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Invited Paper

DESIGN OF CURVED PANEL SOURCES FOR ACTIVE CONTROL OF SOUND RADIATED BY TRANSFORMERS

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ABSTRACT

Traditional means of controlling sound radiated by large electrical power transformers involve the construction of large, expensive ugly barriers or full enclosures which compromise the maintainability of the transformers. One promising alternative is to use active sound cancellation to reduce the noise. In this paper the work was concerned with the development of a resonant curved panel with a backing cavity which could be more effective, more efficient (regarding electrical power requirements) and more rugged than conventional loudspeakers. The reasons for using a curved panel rather than flat panel were twofold: first a curved panel is more easily excited by piezoelectric patch actuators because the bending vibration of the panel couples better with the extensional motion of the actuators; and second, it is not possible to adjust the resonance frequencies of the efficient modes of a flat panel so easily.

An analytical model was developed to design a curved panel with backing cavity system having resonance frequencies of 100 Hz and 200 Hz for the 1,1 mode and 1,3 mode respectively. Of course there are many possible geometric configurations which can achieve the desired result. An example configuration was designed using the analysis described in this paper and the result verified by finite element analysis (FEA). The panel was constructed, resonance frequencies were measured and the sound radiation as a function of piezoelectric patch actuator excitation voltage was determined.

1. INTRODUCTION

To overcome the disadvantage of traditional noise control which involves the construction of a massive enclosure around the transformers, active noise cancellation has been investigated by several researchers[1]-[8]. Some of them, Hesselmann [2] and Angevine [3], demonstrated that global control could be achieved for a transformer in an anechoic room provided that the transformer was completely surrounded by loudspeakers. In Angevine's work, independent single channel controllers were used to minimise the sound level in the vicinity of each control source. He found that the attenuation was dependent on the number of control sources and that this dependence was stronger at lower frequencies.

More recently, Angevine [6,7,8] has reported some success using multiple loudspeaker sources in front of a transformer tank to achieve significant noise reductions over a wide area (15 to 20 dB over an azimuth angle of 35 to 40 degrees).

However, all of the previously mentioned work has relied on the use of loudspeakers to generate the cancelling noise. The principle disadvantage of loudspeakers is that to achieve global noise control (or even control over a significant angle) a large number of loudspeakers (50 to 100) driven by one multi-channel controller are required.

The main aim of this research was to develop a procedure for the optimal design of tuned panel sound sources for use with active systems for controlling transformer sound. In addition to reducing the number of control sources and simplifying their application, tuned panel sound sources could overcome some disadvantages of the loudspeakers such as limited life and deterioration due to the weather.

2. MODEL OF THE CURVED PANEL WITH A BACKING CAVITY

The curved panel coupled with the cavity model is shown in Figure 1.

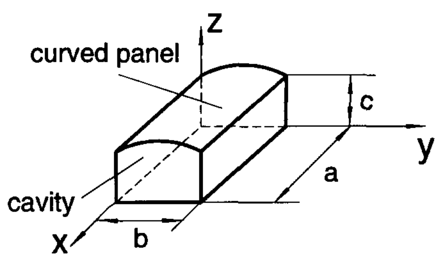


Figure 1. Curved panel with a backing cavity

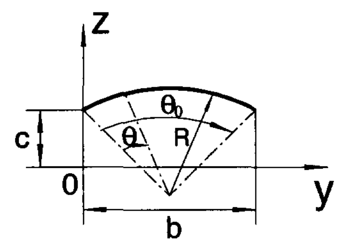


Figure 2. Front view of the curved panel

According to Dowell's work [9], in cases where there is a significant change in a flexible panel resonance frequency due to a backing cavity, the equation of motion of the flexible panel is

$$M_s(\ddot{q}_s + 2\xi_s\omega_s\dot{q}_s + \omega_s^2 q_s) = -(\rho_0 c_0^2 A_F^2 / V) L_{os}^2 q_s + \overline{Q}_s^E e^{i\omega_E t} \tag{1}$$

Rearranging equation (1) gives

$$M_s(\ddot{q}_s + 2\xi_s\omega_s\dot{q}_s + \omega_s^{c\,2} q_s) = \overline{Q}_s^E e^{i\omega_E t} \tag{2}$$

where $\omega_s^{c\,2} = \omega_s^2 + \rho_0 c_0^2 A_F^2 L_{os}^2 / (M_s V)$ (3)

Equation (3) presents the resonance frequency of the panel coupled with a backing cavity.

In the above equations: $M_s \equiv \int_{A_F} m_s \psi_s^2 dA$ is the panel generalised mass for the sth mode, $m_s = \rho_s h$ is panel mass per unit area, ρ_s and h are the density and thickness of the panel, respectively. Thus $M_s = \rho_s h \int_{A_s} \psi_s^2 dA$ for a uniform thickness panel. (4)

The quantities ψ_s and ω_s are the sth natural mode and resonance frequency of the panel, respectively, ρ_0 and c_0 are the mean fluid density and sound velocity within the cavity respectively, A_F is the panel area and V is the volume of the cavity.

The quantity L_{0s} is the generalised panel exciting force and is defined by $L_{0s} = \frac{1}{A_F}\int_{A_F} F_0 \psi_s dA$ (5)

The geometrical parameters of the preceding equations are defined in Figure 2.

Soedel [10] gives an expression for the mode shapes of a curved panel for which the boundary conditions are simply supported on all four edges as follows

$$\psi_{mn}(x,\theta) = \sin\frac{m\pi x}{a}\sin\frac{n\pi\theta}{\theta_0} \tag{6}$$

where n is the number of axial node lines (plus one) and m is the number of circumferential node lines (plus one) on the shell. The quantities θ and θ_0 are defined in the Figure 2 and the quantities x and a are defined in the Figure 1.

Substituting equation (6) into equations (4) and (5), rearranging the geometric parameters, then integrating M_{mn} and L_{0mn} over A_F gives

$$M_{mn} = \frac{\rho_s habR}{2n\pi}\left(\frac{n\pi\theta_0}{2b} - \frac{1}{4}\sin\frac{2n\pi\theta_0}{b}\right) \tag{7}$$

$$L_{0mn} = \frac{2b}{mn\pi^2\theta_0}\sin^2\frac{n\pi\theta_0}{2b}(1-\cos(m\pi)) \tag{8}$$

The resonance frequency ω_{mn} of the curved panel is given by [10],

$$\omega_{mn} = \frac{1}{R}\sqrt{\frac{(m\pi R/a)^4}{[(m\pi R/a)^2 + (n\pi/\theta_0)^2]^2} + \frac{(h/R)^2}{12(1-\mu^2)}\left[\left(\frac{m\pi R}{a}\right)^2 + \left(\frac{n\pi}{\theta_0}\right)^2\right]^2} \times \sqrt{\frac{E}{\rho_s}} \tag{9}$$

Substituting equations (7) and (8) and the square of equation (9) into equation (3) gives

$$\begin{aligned}\omega_{mn}^{c\,2} = {} & \frac{1}{R^2}\left(\frac{(m\pi R/a)^4}{[(m\pi R/a)^2 + (n\pi/\theta_0)^2]^2} + \frac{(h/R)^2}{12(1-\mu^2)}\left[\left(\frac{m\pi R}{a}\right)^2 + \left(\frac{n\pi}{\theta_0}\right)^2\right]^2\right)\frac{E}{\rho_s} \\ & + \frac{32\rho_0 c_0^2 abR}{\rho_s h m^2 n \pi^3}\frac{\sin^4(n\pi\theta_0/(2b))(1-\cos m\pi)^2}{[2n\pi\theta_0/b - \sin(2n\pi\theta_0/b)]V}\end{aligned} \tag{10}$$

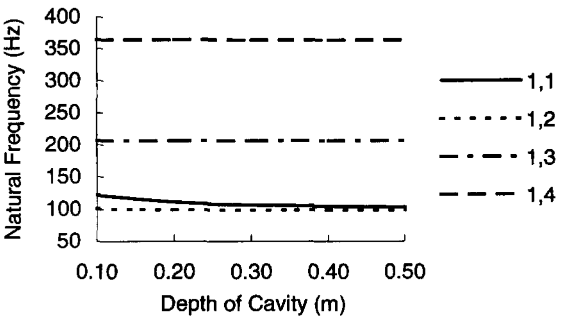
By using equation (9) the curved panel with a backing cavity can be designed to be resonant at the required frequencies.

3. EXAMPLE DESIGN

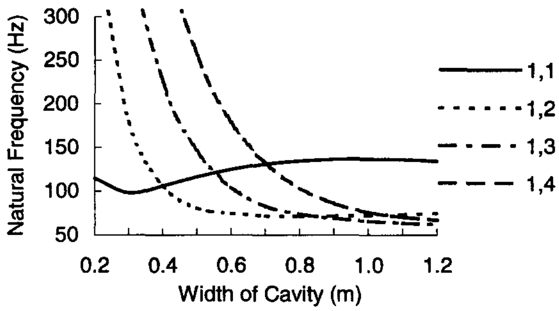
3.1. Specification of the Geometry Size of the System

In this design the material of a curved panel is aluminium and the cavity is made of medium density fibre (MDF) wood with a thickness of 25 mm. The parameters which represent the system properties are ρ_s=2700, E=70300, μ=0.345, ρ_0 = 1.21 and c_0=344.

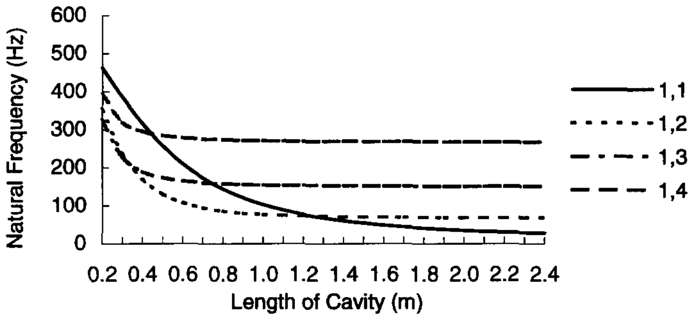
To help optimise the design of the system, the effect of variations in the cavity size on the panel natural frequencies for a panel thickness of 0.0016m are given in Figure 3.



(a) depth of cavity vs natural frequency



(b) width of cavity against natural frequency



(c) length of cavity against natural frequency

Figure 3. Variations of natural frequencies of the panel-cavity system as a function of cavity size

According to Wallace's work [11], the 1,1 and 1,3 modes for a simply supported panel have the highest radiation efficiencies at low frequencies; thus, these two modes have been selected for excitation on the curved panel. The intention is to design a panel such that the 1,1 mode resonance frequency occurs at 100Hz and the 3,1 resonance frequency occurs at 200Hz. The dimensions of the panel-cavity system which give $f_{1,1}$=108 Hz and $f_{1,3}$=206 Hz are a=0.985, b=0.418, c=0.25, h=0.0016 and l_0=0.420 (arc length of the curved panel).

The numerical frequencies have been selected to be higher than required as it is easier to reduce resonance frequencies in practice by adding mass to the panel, but it is not very easy to increase the frequencies by subtracting mass from the panel. The calculated resonance frequencies of the system are given in Table 1.

Table 1 Resonance frequencies of the shallow panel with a backing cavity (Hz)

	n=1	n=2	n=3
m=1	108 (105)	98 (95)	206 (202)
m=2	259 (259)	141 (139)	223 (218)
m=3	382 (379)	217 (214)	256 (250)

Note: Data in brackets represent calculations made using ANSYS.

3.2. Adjusting the Natural Frequencies of the System

To obtain the required natural frequencies ($f_{1,1}$=100 Hz and $f_{1,3}$=200 Hz) after the panel-cavity system is constructed, masses may be attached to the panel or the depth of the cavity may be changed to fine tune the natural frequencies of the system after the system has been constructed. Fortunately, as will be shown, the 1,3 mode is much less sensitive than the 1,1 mode to changes in the depth of the backing cavity and the 1,1 mode can be made relatively insensitive to the presence of point masses attached to the panel in properly chosen locations.

3.2.1 *Mass Attached to the Curved Panel*

The characteristic equation for the curved panel with an attached mass [10], M^*, is

$$\frac{4}{\rho_s haR\theta_0}\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{1}{\omega_{mn}^2-\omega^2}\sin^2\frac{m\pi x^*}{a}\sin^2\frac{n\pi\theta^*}{\theta_0}-\frac{1}{M^*\omega^2}=0 \tag{11}$$

where (x^*,θ^*) is the position of the attached mass and ω_{mn} is the original resonance frequency.

By rearranging equation (11) the equation for the system with a mass attached to the panel may be written in form

$$M^*=\frac{\rho_s haR\theta_0}{4\omega^2}\frac{1}{\sum_{m=1}^{\infty}\sum_{n=1}^{\infty}\frac{1}{\omega_{mn}^2-\omega^2}\sin^2\frac{m\pi x^*}{a}\sin^2\frac{n\pi\theta^*}{\theta_0}} \tag{12}$$

3.2.2. *Adjusting the Depth of the Backing Cavity*

As shown in Figure 3, the variation of the natural frequency of the system with the depth of the cavity is obvious in mode 1,1 but not much in mode 1,3. Thus it is possible to change the resonance frequency of the 1,1 mode by adjusting the cavity volume without affecting the resonance frequency of the 1,3 mode significantly.

4. EXPERIMENTAL WORK

4.1 Resonance Frequency Tests

4.1.1 *Experimental Set-up*

The curved panel with backing cavity was tested using experimental modal analysis to determine the resonance frequencies of the 1,1 mode and 1,3 modes.

The modal testing technique involved the use of a B&K type 8202 impact hammer to excite the panel. Frequency response data were recorded using a B&K type 4394 accelerometer connected through a B&K type 2635 charge amplifier to a B&K type 2032 dual channel signal analyser. The data were transferred into a PC in which the modal analysis software, PC Modal, was installed.

4.1.2 *Experimental Results*

In Figure 4, the measured frequency response of the panel is shown. The first peak (76 Hz) corresponds to the 1,1 mode resonance and the second peak (199 Hz) corresponds to the 1,3 mode resonance. These frequencies can be compared to the corresponding theoretical predictions of 108 Hz and 206 Hz respectively. Although the difference between measured and predicted resonance frequencies for the 1,3 mode is acceptably small, the difference for the 1,1 mode is unacceptably large. The reason for this is the high sensitivity of the 1,1 mode resonance frequency to errors in the radius of curvature of the panel and the difficulty in manufacturing panels of uniform and accurate radii of curvature.

As mentioned in the previous section, the resonance frequency of the 1,1 mode may be adjusted by varying the depth of the backing cavity. This was done experimentally and the effect of backing cavity depth on the 1,1 and 1,3 mode resonance frequencies is shown in Figure 5.

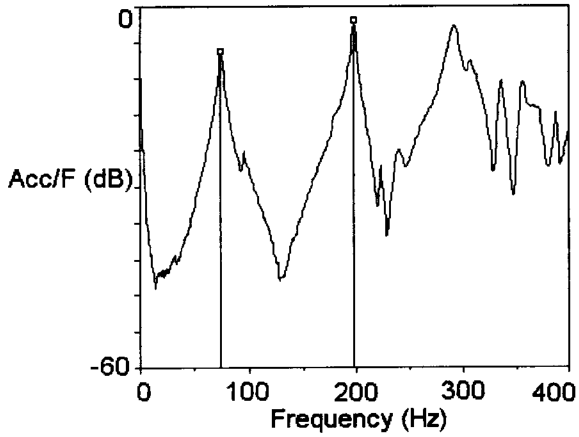


Figure 4 Acceleration response of the curved panel with a backing cavity

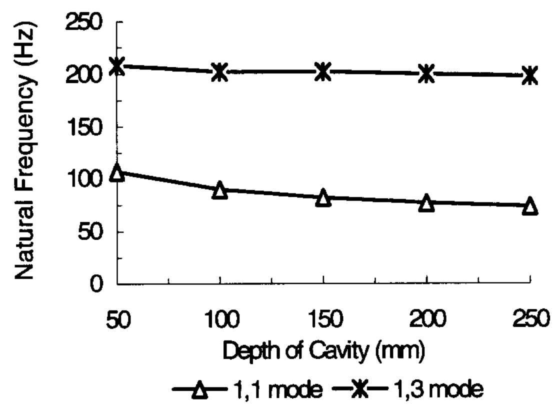


Figure 5 The effect of variations in backing cavity depth on the panel resonance frequency

From Figure 5 it is seen that the required depth of cavity for the 1,1 mode to be resonant at 100Hz is 58 mm. The resonance frequency of the 1,3 mode is only affected by a small amount by variations in the backing cavity depth as predicted theoretically in the previous section.

The resonance frequency of the 1,3 mode was slightly above 200 Hz when the cavity depth was reduced. However this was reduced to 200 Hz by the addition of a 22 gram mass at $a/2$, $b/3$.

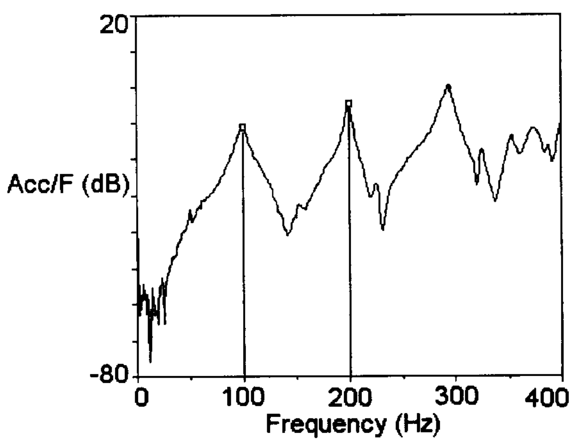


Figure 6 Frequency response of the adjusted curved panel with backing cavity

The measured frequency response for the panel with a 58 mm backing cavity and a small mass of 22 grams fixed at $a/2$, $b/3$ is shown in Figure 6, where it can be seen that adjustment of the experimental model has resulted in the 1,1 mode being resonant at 100 Hz and the 1,3 mode being resonant at 200 Hz.

4.2 Sound Radiation Tests

4.2.1 *Experimental Set-up*

The panels were excited by 6 piezoelectric patch actuators. Three actuators were attached on each side and driven in phase but out of phase with the actuators on the other side. The actuators were driven by an oscillator which generated a sine wave signal through a B&K type 2706 power amplifier and a 7:1 step-up transformer. An HP 54601A oscilloscope was used to monitor the driving signal. To measure the directivity of the radiated sound fields around the panel-cavity system, a B&K type 3921 turntable was used to keep the panel-cavity system turning during the measurement. SPL data were recorded using a B&K type 4133 microphone connected through a B&K type 2604 microphone amplifier and filter, and A/D board to a computer. The reason for using the computer to record data is that it allowed data to be recorded automatically when the panel-cavity system was turning on the turntable. The experimental set-up is shown in Figure 7.

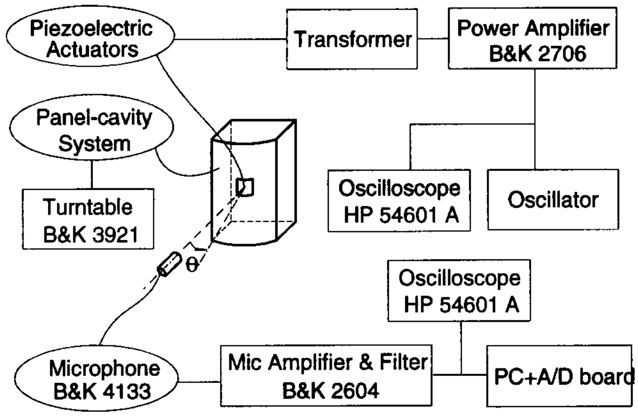


Figure 7 Experimental set-up for sound radiation measurement

4.2.2 *Experimental Results*

The directivities of the radiated sound fields around the panel-cavity system are shown in Figures 8 and 9 using a polar co-ordinate system. The measurements were taken at 100 Hz and 200 Hz corresponding to the 1,1 mode, 1,3 mode respectively. To measure both near-field and far-field conditions, the microphone was located at 0.5 m and 3.44 m from the panel-cavity system respectively and the height was slightly above the centre of the panel. The input peak voltage of the actuators was 37.5 V, which is about 5 times less then the maximum allowed.

To study the effect of the variation of distance between the panel and sensing point on the sound radiation levels, the sound pressure levels of the panel-cavity system were measured as a function of the distance of the microphone location from the panel, with V_p=37.5 V, and with a total of 3 actuator pairs driven simultaneously. The results are shown in Figure 10.

To analyse the effect of the number of actuators on the panel sound radiation, the sound pressure level of the panel-cavity system was measured as a function of number of the actuators, with V_p=37.5 V and microphone at 1 m. The results are given in Figure 11.

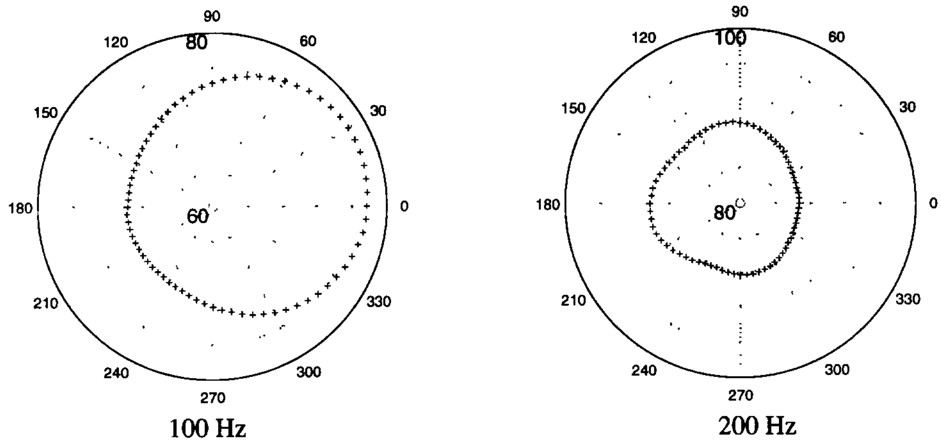


Figure 8 Sound pressure level (dB) as a function of angular location θ, with the microphone at 0.5 m

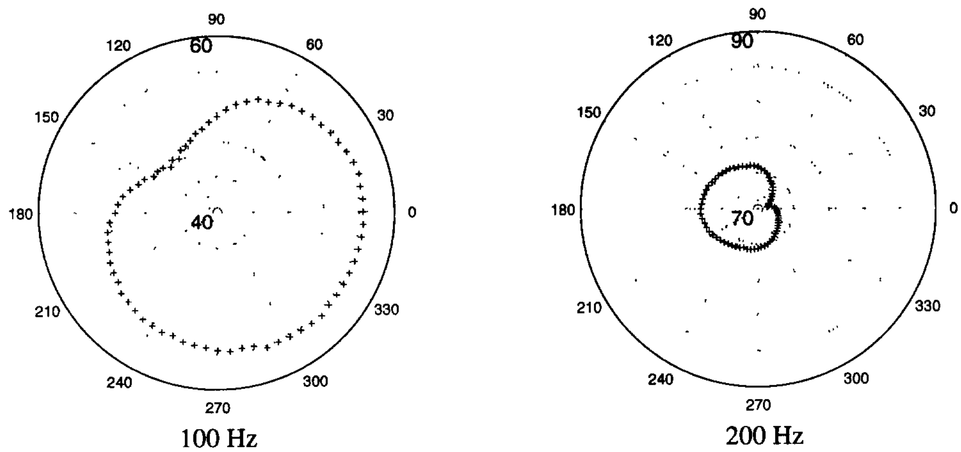


Figure 9 Sound pressure level (dB) as a function of angular location θ, with the microphone at 3.44 m

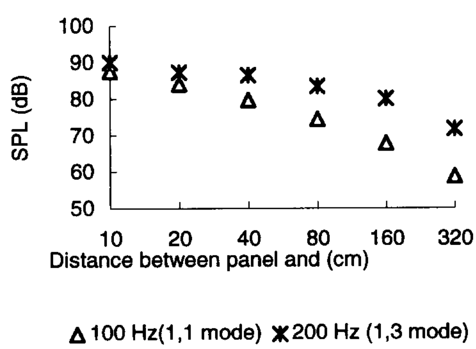


Figure 10 Variation in SPL as a function of distance from the panel

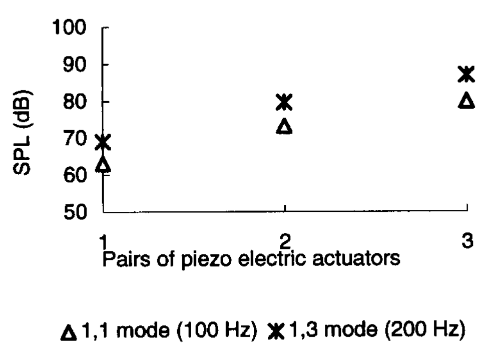


Figure 11 Variation in SPL as a function of the number of actuator pairs driving the panel

5. CONCLUSIONS

The work described here has demonstrated that a curved panel with backing cavity sound source could be used as an alternative to a number of loudspeakers for active cancellation of electric power transformer noise. Furthermore, it has been shown here that a single panel may be tuned to resonate at 100 Hz and 200 Hz, thus enabling one source type to control the two dominant noise radiating frequencies of the transformer.

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