Analysis of the global reduction of broadband noise in a telephone kiosk using a MIMO modal ANC system

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Abstract

The possibility of implementation of ANC system for global reduction of broadband noise in a telephone kiosk application has been studied in this paper. By using a modal model of the acoustic environment of the enclosure the shape modes and their frequencies are obtained and suitable bandwidth for the controller design is calculated. Analysis shows that because of existing of some degenerate modes in this application, the control action is complicated. Simulation results show that 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 show the effectiveness of the proposed system in reduction of the acoustic potential energy in the kiosk.

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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, interests in active control of enclosed sound fields have increased dramatically since early 1980s, focusing on two main applications. One of the frequently reported applications is to reduce the cabin noise in aircrafts [2–4]; such active noise control (ANC) systems are installed and used in production stage as reported in [5]. Another documented application is for the automobile cabin noise [6–8]; 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 enclosed spaces: 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 quite zones at some specific points. Although reducing...
the sound in a global sense has the advantage of reducing the noise at all points of the enclosure, its applicability is restricted with some parameters such as working frequency, the spatial distance between the primary and secondary sources, the number of control channels, and physical characteristics of the enclosure (for example its dimensions and damping ratio of its walls). Difficulties in measuring the required performance index for global control may also pose some additional problems. On the other hand, based on the above limitations, local control and creating small quite zones around the microphones may be the only possible choice in some applications.

Selection of the proper strategy for the application in hand is not straightforward. However, previous works 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 literatures mostly in situations when the sound field in enclosure is described statistically [13–16], or when a large number of modes contribute in response [17,18].

Phones are currently common devices used for communications. However, in many places, such as a noisy street or an industrial place, the communication suffers from excessive noise. A common way to reduce such environmental noise is to place the phone set 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 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 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 in a telephone kiosk for global reduction of broadband noise is investigated. For this purpose, first a model of acoustic environment of enclosure is obtained by solving wave propagation equations in the enclosure in 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 obtained modal information, 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 acoustic potential energy of 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 conclusion comes.

2. Modeling the acoustical environment of a kiosk

What distinguishes most active noise and vibration control problems from other ordinary control problems is that the disturbance propagates from one point of physical system to other points in the form of wave. The governing wave equation in a three dimensional environment, taking into account viscous damping of environment and assuming sound source with specific volume velocity distribution in right-hand side, may be expressed as

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

where \(c_0\) is the speed of sound in enclosure, \(p(x, t)\) is acoustical pressure at point \(x\) and time \(t\), \(\rho_0\) is the density of air, \(\mu\) is a constant indicating damping in air or walls, and \(g(x, t)\) is volume velocity of sound per volume. Since the acoustical pressure \(p\) in comparison with atmosphere pressure is very small, the linearity of (1) is assured. In addition to (1), there are six boundary conditions assuming the walls of enclosure are rigid. As the velocity on walls is zero, acoustical pressure will not transmit outside and hence pressure gradient on walls will become zero. As a result, the boundary conditions of (1) will become:
The experimental kiosk aimed 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 multiplies 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) is standing waves in steady state which can be expressed in terms of some basis functions called the acoustical modes of enclosure. Assuming a harmonic excitation in right-hand side of (1), the amplitude of complex pressure at a specific point and frequency can be obtained as

\[ p(x, \omega) = \sum_{n=1}^{\infty} \psi_n(x) a_n(\omega) = \Psi^T a. \]  

In (3) \( \psi_n(x) \) is the \( n \)th mode shape of enclosure, and \( a_n(\omega) \) is its amplitude. By truncation of (3) to the first \( N \) modes, \( \Psi \) and \( a \) are \( N \times 1 \) vectors of these values. The mode shape and its amplitude can be calculated as follows:

\[ \psi_n(x) = \sqrt{\frac{\psi_n \psi_2 \psi_3}{e_n}} \cos \left( \frac{n_1 \pi x}{L_1} \right) \cos \left( \frac{n_2 \pi y}{L_2} \right) \cos \left( \frac{n_3 \pi z}{L_3} \right), \]

\[ a_n(\omega) = \frac{\rho_0 c_0^2}{V} A_n(\omega) \int_V G(x, \omega) \psi_n(x) \, dV, \]

where in (4):

\[ e_n = \begin{cases} 2 & \text{if } v > 0, \\ 1 & \text{if } v = 0 \end{cases} \]

and \( A_n(\omega) \) in (5) is defined as

\[ A_n(\omega) = \frac{j \omega}{-\omega^2 + j2\zeta_n \omega_n \omega + \omega_n^2}. \]
As it can be seen from (6), each acoustic mode has its corresponding resonance frequency and damping ratio. Here the damping ratio of all different modes is chosen equal to 0.01, and the resonance frequencies of the kiosk in the range of 0–300 Hz are calculated in Table 1. 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. 2–4, the mode shapes at resonant frequencies 255 Hz, 170 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 resonance frequencies at 255 and 294 Hz has one oblique mode.

Table 1
Resonance frequencies calculated between 0 and 300 Hz

<table>
<thead>
<tr>
<th>No.</th>
<th>Freq. (Hz)</th>
<th>Mode contribution</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>85</td>
<td>(0, 0, 1)</td>
</tr>
<tr>
<td>2</td>
<td>170</td>
<td>(1, 0, 0) (0, 0, 1) (0, 0, 2)</td>
</tr>
<tr>
<td>3</td>
<td>190.1</td>
<td>(1, 0, 1) (0, 1, 1)</td>
</tr>
<tr>
<td>4</td>
<td>240.4</td>
<td>(1, 1, 0) (1, 0, 2) (0, 1, 2)</td>
</tr>
<tr>
<td>5</td>
<td>255</td>
<td>(1, 1, 1) (0, 0, 3)</td>
</tr>
<tr>
<td>6</td>
<td>294.4</td>
<td>(1, 1, 2)</td>
</tr>
</tbody>
</table>

Fig. 2. Mode shapes at 255 Hz corresponding to mode nos. (0,0,3) and (1,1,1) from top to bottom.
Fig. 3. Mode shapes at 170 Hz corresponding to mode nos. (0,0,2), (0,1,0) and (1,0,0) from top to bottom.
Fig. 4. Mode shapes at 240 Hz corresponding to mode nos. (1,0,2), (0,1,2) and (1,1,0) from top to bottom.
3. Global control of broadband noise in a kiosk

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 in 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. In order to show how different modes may contribute to form the response of a kiosk, the square amplitude of some of acoustic modes are shown as a function of frequency in Fig. 5.

The parameter selected here for studying the response of the kiosk is the modal overlap. This quantity is a measure of the average number of modes in a specified bandwidth (for example the 3 dB bandwidth) and can be obtained as follows:

![Fig. 5. The potential energy of different modes which may contribute to the response of the enclosure.](image)

![Fig. 6. Potential energy in the enclosure as a function of modal overlap between 0 and 2.](image)
\[ M(\omega) = \frac{\omega^3 V \xi_n}{2\pi c_0^3}. \]  

Fig. 6 shows the potential energy of the kiosk in terms of the modal overlap between 0 and 2 associated with frequencies between 0 and 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 distinction of 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 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 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 distinct number of modes.

4. Calculation of acoustic potential energy in enclosure

A good measure that when minimized will result in the global reduction of sound, is the acoustic potential energy of the enclosure. In fact the suppression of dominant excited modes of enclosure will reduce the annoyance of noise someone may feel, and this is because of flattened spatial variation of acoustic pressure inside the kiosk. Acoustic potential energy in an enclosed space can be written as

\[ E_p = \left( \frac{1}{4\rho_0 c_0^2} \right) \int_V |p(x, \omega)|^2 dV. \]  

Substituting (3) in (8) and considering the orthogonality of mode shapes yields:

\[ E_p = \left( \frac{V}{4\rho_0 c_0^2} \right) \sum_{n=1}^{\infty} |a_n(\omega)|^2 = \left( \frac{V}{4\rho_0 c_0^2} \right) a^H a \]  

when one primary source and \( L \) secondary sources are assumed in the enclosure, due to the linearity of (1), the amplitude of each mode using the principle of superposition can be written as

\[ a_n(\omega) = a_p(\omega) + \sum_{l=1}^{L} B_{nl}(\omega)q_{sl}(\omega), \]  

where \( q_{sl}(\omega) \) is the complex volume velocity of the \( l \)th secondary source and is calculated by integration of the velocity distribution of source over a specified surface or volume. Here, primary and secondary sources are assumed to have uniform volume velocities. In (10) \( a_p(\omega) \), the amplitude of mode \( n \), results from the action of primary source, and \( B_{nl}(\omega) \) shows the effect of the \( l \)th secondary source on the amplitude of \( n \)th mode and is calculated from (11) and (12) as follows:

\[ a_p(\omega) = \frac{\rho_0 c_0^2}{V} A_n(\omega) v_p(\omega) \int_{S_p} \psi_n(y) dS, \]

\[ B_{nl}(\omega) = \frac{\rho_0 c_0^2}{V} A_n(\omega) \frac{1}{S_s} \int_{S_{sl}} \psi_n(y) dS. \]

By rewriting (10) in a matrix form and omitting \( \omega \) for convenience we will have

\[ a = a_p + Bq_s, \]

where \( B \) is an \( N \times L \) matrix with elements of \( B_{nl}(\omega) \). By substituting (13) into (9), the potential energy is expressed directly in terms of complex volume velocity of secondary sources:

\[ E_p = \frac{V}{4\rho_0 c_0^2} \left[ |q^H B^H B q_s + q^H B^H a_p + a_p^H B q_s + a_p^H a_p| \right]. \]
This relation is a quadratic function of the volume velocity, and the optimal value of $q_s$ which minimizes (14) is obtained by least square solution as follows:

$$q_{so} = -[B^H B]^{-1} B^H a_p.$$  \hspace{2cm} (15)

Moreover, by putting (15) in (14) the ratio of minimum potential energy with respect to initial potential energy caused by primary source will be

$$\frac{E_{po}}{E_{pp}} = 1 - a_p^H B[B^H B]^{-1} B^H a_p/a_p^H a_p.$$ \hspace{2cm} (16)

One problem in the calculation of (3) or (9) for the modeled enclosure, is the infinite sum over mode numbers $(n_1,n_2,n_3)$, which is impossible for the simulated model. However, in a rather low modal density enclosure, this summation can be approximated considering a trade-off between the necessary accuracy of response and the reasonable computation time of simulation, over the finite number of modes. In order to determine the number of modes required for convergence of (3), various simulations have been performed for two cases. In the first case, the loudspeaker was modeled as a rectangular piston with dimension of 0.15 cm, while in the second case it modeled 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 1 dB in the responses, all modes below 3000 Hz (i.e. 6384 mode) is considered in future simulations.

5. Evaluation of a MIMO modal ANC for implementation

The use of modal ANC [22,23] for an active control problem has the following advantages:

- If an active control problem is known to involve only a few significant modes, then independent modal control can minimize the number of secondary sources, sensors, and corresponding dimensionality of the controller, as well as the control energy.
- Modal control offers advantages of robustness to system parameter uncertainty and errors arising from spatial discretization.
- If the LMS algorithm is employed, the convergence time problem is minimized by uncoupling the modal responses.

Based on the proposed bandwidth for the controller in Section 3 and its corresponding significant modes, the next step for the implementation of a modal ANC 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.

5.1. 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 enclosure when the secondary source moves toward the primary source in z direction (Fig. 7) is calculated. Fig. 9 shows the maximum potential energy reduction of enclosure for two modes at 85 Hz and 170 Hz, and Fig. 10 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 highly 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. 9, 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 control source for each excited mode, as a function of position of loudspeaker, is shown in Fig. 10. As 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 control source becomes much larger than the primary source. However, since the amplitude of this mode is zero at 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 explains the sudden fall in reduction of acoustic potential energy at this frequency. The phase plot of Fig. 10 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 \( z = 0.5 \) and \( z = 1.5 \), the amplitude of secondary source increases. However, due to 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 secondary source, the optimal phase difference between the primary and secondary source is 180°. 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 of mode numbers (0,0,1) and (0,0,1) will increase the amplitude of (0,0,2) mode by 180° 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.

Fig. 11 compares maximum potential energy reduction at a non-resonant frequency of 180 Hz and a resonant frequency of 190 Hz when control source moves as in Fig. 7. As can be seen in Fig. 11, 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 this will 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 primary source, a reduction of about 15 dB 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 primary source for both frequencies are shown in Fig. 12.
In order to examine how the number of secondary sources will affect the maximum achievable reduction of acoustic potential energy in the bandwidth of 0–300 Hz, the arrangement shown in Fig. 8 is adopted for simulations. If S3 is the only active source in Fig. 8, potential energy will be reduced at 85 Hz and 294 Hz (Fig. 13). This can be explained considering the shape of 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.

Fig. 11. Maximum potential energy reduction at 180 Hz and 190 Hz when secondary source moves as in Fig. 7.

Fig. 12. Amplitude and phase of control source relative to primary source as a function of its position.
Assuming the only active source is S2; Fig. 14 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 $S_p$ 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). Therefore, by optimal tuning of secondary sources, these six modes will be controlled simultaneously. The amplitude and phase of the source S2 relative to primary source at different frequencies is shown in Fig. 15. To have a view of how control system behaves if the secondary source is placed at a non-corner position of enclosure, Fig. 16 shows the potential energy in 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 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. 14, now it is assumed that two sources S2 and S4 are active simultaneously.

Fig. 17 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 mode shapes (85, 190, 255 and 294 Hz), they will cooperate by proper tuning of their amplitude and phase (Fig. 18), 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 be an important task.
5.2. The effect of number and positions of microphones

A more practical solution for control system perspective, where tuning the amplitude and phase of control sources must be done automatically, acoustic potential energy in enclosure is approximated by measurement of acoustic pressure at some discrete points using microphones. In this case, the performance index is the sum of square amplitude of acoustic pressure at selected points, and its minimization will result in the optimum volume velocity of control sources. The approximation of acoustic potential energy in (8) is:

![Graph showing reduced potential energy and potential energy from primary source](image)

**Fig. 17.** Reduced potential energy at enclosure when the sources S2 and S4 act simultaneously (dashed line), and potential energy from primary source $S_p$ (solid line).

![Graph showing optimal amplitude and phase of control sources](image)

**Fig. 18.** Optimal amplitude and phase of control sources S2 and S4 relative to primary source $S_p$ when they are used for potential energy reduction.

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where \( p \) is the vector of acoustic pressure at \( M \) discrete point as
\[
p = \Psi_L^T a
\]
and \( \Psi_L \) is an \( M \times N \) matrix whose element \((m, n)\) is the value of characteristic function of \( n \)th mode at point \( m \).
Replacing \( a \) from (13) to (18) will yield:
\[
p = p_p + Zq_s,
\]
where \( p_p = \Psi_L^T a_p, \ Z = \Psi_L^T B \). Here, \( p_p \) and \( Z \), unlike \( a_p \) and \( B \) are measurable quantities and can be used for practical implementation of ANC system. By combination of (17) and (19) the performance index is computed directly in terms of the volume velocity of secondary sources as follows:
\[
J_p = \frac{V}{4\rho_0 c_0^2 M} [q^H Z^H Z q_s + q^H Z^H p_p + p_p^H Z q_s + p_p^H p_p].
\]
Minimization of this quadratic function will give the optimal value and the minimum of performance index as
\[
q_{so} = -[Z^H Z]^{-1} Z^H p_p,
\]
\[
J_{po} = \frac{V}{4\rho_0 c_0^2 M} (p_p^H p_p - p_p^H Z[Z^H Z]^{-1} Z^H p_p).
\]
Block diagram of the practical Modal ANC system, using the preceding formulations is shown in Fig. 19.

Referring to (18), the approximation of potential energy in enclosure can be obtained with reasonable number of microphones, if they are placed at some proper points in the enclosure. 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 loudspeakers. Based on the simulation study in previous section, here it is assumed that S2 and S4 are control speakers as shown in Fig. 20. Furthermore, six microphones with the arrangement shown in Fig. 20 are used to study the control system.

Fig. 21 shows the level of potential energy in 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 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 enclosure is increased near all other resonance frequencies except at 85 Hz. Because
acoustic modes at the corners of enclosure have maximum amplitudes, it is expected that the placement of microphones at these locations give better approximation of potential energy in enclosure for the control system.

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. 22 shows the level of potential energy in enclosure using a two by six control system (two speakers S2 and S4 and six microphones M1–M6) shown in Fig. 20, and the results are compared with the case when an exact knowledge of potential energy is available.

6. Conclusions

In this paper the possibility of the global control of a broad band noise in a telephone kiosk application using a MIMO Modal ANC system is investigated. This study shows the performance of the control system regarding to the position and number of loudspeakers, and also the position and number of microphones in the kiosk. The dimensions of the selected enclosure are chosen integer multiplies (1 × 1 × 2 here), and so there are some degenerate modes in the response of the enclosure complicating the modal control problem in turn.
With the aim of global sound reduction, modal analysis of such an enclosure has been performed, and the target bandwidth of the controller is selected using 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 modal ANC system requires microphones for adaptation of the controller, the position and number of these microphones have 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.

References