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STRATEGIES FOR ACTIVE CONTROL OF SOUND TRANSMITTED THROUGH A DOUBLE-PANEL PARTITION USING DISTRIBUTED ACTUATORS AND SENSORS

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ABSTRACT

The paper considers the active control of harmonic and random sound transmitted through a double-leaf partition using a single *distributed* actuator and a single distributed sensor. The double-leaf partition consists of a pair of small plates (300×380 mm, separated by a 100 mm air-gap). An actuator made up of shaped, distributed PVDF offers the possibility of controlling the volume velocity of a plate without giving rise to control spillover. The sensor consists of a matched PVDF sensor to detect volume velocity.

For harmonic excitation with the actuator attached to either panel, substantial reductions in the transmitted sound power are possible up to around 350 Hz. A radiation mode analysis of the panels shows that the double-leaf construction provides good passive attenuation of the first radiation mode at high frequencies, so that inefficiently radiating even modes of the radiating panel make a dominant contribution to the radiated sound power. For the random excitation, an internal model control scheme is outlined for the double-panel system in which the transmission of random sound is controlled without the need for a reference signal. Cancellation of volume velocity with the distributed actuator provides reduction in random sound transmission up to 100 Hz.

INTRODUCTION

Double-leaf partitions are often used in noise control engineering when high sound transmission loss has to be achieved with lightweight structures: an example is an aircraft fuselage shell. However, the sound transmission loss decreases rapidly towards low frequencies, at which it could be poorer than that with a single panel[1, 2]. Thus, active noise control is required to solve this problem.

Studies of the active control of sound transmission through double-leaf partitions carried out to date have generally used piezoceramic patches or compact acoustic sources. Carneal and Fuller [3] carried out an experimental study of active control of a double-leaf partition using three PZT actuators for control. Another investigation of active control of double-leaf partitions was carried out by Sas et al [4] who inserted small loudspeakers into the space between the two panels. This approach benefits from the fact for most geometries there are far fewer acoustic modes in the space than there are structural modes on the panels.

The present study considers the case of a shaped, distributed PVDF actuator and sensor for control of sound transmission through a double-leaf partition. The design and use of the distributed PVDF actuator has been discussed in a number of publications [5, 6, 7]. If the PVDF electrode takes the form of a set of quadratic strips, the device can be used as a actuator and/or a sensor[8]. The actuator presents a uniform pressure and the sensor detects the volume velocity [9, 10]. However, to the authors' knowledge, the use of the distributed

PVDF actuator and sensor to control sound transmission through a double-panel partition has not been considered elsewhere but it is examined in this paper. Also the limitation of active control is discussed between harmonic and random sound transmission.

THEORY

Description of double-panel model. Figure 1 shows a double-panel partition in an infinite rigid baffle. An incident plane wave P excites panel 1. Through the coupling of panel 1, the acoustical enclosure between the double panels and panel 2, sound power is radiated from the right hand side of the baffle. The coordinates of panel 1 which is excited by a plane wave (P) are shown in Figure 2(a), and the coordinates of panel 2 are shown in Figure 2(b). The sound radiation from panel 2 is calculated using a number of elemental radiators [9].

Structural-acoustic coupled response. The acoustic pressure and the structural vibration velocities at each panel can be expressed as a function of the amplitudes of the uncoupled mode shape functions[11]. Considering all the modal forces applied to each panel, also the modal velocity of each panel applied to the enclosed volume, the amplitudes of the acoustic modes and the amplitudes of the structural modes on panel 1 and panel 2 can be written respectively as follows [12]:

$$\mathbf{a} = \mathbf{Z_a}(\mathbf{C_1 b_{p1}} + \mathbf{C_2 b_{p2}}), \tag{1}$$

$$\mathbf{b_{p1}} = \mathbf{Y_{p1}}(\mathbf{g_p} - \mathbf{p_{c1}} - \mathbf{C_1}^T\mathbf{a}) \tag{2}$$

and

$$\mathbf{b_{p2}} = \mathbf{Y_{p2}}(-\mathbf{p_{c2}} + \mathbf{C_2}^T\mathbf{a}) \tag{3}$$

where $\mathbf{Z_a}$ is the acoustic resonance matrix, $\mathbf{C_i}$ $(i = 1, 2)$ is the coupling between panel i and the enclosure, $\mathbf{b_{pi}}$ is the amplitude vector of structural modes on panel i, $\mathbf{Y_{pi}}$ is the structural resonance matrix, $\mathbf{g_p}$ is the generalized modal vector due to the primary excitation, and $\mathbf{p_{ci}}$ is the modal vector due to the control excitation.

Substituting equations (2) and (3) into equation (1), we get

$$\begin{aligned} \mathbf{a} \;=\; & [\mathbf{I} + \mathbf{Z_a}(\mathbf{C_1 Y_{p1} C_1}^T - \mathbf{C_2 Y_{p2} C_2}^T)]^{-1} \\ & \mathbf{Z_a}[\mathbf{C_1 Y_{p1}}(\mathbf{g_p} - \mathbf{p_{c1}}) - \mathbf{C_2 Y_{p2} p_{c2}}]. \end{aligned} \tag{4}$$

If one were to assume $\mathbf{C_1 Y_{p1} C_1}^T = 0$ and $\mathbf{C_2 Y_{p2} C_2}^T = 0$, one would get a weakly coupled system. However, the results in this paper have been obtained using a fully-coupled analysis.

Feedforward control system. The control scheme for harmonic sound is shown in Figure 3. This is a feedforward system, in which r is a reference signal to the controller, A is the frequency response of the sensor to the primary wave incident, W is the frequency response of the control filter, C is the frequency response of the sensor to the control actuator and e is the error signal. In this single-channel system the controller W is driven to set error e to zero.

Feedback control system. In many practical applications the incident sound is random and no reference signal is available. In this case a feedback controller is required. For the optimal controller to minimise a chosen cost function, we implement an internal model control scheme as shown in Figure 4. In this configuration the feedback controller consists of a control filter W and a model of the plant system, $\hat{G}$ as shown in Figure 4(a). If the model is perfect, the control scheme reduces to the form shown in Figure 4(b) and the coefficients of the optimal causal FIR control filter W can be identified from the appropriate autocorrelation and crosscorrelation functions using the Wiener equation.

The optimal filter coefficients obtained by the use of Wiener equation are [14]

$$\mathbf{w_o} = -[\mathbf{R_{rr}} + \beta\mathbf{I}]^{-1}\mathbf{R_{rd}} \tag{5}$$

where $\mathbf{R_{rr}}$ and $\mathbf{R_{rd}}$ are autocorrelation and crosscorrelation functions between the filter disturbance signals, β is the effort weighting coefficient and $\mathbf{I}$ is an $N \times N$ unit matrix. As the volume velocity sensor is used, the corresponding power spectrums are (see the sample for a signal panel in[14])

$$\mathbf{S_{rr}} = |\mathbf{c_2}|^2|\mathbf{d_2}|^2 \tag{6}$$

and

$$\mathbf{S_{rd}} = \mathbf{c_2}^*|\mathbf{d_2}|^2 \tag{7}$$

where $\mathbf{c_2}$ is the volume velocity vector of panel 2 due to the unit control force and $\mathbf{d_2}$ is the volume velocity vector of panel 2 due to the unit primary force, so $\mathbf{R_{rr}}$ and $\mathbf{R_{rd}}$ are the inverse Fourier transform of $\mathbf{S_{rr}}$ and $\mathbf{S_{rd}}$ respectively.

Radiated sound power output. The sound power output due to the primary incident wave and control force together at a frequency of ω is[12]

$$W^r = \mathbf{v}^H\mathbf{R}\mathbf{v} \tag{8}$$

where $\mathbf{v} = \mathbf{v_{p2}} + \mathbf{w_o}\mathbf{v_{c2-f}}$ for harmonic excitation, and $\mathbf{v} = \mathbf{v_{p2}} + \mathbf{d_2}\mathbf{w_o}(\omega)\mathbf{v_{c2-f}}$ for random excitation[14]. $\mathbf{v_{p2}}$ is the velocity vector of panel 2 due to primary incident wave, $\mathbf{v_{c2-f}}$ is the controlled velocity vector of paenel 2 due to an unit control force, and $\mathbf{R}$ is a $d \times d$ matrix giving the real part of the acoustic transfer impedance between each pair of elemental areas of the panel.

NUMERICAL RESULTS

Natural resonance and response. The numerical results presented here have been calculated for aluminium panels of thickness 1 mm and 1.1 mm. Each panel has dimensions 300×380 mm and the distance between the double panels is 100 mm. The panel 1 is excited by a plane wave incident at $\theta = 45^0$ and $\phi = 45^0$.

Figure 5 presents the uncontrolled power transmission ratio for both a single panel (panel 1 only) and double-panel partition for the same incident plane wave excitation. In general, there is an overall improvement in power transmission loss from the introduction of the second panel, particularly at high frequencies. However, the double-panel partition still has a relatively poor sound insulation at some low frequencies due to effective coupling of the wall structural modes and acoustical modes in the sound field between them. The sound transmission loss is worst at the mass-air-mass resonance of the partition (49 Hz).

Cancellation of volume velocity with a point force or a uniform-force actuator. Figure 6 shows the results of cancellation of volume velocity on panel 2 with a structural actuator on panel 2. It shows that good reductions in power transmission ratio can be achieved with a single point force in the low frequency range. The results also show that in the low frequency range an even larger reduction of power transmission can be achieved by using a uniform-force actuator. However, in the high frequency range, the reduction of power transmission is very low. There is spillover at a frequency of about 210 Hz when using a single point-force actuator. The spillover disappeared when using a uniform-force actuator.

Active control at high frequencies. Figure 5 shows that virtually no reduction in power transmission is achievable above 400 Hz. This is in contrast with the corresponding result for a single panel having the same dimensions (given in Figure 11 of [9]) where reductions in radiated sound power approaching 10 dB were possible up to 600 Hz.

In order to explain the poor reduction in the high frequency range, radiated power by each radiation mode[13] has been determined for both uncontrolled and controlled cases. Figure 7(a) shows the contribution of each radiation mode at 25 Hz both with and without control using a uniform force on panel 2 to cancel volume velocity on panel 2. 0 dB represents the total power radiated equal to the sound power incident on panel 1. From Figure 7(a), it can be seen that before control the first radiation mode dominates, as would be expected because the plate is very small compared with an acoustic wavelength. After control, all the radiated power is reduced as the volume velocity is driven to a low value. This results in the reduction of total power transmission at this frequency. Figure 7(b) shows the corresponding results at 350 Hz. At this frequency, before control there are two significantly excited radiation modes. After control, the power due to the first mode is reduced significantly. However, the power due to the second radiation mode remains unchanged, so the total radiated power stays almost the same as in the uncontrolled case. It is apparently a feature of the uncontrolled double-panel system that at high frequencies much of the sound radiation is due to dipole-type motions of the radiating panel (which is not detected by a volume velocity sensor). The double-panel construction provides passive attenuation of the first radiation mode in this frequency range. Therefore, in the high frequency range, there is no advantage in controlling only the volume velocity of a double-panel partition; if further attenuation is required, the second and third radiation modes must be controlled as well.

Cancellation of volume velocity for random excitation. Figure 8 shows random sound transmission for both without and with control by cancellation of volume velocity on panel 2 with a uniform-force actuator on panel 2. No controller delays is assumed. The sample rate is 2000 Hz. From Figure 8, it can be seen that after control, the peaks below 300 Hz decrease. However, there is no significant attenuation achieved above 100 Hz. At 68 Hz, the controlled power stays the same as uncontrolled case. This is because that panel 1 and the enclosure form a resonant system at 68 Hz as the control actuator is applied on panel 2.

CONCLUSIONS

The transmission of harmonic sound power through a double-panel partition can be reduced significantly by the cancellation of volume velocity on either panel in a low frequency range. In a high frequency range, inefficiently radiating even modes of the radiating panel make a dominant contribution to the radiated sound power and so there is no advantage in controlling volume velocity alone in this frequency range. However, the transmission of the sound power from the double-panel partition is significantly reduced at high frequencies by purely passive means.

The performance of the feedback controller for random sound transmission is poorer than for harmonic sound. The difference arises because the controller is constrained to be causal and no time-advanced reference signal is available. It was found that the performance only applies to a very low frequency range which is up to 100 Hz. These findings provided an opportunity to design an active control system for sound transmission through a double-panel partition.

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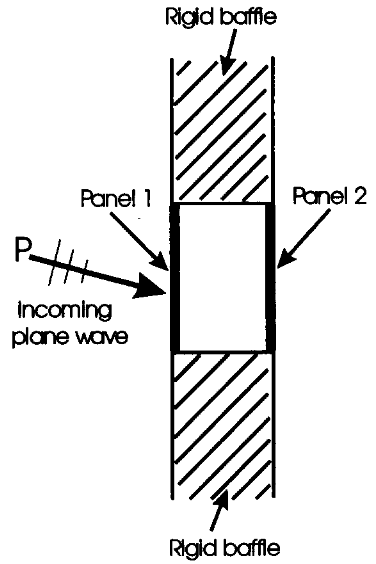


Figure 1. Double-panel partition.

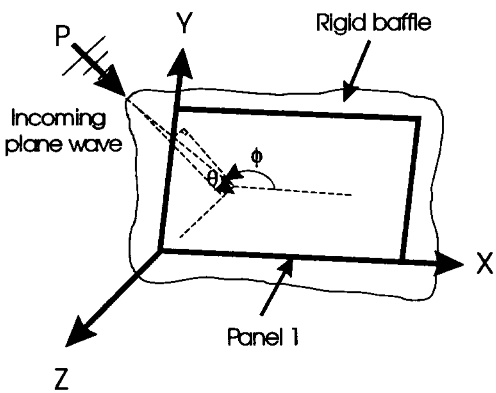


Figure 2(a). Coordinates of panel 1 which is excited by an incident plane wave P.

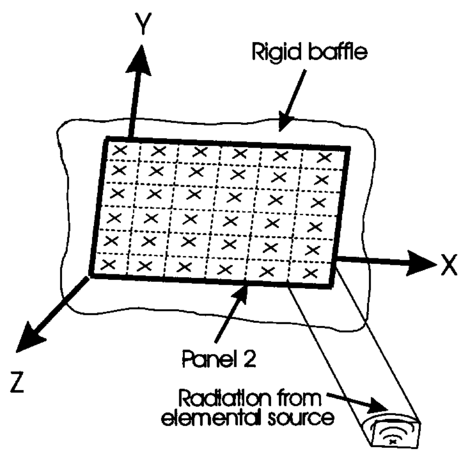


Figure 2(b). Coordinates of panel 2 and sound radiation from the panel using a number of elemental radiators.

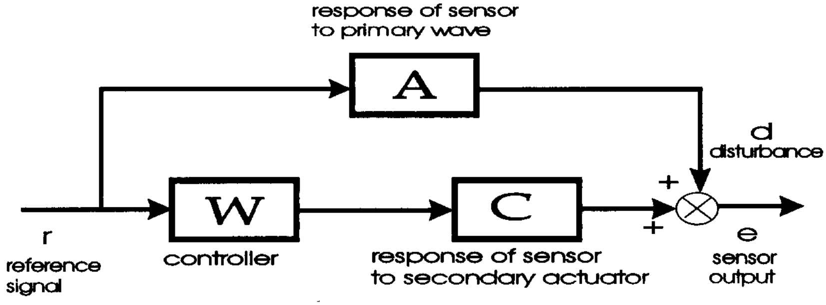
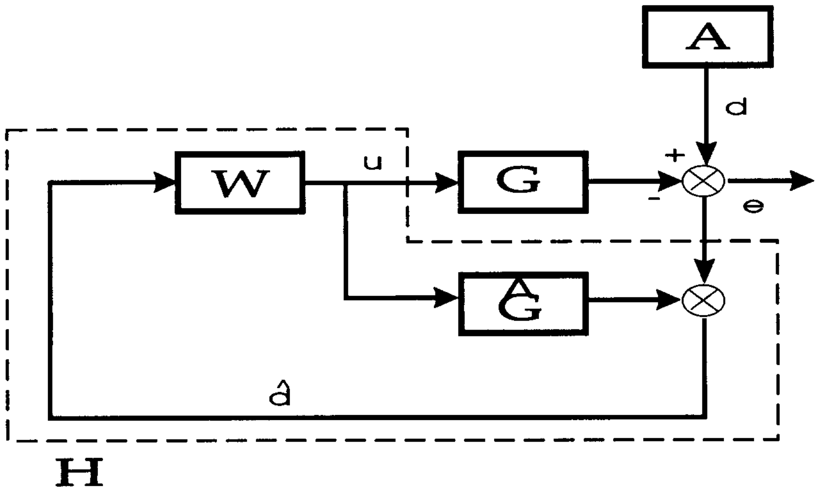
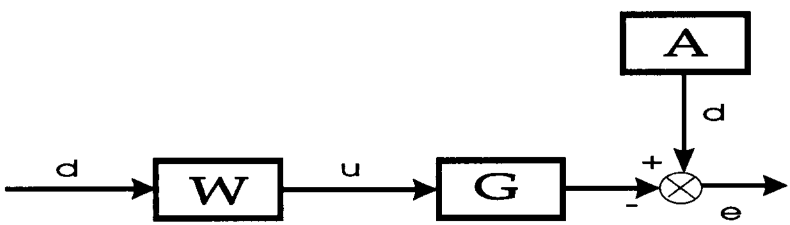


Figure 3: Block diagram of a feedforward control system.



(a)



(b)

Figure 4(a): Block diagram of a feedback control system implemented using internal model control;
(b) as (a) but with an accurate plant model, $G=\hat{G}$.

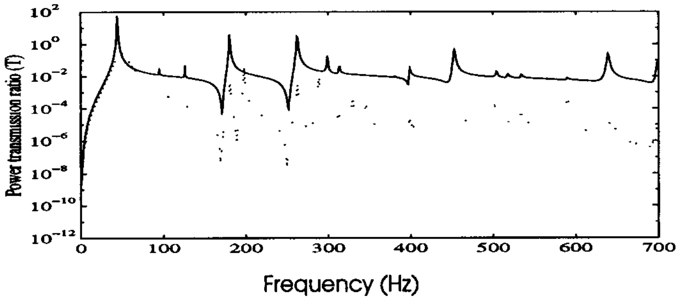


Figure 5. Uncontrolled harmonic power transmission ratios for both a single panel and double-panel partition.
_______ single panel;
.... double-panel partition.

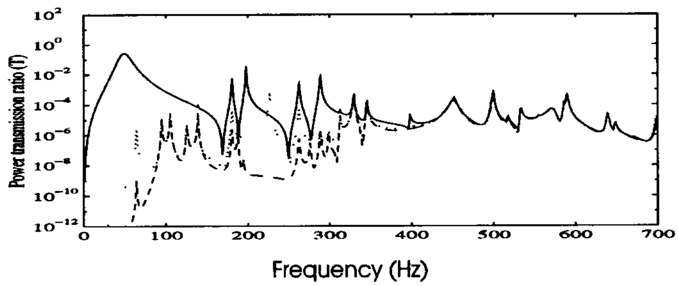
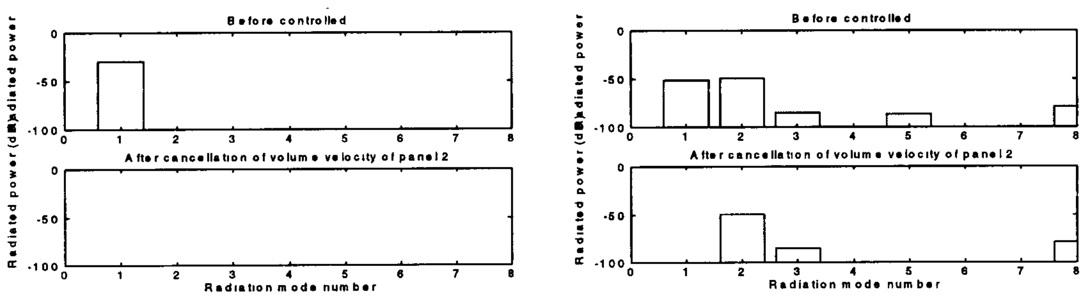


Figure 6. Harmonic power transmission ratios of the double-panel partition, for both without and with control by driving a structural actuator on panel 2 to cancel volume velocty of panel 2;
_______without control;
.............using a central point force;
----------using a uniform-force actuator.



(a) (b)

Figure 7.Harmonic radiated power by the first eight radiation modes.
(a) contribution of each radiation mode at 25 Hz both with and without control;
(b) contribution of each radiation mode at 350 Hz both with and without control.

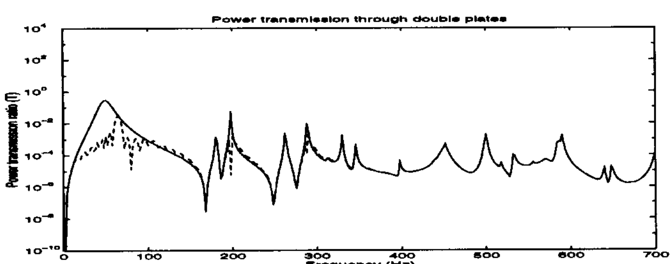


Figure 8. Random power transmission ratios of the double-panel partition, for both without and with control by driving a structural actuator on panel 2 to cancel volume velocity of panel 2;
_______without control;
----------using a uniform-force actuator.