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STUDY ON CALCULATING ACOUSTIC SENSITIVITY OF CAR CAB BASED ON FINITE ELEMENT METHOD

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Abstract

The comfort in the car cab is a key element for modern car, which could determine the customers' viewpoint to the car. The noise of frequency range from 30Hz to 200Hz in the car cab is called booming noise. This kind of noise could make passengers feel oppression, fret and tired. In order to decrease booming noise pressure, acoustic design variables of car cab, such as car cab geometry, the thickness of panel, the surface admittance of panel etc, should be designed optimization. Acoustic sensitivity of car cab is defined as the sensitivity of the interior sound pressure with respect to an acoustic design variable, which is very useful characteristic for design optimization of car cab. In this paper, firstly, finite element method (FEM) model is given for calculating booming noise pressure of car cab. Secondly, based on above FEM model, numerical method for acoustic sensitivity is presented. Thirdly, acoustic sensitivity is decomposed to acoustic magnitude sensitivity and acoustic phase sensitivity. Lastly, acoustic sensitivity of one NASTRAN model of car cab is calculated.

1. INTRODUCTION

The comfort in the car cab is a key element for modern car, which could determine the customers' viewpoint to the car [1]. Two main ways on generation noise in car cab are vibration of car body and propagation sound from opening of car body. Engine and excitation by road produce vibration of car body. Noise in car cab generated by vibration of car body is called structure-borne noise. This kind of noise could make passengers feel oppression, fret and tired. Because frequency of structure-borne noise is always between 30Hz and 200Hz, finite element method (FEM) or boundary element method (BEM) based on wave acoustics is suitable analysis tool [2, 3]. Most research works are to predict noise based on known structure and boundary condition of car body [2, 4]. Designers hope to decrease noise by optimization some kind of structure and boundary condition of car body on design stage.

Acoustic sensitivity of car cab is defined as the sensitivity of the interior sound pressure with respect to an acoustic design variable, which is very useful characteristic for design optimization of car cab. One kind of numerical method for acoustic sensitivity is presented based on FEM model.

2. FEM MODEL

Vibration of car body can be regarded as sound source. Media mass of unit volume provided by the sound source is $\rho_0 q(\vec{r},t)$ in unit time. Then using wave equation about sound pressure $p(\vec{r},t)$, the sound field situation of car cab can be expressed as:

$$\nabla^2 p - \frac{1}{{c_0}^2}\frac{\partial^2 p}{\partial t^2} = -\rho_0 \frac{\partial q}{\partial t} \tag{1}$$

Where c_0 is transmitting velocity of sound in air, ρ_0 is air density.

If sound source makes harmonic vibration, the function of sound source $q(\vec{r},t)$ is written as:

$$q(\vec{r},t) = q_0(\vec{r})\mathrm{e}^{\mathrm{j}\omega t} \tag{2}$$

Where ω is frequency.

Due to the system considered is linear, and then sound pressure in each point of car compartment vibrates with frequency of sound source. Sound pressure is given as

$$p(\vec{r},t) = p(\vec{r},\omega)\mathrm{e}^{\mathrm{j}\omega t} \tag{3}$$

Substituting equations(2) and (3)into (1), equation (1) can be written as Helmholtz equation:

$$\nabla^2 p_\omega(\vec{r}) + \left(\frac{\omega}{c_0}\right)^2 p_\omega(\vec{r}) = -\mathrm{j}\rho_0\omega q_\omega(\vec{r}) \tag{4}$$

After that, the question in time domain can be transferred into frequency domain. At the same time, equation (4) is also the control equation of building FEM equation. In order to simplify the writing, let: $p = p_\omega(\vec{r})$， $f = -\mathrm{j}\rho_0\omega q_\omega(\vec{r})$. Then, Helmholtz equation is simplified as

$$\nabla^2 p + k^2 p = f \tag{5}$$

In our research, two kinds of boundaries, rigid and absorption walls, are taken into account. S_1 is rigid wall, and S_2 is absorption wall. Boundary conditions are given as

$$\frac{\partial p}{\partial n} = 0 \qquad \text{On } S_1 \tag{6}$$

$$\frac{\partial p}{\partial n} = -\mathrm{j}\rho\omega\frac{p}{z_s} \quad \text{On } S_2 \tag{7}$$

Where n is direction of outward normal to the wall, z_s is sound impedance of absorption wall S2

Using variation principle based on equation(5), (6)and(7). Element's FEM equation can be written as:

$$\mathbf{K}^{(e)}\mathbf{p}^{(e)} + \mathrm{j}\rho\omega\mathbf{C}^{(e)}\mathbf{p}^{(e)} - k^2\mathbf{M}^{(e)}\mathbf{p}^{(e)} = \mathbf{F}^{(e)} \tag{8}$$

Where local matrixes are:

$$\mathbf{K}^{(e)} = \int_{V^{(e)}} (\nabla\mathbf{N})(\nabla\mathbf{N})^T dV$$

$$\mathbf{M}^{(e)} = \int_{V^{(e)}} \mathbf{N}\mathbf{N}^T dV$$

$$\mathbf{C}^{(e)} = \int_{S_2^{(e)}} \mathbf{N}\mathbf{N}^T / z_s dS$$

$$\mathbf{F}^{(e)} = \int_{V^{(e)}} f\mathbf{N}dV \tag{9}$$

Where $\mathbf{N}$ is column vector of element shape function, $\nabla\mathbf{N}$ is derivative matrix of $\mathbf{N}$

According to rule of elements' combination, system global matrixes are given as:

$$\mathbf{M} = \sum_e \mathbf{M}^{(e)}, \quad \mathbf{K} = \sum_e \mathbf{K}^{(e)}$$
$$\mathbf{C} = \sum_e \mathbf{C}^{(e)}, \quad \mathbf{F} = \sum_e \mathbf{F}^{(e)} \tag{10}$$

Then, system global FEM equation in car compartment is given as:

$$\left(\mathbf{K} + \mathrm{j}\rho\omega\mathbf{C} - k^2\mathbf{M}\right)\mathbf{p} = \mathbf{F} \tag{11}$$

ACOUSTIC SENSITIVITY OF CAR CAB

If structure and boundary condition of car body are $x_i (i = 1, 2, \cdots n)$, the derivative of equation (11) to x_i is:

$$\left(\mathbf{K} + \mathrm{j}\rho\omega\mathbf{C} - k^2\mathbf{M}\right)\frac{\partial \mathbf{p}}{\partial x_i} + \left(\frac{\partial \mathbf{K}}{\partial x_i} + \mathrm{j}\rho\omega\frac{\partial \mathbf{C}}{\partial x_i} - k^2\frac{\partial \mathbf{M}}{\partial x_i}\right)\mathbf{p} = \frac{\partial \mathbf{F}}{\partial x_i} \tag{12}$$

Because **F** in equation (11) is vector of excitation generated by sound source, **F** is no relation with structure and boundary condition of car body and sensitivity to x_i is zero.

With equation(12), acoustic sensitivity $\mathbf{s}_i$ can be calculated as:

$$\mathbf{s}_i = -\left(\mathbf{K} + \mathrm{j}\rho\omega\mathbf{C} - k^2\mathbf{M}\right)^{-1}\left(\frac{\partial \mathbf{K}}{\partial x_i} + \mathrm{j}\rho\omega\frac{\partial \mathbf{C}}{\partial x_i} - k^2\frac{\partial \mathbf{M}}{\partial x_i}\right)\mathbf{p} \tag{13}$$

If sound pressure $\vec{p} = \left(p_r + j \cdot p_i\right)$ and sensitivity $\vec{s} = \left(s_r + j \cdot s_i\right)$ are respectively calculated based on equation (11) and (12).

Both of figures 1 and 2 describe the relation between sound pressure $\vec{p}$ and sensitivity $\vec{s}$. But one important phenomenon is observed that sound pressures $\vec{p} + \Delta\vec{p}$ of two situations are different even if sensitivity $\vec{s}$ are the same.

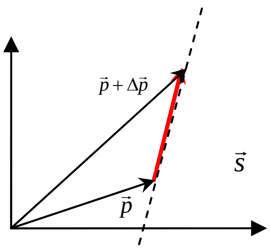

Fig. 1

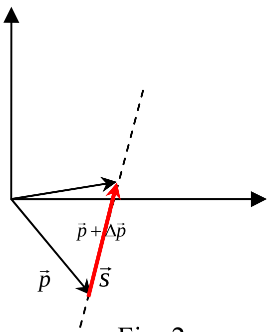

Fig. 2

With geometry knowledge, we can find:

If dot product of sound $\vec{p}$ pressure and sensitivity $\vec{s}$ is right, sound pressure magnitude will increase;

If cross product of sound pressure and sensitivity $\vec{s}$ is right, sound pressure phase will increase;

Based on above analysis, two kinds of sensitivity are proposed. Sound pressure magnitude sensitivity is:

$$s_a = \frac{\vec{p} \cdot \vec{s}}{\|\vec{p}\|} = \frac{p_r s_r + p_i s_i}{\sqrt{p_r^2 + p_i^2}} \tag{14}$$

Sound pressure phase sensitivity is：

$$s_{\phi} = \frac{\left\| \vec{p} \times \vec{s} \right\|}{\left\| \vec{p} \right\|} = \frac{p_r s_i - p_i s_r}{\sqrt{p_r^2 + p_i^2}} \tag{15}$$

3. EXAMPLE

In this example, Sensitivity analysis is taken in one car cab. FEM model is built with NASTRAN. Sound pressure and sensitivity are calculated with SYSNOISE.

Material properties of this car body are:

Young's modulus $E = 2.1 \times 10^5 \, Mpa$;

Poisson ratio $\varepsilon = 0.3$;

Density $\rho = 7800 kg/m^3$

Height of car cab $h = 1m$

Length $l = 2.67m$

Bottom panel of car cab is vibrated and magnitude is 0.001m/s.

Sound pressure sensitivity at node number 299 of FEM Grid to sound impedance of boundary is calculated. Node number 299 is close to drivers' ear. Initial value of Sound impedance at top of car body, front and back seat is $500 \, kg/(m^2 \cdot s)$.

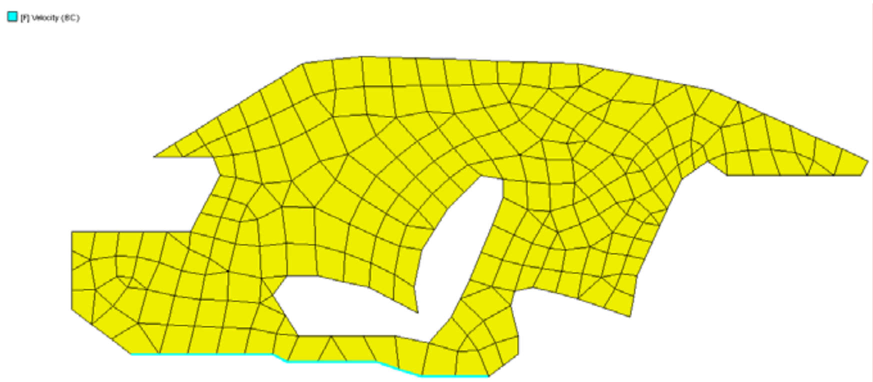

Fig. 3 Grid of Car Cab and vibrated bottom panel

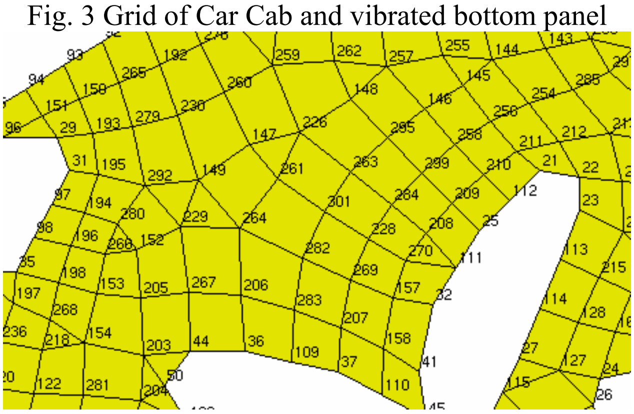

Fig. 4 Node Number of FEM Grid

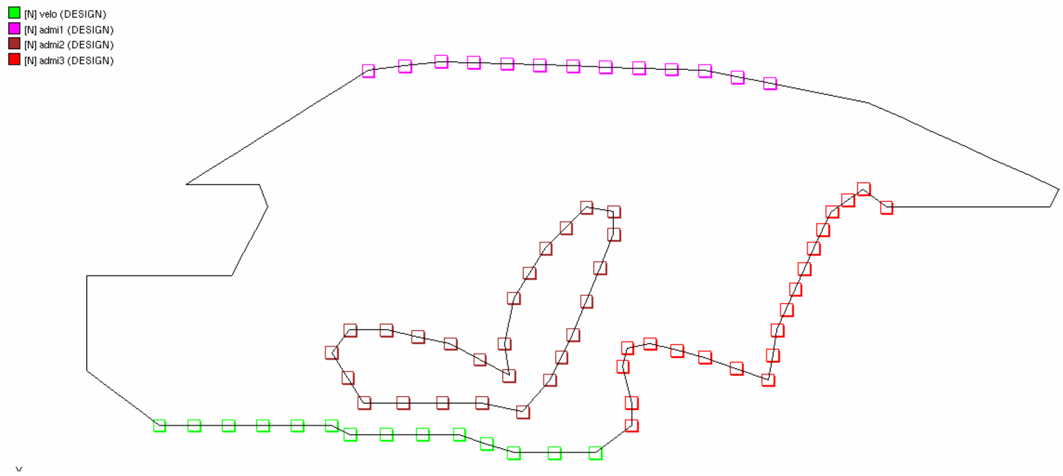

Fig. 5 Sound Impendence Positions

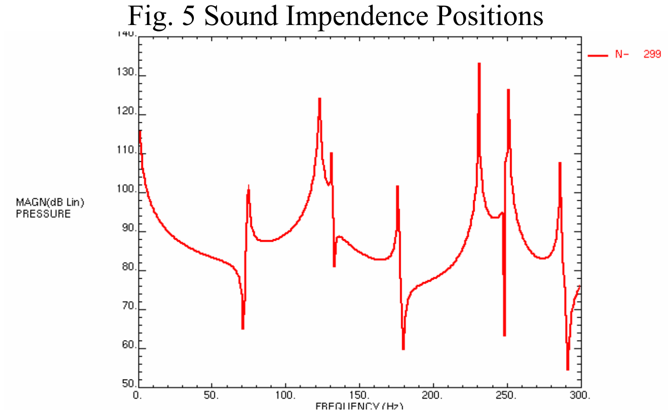

Fig. 6 Sound Pressure Magnitude at node number 299

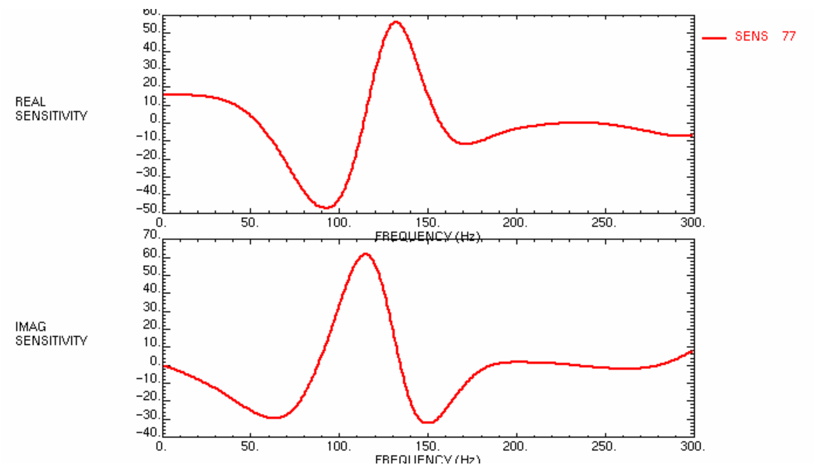

Fig. 7 Sensitivity to sound impendence of top

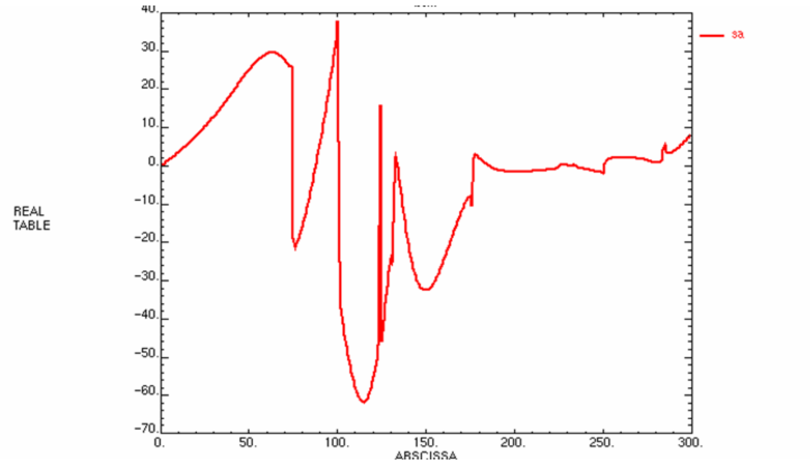

Fig. 8 Sound Pressure Magnitude Sensitivity

Figure 8 show the sound pressure Magnitude sensitivity to sound impendence of top. From this figure, we clearly know how to change sound impendence of top for decreasing noise. For example, in order to decrease noise at 50Hz, sound impendence should be decreased.

4. CONCLUSIONS

In this paper, based on FEM model numerical method for acoustic sensitivity is presented. Comparing two situations, we find that same sensitivity value can not match along with the same change of sound pressure. In order to resolve this problem, we propose to decompose acoustic sensitivity to acoustic magnitude sensitivity and phase sensitivity. At Last, acoustic sensitivity of one NASTRAN model of car cab is calculated.

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REFERENCES

[1] E. Granier, M. Kleinner, B. Dalerbäck, P. Svensson, "Experimental Auralization of Car Audio Installations", J. Audio Eng. Soc., 44, 835-849 (1996)

[2] K. Kleiner, B.-I. Dalenbäck, P. Svensson, "Auralization—Overview", J. Audio Eng. Soc., 41,816-875 (1993)

[3] Fangli Ning, Juan Wei, "One New Adaptive Finite Element for Calculating Sound Transfer Function in Small Enclosures", *Proceedings of the Tenth International Congress on Sound and Vibration* (ICSV10), 7-10 July 2003, Stockholm, Sweden

[4] Polack J D, "Auralization in Scale Models: Processing of Impulse Response", J. Audio Eng. Soc., 41,939-945(1993)