inter•noise 96
25th Anniversary Congress – Liverpool

INCE: 25.2

EFFECT OF AIR-STRUCTURAL COUPLING ON THE MEDIUM FREQUENCY SOUND FIELD RESPONSE IN ENCLOSURES

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1. INTRODUCTION

The characteristics of sound field in enclosures are affected by absorption and vibrational properties of boundary structures from which they are formed. Most of these structures such as panels and partitions found in real enclosures like buildings and vehicles are often elastic and their perturbational/coupling effect cannot be isolated from the vibrational motion of boundary-induced acoustic modes in the enclosed sound field. Acoustic-structural coupling problems have been well studied in both the low and high frequency regimes. The sparse distribution of acoustic modes in the low frequency regime indicates that the sound field response can be characterised in terms of modal resonance, modal decay time and mode shape of the dominant modes of the acoustic-structural coupled system [1]. In the high frequency regime, there are difficulties in the determination of modal information of each dominant mode due to the large modal density of the sound field and the Statistical Energy Analysis (S.E.A.) suggests that modal details are not required. The effect of acoustic-structural coupling on the sound field response behaviours in this frequency regime is related to the averaged behaviour of the modes [2]. At medium frequencies, the sound field behaviour can neither be described fully by a limited number of individual modes nor by the statistical analyses. A good understanding of the medium frequency response of the sound field in enclosures becomes necessary before any effective model for practical prediction can be developed. In this paper, the effect of acoustic-structural coupling on the medium frequency steady-state sound pressure response in an enclosure is analysed using the Modal-Interaction Model and explained using numerical examples.

2. REVIEW OF THE MODAL-INTERACTION MODEL

An acoustic-structural coupled system consists of a sound field described by the sound pressure $p(\vec{r},\omega)$ and a vibration field in the flexible boundary structure described by the structural velocity $v(\vec{\sigma},\omega)$. By expressing these two quantities in terms of their orthogonal expansions and using the Green's function method [3], the complex modal coefficients of the sound pressure in the enclosure can be derived as :

$$\begin{bmatrix} P_1 \\ \vdots \\ P_N \end{bmatrix} = \begin{bmatrix} 1+\sum_{j=1}^{M}\frac{B_{1,j}B_{j,1}}{\chi_{a1}\chi_{sj}} & \cdots & \sum_{j=1}^{M}\frac{B_{1,j}B_{j,N}}{\chi_{a1}\chi_{sj}} \\ \vdots & \ddots & \\ \sum_{j=1}^{M}\frac{B_{N,j}B_{j,1}}{\chi_{aN}\chi_{sj}} & & 1+\sum_{j=1}^{M}\frac{B_{N,j}B_{j,N}}{\chi_{aN}\chi_{sj}} \end{bmatrix}^{-1} \begin{bmatrix} Q_1 \\ \vdots \\ Q_N \end{bmatrix}. \tag{1}$$

Here the sound field is directly excited using a distributed steady-state acoustic source. The quantities in Eq.(1) are as follows :

$$\chi_{ai} = \frac{jM_{ai}}{\omega\rho_0 c_0 A_s}(\omega_{ai}^2 - \omega^2 + j\eta_{ai}\omega_{ai}^2), \qquad Q_i = \frac{-\rho_0 c_0}{A_s\chi_{ai}}\int_{V_0} q\Phi_i \ dV,$$

$$\chi_{sj} = \frac{jM_{sj}}{\omega\rho_0 c_0 A_s}(\omega_{sj}^2 - \omega^2 + j\eta_{sj}\omega_{sj}^2), \qquad B_{j,i} = \frac{1}{A_s}\int_{A_s} \Phi_i(\vec{r})S_j(\vec{\sigma}) \ d\sigma$$

where $M_{ai}, M_{sj}, \omega_{ai}, \omega_{sj}, \eta_{ai}, \eta_{sj}, \Phi_i(\vec{r}), S_j(\vec{\sigma})$ are acoustic and structural modal masses, resonance frequencies, loss factors and mode shape functions respectively. $q(\vec{r},\omega)$ is the volume velocity of the sound source per unit volume of the enclosure. The acoustic amplitude and phase response at any position in the enclosure can then be evaluated from

$$p(\vec{r},\omega) = \sum_{i=1}^{N} P_i(\omega)\Phi_i(\vec{r}). \tag{2}$$

3. RESULTS AND DISCUSSION

A rectangular panel-cavity system similar to that in reference [1] is used in this study. A monopole source located at a cavity corner is used to drive the enclosed sound field. For medium frequency analysis, the excitation frequencies are chosen such that they are below the Schroeder cut-off frequency [4] of the cavity but above which acoustic modes are no longer well-separated.

Figure 1 shows the frequency response of the uncoupled sound field at a cavity corner in a medium frequency range. Also shown in Fig.1 is the frequency response of the sound field in a coupled panel-cavity system for several different panel thickness. It can be seen that the sound field response is extremely sensitive to the panel modal density (even a change of $2\text{x}10^{-4}$ Hz^{-1} in modal density or $2\text{x}10^{-5}$ m in panel thickness can cause significant changes in the sound field response). The characteristic of the response is controlled by either the modal domination mechanism or by the modal superposition mechanism and dependent on the modal distribution of the panel. The previous mechanism is defined as the case where there is only one dominant acoustic mode to the sound field response with significant contribution from other overlapped modes. The latter mechanism is the case where the amplitude responses of the overlapped acoustic modes are comparable. The total sound field response is then related to the averaged superposition of the modes. Results for the modal decomposition of the sound field for the overlapped modes at the uncoupled response maximum (f=825 Hz) are shown in Fig.2 for different panel thickness. At a certain panel thickness (eg. 5.76 mm), the amplitude of one mode is greater than the others and the resultant response is a maximum although the modes have out-of-phase superposition (modal domination mechanism). When the panel modal density is adjusted (eg. 5.8 mm) such that the amplitudes of the

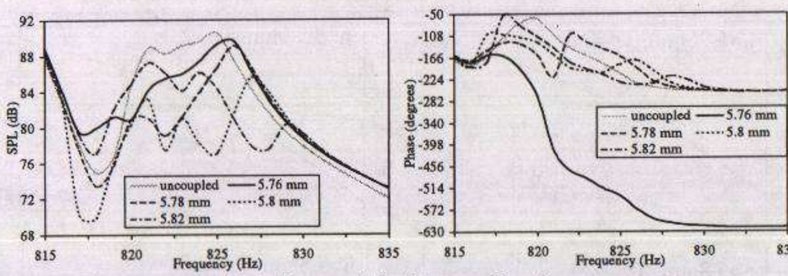

Fig. 1: Frequency response of the sound field in a medium frequency range.

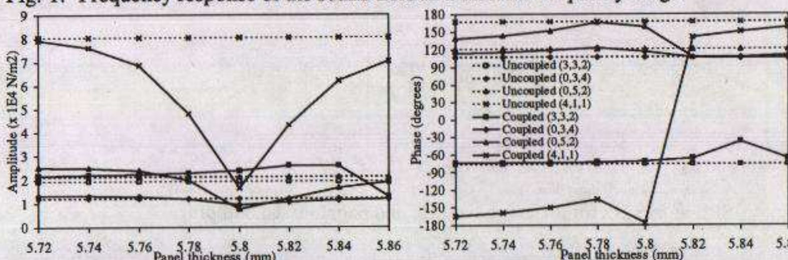

Fig. 2: Modal amplitude and phase of sound pressure at 825 Hz.

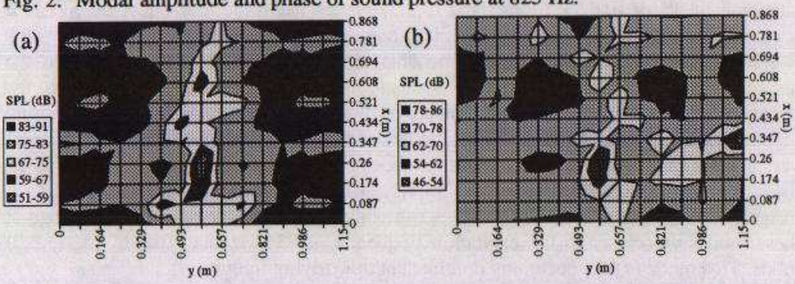

Fig. 3: Sound pressure level distributions of the sound field across the $z=L_z$ plane for (a)h=5.72 mm and (b)h=5.8 mm both at 825 Hz.

modes are comparable, out-of-phase superposition now yields a minimum in the total response (modal superposition mechanism). SPL distribution of the sound field is also affected by the change in panel modal density. When the sound field is controlled by the modal domination mechanism, the mode shape of the dominant mode (mode (4,1,1)) can be observed in the SPL response (see Fig.3a) which indicates that the sound field is still responding modally. However the disturbance from modal superposition with other overlapped modes causes this mode shape to be distorted and thus the modal response is termed as weak. When modal superposition mechanism is dominant, the response becomes more uniform (Fig.3b).

Fig.4 shows the significance of both acoustic and panel modal damping to the sound field response. If the overlapped acoustic modes are not well-coupled to the panel modes in terms of natural frequencies and mode shape matching , the effect of panel modal

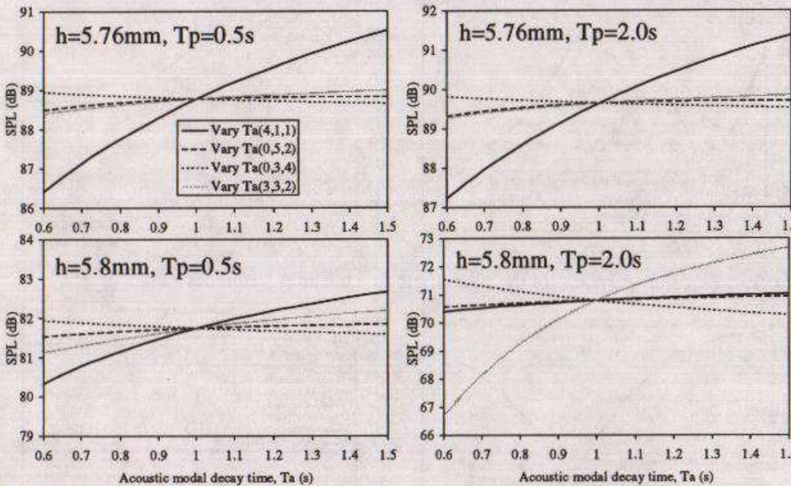

Fig. 4: SPL at 825 Hz for different acoustic and panel modal damping.

damping on the sound field control mechanism is insignificant (eg. for h=5.76 mm, changing of Ta of (0,5,2), (0,3,4) and (3,3,2) modes has minimum effect on SPL). If acoustic-panel modes are well-coupled, it is possible for the sound field response which is modal superposition controlled to become modal domination controlled when the response amplitudes of well-coupled acoustic modes are all attenuated due to the decrease in panel damping which indicates that the panel resistance to acquiring vibrational energy has decreased (eg. for h=5.8 mm and Tp=0.5s, the significance of variation in Ta of the overlapped acoustic modes on SPL is comparable. When Tp=2.0s, the effect of variation in Ta of the (4,1,1), (0,5,2) and (0,3,4) modes on SPL is insignificant compared to that of the (3,3,2) mode even though the natural frequencies of these modes are closer to the excitation frequency of 825 Hz than that of the (3,3,2) mode. This mode is thus becoming dominant at this driving frequency).

4. CONCLUSIONS

The effect of acoustic-structural coupling on the steady-state medium frequency sound field response has been described. It has been shown that the characteristics of boundary structures determine if the enclosed sound field is controlled by the modal domination mechanism or by the modal superposition mechanism. Thus unlike the low frequency sound field which is controlled by the modal domination mechanism and high frequency sound field which is controlled by the modal superposition mechanism, both mechanisms are at work at medium frequencies.

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