Beam steering of sound from flat panels using spatially-averaged objective functions

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ABSTRACT

The paper proposes a beam steering method for regulating sound radiation from flat panel structures using multiple structural velocity sensors. Velocity measurements from the structural sensors are used to estimate the velocity profile of the panel, which is then used to estimate the acoustic beam pattern of radiated sound. An objective function is defined for active beam steering purposes, representing the spatially-averaged error between the reference beam pattern and the estimated beam pattern of sound radiation in the far-field. Numerical studies on a rectangular flat panel are used to demonstrate the ability of the proposed method to regulate a beam pattern of sound for steering a beam to different directions in the far-field. It is demonstrated that the proposed method can modify the beam pattern of tonal sound radiation by modifying the vibration velocity profile of the panel.

INTRODUCTION

Over the past few decades, numerous research has been done in the area of structural-acoustics for analysing and controlling the radiated sound from vibrating structures. Most of the research in active structural-acoustics control concentrates on attempts to reduce the noise radiated from structures (Dimitriadis and Fuller 1992; Clark and Fuller 1991; Burgan et al. 2002). However, it is also beneficial to be able to actively control the pattern of sound radiation from structures since it can be used to steer beams of sound radiation to certain desired directions. One of the potential applications of beam steering is to steer the sound radiation from flat panel speakers to a desired direction without physically moving the panels. Motivated by this, the work in this paper will concentrate on attempting to regulate beam pattern of panel structures for beam steering purposes.

In order to produce a practical and compact active controller for beam steering, it is useful to use vibration sensing for estimating the sound radiation from structures. Vibration sensing using multiple structural sensors for vibration or sound radiation control has been used in various research (Burgan et al. 2002; Meirovitch and Baruh 1982; Pajunen et al. 1994). In the work presented in this paper, vibration sensing using multiple sensors is also used for estimating the sound radiation from panel structures. However, the main difference of the proposed method with the previous research is that the vibration sensing is implemented specifically to allow the regulation of beam pattern of sound radiation.

VIBRATION & SOUND RADIATION SENSING OF PANEL STRUCTURES

In order to steer the beam of sound radiation from a panel structure, it is necessary to be able to estimate the sound radiation from the panel. The method used in this work is to estimate the panel's radiated sound from the velocity vibration profile of the panel.

Vibration sensing of a panel structure

To measure the vibration profile of a panel, system identification methods, such as modal analysis tests can be performed so that the vibration of the panel can be characterised. However, the identification methods rely on the assumption that the system's dynamics would not change significantly during the control operation. In addition, the methods might require a significant amount of time to complete due to the requirements for measuring and processing the vibration data at a number of locations over the panel.

In contrast, the proposed method here utilises multiple structural sensors that are distributed across the panel, which provides the required direct vibration information from the panel. This vibration information, in terms of the vibration velocity profile of the panel, would be utilised for estimating the radiated sound. The method thus does not require an off-line system identification for obtaining the characteristics of the panel vibration. Hence, when the panel undergoes a change in dynamics, the sensing method would still be able to provide accurate information about the panel's vibration, and consequently the sound radiated from the panel.

Consider a plate/panel structure in Figure 1 where N discrete structural sensors are used to measure the normal vibration velocity at a number of locations over the panel. The vibration sensing method is implemented by using the vibration information from the sensors and spatially interpolating the vibration at other locations on the panel. The spatial interpolation is implemented in a way similar to the approach used in finite element method (Meirovitch 1975; Cheung and Leung 1991). An element is created by the adjacent sensors that are acting as 'nodes'. To improve the vibration sensing across the entire panel, N_o additional nodes are added at the structural boundaries where the vibration is expected to be minimal (Halim and Cazzolato 2005).

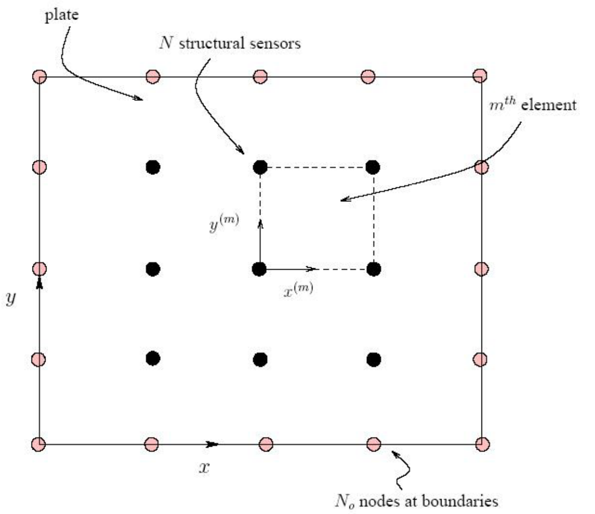

Figure 1 A panel structure with N structural sensor nodes and N_o structural boundary nodes.

Suppose there are M elements constructed on the panel, where each m^{th} element has local coordinates ($x^{(m)}$,$y^{(m)}$). For the particular example in Figure 1, rectangular elements with 4 nodes for each element are used. However, other element shapes and node numbers can also be used for this purpose.

For each element, a linear transformation matrix $A^{(m)}$ is used to transform the local coordinates to the global coordinates (x,y). The normal velocity vibration profile $v(x,y)$ can then be estimated from a set of velocity measurements from N structural sensors, w (Halim and Cazzolato 2005):

$$v(x,y) = H(x^{(m)}, y^{(m)})A^{(m)}w \tag{1}$$

where $H(x^{(m)},y^{(m)})$ is the spatial interpolation matrix used to estimate the velocity profile within each element, such as the interpolation matrices used in the finite element method (Meirovitch 1975).

Sound radiation sensing of a panel

Having obtained the velocity profile of a panel, the panel's radiated sound can then be estimated. The Rayleigh integral (Wallace 1972; Fuller et al. 1996) is used to approximate the sound radiation from the velocity profile of the panel. The interest of this work is for the far-field sound radiation, a situation many practical sound radiation problem can be approximated by.

Consider the spherical coordinates in Figure 2. The far-field complex sound pressure p at a point of interest, a distance r away from the coordinate origins on the plane of panel, can be described by:

$$\begin{aligned} p(r,\theta,\phi) &= \frac{j\omega_o\rho}{2\pi r}\int_R f(r,\theta,\phi,x,y)v(x,y)dR \\ f &= \exp\{-jkr(r - x\sin\theta\cos\phi - y\sin\theta\sin\phi)\} \end{aligned} \tag{2}$$

where R is the surface of the panel, $v(x,y)$ is the normal surface velocity of the panel, ρ is the density of the acoustic medium, and j is the unit imaginary number. Here, k is the acoustic wave number that is related to the speed of sound c and the frequency of sound radiation ω_o by

$$k = \frac{\omega_o}{c}. \tag{3}$$

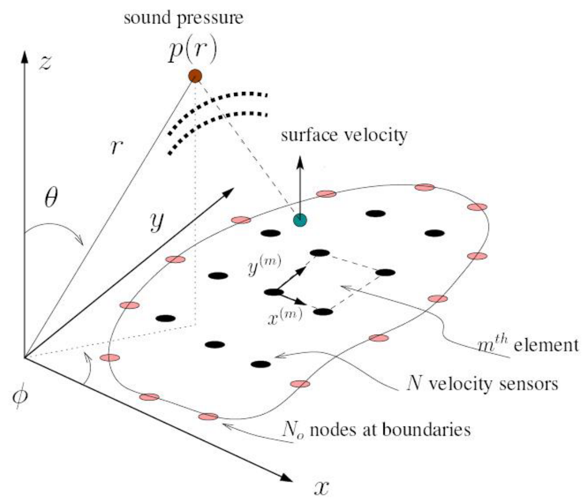

Figure 2 Sound radiation from a panel structure with N structural sensor nodes and N_o structural boundary nodes.

The estimated normal surface velocity $v(x,y)$ in Equation (1) can be substituted to Equation (2) so the far-field sound pressure can be estimated from the vibration measurements from all M elements (Halim and Cazzolato 2005b):

$$p(r,\theta,\phi) = \kappa\exp(-jkr)\left\{\sum_{m=1}^{M} b^{(m)}P^{(m)}A^{(m)}\right\}w \tag{4}$$

where

$$\begin{aligned} \kappa &= \frac{j\omega_o\rho}{2\pi r} \\ b^{(m)} &= \exp\left(j\left(\alpha x_o^{(m)} + \beta y_o^{(m)}\right)\right) \\ P^{(m)} &= \int_{y^{(m)}}\int_{x^{(m)}} H\left(x^{(m)}, y^{(m)}\right)\exp\left(j\left(\alpha x^{(m)} + \beta y^{(m)}\right)\right)dx^{(m)}dy^{(m)}. \end{aligned} \tag{5}$$

Here,

$$\begin{aligned} \alpha &= k\sin\theta\cos\phi \\ \beta &= k\sin\theta\sin\phi \end{aligned}$$

and $(x_o^{(m)}, y_o^{(m)})$ is the location of the m[th] element's origin.

The far-field sound pressure estimation can then be used for acoustic beam steering from a panel structure as discussed in the following section.

ACTIVE CONTROL METHOD FOR ACOUSTIC BEAM STEERING

Description of the objective function for acoustic beam steering

In order to achieve an acoustic beam steering of sound radiation from a panel, it is necessary to define an objective function that fits to this purpose. In this case, a reference 'spatial' beam pattern of sound radiation is used with can be tailored according to the requirement for beam steering. The objective function is defined by considering a continuous spatial error e between the reference 'spatial' beam pattern p_r and the estimated 'spatial' beam pattern p from vibration measurements:

$$e(r,\theta,\phi) = p(r,\theta,\phi) - p_r(r,\theta,\phi). \tag{6}$$

For the far-field condition, the radius r can be set at a particular value, i.e. $r=r_o$. A spatial averaging over the hemisphere of far-field sound field is performed to obtain an objective function that can be used for beam steering via active control. The objective function is defined as:

$$W = \int_0^{2\pi}\int_0^{\pi/2} e(r_o,\theta,\phi)^H e(r_o,\theta,\phi)d\theta d\phi \qquad (7)$$

where G^H is the Hermitian transpose of matrix G. Let the estimated far-field sound pressure p in Equation (4) be expressed as $p = C_p(r,\theta,\phi)w$, which then substituted to Equations (6) and (7). After some algebra, the objective function can be expressed as:

$$W = w^H Q_1 w - w^H Q_2 - Q_2^H w + Q_3 \qquad (8)$$

where

$$Q_1 = \int_0^{2\pi}\int_0^{\pi/2} C_p(r_o,\theta,\phi)^H C_p(r_o,\theta,\phi)d\theta d\phi$$
$$Q_2 = \int_0^{2\pi}\int_0^{\pi/2} C_p(r_o,\theta,\phi)^H p_r(r_o,\theta,\phi)d\theta d\phi \qquad (9)$$
$$Q_3 = \int_0^{2\pi}\int_0^{\pi/2} p_r(r_o,\theta,\phi)^H p_r(r_o,\theta,\phi)d\theta d\phi.$$

The significance of this result is that the objective function W is a quadratic function of velocity measurements from structural sensors w. Hence, an active control can be designed to globally minimise this objective function, which consequently will attempt to regulate the acoustic beam pattern to be optimally close to the desired beam pattern p_r.

Figure 3 shows how a controller can be set-up based on the velocity measurements from the panel structure. The primary input disturbance generates a primary vibration on the panel P. The sensors detect the vibration and the velocity information is sent to the controller K where the controller attempts to generate a control input that minimises the objective function W in Equation (8). The minimisation can be done via an adaptive algorithm, for instance. The control input signal then generates a secondary input to the panel in order to modify the vibration characteristics of the panel to achieve the desired acoustic beam pattern.

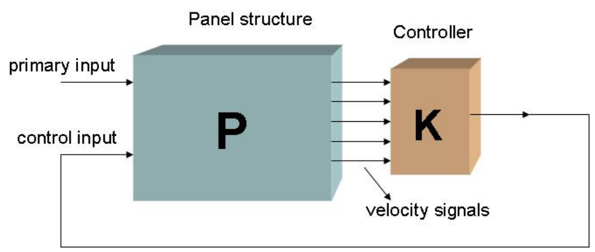

Figure 3 An active control diagram for regulating the acoustic beam pattern of radiated sound.

For a general active control for steering the beam, it can be noticed that the objective function in Equation (8) depends on the reference acoustic beam pattern p_r that is contained in Q_2 and Q_3 in Equation (9). Thus, active beam steering can be achieved by using a number of different objective functions for various reference beam patterns. By switching to the controller that minimises the appropriate objective function, the acoustic beam will be able to be steered according to the requirements. This active control method is illustrated in Figure 4 where a number of different controllers K_1, K_2 and K_3 and a switching method can be used.

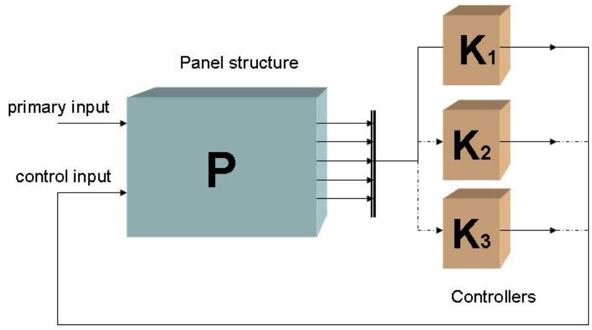

Figure 4 Beam steering method by switching to the required active controller.

Optimal tonal acoustic beam steering

For the purpose of demonstrating the effectiveness of the proposed beam pattern regulation approach, optimal tonal beam steering control is analysed using a feedforward configuration.

Consider a tonal system described as follows:

$$w = G_{wd}d + G_{wu}u \qquad (10)$$

where G_{ji} is the transfer from i to j with primary disturbance and control input denoted as d and u respectively.

Equation (10) can be substituted to the objective function W in Equation (8) and the optimal control can be computed using the quadratic minimisation approach (Elliot 2001):

$$u_{opt} = -\left(G_{wu}{}^H Q_1 G_{wu}\right)^{-1} G_{wu}{}^H \left(Q_1 G_{wd} d - Q_2\right). \qquad (11)$$

In the next section, the tonal optimal controller will be implemented on a panel structure.

BEAM STEERING ANALYSIS OF RADIATED SOUND FROM A RECTANGULAR PANEL

The results that will be presented here demonstrate how the acoustic beam pattern in the far-field can be regulated for beam steering. Consider a simply-supported rectangular aluminium panel with 5x5 structural velocity sensors distributed over the panel, using the sensor configuration similar to the one shown in Figure 1. The dimensions of the panel are 0.450m x 0.300mm x 0.004m. Point sources are used as primary disturbance and control sources, where the disturbance source is located at (x,y) = (0.129m,0.107m) based on the coordinates shown in Figure 1. The locations of control sources are described in Table 1.

Table 1. Locations of point control sources on the panel.

Control	x [m]	y [m]
1	0.141	0.091
2	0.250	0.143
3	0.375	0.094
4	0.346	0.246
5	0.300	0.136

Modal analysis (de Silva 2000) is used to obtain the panel model, with the first 20 vibration modes are considered for this analysis. The natural frequencies of the modes are shown in Table 2. The interpolation function used is a linear function similar to the one used in (Halim and Cazzolato 2005b).

Table 2.The first 7 vibration modes of the panel.

Mode	Frequency [Hz]
(1,1)	154.0
(2,1)	296.1
(1,2)	473.7
(3,1)	532.9
(2,2)	615.8
(3,2)	852.7
(4,1)	864.5

Results for beam pattern I at 160 Hz

In the first numerical analysis, a particular reference beam pattern in the far-field at r=8m is used as shown in Figure 5, using the spherical coordinates described in Figure 2. For this particular beam pattern, the purpose is to generate a beam pattern that is concentrated on the far-field region centred at $\phi = 200^o$.

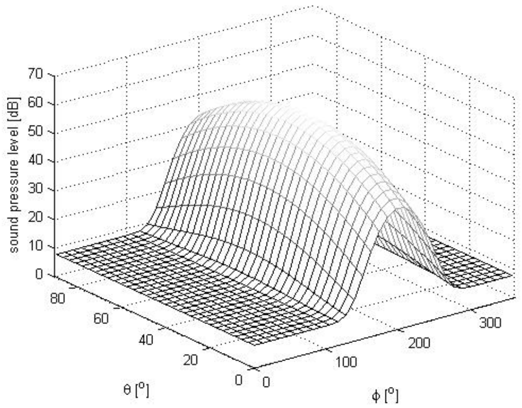

Figure 5 Reference acoustic beam pattern 1.

Figure 6 shows the far-field sound radiation at 160 Hz, which is just above the natural frequency of mode (1,1), where the panel behaves like a monopole source. An optimal control is implemented on the panel using the first 3 control sources whose locations are described in Table 1. Figure 7 shows the results where the acoustic beam pattern has been modified, closer to the reference beam pattern shown in Figure 5. Using 5 control sources (see Table 1 for their locations), the controlled beam pattern in Figure 8 is even closer to the desired reference beam pattern where the controller attempts to reduce the sound radiation at the regions of low and high ϕ. In general, more freedom in modifying the beam pattern can be expected by using more control sources, although some computational issues may also arise.

It can be observed that the sound radiation characteristics can be quite complicated since it depends on velocity profile of the panel vibration. The reference beam pattern used may also affect whether a practical beam pattern can be achieved effectively. Figures 9 and 10 depict the magnitude of the complex vibration velocity profiles of the panel, for without-control and with-control cases.

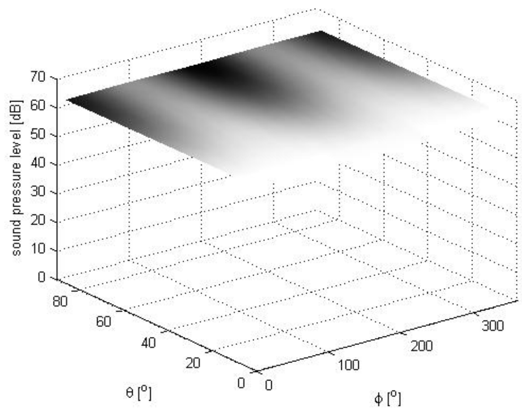

Figure 6 Far-field sound radiation at 160 Hz, without control.

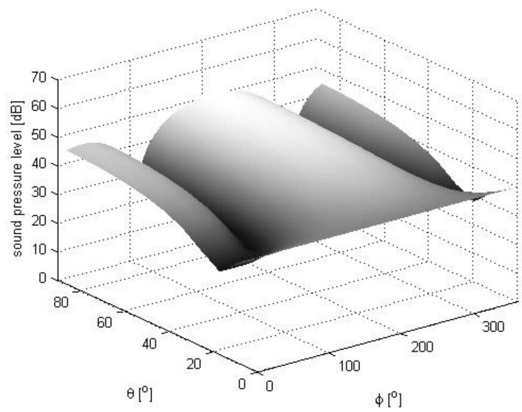

Figure 7 Far-field sound radiation at 160 Hz, with control using 3 control sources.

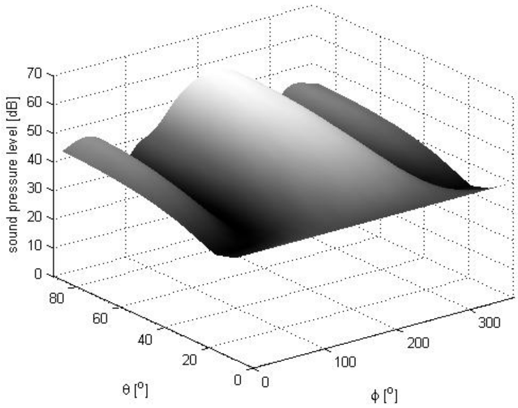

Figure 8 Far-field sound radiation at 160 Hz, with control using 5 control sources.

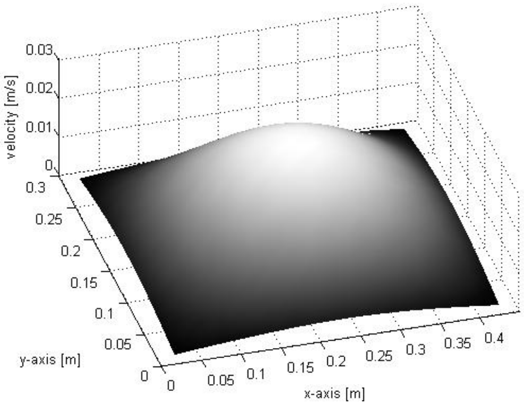

Figure 9 Vibration velocity profile of the panel at 160 Hz, without control.

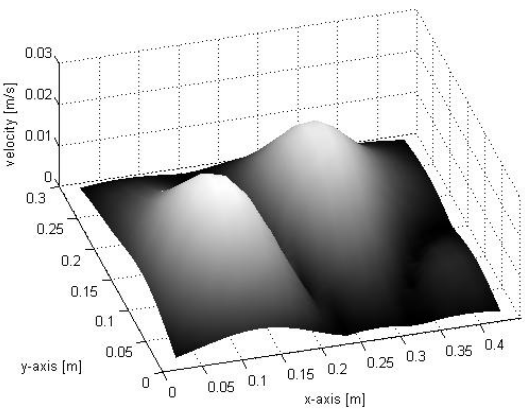

Figure 10 Vibration velocity profile of the panel at 160 Hz, with 3 control sources.

Results for beam pattern II at 160 Hz

In the second analysis, a different reference beam pattern is used as shown in Figure 11. Figure 12 illustrates how the beam pattern is modified by 3 control sources, as the velocity profile of the panel is also modified (see Figure 13). Comparing Figures 10 and 13, it can be seen that the velocity profile can be quite different for a different beam pattern, which can make beam pattern regulation complicated. The beam pattern result using 5 control sources is depicted in Figure 14, where the controller attempts to reduce the sound pressure at regions of high ϕ. It can be noticed that for two difference beam patterns presented in the numerical analysis, two different controllers arise. By switching between the two controllers, the sound radiation will be able to be steered between two different regions with low or high ϕ.

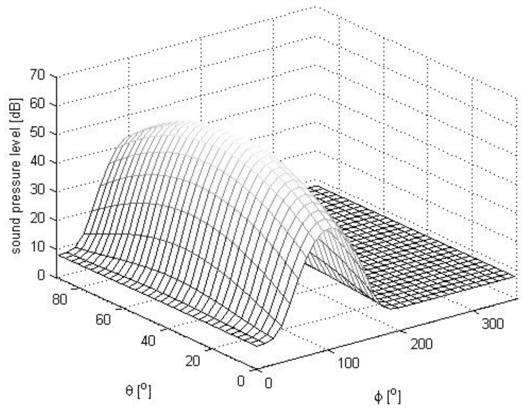

Figure 11 Reference beam pattern 2.

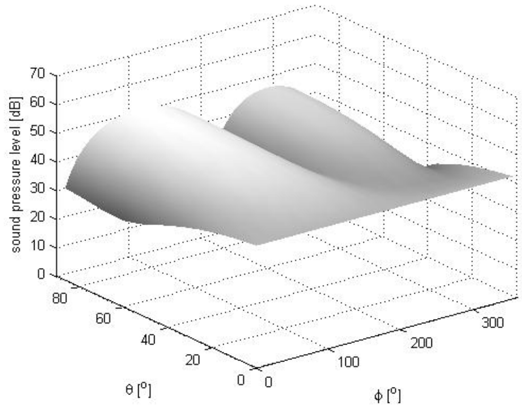

Figure 12 Far-field sound radiation at 160 Hz, with control using 3 control sources.

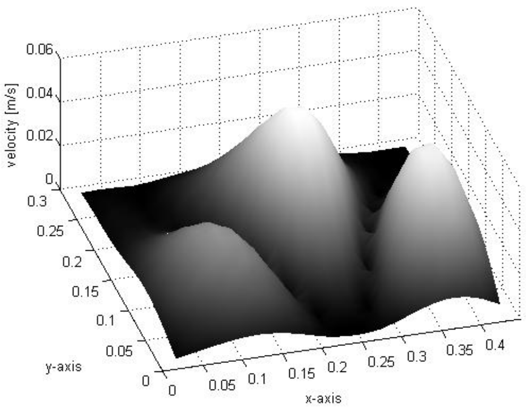

Figure 13 Vibration velocity profile of the panel at 160 Hz, with 3 control sources.

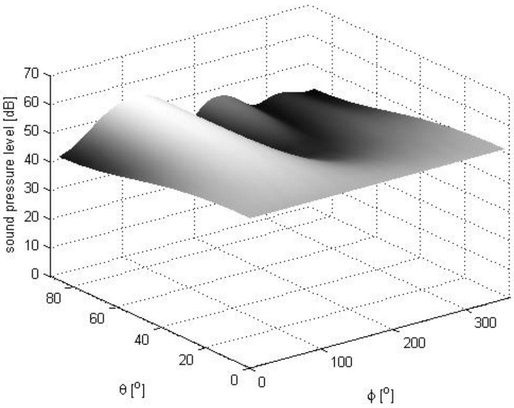

Figure 14 Far-field sound radiation at 160 Hz, with control using 5 control sources.

CONCLUSIONS

The proposed control approach for beam steering of sound radiation from a panel structure has been presented. The proposed approach utilises multiple structural sensors distributed over the panel, together with spatial interpolation functions, to estimate the sound radiation/beam pattern from the panel. An objective function that incorporates the estimated beam pattern of sound radiation has been defined, which can be used with active control methods for regulating the beam pattern. Numerical analysis on a rectangular panel showed that the beam pattern of sound radiation can be regulated for beam steering purposes by modifying the vibration characteristics of the panel.

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