Modal Analysis for Global Control of Broadband Noise in a Rectangular Enclosure

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Modal analysis for examination of the possibility of the global reduction of broadband noise in a rectangular enclosure has been studied in this paper. By developing a modal model of the acoustic environment of the enclosure the shape mode and their frequencies are obtained and on suitable bandwidth for the controller design is calculated. In order to simplify the analysis the noise sources are assumed localized and internal in one corner of the kiosk. The analysis results show that due to the existence of some degenerate modes in this application, the control action is complicated. Simulation results demonstrate how positions and number of loudspeakers and microphones will change the performance of the controller and its corresponding control effort. Based on these results and analysis a reasonable size for the controller and proper locations for sensors and actuators are proposed. Simulation results also confirm the effectiveness of the proposed system in reduction of the acoustic potential energy in the kiosk.

1. INTRODUCTION

Initial consideration of the active control of an enclosed sound field dates back to the work by Olson and May [1]. Because of advances in microprocessor technology, interest in active control of enclosed sound fields has increased dramatically since the early 1980s, focusing on two main applications. One of the frequently reported applications is to reduce the cabin noise in aircraft [2, 3, 4]; such active noise control (ANC) systems are installed and used in production as reported in [5]. Another documented application is for automobile cabin noise [6, 7, 8]; now it is possible to purchase an automobile equipped with an ANC system installed [9].

There are two major strategies for the implementation of an ANC system in an enclosed space: local control and global control [10]. The aim in global control is to reduce sound at all points in an enclosure, while the main objective in local control is to generate a series of quiet zones at some specific points. Selection of the proper strategy for the application in hand is not straightforward. However, previous work in this area may give some views in the possibility of each strategy. In [11], it was shown that when modal densities in enclosure are high, global control is possible if the distance between control source and primary source is less than the wavelength of the noise in enclosure, otherwise local control is the only possible option.

Practical results show that global reduction of sound in an enclosed space is feasible when the enclosure dimensions are smaller than one half the wavelength of sound which is intended to be reduced [12]. Local control is studied in the literature mostly in situations when the sound field in the enclosure is described statistically [13, 14,15,16], or when a large number of modes contribute [17, 18].

Phones are currently common devices used for communications. However, in many places, such as a noisy street or in an industrial place, communication suffers from excessive noise. A common way to reduce such environmental noise is to place the phone in an enclosure. However, since the main components of street or industrial noise cover frequencies from below 100 Hz to over 500 Hz, active noise control (ANC) systems are a potential solution to improve the quality of communication in telephone kiosks by reducing the noise inside the cabin. Since the sound fields in most practical enclosures such as rooms, classrooms or telephone kiosks considered in this study are neither of low modal density nor diffuse [19], the feasibility of the practical implementation of ANC systems for global reduction of sound in each application needs to be investigated.

In this paper, a feasibility study for the implementation of an ANC system for global reduction of broadband noise in a rectangular enclosure with a telephone kiosk application is investigated. For this purpose, first a simplified model of the acoustic environment of enclosure is obtained by solving the wave propagation equations in the enclosure in the frequency domain and using modal analysis techniques. Because of the selected dimensions for the enclosure, the existence of some degenerate modes has complicated the analysis, and led to some new results which have not been studied previously. Based on the modal information obtained a measure of the bandwidth of the ANC system where global control of noise may be effective is proposed in section 3. By formulation of the acoustic potential energy of the enclosure in section 4, the effect of positions and number of loudspeakers and microphones in a multi-channel ANC system for broadband control of noise is studied in section 5 and finally a conclusion is reached.

2. MODAL DESCRIPTION OF ENCLOSURE

Although in a real kiosk the walls may be flexible and the noise sources are external, as a preliminary work to simplify the analysis the walls are assumed to be rigid and the noise source is local and internal in one corner of the enclosure. The governing wave equation in a three dimensional environment, taking into account viscous damping of the environment and assuming a sound source with specific volume velocity distribution in the right hand side, may be expressed as:

$$\frac{1}{c_0^{\,2}}\frac{\partial^2}{\partial t^2}p(\mathbf{x},t)-\mu\nabla^2\frac{\partial}{\partial t}p(\mathbf{x},t)-\nabla^2 p(\mathbf{x},t)=\rho_0\frac{\partial}{\partial t}g(\mathbf{x},t) \qquad (1)$$

where C_0 is the speed of sound in the enclosure, $p(\mathbf{x},t)$ is the acoustical pressure at point $\mathbf{x}$ and time t, ρ_0 is the density of air, μ is a constant indicating damping in the air or walls, and g(x,t) is the volume velocity of sound per unit volume. Since the acoustical pressure p in comparison with the atmosphere pressure is very small, the linearity of (1) is assumed. In addition to (1), there are six boundary conditions assuming the walls of the enclosure are rigid. As the velocity on the walls is zero, the acoustical pressure will not transmit outside and hence the pressure gradient on the walls will become zero. As a result, the boundary conditions of (1) will become:

$$\begin{aligned} p_x(x,y,z,t) &= 0 \qquad x = 0,L_1 \\ p_y(x,y,z,t) &= 0 \qquad y = 0,L_2 \\ p_z(x,y,z,t) &= 0 \qquad z = 0, L_3 \end{aligned} \qquad (2)$$

The experimental kiosk planned for implementation of the ANC system, and its model is shown in Fig. 1. The main difference between the current analysis and the previous ones in this field [20, 21] is that the dimensions of the selected enclosure are proper multiples of each other. This property will result in some degenerate modes in the analysis, and this will complicate the control action. In [20] with the assumption that the variations of mode shapes are negligible in comparison with other two dimensions, the problem is studied in a two dimensional case, and in [21] the dimensions of enclosure are chosen such that each resonance frequency corresponds to one mode shape, and hence there are no degenerate modes. The solution of (1) with boundary conditions in (2) can be expressed in terms of some basic functions, called the acoustical modes of enclosure. The formulations used here for this purpose are the same as in [11], and are not included here for the sake of brevity. The damping ratio of all the different modes is chosen equal to 0.01, and the resonance frequencies of the kiosk in the range of 0 to 300 Hz are calculated in Table I. Having a graphical view of mode shapes of enclosure will be of great help for suitable placement of loudspeakers and microphones in an ANC system.

In Figs 2 and 3, the mode shapes at resonant frequencies 255 Hz and 240 Hz are illustrated respectively. Among the mode shapes of enclosure, the one that has resonance at 170 Hz is the sum of three axial modes, and others at 190.1 Hz and 240.4 Hz have two and three tangential modes correspondingly. Moreover, each of the resonance frequencies at 255 and 294 Hz have one oblique mode.

Table I Resonance frequencies calculated between 0 to 300 Hz

No.	Freq. (Hz)	Mode Contribution
1	85	(0,0,1)
2	170	(1,0,0)(0,1,0)(0,0.2)
3	190,1	(1,0,1)(0,1,1)
4	240.4	(1,1,0)(1,0,2) (0,1,2)
5	255	(1,1,1)(0,0,3)
6	294.4	(1,1,2)

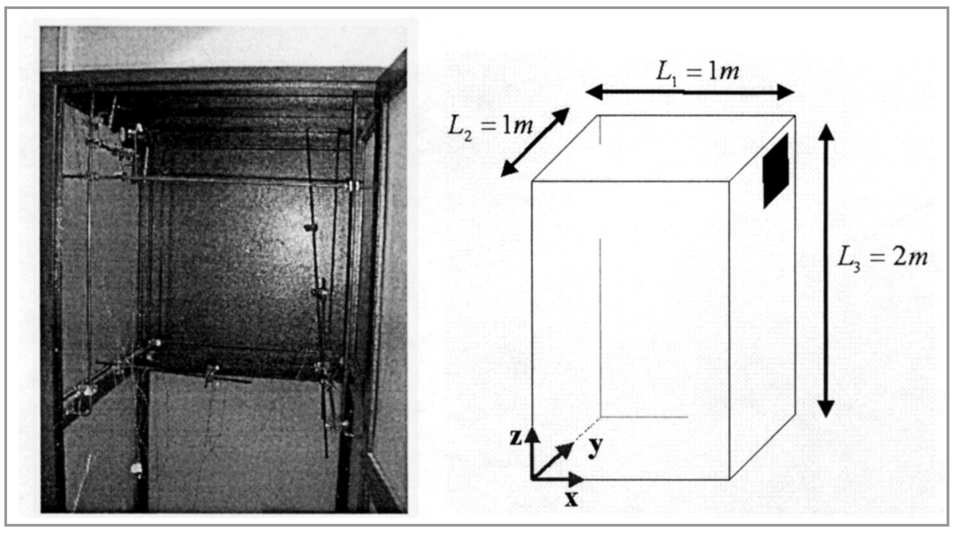

Figure 1 Experimental kiosk and its model with primary source and coordinate system

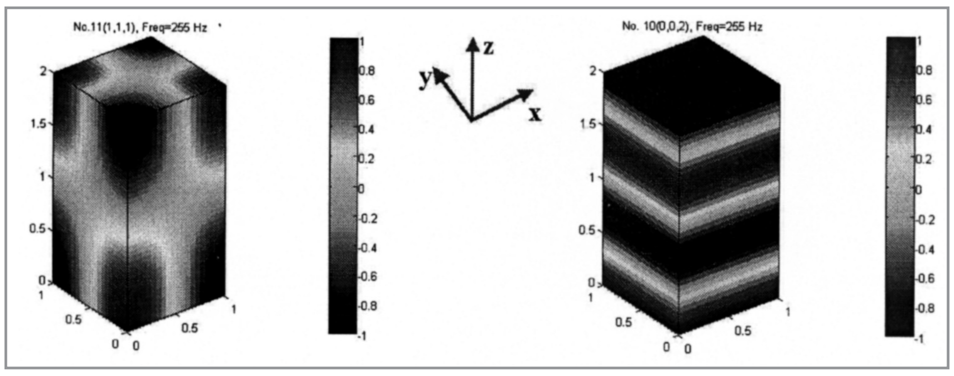

Figure 2 *Mode shapes at 255 Hz corresponding to mode No.(0,0,3) and (1,1,1)*

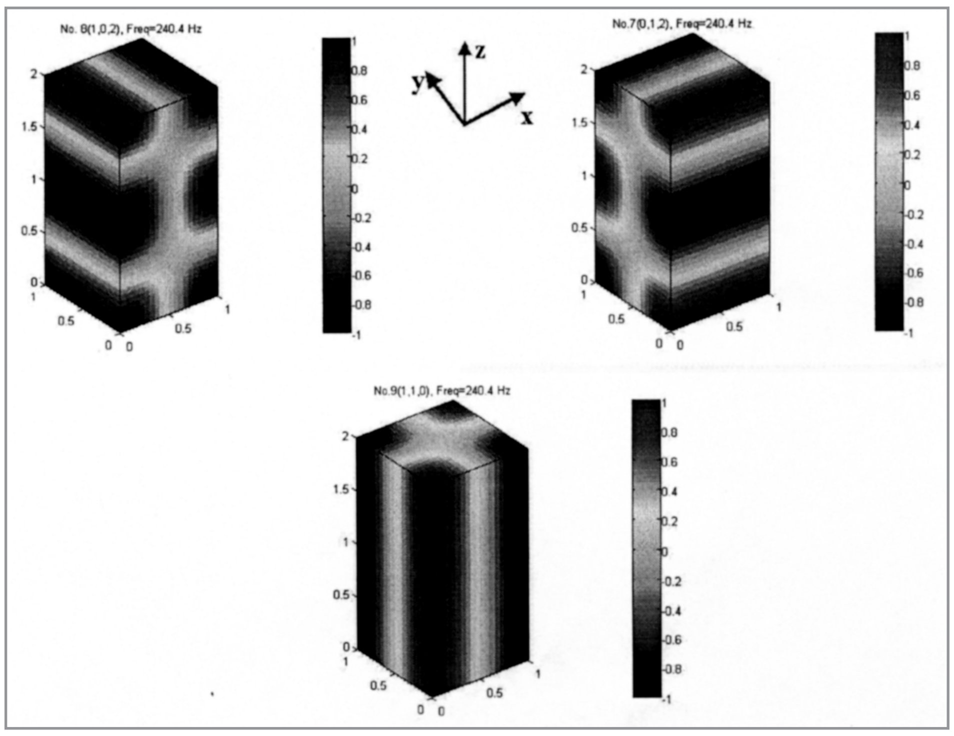

Figure 3 *Mode shapes at 240 Hz corresponding to mode No. (1,0,2),(0,1,2) and (1,1,0)*

3. GLOBAL NOISE CONTROL IN THE ENCLOSURE

The level of global reduction of sound that may be achievable with active control methods in an enclosure under the broadband excitation depends highly on the contribution of different modes to the response of the enclosure. For example, the frequency response below 100 Hz of a big room with high damping is the result of the contribution of many excited modes, and hence it is easier to analyze such a room statistically. In contrast, the response of a small room with low damping which is excited below 100 Hz is the result of the contribution of some distinct modes, and hence modal analysis is suitable for this purpose. The parameter selected here for studying the response of the enclosure is the modal overlap. This quantity is a measure of the average number of modes in a specified bandwidth (for example the 3dB bandwidth) and can be obtained as follows:

$$M(\omega) = \frac{\omega^3 V \xi_n}{2\pi {c_0}^3}. \qquad (3)$$

where ω (rad/s) is the frequency of calculation, ξ_n is the damping of each mode, V (m^3) is the volume of the enclosure, and c_0 (m/s) is the speed of sound in the enclosure.

Figure 4 shows the potential energy of the kiosk in terms of the modal overlap between 0 and 2 associated with frequencies between 0 to 464 Hz. As can be seen from this figure, when the modal overlap is below 0.5, potential energy in the enclosure has sharp peaks; and this is a sign of distinct contributive modes in the response. However, for values over 0.5, the sharpness in the amplitude of the potential energy is gradually reduced; this shows the increasing number of contributive modes in the response of the kiosk. In this case, it may be said that the response of the enclosure behaves randomly, and the diffused field assumption is reasonable. Since the modal overlap equal to 0.5 corresponds to the frequency 294 Hz, the upper limit of frequency band that is predicted to have good results for global reduction of noise in the kiosk is chosen as 300 Hz. This analysis implies the ability to use an adaptive modal-based active control system for a practical implementation of the controller. Such a modal controller requires that the response of enclosure be made of a distinct number of modes. The calculation of acoustical potential energy in the enclosure is based on the formulation used in [11].

Because theoretically, an infinite number of modes will contribute to the response of the enclosure, in a rather low modal density enclosure, this can be approximated considering a trade-off between the necessary accuracy of the response and the reasonable computation time of simulation, over the finite number of modes. In order to determine the number of required modes, various simulations have been performed for two cases. In the first case, the loudspeaker was modelled as a rectangular piston with dimension of 0.15cm, while in the second case it was modelled as a point source. These simulations suggest that a rectangular model for loudspeaker results in faster convergence of response than that of point source, and hence this model will be used in the following sections. As a consequence of this convergence analysis, in order to have an accuracy of more than 1dB in the responses, all modes below 3000 Hz (i.e. 6384 mode) are considered in future simulations.

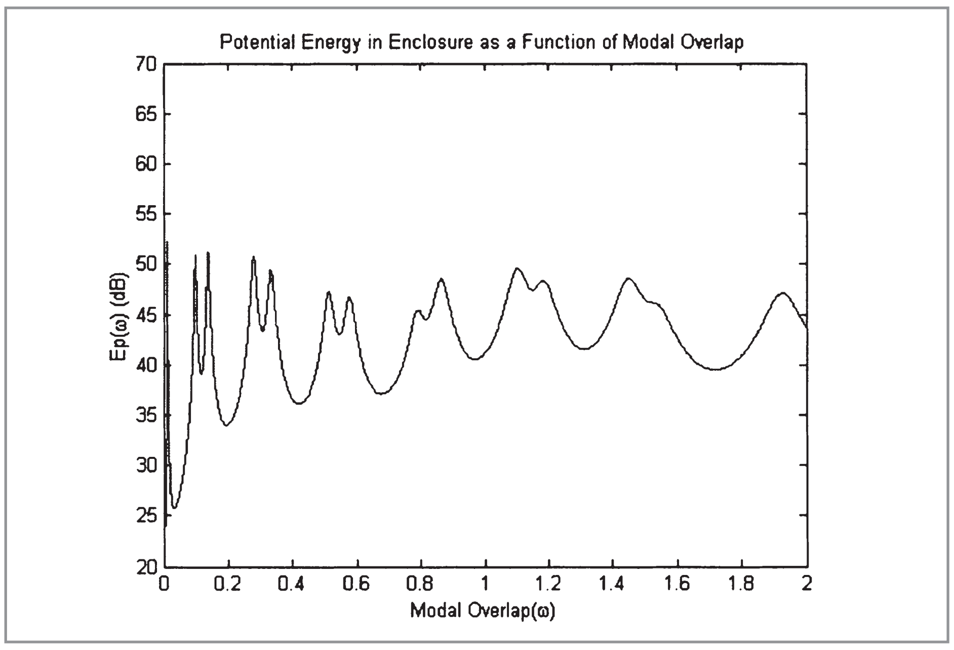

Figure 4 *Potential energy in the enclosure as a function of modal overlap between 0 to 2*

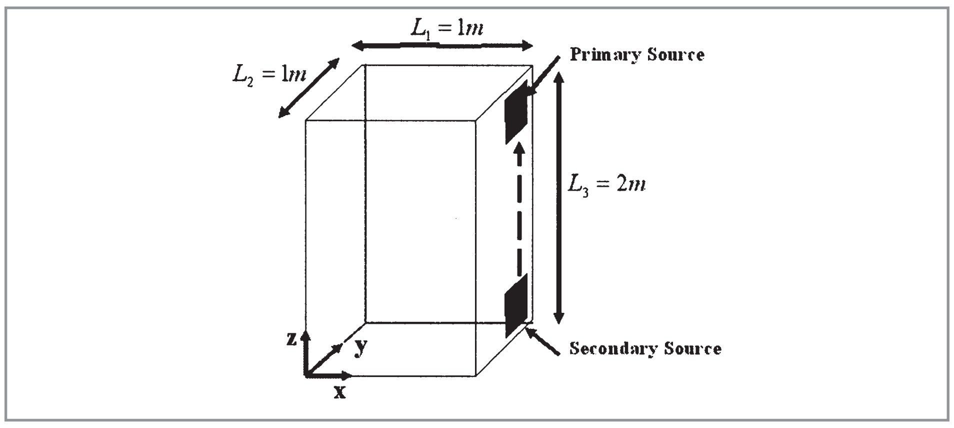

Figure 5 *Movement path of secondary source from [1, 0.925, 0] to [1, 0.925, 1.925] in z direction*

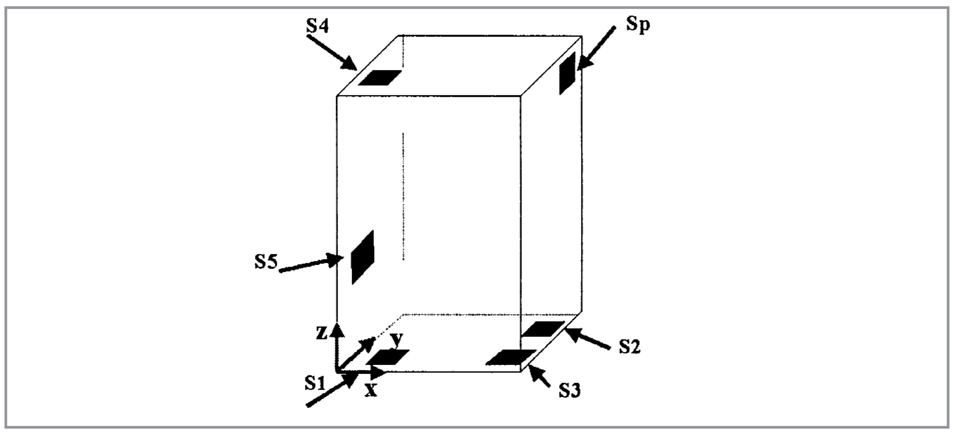

Figure 6 *Arrangement of primary and secondary sources in kiosk*

4. EVALUATION OF THE LOCATIONS OF TRANSDUCERS

Based on the proposed bandwidth for the controller in section 3 and its corresponding significant modes, the next step in the implementation is to determine the number and positions of secondary sources and error microphones required to control all modes in this band. The following two subsections deal with these problems.

4.1. THE EFFECT OF NUMBER AND POSITIONS OF LOUDSPEAKERS

Since in this subsection we intend to show only the effect of position and number of secondary sources on the performance of a modal ANC system, it is assumed that the modal properties of the enclosure are known, and the simulations are based on formulations presented in the previous section. As these modal properties must be estimated in a real ANC system, it is obvious the performance will fall if there are some degrees of mismatch between the actual and estimated parameters.

In order to have an idea how influential the position of secondary sources are for different excited modes, the level of reduction of potential energy in the enclosure when the secondary source moves toward the primary source in the z direction (Fig. 5) is calculated. Fig. 7 shows the maximum potential energy reduction of enclosure for two modes at 85 Hz and 170 Hz, and Fig. 8 shows their corresponding control efforts. Since at 85 Hz there is only one dominant mode (mode number (0, 0, 1)), the level of reduction is much greater than that of 170 Hz where three dominant modes (mode numbers (1, 0, 0), (0, 1, 0) and (0, 0, 2)) contribute in the response. This is mainly because of the spillover effect at 170 Hz. As can be seen in Fig. 7, maximum reduction occurs in both cases when the secondary source is placed near the primary source. In fact, by tuning the amplitude and phase of secondary sources, the amplitude of all dominant modes can be suppressed simultaneously near the noise source. The amplitude and phase of the control source for each excited modes, as a function of the position of the loudspeaker, is shown in Fig. 8. As it can be seen, for mode 85 Hz, near z=1 where the nodal plane occurs, because of the small amplitude of this mode, the amplitude of the control source becomes much larger than the primary source. However, since the amplitude of this mode is zero at the nodal plane, it may not be controlled by the secondary source. Therefore, in order not to excite other modes, no control signal should be generated by the secondary loudspeaker for an optimal performance. This explain the sudden fall in reduction of acoustic potential energy at this frequency. The phase plot of Fig. 8 at 85 Hz can be explained as follows. Below z=1, because the primary and secondary sources are positioned in two opposite phases of this mode, for control their relative phase is zero. Above z=1, the position of the sources are at the same phase of 85 Hz mode, and hence to eliminate this mode they are out of phase. The same analysis at 170 Hz, shows that near the nodal plane of mode number (0, 0, 2) at z=.5 and z=1.5, the amplitude of the secondary source increases. However, due to the extra excitation of mode numbers (0, 0 1) and (0, 1, 0), this is not as much as for 85 Hz. Moreover, at all positions of the secondary source, the optimal phase difference between the primary and secondary source is 180 degrees. The reason for the small potential energy reduction near z=1 can be explained as follows. Because the minimum of mode number (0,0,2) is at z=1, reduction mode numbers (0,0,1) and (0,1,0) will increase the amplitude of the (0,0,2) mode by 180 degree phase difference between primary and secondary sources in these positions, and vice versa. This implies that a better control of these three modes is not possible at the same time with one speaker, and it is necessary to increase the number of secondary sources for a better performance, that is a larger reduction in acoustic potential energy of the enclosure.

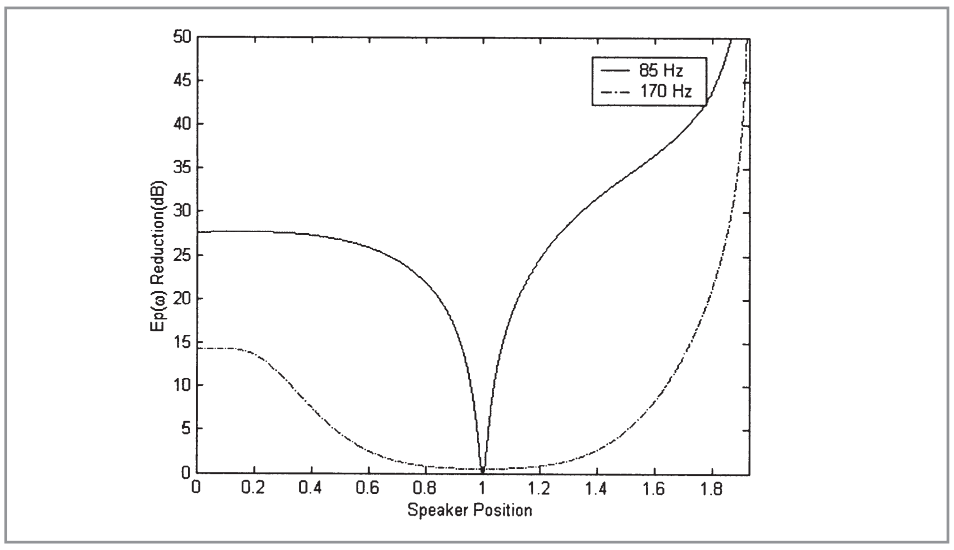

Figure 7 *Maximum potential energy reduction at 85 Hz and 170 Hz when secondary source moves as in Fig.5*

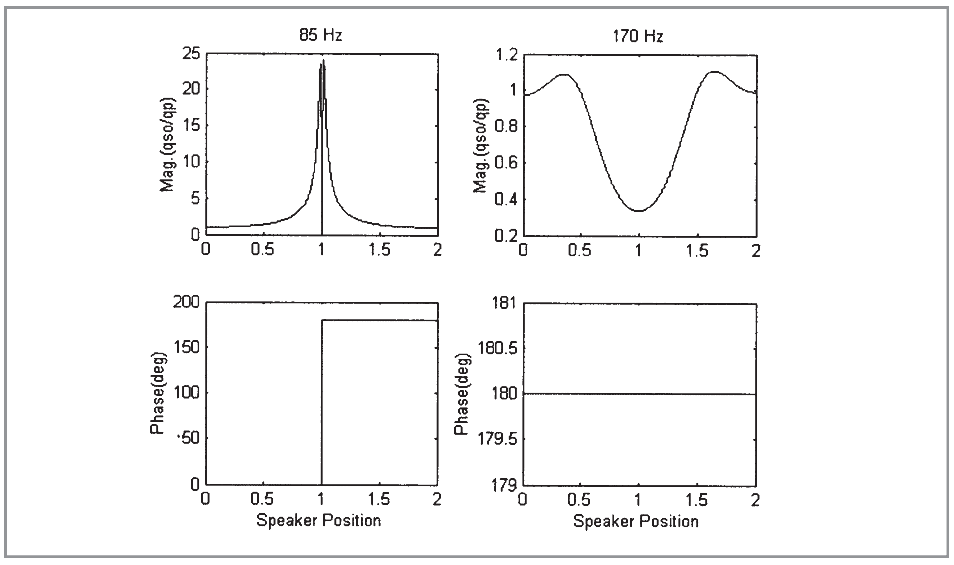

Figure 8 *Amplitude and phase of control source relative to primary source as a function of its position*

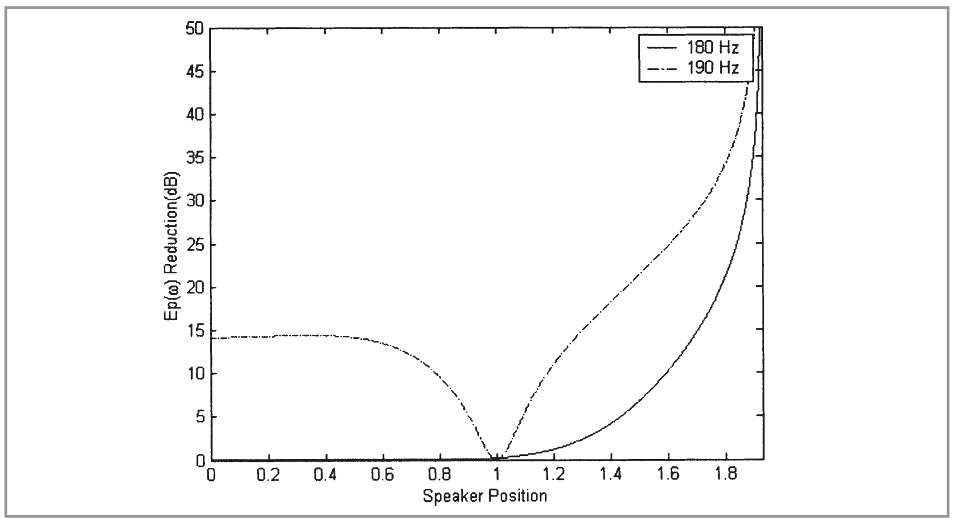

Figure 9 *Maximum potential energy reduction at 180 Hz and 190 Hz when secondary source moves as in Fig.7*

Fig. 9 compares maximum potential energy reduction at a non-resonant frequency of 180 Hz and a resonant frequency of 190 Hz when the control source moves as in Fig. 5. As can be seen in Fig. 9, the level of reduction below z=1 is negligible when compared with 190 Hz and only when the control source is near the primary source will this be increased. Actually, because of two dominant modes at 190 Hz (mode numbers (1, 0, 1) and (0, 1, 1)) when the control source is far from the primary source, a reduction of about 15dB is still achievable. However, since most of acoustic potential energy is stored in resonant frequencies, this will not pose a strict limit. The amplitude and phase of the control source relative to the primary source for both frequencies are shown in Fig. 10.

In order to examine how the number of secondary sources will affect the maximum achievable reduction of acoustic potential energy in the bandwidth of 0-300Hz, the arrangement shown in Fig. 6 is adopted for simulations. If S3 is the only active source in Fig. 6, potential energy will be reduced at 85 Hz and 294 Hz (Fig. 11). This can be explained considering the shape of the resonant modes. For example, the abatement of first and second modes requires the secondary and primary sources to operate at the same phase, while for controlling the third mode two speakers must operate at opposite phase; hence the reduction of the first two modes will amplify both the third mode and the net potential energy in enclosure. The same story will be true for other resonant frequencies. Assuming the only active source is S2; Fig. 12 shows that all resonant frequencies can be controlled. This could be explained referring to mode shapes depicted in section 2. At this position, sources S2 and Sp are either at the same phase (170 Hz, 240 Hz, 295 Hz) or at the opposite phase of enclosure modes (85 Hz, 190 Hz, 255 Hz) or at the opposite phase of enclosure modes (85 Hz, 190 Hz, 255 Hz). Therefore, by optimal tuning of secondary sources, dependent on the position of localized noise source Sp, and their relative positions will affect the maximum achievable performance.

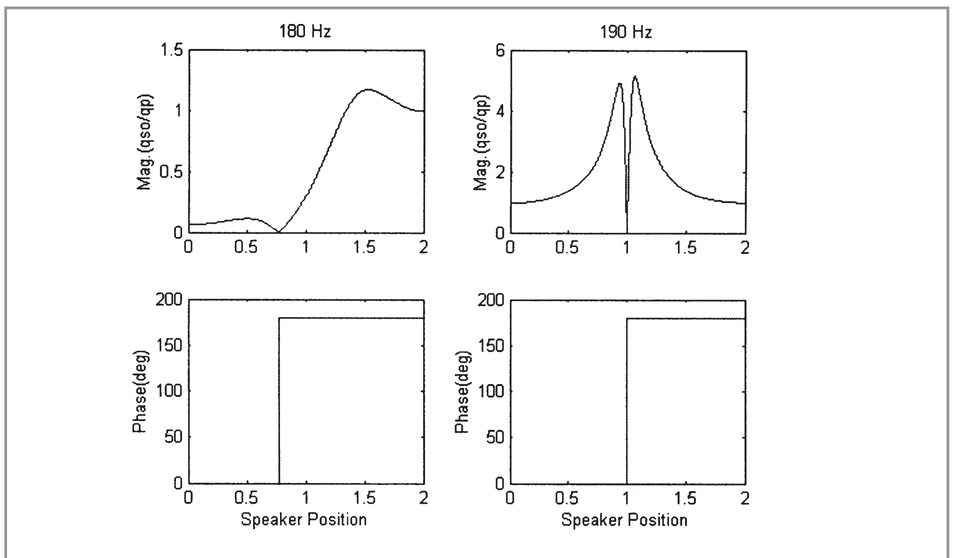

Figure 10 *Amplitude and phase of control source relative to primary source as a function of its position*

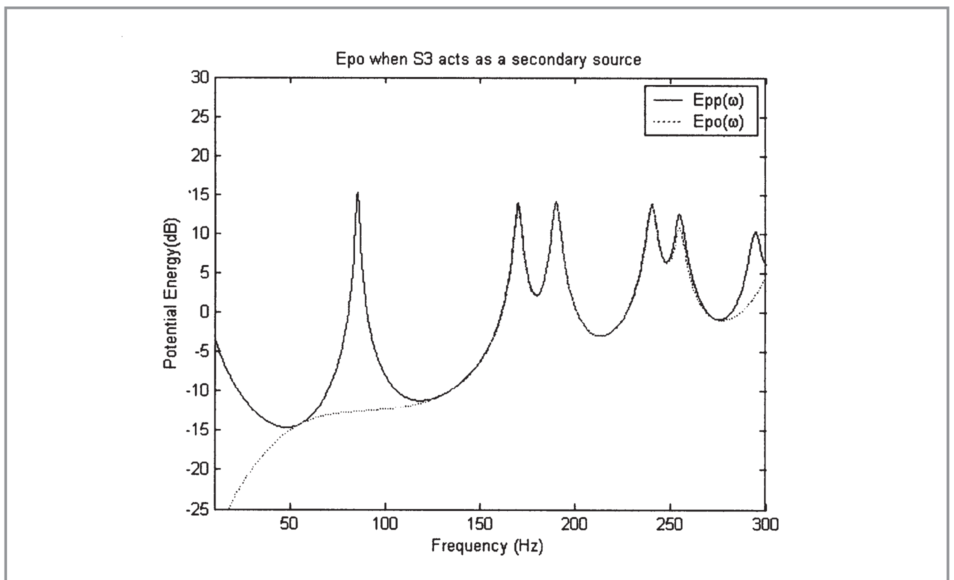

Figure 11 *Reduced potential energy in enclosure when only S3 operates with primary source (dashed line), and potential energy when primary source only is active (solid line)*

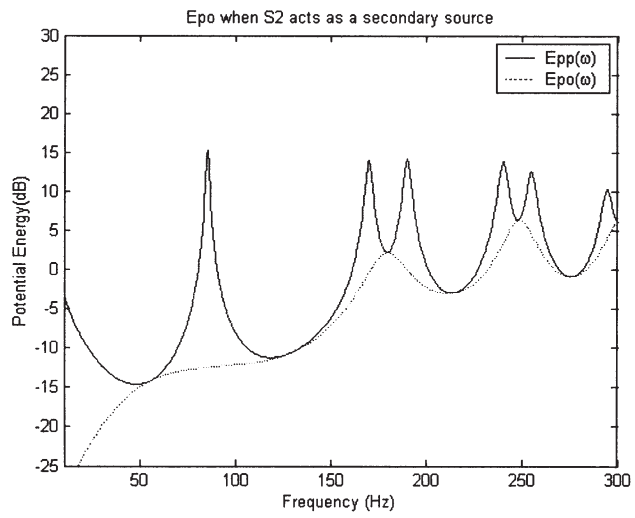

Figure 12 *Reduced potential energy in enclosure when only S2 operates with primary source (dashed line), and potential energy when primary source only is active (solid line)*

The amplitude and phase of the source S2 relative to the primary source at different frequencies are shown in Fig. 13. To have a view of how the control system behaves if the secondary source is placed at a non-corner position of enclosure. Fig. 14 shows the potential energy in the enclosure when the secondary source is placed at point [0.1, 0.1, 0.9]. As can be seen in this figure, control effort of the secondary source at 85 Hz is about 6 times the amplitude of primary source. This is due to the position of the control source which is near the nodal plane of this mode. The last simulation in this section investigates how an increase in the number of secondary sources may help to improve the results. Based on the results of Fig. 12, now it is assumed that two sources S2 and S4 are active simultaneously.

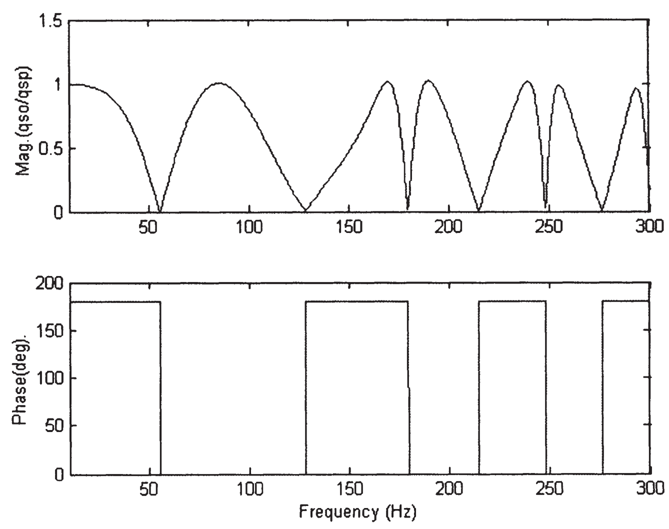

Figure 13 *Optimal amplitude and phase of secondary source S2 relative to primary source for acoustic potential energy in enclosure*

Figure 15 shows the potential energy of enclosure before and after the action of these sources. As can be seen in this figure, at frequencies where the two sources are at suitable positions relative to the mode shapes (85, 190, 255 and 294 Hz), they will cooperate by proper tuning of their amplitude and phase (Fig. 16), resulting in a higher reduction of acoustic potential energy than when only the source S2 is active. However, at frequencies where only one of the sources (i.e. S2) is placed at a suitable position (170 Hz and 240 Hz), and hence works effectively, the amplitude of the other source (i.e. S4) should be reduced down to zero for the optimal performance. One important result of these simulations is that, in an enclosure with degenerate modes, where at one resonance frequency several mode numbers may contribute in response simultaneously, placement of control sources at corners of enclosure will not necessarily result in global reduction of noise at all frequencies, and even if this will be achieved, the choice of the proper corner will significant.

4.2 THE EFFECT OF POSITION AND NUMBER OF MICROPHONES

A more practical solution for a control system perspective where tuning the amplitude and phase of the control sources must be done automatically, the acoustic potential energy in the enclosure is approximated by the measurement of acoustic pressure at some discrete points using microphones. In this case, the performance index is the sum of squared amplitude of acoustic pressure at selected points, and its minimization will result in the optimum volume velocity of the control sources. This approximation of acoustic potential energy is stated in [21] and is not repeated here. The approximation of potential energy in enclosure can be obtained with a reasonable number of microphones, if they are placed at some proper points in the enclosure.

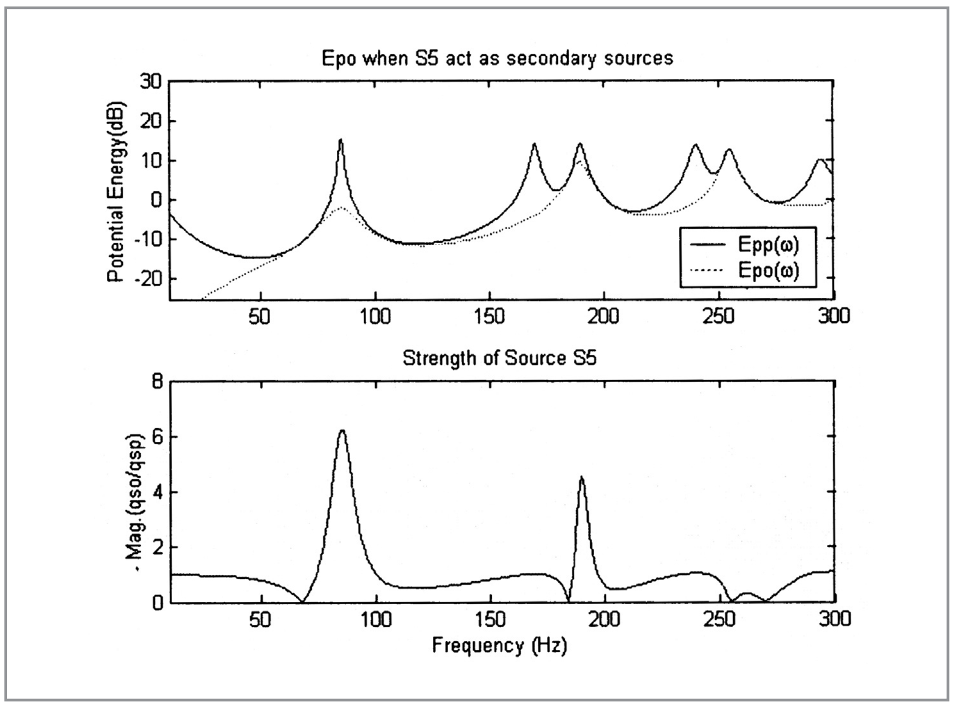

Figure 14 *Top: Reduced potential energy in an enclosure when the secondary source S5 only is active (dashed line) and potential energy from the primary source Sp (solid line). Bottom: the ratio of volume velocity of secondary source to primary source*

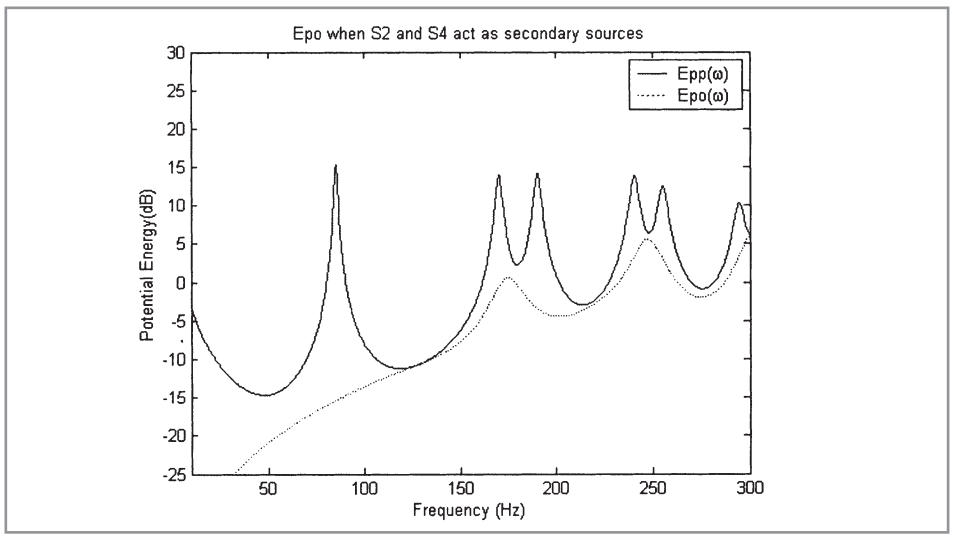

Figure 15 *Reduced potential energy in the enclosure when the sources S2 and S4 act simultaneously (dashed line), and Potential energy form the primary source Sp (solid line)*

For this purpose, several arrangements of microphones must be tested. Since the location of secondary sources may have major effects on finding the suitable locations for microphones the strategy adopted here is first to fix positions of the loudspeakers. Based on the simulation study in a previous section, here it is assumed that S2 and S4 are control speakers as shown in Fig. 17. Furthermore, six microphones with the arrangement shown in Fig. 17 are used to study the control system.

Figure 18 shows the level of potential energy in a kiosk for a two by two control system using M1 and M2 as error microphones. These microphones are placed near the head of a person who is standing in kiosk. However, as can be seen in this figure, although this control system may result in good performance in local reduction of sound, potential energy in the enclosure is increased near all other resonance frequencies except at 85Hz. Because acoustic modes at the corners of the enclosure have maximum amplitudes, it is expected that the placement of microphones at these locations will give a better approximation of potential energy in the enclosure for the control system.

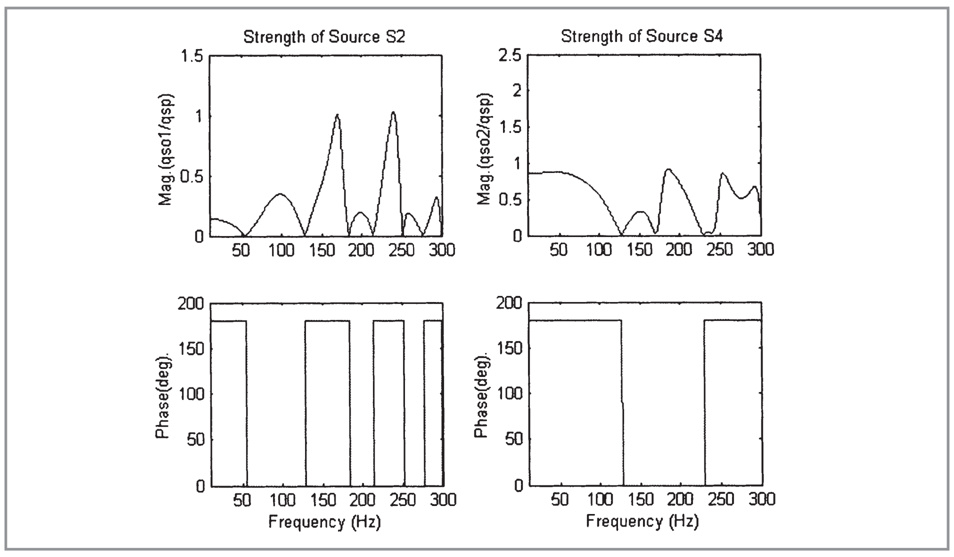

Figure 16 *Optimal amplitude and phase of control sources S2 and S4 relative to primary source Sp when they are used for potential energy reduction*

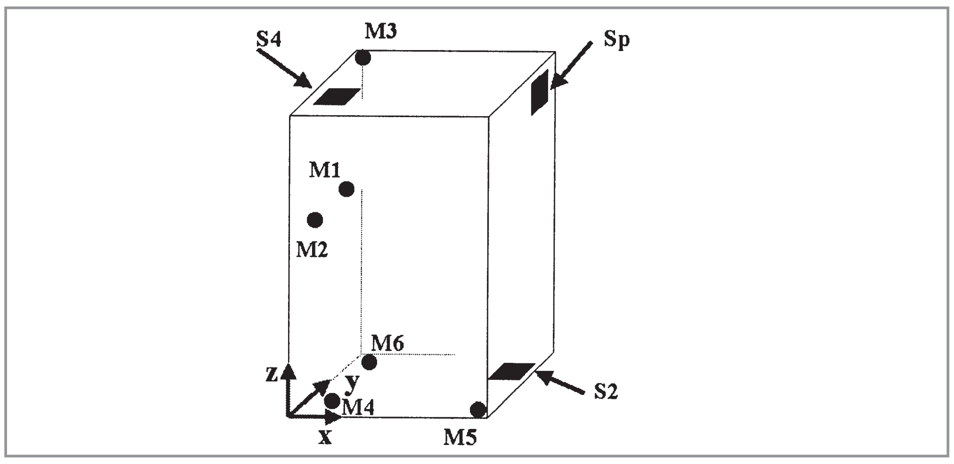

Figure 17 *Arrangement of speakers and microphones in kiosk*

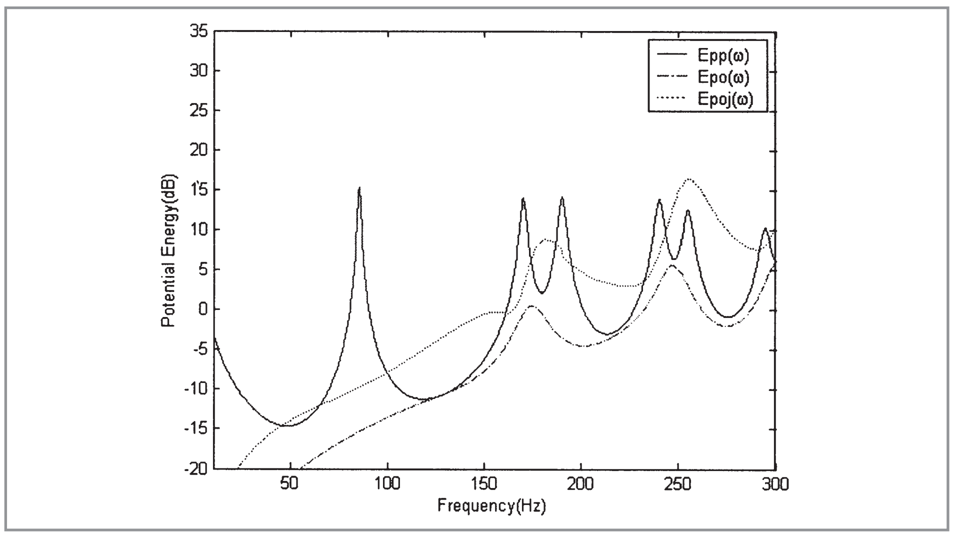

Figure 18 *Potential energy in an enclosure from a primary source (solid line), Potential energy when two speakers S2 and S4 and two microphones M1 and M2 are used to minimize J_p (dotted line), and Potential energy when Ep is miminized using two speakers S2 and S4 (dashed line)*

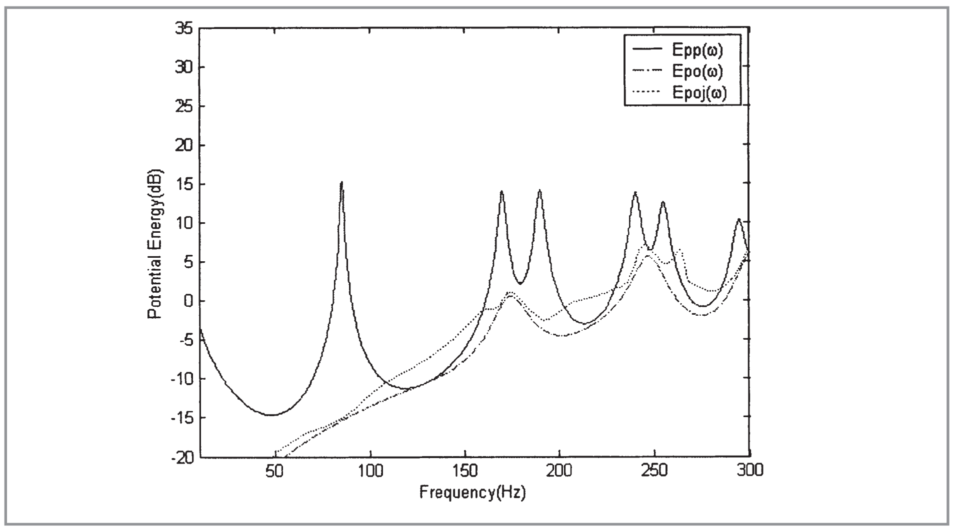

Figure 19 *Potential energy in enclosure from a primary source (solid line), Potential energy when two speakers S2 and S4 and six microphones M1 - M6 are used to minimize J_p (dotted line), and Potential energy when E_p is minimized using two speakers S2 and S4 (dashed line)*

By increasing the number of microphones a better approximation will be achieved. In fact, this is a compromise between the dimension of the control system and the performance of the system. Fig. 19 shows the level of potential energy in the enclosure using a two by six control system (two speakers S2 and S4 and six microphones M1-M6) shown in Fig. 17, and the results are compared with the case when an exact knowledge of potential energy is available.

5 CONCLUSIONS

In this paper modal analysis for the examination of the possibility of global control of a broadband noise in a rectangular enclosure with a telephone kiosk application is investigated. This analysis shows the performance of the control system regarding the position and number of loudspeakers/microphones in the kiosk. The dimensions of the selected enclosure are chosen integer multiples (1*m* x 1*m* x 2*m* here), and hence there are some degenerate modes in the response of the enclosure complicating the modal control problem in turn. Some simplifying assumptions such as rigid walls of the enclosure and localized and internal noise source are adopted, and hence further steps have to be taken for a real kiosk problem with external and distributed noise sources. These issues are under investigation by the authors.

With the aim of global sound reduction, the target band-width of the controller is selected using the modal overlap technique. Simulation results show that the placement of loudspeakers at corners will not necessarily result in the best performance, as previously claimed in some papers, and among all tested points, only the corner where S2 is located has resulted in successful reduction of all modes. Although using further speakers improve the simulation results, because of practical implementation issues however, the number of control speakers was limited to two. Since a practical ANC system requires microphones for adaptation of the controller, the position and number of these microphones have a great effect on the performance of the system. Simulation results show that by increasing the number of microphones a better approximation of potential energy in enclosure can be obtained and, hence the reduction of potential energy, obtained based on the measured performance index, will be close to the level of reduction obtained when exact modal information of enclosure is known.

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STANSTED

A Wareside man found guilty of harassing the manager of Stansted Airport's noise complaint line has defended his actions and intends to appeal the decision. John McCaskie, 54, admits that he swore over the phone when making complaints about low-lying aircraft but claims he did not directly abuse staff. "I never called anyone any names. As far as I know, harassment is a continuous and designed attack on someone - I phoned four times and emailed twice in six months," said Mr McCaskie, who was found guilty of harassment between July and October last year, at Harlow Magistrates' Court. "I'm the one being harassed by these planes 170 times a year. This has been my worst nightmare, having my peace and quiet disturbed, and my health has suffered." Mr McCaskie has lived at St George's Cottages since 1998, but only began to have a problem with planes a year ago. "A plane flew right by my house and I thought 'That was a bit low, thank heavens that doesn't happen very often', but then another one flew by my window a couple of minutes later - that was the beginning," he said. "All day long these planes started flying over, it was totally bizarre." Mr McCaskie, a taxi driver, phoned the noise complaint line. "I was told that I was more conscious of the noise due to the publicity of the problem - I thought they were wasting my time," he said. He got through to the manager, whom he claims regularly failed to call him back. At one stage, he had a 40-minute conversation with her, during which he was told that nothing had changed. He claimed: "If you lived next door to me and after 10 years I started blaring out loud music, then when you complained I told you that nothing had changed, you'd think I was taking the mick!" A Stansted Airport spokesman said "The airport's flight evaluation unit works extremely hard to help people better understand what can be complex and sometimes sensitive areas regarding the airport's operation. However, it's totally unacceptable when individuals overstep the mark and abuse staff and the service they provide."

NORTH DORSET

People suffering from noisy neighbours in Dorset must not complain anonymously if they want the matter to be investigated. And, as the north of the county does not have a responsive out-of-hours service, victims have to use noise monitoring and recording equipment to capture evidence of the noise nuisance they're suffering, even though it is less favoured by the courts. North Dorset council has no plans to introduce night-time call-outs because the council's environmental chiefs say it would be too expensive and the existing daytime service would suffer. The council's procedures are that environmental officers will not usually act on anonymous complaints or those via a third party, unless the complainant can then be established and contacted to confirm the details. In keeping with local authorities across the country, victims in north Dorset are encouraged to keep a diary of noise nuisance and in severe cases officers will visit. When officers are satisfied that the noise amounts to a statutory nuisance, an abatement notice will be served and failure to comply is an offence. The guidelines also state that offices will respond within three working days for non-urgent cases, and within one for urgent ones.