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ACTIVE ATTENUATION OF NOISE TRANSMISSION THROUGH ELASTIC PARTITIONS WITH HIGH MODAL DENSITIES

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1. INTRODUCTION

Low frequency sound transmission into a room through lightly-weighted partition structures is characterised by the coupling between many structural modes and a few room modes. Active structural acoustic attenuation of the sound transmission using one or two actuators on the partition structures relies on the mechanism of rearrangements of the structural mode to minimise the overall sound radiation into the room-controlled-modes. However, the achievable attenuation level decreases as the structural modal overlap factor increases because of the difficulties in optimally rearranging the phases and magnitudes of many structural modes by using only a couple of control actuators. In this paper three control arrangements are considered for a double wall sound transmission model. They are (1) to directly attenuate the room-controlled-modes by using one control loudspeaker in the room, (2) to control the sound radiation into the room using a vibrational actuator on the radiating structure and (3) to break the noise transmission path by inserting one control loudspeaker between the double wall partitions. The mechanisms of the active acoustic attenuation associated with these three arrangements are illustrated and the performance of the control system reviewed.

2. DESCRIPTION OF THE CONTROL SYSTEM

Figure 1 is a double wall partition model, where an incident sound wave p_{in} excites the top plate (plate 1). Through the coupling of four sub-systems including the top plate, acoustical cavity between the double plates, the internal plate (plate 2) and the room, the internal sound field is generated. Using the modal expansion method for each sub-system, coupled equations for the modal components of each sub-system are obtained. If the sound field in the room is controlled directly by acoustical actuators in the room only, the pressure components $[P_{N2}]=[P_1^{(2)},P_2^{(2)},...,P_{N2}^{(2)}]^T$ are contributed by the velocity of plate 2 ($[V_{M2}]=[V_1^{(2)},V_2^{(2)},...,V_{M2}^{(2)}]^T$) and the velocity of the control sources ($[V_{Nr2}^{(c)}]\neq 0$):

$$[P_{N2}]=[Z_A^{(2)}][V_{M2}]+[Z_{Q2}^{(2)}][V_{Nr2}^{(c)}], \qquad (1)$$

where $[Z_A^{(2)}]$ is the modal transfer impedance matrix from plate 2 to the room and

$[Z^{(2)}_{Q2}]$ the matrix from the acoustical control source to the room. If the transmitted sound pressure is controlled by vibrational actuators on plate 2 only ($[V^{(c)}_{Nr2}]=0$), the velocity components of plate 2 are related to $[P_{N2}]$, the sound pressure components of the cavity $[P_{N1}]$ and the pressure due to the vibration actuators ($[P^{(c)}_{Mp2}]\neq 0$):

$$[V_{M2}]=[Y^{(2)}_{p2}][P_{N2}]-[Y^{(2)}_{p1}][P_{N1}]+[Y^{(c)}_{F}][P^{(c)}_{Mp2}], \quad (2)$$

where $[Y^{(2)}_{p2}]$, $[Y^{(2)}_{p1}]$ and $[Y^{(c)}_{F}]$ are respectively the modal receptance matrices between each pressure vector and the velocity vector of plate 2. For this case, the room sound pressure is expressed as:

$$[P_{N2}]=\{[I]-[Z^{(2)}_{A}][Y^{(2)}_{p2}]\}^{-1}[Z^{(2)}_{A}]\{\underline{[Y^{(c)}_{F}][P^{(c)}_{Mp2}]-[Y^{(2)}_{p1}][P_{N1}]}\}. \quad (3)$$

If acoustical actuators are installed in the double panel cavity only ($[V^{(c)}_{Nr2}]=0$ and $[P^{(c)}_{Mp2}]=0$) to attenuate the sound transmission, the pressure components in the cavity are not only related to the velocity components of the two plates but also to the velocity of control sources ($[V^{(c)}_{Nr1}]\neq 0$):

$$[P_{N1}]=[Z^{(1)}_{A1}][V_{M1}]-[Z^{(1)}_{A2}][V_{M2}]+[Z^{(c)}_{Q1}][V^{(c)}_{Nr1}] \quad (4)$$

where $[Z^{(1)}_{A1}]$, $[Z^{(1)}_{A2}]$ and $[Z^{(c)}_{Q1}]$ are respectively the modal impedance matrices between each velocity vectors and the pressure vector of the cavity. In Eq.(4) the primary excitation comes from plate 1, which is described as:

$$[V_{M1}]=[Y^{(1)}_{p1}][P_{N1}]-[Y^{(in)}_{p}][P_{in}] \quad (5)$$

where $[Y^{(1)}_{p1}]$ and $[Y^{(in)}_{p}]$ are respectively the modal receptance matrices between each pressure vector on plate 1 and the velocity vector of plate 1. Using Eqs.(4), (5), (2) and (1) and letting $[V^{(c)}_{Nr2}]=0$ and $[P^{(c)}_{Mp2}]=0$, we can obtain:

$$[P_{N2}]=[\overline{P}_{N2}]\{\underline{[Z^{(1)}_{A1}][Y^{(in)}_{p}][P_{in}]-[Z^{(c)}_{Q1}][V^{(c)}_{Nr1}]}\} \quad (6)$$

where

$$[\overline{P}_{N2}]=\{[I]-[Z^{(2)}_{A}][Y^{(2)}_{p2}]-[Z^{(2)}_{A}][Y^{(2)}_{p1}][X]^{-1}[Z^{(1)}_{A2}][Y^{(2)}_{p2}]\}^{-1}[Z^{(2)}_{A}][Y^{(2)}_{p1}][X]^{-1},$$

$$[X]=[I]-[Z^{(1)}_{A1}][Y^{(1)}_{p1}]-[Z^{(1)}_{A2}][Y^{(2)}_{p1}].$$

It is useful to analyse the underlined terms in Eqs.(1), (3) and (6) because they determine the performance of the control system. For the simplicity of the analysis, only one control actuator is assumed in use each time and the acoustic potential energy in the room is used as the cost function.

3. ANALYSIS OF CONTROL MECHANISMS

Acoustic control in the room

When only one acoustic actuator is used in the room to control the transmitted sound field, Eq. (1) can be expressed as:

$$[P_{N2}]=X_p+X_c V^{(c)}_r \quad (7)$$

where the elements of X_p and X_c are:

$$X^{(p)}_I=\frac{\rho_o c_o}{\chi^{(2)}_{AI}}\sum_{J=1}^{M_2}B^{(2)}_{I,J}V^{(2)}_J \quad \text{and} \quad X^{(c)}_I=\frac{\rho_o c_o}{\chi^{(2)}_{AI}}\phi^{(2)}_I(r^{(2)}_c) \quad (8)$$

and $B_{l,j}^{(2)}$, $\phi_l^{(2)}(r_c^{(2)})$ and $\chi_{Al}^{(2)}$ are respectively coupling coefficient, mode shape function and modal impedance. Because the elements of $[P_{N2}]$ are all orthogonal to each other in terms of the modal shape functions of the room, the mechanism involved in this arrangement is purely modal suppression. The minimisation of the potential energy in the room suggests that:

$$E_{min} \propto X_p^H \Lambda_{N2} X_p - \frac{|X_p^H \Lambda_{N2} X_c|^2}{X_c^H \Lambda_{N2} X_c} \tag{9}$$

where Λ_{N2} is a diagonal normalisation matrix of room modes. To achieve a large reduction in the potential energy at a frequency, the following relationship is required:

$$|X_p^H \Lambda_{N2} X_c|^2 \to (X_c^H \Lambda_{N2} X_c)(X_p^H \Lambda_{N2} X_p) \tag{10}$$

Mathematically, the above relationship is satisfied only if the two vectors are proportional ie:

$$X_c = cX_p \tag{11}$$

where c is a constant. As X_c is due to a point sound source and X_p to a distributed structural vibration, Eq.(11) can only be satisfied in a very low frequency range where the modal overlap is low, (sound field is dominated by a single mode and the contribution of non-resonance modes is small). With increased modal overlap, the effectively control will rely on that the primary and secondary sources are close to allow a similar generation of all adjacent modes. Because of the large size and absorption of the rooms examined in this paper, the frequency range where global sound reduction can be achieved is around the Helmholze mode and the first acoustic mode. Figure 2 shows the controlled and uncontrolled acoustic potential energy due to a primary panel ($2 \times 1.2 \times 0.001 m^3$ steel panel) radiation and an optimal point acoustical control source in highly damped room ($V_o^{(2)} = 24 m^3$ and the equivalent specific acoustic admittance of the boundary $\beta_e = 0.067$). For this arrangement, a large reduction of potential energy is only achieved at the very low frequencies. In the low frequency rang, the one with the acoustic control source near the radiating plate gives larger reduction than that with control source located at the far corner of the room.

Vibration control on plate 2

When only one point vibration actuator is used on plate 2 to control the transmitted sound field, Eq. (3) can be expressed as:

$$[P_{N2}] = [A]\{Y_p + Y_c P_{p2}^{(c)}\} \tag{12}$$

where $[A]$ representing the coefficient matrix of the underlined vector in Eq.(3), and the elements of Y_p and Y_c are respectively:

$$Y_l^{(p)} = -\frac{1}{\rho_o c_o \chi_{pl}^{(2)}} \sum_{J=1}^{N_1} B_{l,J}^{(2,1)} P_J^{(1)} \text{ and } Y_l^{(c)} = -\frac{1}{\rho_o c_o \chi_{pl}^{(2)}} S_l^{(2)}(\sigma_c)$$

and $B_{l,J}^{(2,1)}$, $S_l^{(2)}(\sigma_c)$ and $\chi_{pl}^{(2)}$ are respectively the coupling coefficient from the cavity modes to that of plate 2, the mode shape function and the modal impedance of plate 2. Eq.(12) suggests that the sound field in the room can be attenuated by two mechanisms:

(i) Suppressing the modal amplitudes of the vibration in plate 2, which involves the reduction of the large modal elements of Y_p using the control elements of $Y_c P_{p2}^{(c)}$. Previous work suggested that the modal suppressing mechanism is associated with the sound transmission at the panel controlled mode where the transmitted energy is

carried mainly by one uncoupled panel mode.
(ii) As each pressure component in the room is the superimposed radiation of many modes in plate 2 shown in Eqs.(3) and (12), the re-arrangement of the magnitudes and phases of the modes in plate 2 can also result in reduced sound pressure in the room. The modal re-arrangement mechanism is associated with the sound transmission into the room controlled modes where energy is transmitted from several plate modes.

To achieve a total reduction of the potential energy in the room by either of the mechanisms, following relationship has to be satisfied:

$$[A]Y_c = b[A]Y_p \qquad (13)$$

where b is a constant. Eq.(13) indicates that the transformed vectors Y_p and Y_c subject to $[A]$ have to be proportional. As Y_c is due to a point control force and Y_p to a distributed sound field $[P_{NI}]$, Eq.(13) can only be approximately satisfied when the modal overlap of plate 2 is low. Because (i) the panel modal density is independent of the frequency, (ii) the second mechanism will be at work when the plate response is dominated by several modes, and (iii) the control of sound field is right at the noise source (the near field control condition of the sound sources is satisfied), the control of sound transmission through panels with small modal overlap was superior to that using acoustic source in the room in terms of the frequency range for the sufficient noise reduction. Figure 3 shows the active control of sound transmission through a panel wall into an office ($V_o^{(2)} = 34m^3$) by using a control force in the plate ($2mm$). Comparing with the result in Fig. 2, it can be seen that ASAC results in better sound reduction in terms of both frequency range and level of reduction. The modal density of plate 2 will also affect the control of sound transmission. Figure 4 shows the noise reduction levels for three plate thickness by one vibration actuator. Generally a less reduce noise reduction level can be observed when ASAC is applied to plate 2 as its modal density increases. For high modal density and except at some discrete frequencies (where modal suppression and re-arrangement are still effective), one vibration actuator will not be able to effectively suppress or re-arrange so many plate modes (which are well coupled with the room modes) simultaneously and to give sufficient noise reduction.

Acoustic control in the cavity

When only one acoustic actuator is used in the double wall cavity to control the transmitted sound field, Eq. (6) can be expressed as:

$$[P_{N2}] = [\overline{P}_{N2}](Z_p + Z_c V_c^{(c)}) \qquad (14)$$

where the elements of Z_p and Z_c are respectively:

$$Z_I^{(p)} = \frac{\rho_o c_o}{\chi_{AI}^{(1)}} \sum_{J=1}^{M_1} B_{I,J}^{(1,1)} V_J^{(1)} \text{ and } Z_I^{(c)} = -\frac{\rho_o c_o}{\chi_{AI}^{(1)}} \phi_I^{(1)}(r_c^{(1)})$$

and $B_{I,J}^{(1,1)}$, $\phi_I^{(1)}(r_c^{(1)})$ and $\chi_{AI}^{(1)}$ are respectively the coupling coefficient, mode shape function and the modal impedance of the cavity. Eq.(14) suggests that there also exist two possible control mechanisms in attenuating the sound transmission by using an acoustic actuator in the cavity. At frequencies where the sound field in the cavity is dominated by a single mode, the suppression of this mode will result in a reduced potential energy in the room. Shown in Eq.(14) is that each pressure component in the room is due to the linear combination of the pressure modal components of the cavity. This suggests the possible re-arrangement of the cavity modes by the acoustic actuator in the cavity and their superposition gives minimum sound pressure components in the room. For cavities with small volume size, the modal density of the cavity will be very small. For this case, there will be a broad low frequency range where the sound field in the cavity is dominated by individual cavity mode and the influence of other non-resonance modes is little (assuming small modal damping). Therefore, in this frequency range, the following proportional relationship:

$$[\overline{P}_{N2}]Z_c = a[\overline{P}_{N2}]Z_p \qquad (15)$$

can be approximately satisfied and large reduction of potential energy in the room is expected regardless of the large modal density in the partition structures. As the modal arrangement requires comparable magnitudes of at least two cavity modes, it is expected that modal re-arrangement mechanism may be observed at the frequencies between the resonance frequencies of the cavity modes and the location of the actuator is capable of exciting them.

4. EXPERIMENTAL RESULTS AND DISCUSSION

A laboratory experiment is also conducted to investigate the active control of sound transmission through elastic partitions with high modal density. The incident sound field generated by speakers in an anechoic chamber drives a double plate partition which separates the chamber and a control room ($V_o^{(2)} = 56m^3$). The partition structure consists of two identical aluminium plates ($2.035 \times 0.78 \times 0.002m^3$), and the cavity between the plates has a volume of $V_o^{(1)} = 0.5m^3$. The sound field in the room is controlled at selected frequencies using either one control loudspeaker near plate 2, or one PZT actuator located close to the centre of plate 2, or one control loudspeaker inside of the cavity at a corner. The sound pressure at a corner of the room is used as an error signal for a single channel adaptive feedforward controller. Although the control output at the actuators is not according to the minimisation of the potential energy in the room, the observation of the controlled and uncontrolled system response can still shed some light to the understanding of the mechanisms involved in each arrangement. The reduction levels of the room acoustic potential energy (measured by averaging sound pressure in the room) are listed in table 1. The three control arrangements are compared experimentally at several frequencies. It can be seen that the control of sound transmission at the cavity gives a sufficient reduction in a broad low frequency range (up to $100Hz$). The ASAC at plate 2 gives sufficient noise reduction below $74Hz$ where the coupling between the plate modes with a couple of acoustic mode allows both mechanisms of the control work effectively. Above this frequency, the global control can be achieved only at certain frequencies where Eq.(13) is approximately satisfied. Because of the large modal overlap in the room, the ANC in the room using one loudspeaker only results in global attenuation of sound field below $62Hz$ where four room modes are well seperated. Below this frequency, the room modes are lightly damped and well separated in frequency. In addition the distance between the radiating plate and the control speaker is close compared with the wavelength of the sound waves. All these made the attenuation of acoustical potential energy possible at the frequencies selected.

Table 1. Reduction of potential energy in the room.

Frequency(Hz)		29	48	51	55	62	74	80	85	100	155
Noise Reduction (dB)	Room ANC	5	8	14	12	10	no	no	no	no	no
	Plate 2 ASAC	8	15	28	16	13	7	no	5	no	6
	Cavity ANC		17	26	15	13	16	6	2	9	no

6. CONCLUSIONS

This paper presents a model for sound transmission through double partitions with high modal density in such a way that the mechanisms of the active control can be analysed. For either acoustic control in the room, or vibration control on plate 2 or acoustic control in the partition cavity, the criteria of a sufficient reduction of acoustic potential energy in the room is if the resultant primary and secondary pressure vectors are proportional (described by Eqs.(11), (13) and (15)). It is shown that the physical nature of the sub-systems and their coupling, and the location of the control actuator determine the amount of the satisfaction of the proportional relationships. The analysis also shown that except the acoustic control in the room can only be achieved by the modal suppression mechanism, both ASA control at plate 2 and AN control in the cavity can be associated with two mechanisms (modal suppression and modal rearrangement).

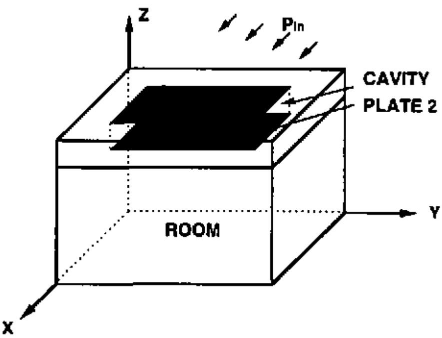

Fig. 1 Double wall model for the active control of sound transmission.

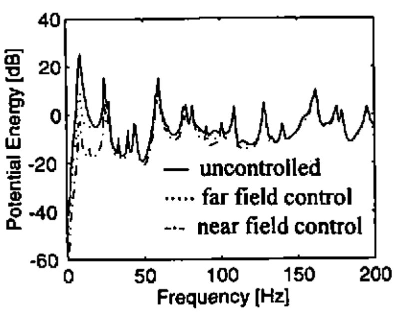

Fig. 2 Room ANC of plate 2 ($t = 2mm$) radiation using an acoustic actuator.

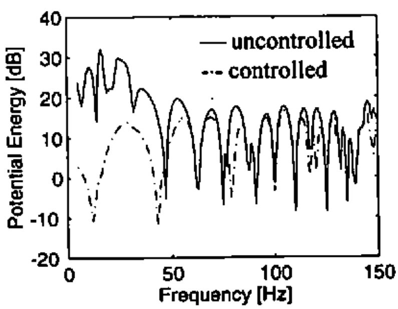

Fig. 3 ASAC of plate radiation at plate 2 ($t = 2mm$) using a point control force.

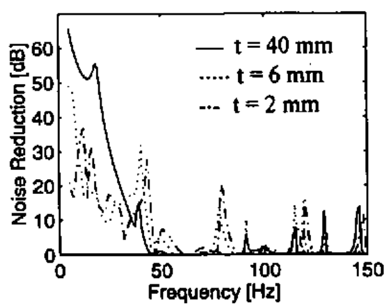

Fig. 4 Noise reduction level using ASAC at plate 2 with different modal densities.