# THE ACTIVE CONTROL OF THE TRANSMISSION OF SOUND

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## 1.INTRODUCTION

There is currently a great deal of interest in the active control of the sound field within the cabins of propeller driven aircraft. It has been demonstrated that the sound field produced in such an aircraft cabin at the blade passing frequency and its second harmonic can be controlled using a number of loudspeakers to minimise the sum of the squares of the pressures at a number of microphone positions (1). An alternative approach (suggested, for example, by Fuller (2)) is to control the internal sound field by use of secondary control forces on the fuselage. Most of the work in this area to date has consisted of a number of computer simulations investigating the active control of the transmission of sound through a thin cylindrical shell model of an aircraft fuselage (3). In order to investigate the basic problem of the active control of the transmission of sound, a series of experiments investigating the active control of sound transmission through a clamped rectangular steel plate have been undertaken. The results from the first of these experiments are presented here . A brief, and by no means complete explanation of these results is attempted using an approximate modal analysis of the plate vibrations, with and without control. With regard to the the active control of the transmission of sound the results are considered to be both interesting and encouraging.

## 2.EXPERIMENTAL RESULTS

The experiment was carried out using an existing multi-channel active control system originally built for use in flight trials carried out in 1988 (1). The system uses a variant of the L.M.S. algorithm (4) to minimise the sum of the squares of the pressures at, in this case, 24 microphone positions by varying the forces applied to the plate through three coil in magnet actuators. The 24 microphones were randomly distributed about the larger of the two reverberation chambers between which the plate was mounted (Fig. 1). The reverberation time in the large chamber (of volume $348m^2$), in the 80Hz third octave band is approximately 11.5s, which gives a Schroeder cut-off frequency of 60Hz. The frequencies at which the experiments were carried out were all above 80Hz, so, for these experiments the sound field in the large reverberation chamber can be considered diffuse. A single frequency noise source was placed in the smaller of the two chambers. The principal measurement made was the sum of the squares of the pressures at the 24 control microphones defined as

$$J_{p24} = \sum_{n=1}^{24} | p_n |^2. \qquad (1)$$

The reduction in this quantity on the introduction of active control is a measure of the effectiveness of the control, as it provides an estimate of the sound power radiated by the plate.

The active control system could only provide stable control when set to converge very slowly. Even with such a slow rate of convergence, however, the system would not always converge to a steady level of $J_p$, which may have produced error in some of these results. The results were, in any case, subject to a variability between the same experiments carried out at different times of up to 2dB.

The experiments were carried out at a range of frequencies between 80 and 100 Hz with all possible combinations of the the three coils mounted on the plate (Fig. 2). Coil 1 was at x=48cm, y=25cm; coil two was at x=16.5cm, y=20cm; coil three was at x=27cm, y=63cm. The plate has a low modal density in this frequency range and hence any dependence of the active control on the resonant behaviour of the plate could hopefully be seen. A number of different primary source positions were used to investigate the sensitivity of results to different acoustic exitation. Figure 1 shows the four different primary source positions that were tried. Figure 3 shows the variation of the reduction in $J_{p24}$ with primary source position and frequency using all three coils. It evident that that $\Delta J_{p24}$ varies greatly with both frequency and source position.

Figure 4 shows the variation of $\Delta J_{p24}$ with frequency and secondary force position when only one controlling force is used. The single primary source position used in this instance is position 1. Figure 5 shows the maximum $\Delta J_{p24}$ obtainable with one, two or three secondary forces in use over the frequency range of the experiment with the primary source in position 1. It can be seen that the number of forces used makes little difference to the reductions in transmitted sound in this frequency range.

## 2.MODAL ANALYSIS

In order to gain some understanding of the mechanism of control in terms of the vibration of the plate, modal decomposition similar to that used by Fuller *et al* (5) was carried out. To carry out the modal decomposition 12 accelerometers were distributed over the surface of the plate.Their positions are shown in figure 6. The relationship between the mode amplitudes and the displacement amplitudes at these 12 positions can be written

$$\mathbf{\Phi}\,\mathbf{a} = \mathbf{x} \tag{2}$$

where a is the vector of N complex mode amplitudes, x is the vector of 12 complex displacement amplitudes at the accelerometer positions, and $\Phi$ is the 12xN matrix of mode-shape function values at the 12 accelerometer positions, which is of the form

$$\mathbf{\Phi} = \begin{bmatrix} \phi_1(x_1,y_1) & \phi_2(x_2,y_2) & \cdots & \phi_1(x_m,y_m) & \cdots & \phi_2(x_{12},y_{12}) \\ \cdot & \cdot & & & & \\ \phi_r(x_1,y_1) & \phi_r(x_2,y_2) & \cdots & \cdots & & \\ \cdot & \cdot & & & & \\ \cdot & \cdot & & & & \\ \phi_N(x_1,y_1) & \phi_N(x_2,y_2) & \cdots & \cdots & & \cdots\phi_N(x_{12},y_{12}) \end{bmatrix}$$

where $\phi_n(x_m, y_m)$ is the value of the $n^{th}$ mode-shape function at the $m^{th}$ accelerometer position. If we premultiply equation (2) by $\Phi^T$ we obtain

$$\Phi^T \Phi\, a = \Phi^T x. \qquad (3)$$

Hence a is given by

$$a = (\Phi^T \Phi)^{-1} \Phi^T x \qquad (4).$$

The estimate of a given by this expression obtained by using a QR decomposition of $\Phi^T\Phi$. In order to carry out this decomposition one needs a set of modeshape functions for the plate. Approximate values for the relative mode amplitudes may be obtained by using the mode-shape functions for a plate simply supported on all edges. These can be written as

$$\phi_{r_1 r_2}(x,y) = \sin\left(\frac{r_1 \pi x}{L_x}\right)\sin\left(\frac{r_2 \pi y}{L_y}\right) \qquad (5)$$

where $L_x$ is the longest dimension of the plate, $L_y$ is the shorter dimension, $r_1$ and $r_2$ are the mode integers for the x and y directions respectively. Note that these functions do not satisfy the conditions of zero slope at the edges of the plate, but they provide a first approximation of the relative mode amplitudes which reveal something of the mechanism of control.

Figure 6 shows the relative mode amplitudes of the first nine modes (thus $\Phi$ was assumed to be 9x12), given by this method, for the four different primary source positions. Figures 7 and 8 show the relative mode amplitudes of the first nine modes with and without control at 80Hz and 85Hz respectively.

## 3.DISCUSSION

The first thing to note is the magnitude of the variation in $\Delta J_{p24}$ with primary source position,which is, perhaps, a little surprising. It can, in part, be explained by the fact that in this frequency range the sound field in the source reverberation chamber produces an incident pressure field that is not uniform. This being the case, then the extent to which a particular plate mode is excited is not merely dependent on frequency, but also on this variation in incident pressure across the plate, which in turn is dependent on the position of the primary source. The dependence of the relative mode amplitudes on source position is shown by the modal decomposition result given in figure 6. Measurements were carried out and it was found that at 100Hz there were variations of up to 60° in the phase of the incident sound pressure from one point on the plate surface to another depending on source position and frequency. At 80Hz, however, there was a uniform incident pressure, no matter what the source position. The modal decomposition also showed that there is little variation in relative mode amplitudes at 80Hz. There are, however still variations in ΔJp24 at this frequency, so it appears that the non-diffuse nature of the source room is not a sufficient explanation for the large variations of he results with source position.

The plate has two modes in the frequency range of the experiment, one at 80Hz, and one at 88.5Hz. However the highest reductions were obtained at 82Hz,which is close to the natural frequency of 80Hz corresponding to the mode depicted in figure 9. It appears that there is no simple relationship between the effectiveness of active control and the relative contribution of these modes. One reason for this is the differing radiation efficiencies of the two modes in the frequency range of interest. Figure 7 shows that at 80 Hz the most dominant mode is that with a natural frequency of 80Hz. This mode is a monopole type mode (6) and is therefore a more efficiently radiating mode than that with a natural frequency of 88.5Hz, which is a quadrapole type mode. The 80Hz mode dominates the response through most of the frequency range of interest. Figure 7 shows us that the introduction of active control shifts the vibrational energy of the plate from the efficiently radiating (2,4) mode to the less efficiently radiating, dipole type, (3,2) mode, also depicted in figure 9. This is also the case with an 85Hz exciting frequency as is shown by figure 8.

The effect of control force position (Fig 4) is not particularly dramatic near the plate resonances, but as frequency increases the position of a single controlling force seems to become more critical. It has been mentioned that at most frequencies in this range there is one mode which dominates the plate response and hence the only position at which a single force could not significantly control the plate's vibration would be on a node of that mode. Figure 5 shows that it makes little difference how many forces are used to control the plate. This also indicates that for most of the frequency range used there is one dominant mode. This also, perhaps, explains the large variations in $\Delta J_{p24}$ that seem to occur with different force positions at 100Hz where one mode does not dominate the response.

## 4.CONCLUSIONS

These experiments have shown that it is possible to use active control forces to control the transmission of sound through a clamped plate, both when the plate is on resonance and off resonance. It is difficult to draw any other firm conclusions from these results because changing the primary source position brought about such large variations in the results, but it is clear that the relationship between the active control and the modal response of the plate is complex. Further investigation involving analysis of the vibration of the plate may help explain the nature of the results and also shed light on the mechanism of control.

## 5.REFERENCES

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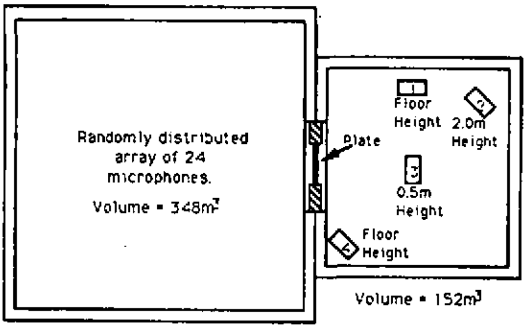


Figure 1. The reverberation suite, showing the clamped steel plate and four primary source positions.

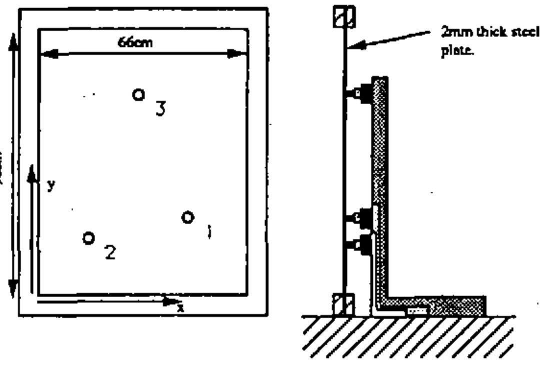


Figure 2. Positions of the three coils and the mounting of the associated magnets.

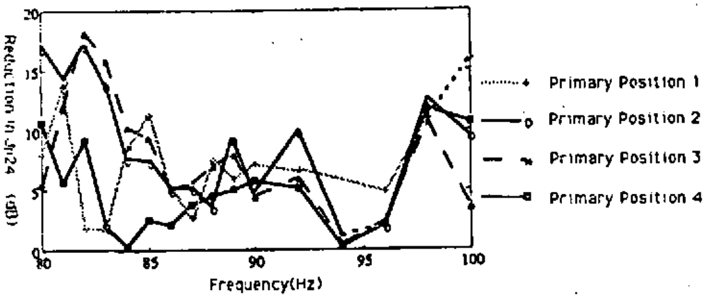


Figure 3. The variation ΔJp24 with frequency and primary source position.

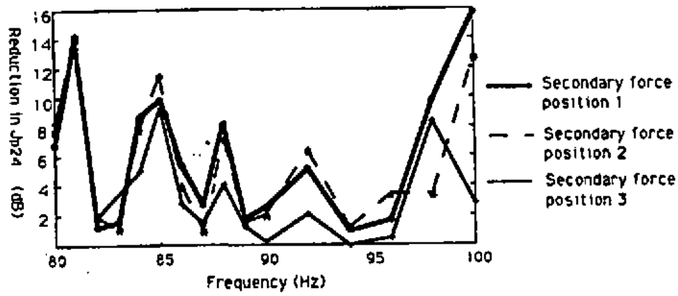


Figure 4. Variation ΔJp24 with frequency and secondary force position.

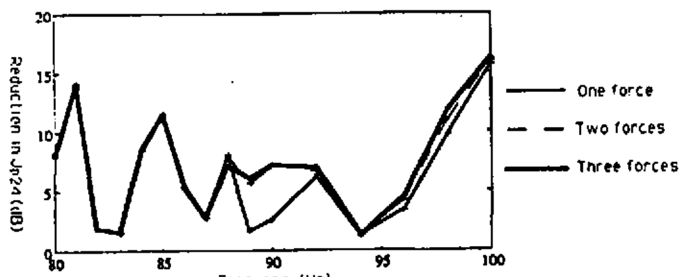


Figure 5. The maximum attenuation obtainable with one, two and three control forces over the frequency range of interest.

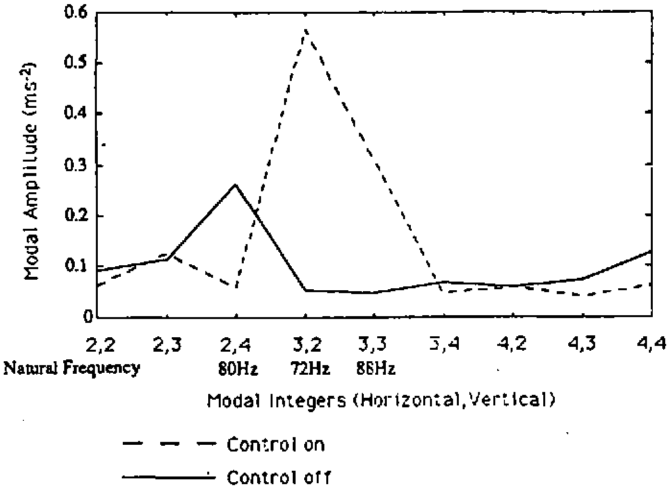


Figure 7. Modal amplitudes of the first nine modes, with and without control, at 80 Hz.

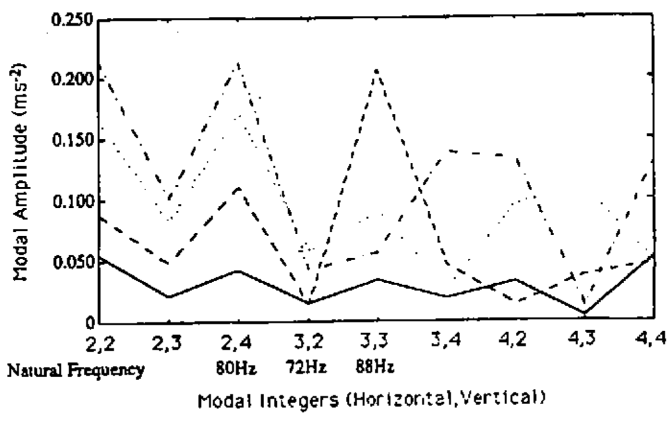


Figure 6. Modal amplitudes of the first nine modes for four different primary source positions at 100Hz.

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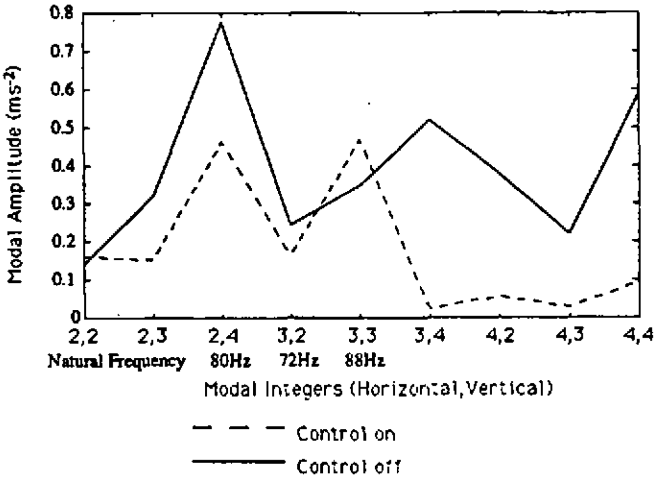


Figure 8. Modal amplitudes of the first nine modes, with and without control, at 85Hz.

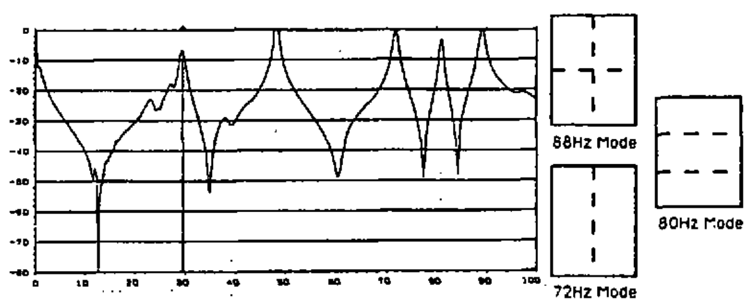


Figure 9. The input impedence of the plate with modeshape diagrams for the three modes of interest.