levels by employing the following formula, upon the completion of the computation with the finite element mode.

$$L_{SP} = 20 \times \log\left(\frac{p_{rms}}{\left|p_{ref}\right|}\right), \quad (12)$$

where L_{SP} is the sound pressure level, p_{rms} is the root mean square pressure, p_{ref} represents the reference pressure (in air, it defaults to 20 x 10^{-6} P_a) [5]. The sound pressure levels are plotted in the following Figure 6, based on the FE results obtained.

As can be seen from Figure 6, the peak sound pressure level in this frequency range locates at 41 Hz with a sound pressure level of 65.5 dB. The sound pressure level distribution inside the cabin is displayed in Figure 7. The maximum sound pressure level takes place in the front part of the vehicle cabin as indicated in Figure 7.

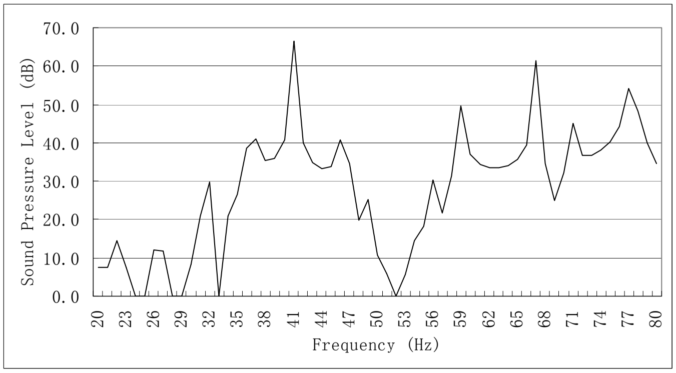

Figure 6: Sound Pressure Level at Driver's Right Ear Position.

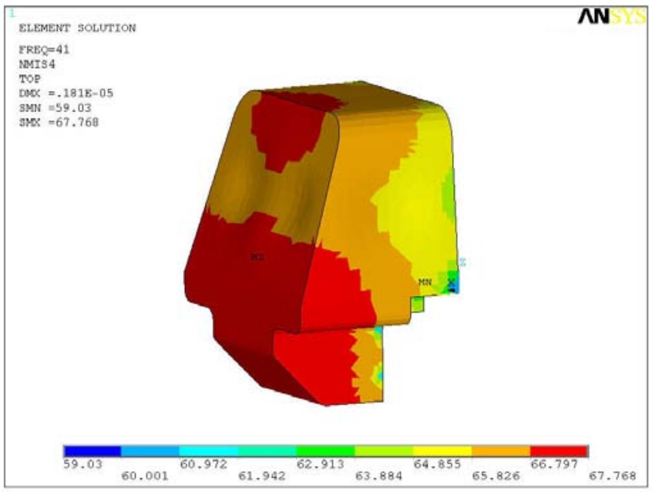

Figure 7: Sound Pressure Level Distribution in the Cabin.

The peak sound pressure level values reflect where the resonant frequencies of the structural-air model are located. For a worst scenario, the excitation acted on the cabin structure is in frequencies that coincide with or are very close to those with peak sound pressure levels. The noise in the cabin will be dramatically amplified in this scenario to produce extremely annoying booming in the tractor cabin. For example, the spectrum of vibration excitation shows a peak value at the frequency of 41 Hz, resonance occurs and the interior noise level of the same frequency will be dominant. To avoid the resonance, it is necessary to control the vibration of this frequency. The most effective method to reduce the noise level for this case is to make the resonant frequency to be relocated. Further investigation on the structural strains of the model vibrating at 41 Hz reveals that the resonant vibration takes place at the windshield and backlight, which is illustrated in Figure 8. This naturally leads to the considerations for stiffening the windshield and backlight. The two panels are modified by adding some beams on the local panels, shown in Figure 9.

Afterwards, the harmonic analysis is conducted for the modified model in the same procedure. The frequency response of the new model is presented in Figure 10, together with the curve from the previous model plotted for comparison. The acoustic response at 41 Hz is lowered from 66.5 dB to 27.8 dB. The resonant frequency is moved to 38 Hz.

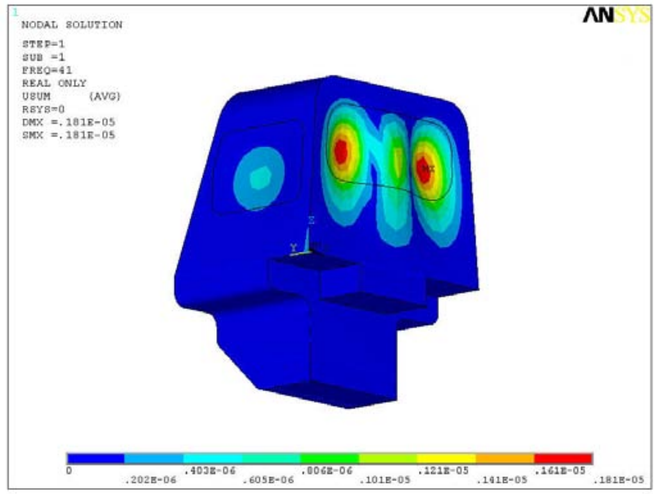

Figure 8: Displacement Diagram of the Cabin.

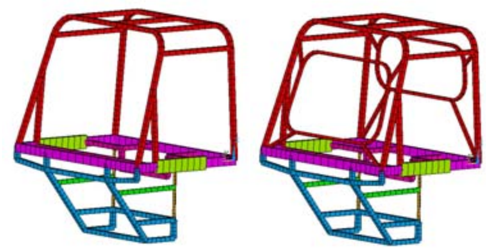

Figure 9: Model with and without Structure Stiffeners.

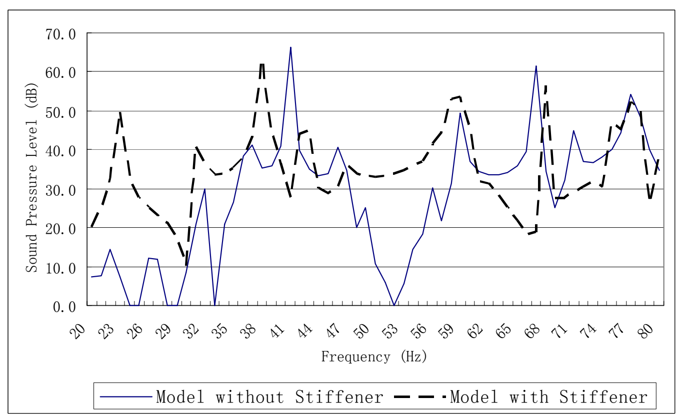

Figure 10: Sound Pressure Levels at Driver's Right Ear Position.

For further noise reduction, damping layer can be applied to the panels. Damping layers adhered on the surfaces the structural panels provide extra damping to the cabin. In this paper, only structural damping effect is considered. In low frequency range, the structural damping effect brought by viscoelastic damping layer can be considered as a constant [10, 11]. Different damping produces vary in the application. Normally, the damping ratio falls in the scope of 4% to 8% [12]. The damping layer is applied to aluminum panels only in the cabin model.

4% and 8% of damping ratios are taken into consideration in the computation. The results are presented in the below Figure 11.

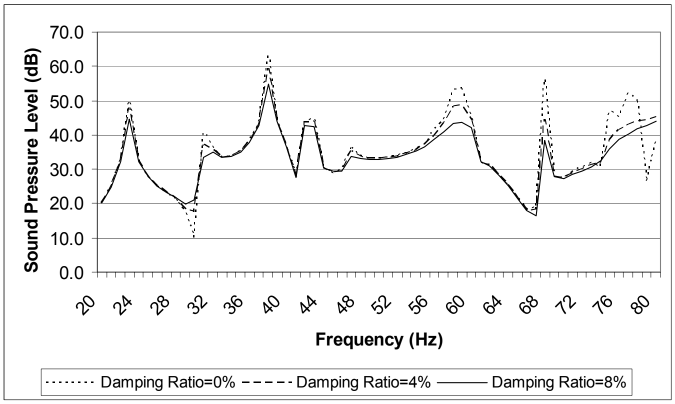

Figure 11: Acoustic Responses of the Stiffened Model with Damping Layer Attached.

The structural damping helps to reduce the resonant peak values dramatically as shown in Figure 11. The reduction reaches 5 to 18 dB in various resonant frequencies. But in non-resonance frequencies, the noise reduction is not obvious.

CONCLUSIVE REMARKS

The methodology and techniques presented in the paper provide useful acoustic assessment tools for vehicles designs at the stage of NVH (Noise, Vibration, Harshness) analysis, especially for the vehicle designs to which noise reduction is required. The finite element model established can be applied to compute cavity resonances with vehicle structure design data. The coupled structural-acoustic model developed is proven to be helpful in analyzing the vibration-induced noise in a vehicle cabin. The interior acoustic response of the cabin excited by harmonic forces can be conveniently predicted with the models established. As indicated in the research results, noise level of a cabin at the resonant frequency can be reduced with the supplementary of structure stiffeners and/or damping layers. The results obtained in the present research are instructive to structure modification for acoustic reason. Findings of the present research may also be implemented in the vibration and acoustic experiments for testing a prototype vehicle, as the communications between the numerical results and experimental data will make the modifications more efficient and effective.

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