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Short Communication

Active noise control of a mechanically linked double panel system coupled with an acoustic enclosure

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Abstract

Active control of sound transmission through a mechanically linked double-wall structure into an acoustic enclosure is investigated in this paper. Based on a fully coupled vibro-acoustic model, the effect of mechanical links on the selection of control strategies is studied by examining (a) cavity control using acoustic sources inside the air gap and (b) structural control using structural actuators between the two panels. The relationship between the transmission path and the control strategies is explored. Numerical results show that cavity control can provide good noise attenuation for soft links when acoustic transmitting path dominates, while either structural or cavity controls can be used with the increase of stiffness of links depending on the frequency range of interest. For each case, the dominant control mechanism is examined and the alteration in the structural–acoustic coupling is analyzed to explain the mechanisms of attenuation. The effect of the acoustic mode (0,0,0) on active control of energy transmission is also discussed, giving guidance to choosing the appropriate control arrangement to ensure the maximum control performance.

1. Introduction

Noise insulation performance of double-wall structures usually deteriorates at low frequencies. Sound absorption materials in the gap in passive control also fail to provide enough absorption in the low-frequency range. As an alternative method, active control techniques have been explored to increase the noise transmission loss of this kind structure [1–3].

In many applications, there exist mechanical links to connect the two walls. In such case, energy can be transmitted into the enclosure either from the acoustic path through the air gap, or from the structural path through links. Different transmitting paths certainly affect the coupling between the panels and the gap cavity, and subsequently the control strategy. For instance, Bao and Pan [4] experimentally verified that the existence of the structural transmitting path could change both sensing arrangement and actuation mechanism. In our recent work [5], the effect of the air gap and mechanical links on the energy transmission and noise insulation properties has been investigated, and a criterion was presented to predict the dominant transmitting path. As far as control actions is concerned, however, it is unknown whether there exists a possible correlation between

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the dominant transmitting path and the choice of control strategy, and subsequently on the control mechanism, which affects the actuator/sensor arrangements during the control design.

This paper addresses these issues, illustrated by the case of a mechanically linked double wall structure radiating sound into a rectangular acoustic enclosure. Based on our previous work [5], the effect of mechanical links between the two panels on the selection of control strategies is studied. Two control strategies are examined and numerical simulations are carried out to reveal the relationship between control strategies and transmitting paths. Dominant control mechanisms in the low- and high-frequency ranges are examined and the alteration in the structural–acoustic coupling is analyzed. In addition, the effect of the acoustic mode (0,0,0) on noise attenuation at low frequencies is also explored, giving guidance to choosing the appropriate control arrangement.

2. System modeling and control

Fig. 1 shows the schematic diagram of a double-wall structure radiating sound into a rectangular acoustic enclosure V. The two panels, a and b, are simply supported along their boundaries and separated by an air gap cavity. A mechanical link, which is simulated by a spring with a translational stiffnessK_m, connects the two panels at location (x_m, y_m). Apart from the surfaces occupied by the two panels, all other surrounding walls of both the air gap and the enclosure are acoustically rigid. Panel a is subjected to an acoustic excitation $\tilde{P}$. Control actions can be provided by P_{con} generated by a sound source inside the air gap or mechanical forces F_{con} applied between the two panels at location (x_l, y_l).

System modeling includes the vibration of two panels and the acoustic pressure inside the air gap and the enclosure, which is similar to those presented in the literatures for the vibro-acoustic modeling of double-wall structures [5,6]. For the two panels, the equations of motion can be described as

$$D_a \nabla^4 w_a + \rho_a h_a \frac{\partial^2 w_a}{\partial t^2} = \tilde{P} - K_m[w_a(x_m, y_m) - w_b(x_m, y_m)] - P_g(z = h_g) - F_{con}\delta(x - x_l, y - y_l) \tag{1}$$

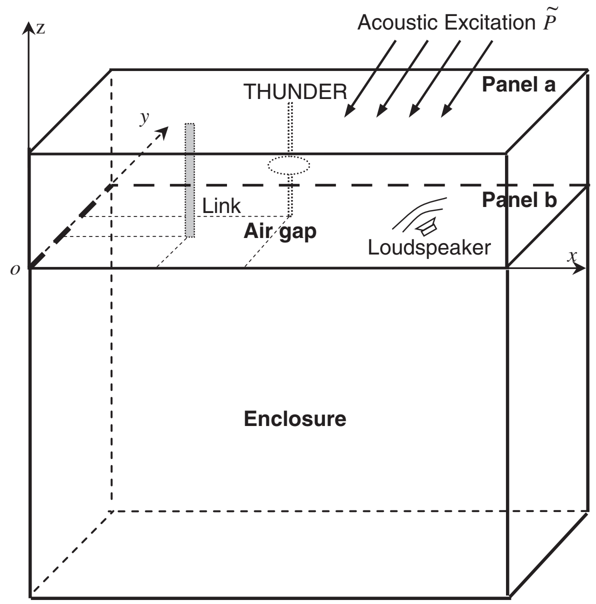

Fig. 1. Schematic diagram of a mechanically linked double-wall structure with an acoustic enclosure.

for panel a and

$$D_b\nabla^4 w_b + \rho_b h_b \frac{\partial^2 w_b}{\partial t^2} = K_m[w_a(x_m, y_m) - w_b(x_m, y_m)] + P_g(z=0) - P_e(z=0) + F_{\mathrm{con}}\delta(x - x_l, y - y_l) \tag{2}$$

for panel b. In the above expression, (w_a,D_a,ρ_a,h_a) and (w_b,D_b,ρ_b,h_b) are the transverse displacement (positive downwards), the flexible rigidity, the density and the thickness of panels a and b, respectively. P_g and P_e are the sound pressures inside the air gap and the enclosure, respectively.

The acoustic pressure inside the cavity can be derived from the classical wave equation and the constraint of the continuity of velocity on different parts of the cavity walls, i.e.,

$$\nabla^2 P_g(r,t) - \frac{1}{c^2}\frac{\partial^2 P_g(r,t)}{\partial t^2} = P_{\mathrm{con}}, \quad \frac{\partial P_g(r,t)}{\partial \mathbf{n}} = \begin{cases} \rho \ddot{w}_a & \text{on panel } a, \\ -\rho \ddot{w}_b & \text{on panel b}, \\ 0 & \text{on the rigid wall} \end{cases} \tag{3}$$

for the air gap and

$$\nabla^2 P_e - \frac{1}{c^2}\frac{\partial^2 P_e}{\partial t^2} = 0, \quad \frac{\partial P_e}{\partial \mathbf{n}} = \begin{cases} \rho \ddot{w}_b & \text{on panel } b, \\ 0 & \text{on the rigid wall} \end{cases} \tag{4}$$

for the enclosure. c and ρ are the sound velocity within the enclosure and the equilibrium fluid density, respectively. $\mathbf{n}$ is the positive outward normal component. In the case of harmonic excitation, Eqs. (1)–(4) can be combined in matrix form after derivations [5]. The whole procedure leads to four sets of coupled equations, which describe the vibro-acoustic behavior of the coupled system under active control.

As is shown in Ref. [5], the dominant energy transmitting path depends on the stiffness of the mechanical link, K_m, and the aerostatic stiffness of the air gap, K_g. It was verified that the energy is mainly transmitted through the air gap when $K_m/K_g < 0.1$, whereas through the link when $K_m/K_g > 10$. Correspondingly, the following control strategies are investigated: (a) cavity control by placing acoustic sources inside the air gap; and (b) structural control by applying mechanical forces between the two panels. Main emphasizes will be put on the physical phenomena which will impact on the choice of control strategies. For this reason, a simple control method, i.e., the optimal control, is adopted to analyze the two mentioned control strategies. The cost function is defined as the total acoustic potential energy inside the enclosure as [7]

$$L_p = \frac{1}{4\rho c^2}\int_V P_e^*(r) P_e(r)\,\mathrm{d}\upsilon. \tag{5}$$

L_p can be minimized by optimizing the control pressure for the cavity control using acoustic sources, or the control voltage for the structural control using actuators.

3. Simulation results and discussion

Numerical simulations are performed to analyze the effect of transmitting paths on the selection of control strategies. The upper and lower aluminum plates are with dimensions of $2.15 \times 0.78 \times 0.004$ m and $2.15 \times 0.78 \times 0.006$ m, respectively. A mechanical link is located at (0.86,0.31). Two cases of link, that is, a soft one with $K_m = 10^2$ N/m and a hard one with $K_m = 5 \times 10^6$ N/m, are taken to analyze different energy transmitting paths. The depths of the enclosure and the gap cavity are set as $h_e = 0.55$ m and $h_g = 0.11$ m. The aerostatic stiffness of the air gap is $K_g \approx 3.5 \times 10^5$ N/m. The modal loss factor is assumed as 0.005 for the two panels and 0.001 for the cavities. The excitation pressure is an oblique plane wave with the amplitude of 1 Pa, an azimuth angle $\theta = 60^\circ$ and an elevation angle $\phi = 30^\circ$. Simulations on different actuator locations are conducted, leading to similar observations. In the following discussions, as an example, the actuation location is set at (0.65,0.31) for structural control using the THUNDER actuator [8], or at (0.65,0.31,0.03) for cavity control using the loudspeaker unless otherwise stated.

Fig. 2(a) illustrates the total acoustic potential energy L_p inside the enclosure without and with control for $K_m = 10^2$ N/m. It can be observed that for both cavity control and structural control, the potential energy

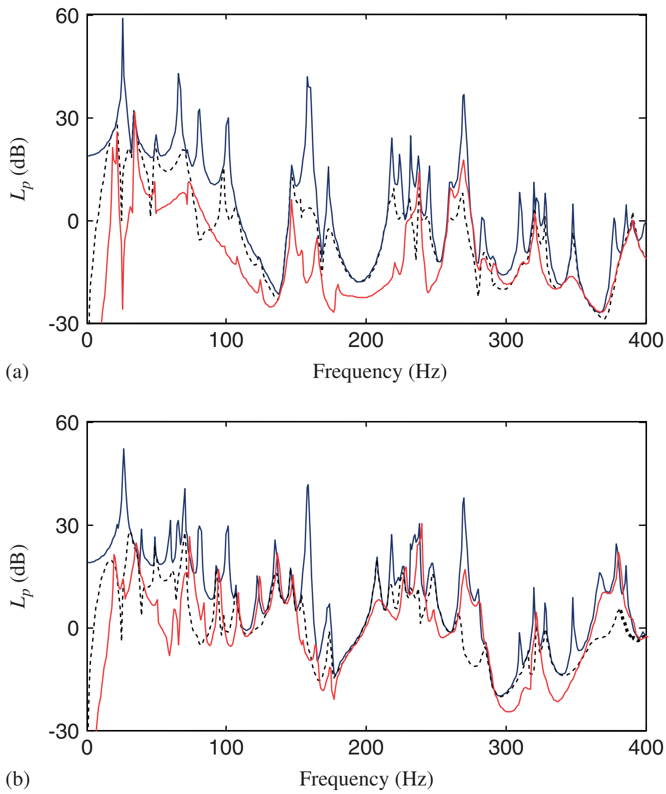

Fig. 2. Total acoustic potential energy L_p inside the enclosure without and with control: (a) $K_m = 10^2$ N/m and (b) $K_m = 5 \times 10^6$ N/m. ——Uncontrolled; • • • • Structural control; ▬▬ Cavity control.

inside the enclosure is attenuated significantly. In the frequency range 0–400 Hz, an overall noise reduction of up to 25dB can be achieved for cavity control while 14 dB for structural control. The reason is that for a soft link, the energy is mostly transmitted acoustically. In such case, cavity control (solid line) reduces the strength of the acoustic field inside the air gap, leading to a vibration suppression of the radiating panel and therefore an attenuation of L_p. As for structural control (dotted line), the actuator is not only applied to the radiating panel for vibration suppression, but also affects the upper panel through the connection, the sound transmission in the air gap can also be weakened, and the energy in the enclosure is attenuated accordingly. However, since the use of actuator cannot completely truncate the energy transmission path via the air gap, and moreover only the vibration of the dominant panel modes can be suppressed using single actuator, the control effect is limited compared with that obtained using the loudspeaker.

For the hard link, Fig. 2(b) shows L_p for the case of $K_m = 5 \times 10^6$ N/m, in which the energy is mainly transmitted from the link. It can be seen that a strong coupling between the two panels occurs, resulting in a remarkable energy transmission through the link. In light of Fig. 2(b), both control strategies are effective for noise reduction, i.e., an overall noise attenuation reaches 17 dB for cavity control while 16 dB for structural control in [0 400] Hz. Although there is no obvious evidence to judge the superiority of one strategy over the other, cavity control seems to be slightly better than structural control in the low-frequency range but worse in the high-frequency one. It is based on the fact that cavity control reduces acoustic field at low frequencies while increases at high frequencies, and accordingly alters the vibration of the radiating panel in the same way shown in Fig. 3. As for structural control, the force exerted on the radiating panel reduces its vibration, especially at high frequencies, and consequently attenuates noise level inside the enclosure.

Another observation in Figs. 3 and 2(b) shows that at low frequencies, the reduction in L_p is due to the suppression of the vibration level of the radiating panel. At high frequencies, although L_p is attenuated, this

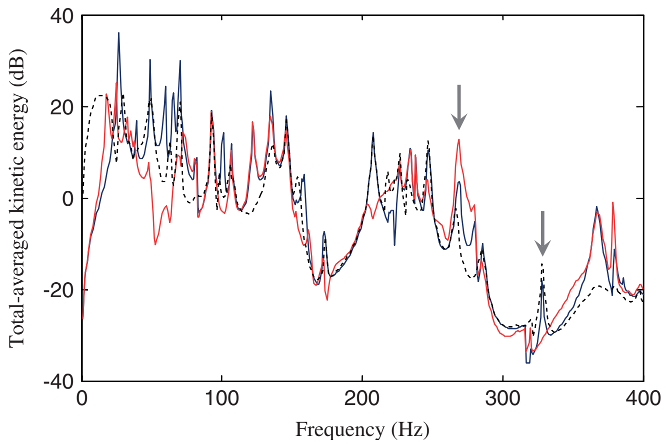

Fig. 3. Total averaged kinetic energy received by the radiating panel for $K_m = 5 \times 10^6$ N/m. ——— Uncontrolled; • • • • Structural control; ——— Cavity control.

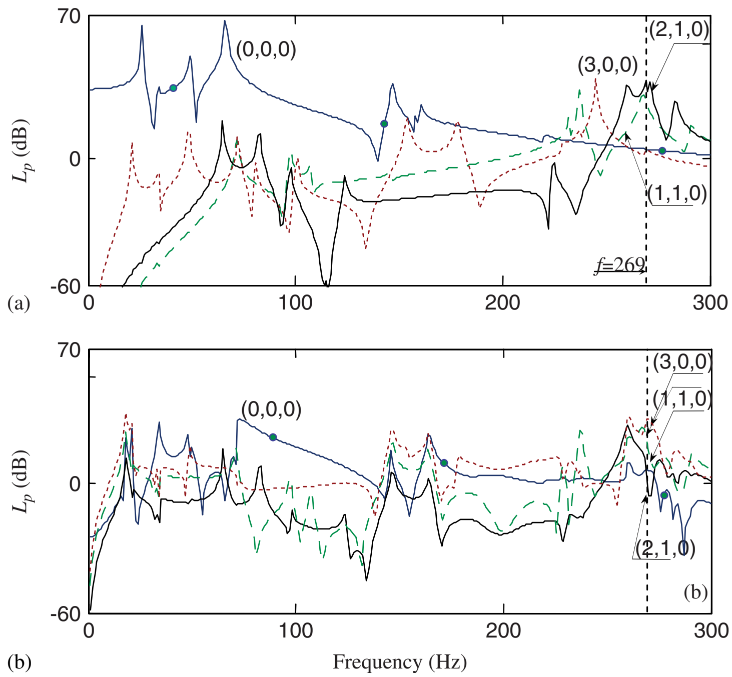

Fig. 4. Contribution of cavity modes on L_p: (a) without control; and (b) with control. Mode (0,0,0): —●—; mode (1,1,0): – – – –; mode (2,1,0): ———; mode (3,0,0): ·········.

change does not come from a reduced vibration level. Instead, the vibration at some frequencies is exacerbated (marked with arrows in Fig. 3), showing a possible change in modal coupling at these frequencies. Obviously, there exist two control mechanisms, i.e., modal suppression and modal rearrangement, at different frequency ranges, irrespective of the type of control strategies. Since modal suppression is the dominant control mechanism in the low-frequency range, this conclusion suggests the possibility of attenuating the sound inside the enclosure by only using vibration sensors instead of acoustic sensors, which can significantly simplify the design and implementation of control systems.

To understand the alteration in structural–acoustic coupling, the effect of cavity modes inside the air gap on L_p is examined under cavity control. Figs. 4(a) and (b) illustrate the effect of acoustic modes on L_p for the

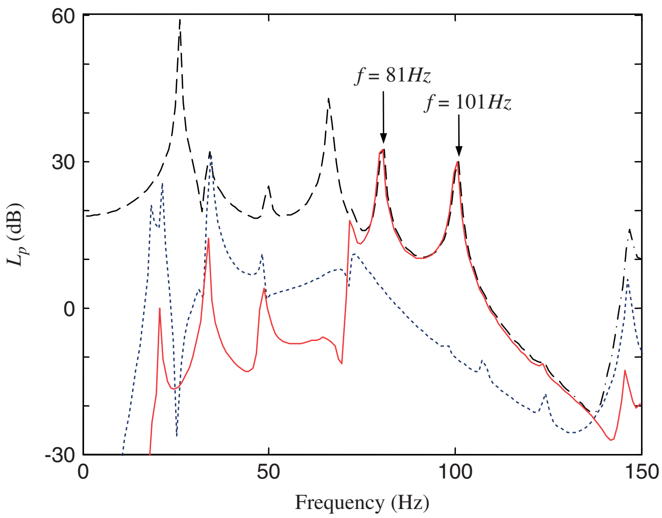

Fig. 5. Cavity control using: (a) a loudspeaker at (0.65,0.31,0.03); and (b) four synchronized loudspeakers located at (0.65,0.31,0.03), (0.65,0.47,0.03), (1.51, 0.31,0.03) and (1.51,0.47,0.03). - - - Uncontrolled; •••• Single loudspeaker; ▬▬ Four synchronized loudspeakers.

structure without and with control, respectively. It can be seen that before control, L_p at $f = 269$ Hz is mainly dominated by the cavity modes (2,1,0) and (1,1,0). The influence of other modes is trivial. The situation is different when the control is deployed. The effect of mode (2,1,0) is significantly weakened. Instead, the mode (3,0,0) becomes the most dominant, followed by the mode (1,1,0). This change in the air gap subsequently affects the modal responses of the radiating panel and therefore its coupling with the enclosure.

An interesting phenomenon observed from Fig. 4(a) is that the contribution of cavity mode (0,0,0) on L_p is dominant in the low-frequency range when energy is mainly transmitted from the acoustic path. This observation can be made use of for a better actuator arrangement during controller design. For example, a uniform control pressure field should be better than a point source at low frequencies. Synchronized multi-source actuation promotes a more homogeneous sound field without increasing the number of the control channels. This idea is tested in Fig. 5, which compares the effect of using (a) one control loudspeaker and (b) four synchronized control loudspeakers symmetrically located within the air gap. The synchronized multi-source actuation obviously out-performs the single-source control scheme, except for a region controlled by $f = 81$ and 101 Hz modes. This is due to the fact that $f = 81$ and 101 Hz corresponds to the acoustic mode (1,0,0) of the enclosure and the structural mode (1,2) of panel b, respectively. Symmetrical arrangement of the control sources has therefore no effect on modes.

4. Conclusions

(1) The existence of mechanical links between the two panels results in different energy transmission paths so as to affect the selection of control strategies. For a soft link resulting in acoustic transmission, effort should be put on the cavity control to weaken the acoustic energy inside the air gap, so that the vibration of the radiating panel can be suppressed, resulting in a reduced sound transmission; whereas for a hard link forming structural transmission, both structural control and cavity control can be used, depending on the frequency range to be controlled.
(2) Two control mechanisms, i.e., modal suppression and modal rearrangement, exist simultaneously for both control strategies. Modal suppression occurs mainly in the low-frequency range while modal rearrangement in the high-frequency one. This analysis has significant value in the practical implementation of the active control. When the suppression mechanism dominates, sound reduction is a by-product of the vibration suppression in the radiating panel. Since in most cases, vibration sensors and actuators can be more easily embedded into the structures, the use of acoustic sensors inside the enclosure

can be avoided, which can greatly simplify the control system. When the restructuring mechanism dominates, structural sensors alone are not enough to reflect the sound–structural interaction.

(3) The cavity mode (0,0,0) plays an important role in the energy transmission process at low frequencies. This observation suggests the use of any actuator arrangement which would promote the response from (0,0,0) mode inside the air gap. In particular, synchronized single-channel control with multi-control sources yields a better control result than a single-point source.

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