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Active control of structural sound radiation in an acoustic enclosure consisting of flexible structure

Chen Dalin Chen Nan

(School of Mechanical Engineering, Southeast University, Nanjing 211189, China)

Abstract: The active control of structural sound radiation in an acoustic enclosure is studied by using distributed point force actuators as the secondary control force, and the control mechanisms for the radiated noise in the cavity are analyzed. A rectangular enclosure involving two simply supported flexible plates is created for this investigation. The characteristics of the primary and secondary sound field and the structural-acoustic coupled system are analyzed, and the optimal control objective for reducing the sound pressure level (SPL) in the cavity is derived. The response of the SPL in the cavity is analyzed and compared when the secondary point force actuators with different locations and parameters are applied to the two flexible plates. The results indicate that the noise in the cavity can be better controlled when some point force actuators are applied to two flexible plates for cooperative control rather than the point force actuators being only applied to the excited flexible plate.

Key words: active control; acoustic mode; radiated noise; control mechanism; sound pressure level (SPL)

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The study of the active control of structural sound radiation in the enclosure cavity is popular, for example, its application in the reduction of noise levels in aircraft fuselages or vehicle cabins[1−2]. Pan et al.[3] demonstrated that the optimal control of radiated noise in the enclosure does not only depend upon the strength of the control forces but also upon the location of the actuators and other parameters. Kim et al.[4] presented an analytical and experimental investigation into the active control of harmonic sound transmission in a structural-acoustic coupled system. They indicated that the number of control actuators must be equal to the number of modes being controlled[5−8]. Many researchers consider that the radiated noise in the interior can be well controlled by locating the actuators only on the excited flexible plate surface, but sometimes the noise cannot be well reduced at these frequencies which are dominated by the coupled plates. So we consider that we can put some point force actuators on both the excited plate and the coupled plate to minimize the SPL in the cavity.

A rectangular enclosure with two simply supported flexible plates and four rigid walls is used as a model for this investigation. Some equations for the sound field in the cavity and the structural vibration field in the panel are used to represent the system behavior, and Green's function technique and boundary integrations are used to describe the sound pressure distribution in the cavity and the panel velocity distribution. The response of the SPL in the cavity is analyzed and compared with different control strategies used to reduce the radiated noise.

1 Analytical Model and Formulation

The panel-cavity system is used for this investigation (see Fig. 1). The dimensions of the cavity are $L_X = 0.868$ m, $L_Y = 1.150$ m, $L_Z = 1.0$ m. The top and right aluminum panels are simply supported and their thicknesses $h = 0.006$ m. The primary point force F_p is expressed as the actuator locations and the complex force amplitudes, and the control point force actuators are described by the locations, numbers and the complex force amplitudes.

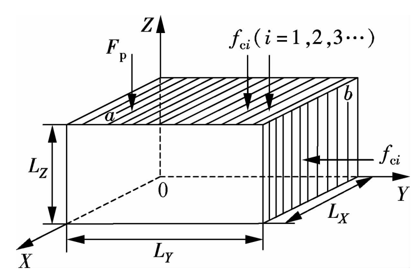

Fig. 1 Enclosure geometry

The sound pressure p in the cavity is described as

$$\nabla^2 p - \frac{1}{c_0^2}\frac{\partial^2 p}{\partial t^2} = \rho_0 \frac{\partial^2 w_a}{\partial t^2}\delta(v - v_a) + \rho_0 \frac{\partial^2 w_b}{\partial t^2}\delta(v - v_b) \tag{1}$$

where $\delta(\cdot)$ is the Dirac delta function; c_0 is the speed of sound in air; ρ_0 is the air density; w_a and w_b are the normal displacement of panel a and b; v_a and v_b are the vibration velocity of panel a and b; σ_i denotes the i-th point force applied at position on the surface of plate a. For the thin isotropic panel a and panel b, the flexural vibration equations are

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Biographies: Chen Dalin (1980—), male, graduate; Chen Nan (corresponding author), male, doctor, professor, nchen@seu.edu.cn.

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$$\rho h \frac{\partial^2 w_a}{\partial t^2} + \frac{Eh^3}{12(1-\mu^2)} \nabla^4 w_a = p - F_p \delta(\boldsymbol{\sigma} - \boldsymbol{\sigma}_0) - \sum_i f_{ci} \delta(\boldsymbol{\sigma} - \boldsymbol{\sigma}_i) \tag{2}$$

$$\rho h \frac{\partial^2 w_b}{\partial t^2} + \frac{Eh^3}{12(1-\mu^2)} \nabla^4 w_b = p \tag{3}$$

where ρ, E, μ, h are the density, Young's modulus, Poisson's ratio and the thickness of the panel, respectively. The Green functions are used to describe the sound field in the cavity and the vibration on the panel[3], and the sound pressure can be expressed as

$$p(\boldsymbol{r},\omega) = \mathrm{j}\rho_0\omega \int_{A_f} G_A v(\boldsymbol{r}_0,\omega)\,\mathrm{d}\boldsymbol{r} \tag{4}$$

$$G_A(\boldsymbol{r},\boldsymbol{r}_0) = \sum_{m=0}^{\infty} \frac{\phi_m(\boldsymbol{r})\phi_m(\boldsymbol{r}_0)}{M_{2m}(k^2 - k_m^2)} \tag{5}$$

$$G_s = -\sum_{n=1}^{\infty} \frac{\psi_n(\boldsymbol{\sigma})\psi_n(\boldsymbol{\sigma}_0)}{M_{1n}(k^2 - k_n^2)} \tag{6}$$

$$M_{1n} = \int_{A_f} \rho_1 h \psi_n^2(\boldsymbol{\sigma})\,\mathrm{d}\boldsymbol{\sigma} \tag{7}$$

$$M_{2m} = \frac{1}{V}\int_V \phi_m^2(\boldsymbol{r})\,\mathrm{d}\boldsymbol{r} \tag{8}$$

where A_f is the flexible panel surface area; $\boldsymbol{r}_i$ is the position in the cavity; ω is the angular frequency; G_A is the Green function for the sound field; M_{2m} is the cavity modal mass, k is the wave number in air; k_m is the wave number of the m-th cavity mode; $\phi_m(\boldsymbol{r})$ is the mode shape of the rectangular cavity; k_n is the wave number of the n-th panel mode; $\psi_n(\boldsymbol{\sigma})$ is the panel mode shape; G_s is the Green function for the simply supported panel; M_{1n} is the panel modal mass. The panel velocity at location $\boldsymbol{\sigma}$ can be expressed as

$$v(\boldsymbol{\sigma},\omega) = \frac{\mathrm{j}\omega}{\rho h}\int_{A_f} G_s \Big[p(\boldsymbol{\sigma}_0,\omega) - F_p\delta(\boldsymbol{\sigma}-\boldsymbol{\sigma}_0) - \sum_i f_{ci}\delta(\boldsymbol{\sigma}-\boldsymbol{\sigma}_i) \Big]\mathrm{d}\boldsymbol{\sigma} \tag{9}$$

According to the modal synthesis method, the interior sound pressure and the panel velocity can be written as

$$p = \boldsymbol{\phi}_m^{\mathrm{T}} \boldsymbol{P} \tag{10}$$

$$v = \boldsymbol{\psi}_n^{\mathrm{T}} \boldsymbol{V}_n \tag{11}$$

Substituting Eq. (10) into Eqs. (4) and (9), we can obtain

$$\boldsymbol{P} = \boldsymbol{Z}_A \boldsymbol{V}_m \tag{12}$$

$$\boldsymbol{P}^{\mathrm{ext}} = \boldsymbol{Z}_s \boldsymbol{V}_m \tag{13}$$

where $\boldsymbol{V}_m$ is the panel velocity distribution; $\boldsymbol{Z}_A$ is the internal modal radiation impedance matrix of the panel; and $\boldsymbol{Z}_s$ is the panel model input impedance matrix[3]; L_{nm} is the modal coupling coefficient between the m-th cavity mode and the n-th panel mode. Here, L_{nm} can be written as[9]

$$L_{nm} = \frac{1}{A_f}\int_{A_f} \psi_n \phi_m \,\mathrm{d}A \tag{14}$$

The modal amplitude of the primary sound pressure vectors due to the force F_p is given by

$$\boldsymbol{P}_k^{\mathrm{p}} = -\left[\frac{2}{A_f}\int_{A_f}\psi_1 F_p \mathrm{d}A,\ \frac{2}{A_f}\int_{A_f}\psi_2 F_p \mathrm{d}A,\ \cdots,\ \frac{2}{A_f}\int_{A_f}\psi_k F_p \mathrm{d}A \right]^{\mathrm{T}} \tag{15}$$

The sound pressure in the cavity without the influence of the point force actuators can be written as

$$\boldsymbol{p}_p = \boldsymbol{\phi}(\boldsymbol{r})^{\mathrm{T}} \boldsymbol{Z}_A \boldsymbol{Z}_s^{-1} \boldsymbol{P}_k^{\mathrm{p}} \tag{16}$$

When some point force actuators are applied on flexible plates, and these actuators are located at $\boldsymbol{\sigma}_k$, the secondary sound pressure in the cavity can be written as

$$\boldsymbol{p}_s = \boldsymbol{\phi}(\boldsymbol{r})^{\mathrm{T}} \boldsymbol{Z}_A \boldsymbol{Z}_s^{-1} \boldsymbol{\psi}_k(\boldsymbol{\sigma}_k) \boldsymbol{f}_c \tag{17}$$

The total sound pressure in the cavity can be expressed by the linear superposition of the primary and secondary sound pressure

$$\boldsymbol{p}_{\mathrm{total}} = \boldsymbol{p}_p + \boldsymbol{p}_s \tag{18}$$

The SPL is used to calculate the effect of constructing the acoustic pressure within the enclosure. The SPL and the panel averaged velocity can be written as

$$\mathrm{SPL} = 20\lg \frac{p_{\mathrm{total}}}{p_{\mathrm{ref}}} \tag{19}$$

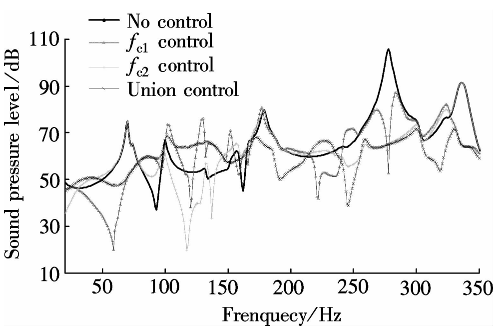
$$V_i^2 = \frac{1}{2A_f}\int_{A_f} v v^* \,\mathrm{d}s \tag{20}$$

2 Results and Discussion

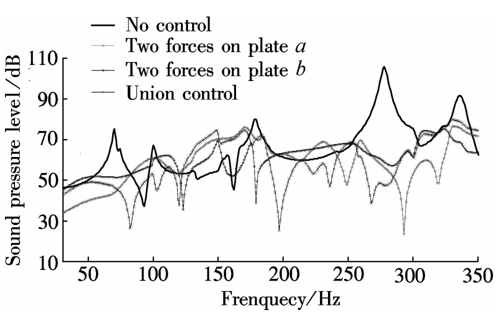
Numerical analyses are conducted using the configuration shown in Fig. 1. The model parameters are as follows: $\rho_0 = 1.21$ kg/m^3, $\rho_1 = 2\ 700$ kg/m^3, $E = 68.5$ GPa, $\eta_1 = \eta_2 = 0.01$, $\mu = 0.33$, $c_0 = 344$ m/s, $c_1 = 6\ 260$ m/s. The primary unit harmonic force F_p is located at (0.3 m, 0.4 m) on plate a. We take some control strategies to reduce the SPL in the cavity. First, we obtain the optimal location of each actuator, and then we use multiple actuators that are located on two flexible plates to obtain better control effort.

The region that we obtain the best noise control effort is just near the middle of the cavity. So we choose the region around S point at the coordinate (0.4 m, 0.5 m, 0.5 m) to analyze the control effort.

Fig. 2(a) shows the SPL in the enclosure at S point. We can see that high resonant peaks (without control forces) are mainly related to acoustic natural frequencies and some structural frequencies of the panels. The noise at 274 Hz is the largest under uncontrolled conditions (see Fig. 2(a)). We know that the frequency 274 Hz is close to the 11th panel mode of plate *a*. We can conclude that the biggest effect on the noise in the cavity is the panel vibration that is caused by the 11th panel mode of plate *a*, so we want to reduce the noise in the cavity by cancelling the vibration of plate *a* that is caused by it. One-point force f_{c1} is used to reduce this noise. The location of f_{c1} on the surface is at the coordinate (0.65 m, 1.01 m), which is close to the high amplitude region of the 11th panel mode. Fig. 3 shows the SPL and the velocity level (VL), $\mathrm{VL} = 10\log(V_i^2/V_{\mathrm{ref}}^2)$ on the receiving plate. We can see that location of f_{c1} is also in the high SPL region and the high VL region on the panel. Fig. 2 (a) shows that noise reduction at 63, 122, 219 and 274 Hz are achieved when the control force f_{c1} is working. In addition, the amplitudes of these contributing modes of plate *a* are minimized.

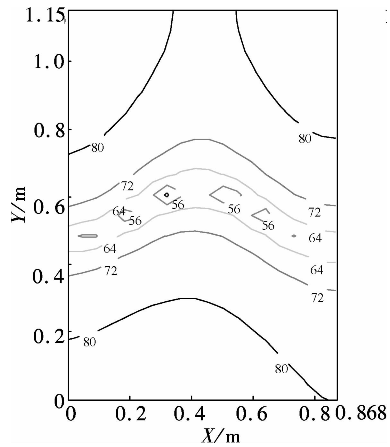


(a)

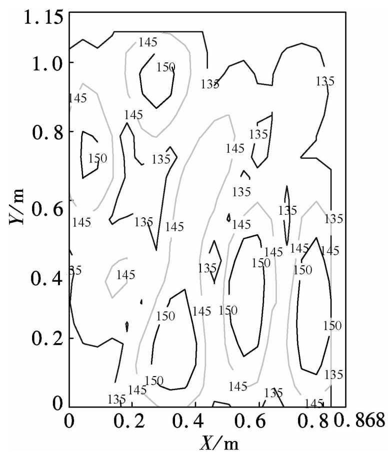


(b)

Fig. 2 Control performances under different control strategies. (a) Control forces located on plate *a*; (b) Union control forces located on two plates



(a)



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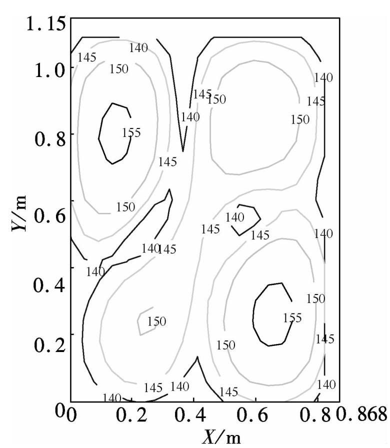

(c)

Fig. 3 Contour plots of different vectors at some frequency on plate *a*. (a) SPL (at $f = 274$ Hz); (b) VL (at $f = 274$ Hz); (c) VL (at $f = 118$ Hz)

However, the noise interior is still high at 118, 184 and 252 Hz. Consequently, f_{c2} is used to control the SPL in the interior at these frequencies. According to the VL equations and the rules on how the location of f_{c1} is selected, we obtain the location of f_{c2} at the high VL region. Fig. 3(c) indicates that there are four regions where the VL is high on the plate. f_{c2} is located at (0.5 m, 0.7 m) in the region that is diagonal from F_p. We can see that the SPL in the cavity is reduced by more than 20 dB at 118 and 252 Hz when f_{c2} is functioning. The results indicate that when the response noise is dominated by panel-controlled modes, the sound pressure level in the cavity is minimized by suppressing these pane modes.

However, when the union control of f_{c1} and f_{c2} are having an effective functioning, the noise reduction at 91 and 135 Hz is small, and the SPL is increased at some frequencies. As plate *b* plays a leading role to the cavity noise at these frequencies, we need some new control mechanisms. So we consider to set some point force actuators on plate *b* to reduce the SPL in the cavity.

Fig. 2(b) and Fig. 4(a) indicate that the 3th, 7th and 12th structural modes of plate *b* play a contributing role to the noise in the cavity when the primary forces are located on plate *a*. So f_{c3} is located at (0.72 m, 0.25 m) on panel *b*, and f_{c4} is located at (0.72 m, 0.75 m) on panel *b*. Fig. 2(b) shows that when there are two forces on panel *b*, the noise reduction is approximately 10 dB at 91 Hz and 135 Hz. It can be found that the cooperative control effect on panel *a* and panel *b* is better than only the control forces located on plate *a*, as shown in Figs. 2(a) and (b). Also, it can be seen that the SPL in the cavity is reduced at 172 Hz which is resonated by the cavity acoustic, and we can predict that when the noise is dominated by cavity-controlled modes, the control forces on panel *b* are used to change the panel velocity distribution and adjust the radiation of each panel mode for reducing the noise in the cavity.

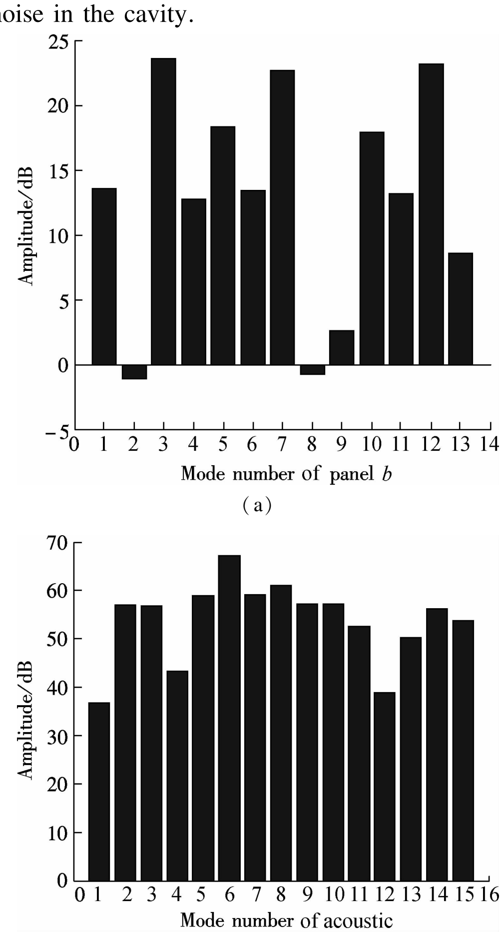

Fig. 4 Modal amplitude of different structures when the primary forces are located on panel *a*. (a) Panel *b*; (b) Acoustic in the cavity

Meanwhile, the increase of the SPL can reach 20 dB merely at 149 Hz, which is the resonant frequency of panel *b* and the cavity acoustic. Fig. 4(b) indicates that the 1st modal amplitude of acoustic plays a less important role when the primary force actuator is working on panel *a* at $f = 274$ Hz. So the noise in the cavity can be better controlled when the control point force actuators are applied to two flexible plates for cooperative control.

3 Conclusion

The theoretical model of sound radiation in an enclosure surrounded by two flexible plates has been formulated. The simulations indicate that the excited flexible plate plays an important role for the noise in the cavity, and the coupled flexible plate also has some influences on the noise. The optimal control actuator location is obtained by the modal analysis method and contour plots; and multiple actuators are used on two flexible plates to reduce the SPL in the cavity. The results indicate that the cooperative control can obtain better control effects rather than the point force actuators that are only applied to the excited flexible plate.

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弹性结构封闭空间中结构辐射噪声的有源控制

陈大林　陈　南

（东南大学机械工程学院，南京 211189）

摘要：研究了利用分布式点力源作为次级力源对封闭空腔内的结构辐射噪声进行控制的问题，并对封闭空腔中结构辐射噪声的控制机制进行了分析. 建立了包含 2 块简支弹性板的矩形封闭空腔，并将其作为研究对象. 通过对初、次级声场以及结构-声耦合的特性分析，以减小腔内声场的声压级为最优控制目标，分析比较了在不同位置及参数的次级点力激励下腔内声场的声压级响应. 仿真结果表明，在 2 块弹性板上施加联合激振力控制腔内噪声的效果要明显好于激振力只作用在受激弹性板上.

关键词：有源控制；声模态；辐射噪声；控制机制；声压级

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