A Generalized Approach to Modal Filtering for Active Noise Control—Part II: Acoustic Sensing

Simon G. Hill, Scott D. Snyder, Ben S. Cazzolato, Nobuo Tanaka, and Ryoji Fukuda

***Abstract*—Scaling laboratory-sized active noise control systems into industrial-sized implementations is a difficult exercise. Problems relating to sensing system design account for some of the difficulty. In the first part of this paper, an alternative approach to sensing system design was presented where acoustic radiation patterns are decomposed using fundamental acoustic quantities, rather than structural modal based quantities. In this paper, the approach is tackled in the acoustic domain instead of structual vibration. The technique has been simulated in the time and frequency domains using acoustic sensors and implemented experimentally.**

I. Introduction

When approaching the development of an active control system for a given structural acoustic radiation problem, two design ideals are 1) to be able to measure, and so attenuate, an error criterion that is directly related to a global quantity such as acoustic power or energy and 2) to minimize the number of input signals that must be handled by the controller. The former ideal may lead to the greatest levels of global disturbance attenuation [1], while the latter will minimize the complexity and maximize the speed of the control system [2]. These ideals often come into conflict.

In an attempt to reconcile the two ideals above, many active control researchers have turned their attention to variants of modal filtering. In modal filtering [3]–[9], the aim is to resolve global quantities, traditionally the amplitude of structural modes, from a large number of point sensor measurements. While free space sound fields do not have any true modes, it is possible to mathematically express the acoustic power radiated from a vibrating structure as the sum of contributions from orthogonal combinations of structural modes [1], [4], [10]–[15] or structural elements [16] by expressing acoustic power as a quadratic function of modes or elements and calculating the eigenvalues and eigenvectors of the expression. The eigenvectors effectively take on the role of modes in the modal filtering exercise, hence, the term ”radiation modes” [16].

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While the above approach to sensing system design works well in the laboratory on simple structures, it is difficult to apply to practical problems. Current design methodologies require detailed knowledge of the target structure to construct the acoustic power expression at the heart of the process. The techniques explored to date are explicitly vibration-based, to be used with vibration sensors and actuators, and have not been applied to acoustic sensing arrays. This greatly limits the scope of application of the techniques.

The aim of this work is to overcome these problems through an alternative sensing system design strategy, formulated in the acoustic space as opposed to on the surface of the structure. Referring to Fig. 1, a set of modal filter weights will be derived to facilitate reduction of a large array of acoustic sensor signals into a much smaller number of outputs that accurately reflect the global quantity, acoustic power. This greatly reduced set of signals can then be easily incorporated into a practical control system, where the unfiltered, large array of raw acoustic signals could not. A sensing system in the acoustic domain expands the scope of application to free-space radiation from both complicated structures and nonstructural sources such as duct outlets by removing the need for knowledge of structural information such as mode shapes, while providing the flexibility to use either vibration or acoustic actuators in the problem.

What will be presented here is an abbreviated derivation of the fundamental modal filtering problem. Some of the detailed rationale and development of the theory has been presented elsewhere [17], [18], and so will not be repeated. Of chief interest are a number of practical issues associated with implementation of the sensing system scheme in the time domain, as will be required in a practical setting. Simulations of the sensing system use a novel time-domain approach and will be supported by experimental results.

II. Theoretical Overview

For systems of interest here, the global performance measure J can be expressed as a quadratic function

$$J(t) = \boldsymbol{q}^T(t)\boldsymbol{A}(\omega)\boldsymbol{q}(t). \tag{1}$$

In (1), $\boldsymbol{q}$ is a state vector of quantities to be measured and $\boldsymbol{A}$ is a positive definite weighting matrix, often frequency-dependent. There are a wide variety of control system design methodologies that can be applied when the performance criteria is in this quadratic form.

In terms of sensing system design, stating the global performance measure in the form of (1) provides the designer with a target: measurement of the quantities in $\boldsymbol{q}$. The aim here is derivation of a quadratic expression of the form in (1) for radi-

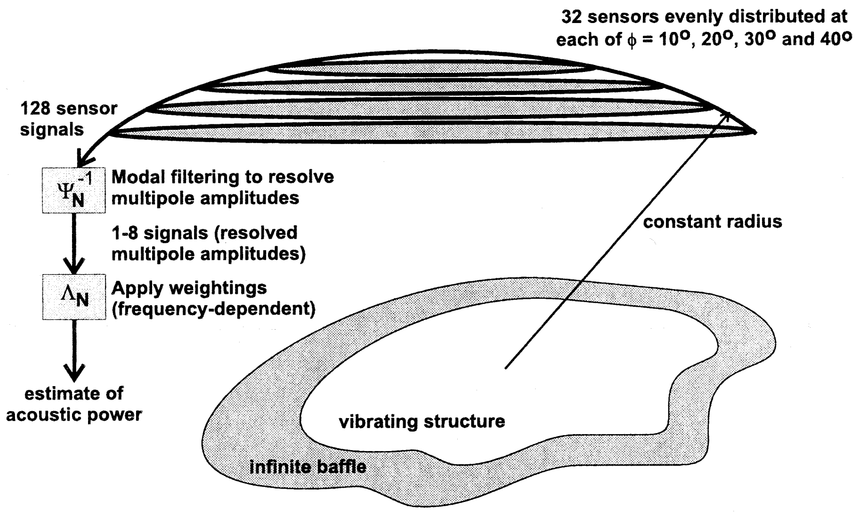


Fig. 1. Example sensing system.

ated acoustic power, in terms of some fundamental, "mode-like" quantity. This provides the basis for a modal filtering problem. In previous work, the goal has been to derive a performance measure in terms of a state vector $\boldsymbol{q}$ containing structural modes [1] or structural element [16] velocities, which can be measured using vibration sensor data.

Consider a planar structure in an infinite baffle subject to harmonic excitation and radiating into air, as shown in the companion paper [18, Fig. 2]. The acoustic power W radiated by this source can be evaluated by integrating the far-field acoustic intensity over a hemisphere enclosing the source. Using the geometry of [18, Fig. 2], acoustic power can be written as

$$W = \int_0^{2\pi} \int_0^{\pi/2} \frac{|p(\boldsymbol{r})|^2}{2\rho_0 c_0} |\boldsymbol{r}|^2 \sin\theta d\theta d\varphi. \quad (2)$$

In (2), $p(\boldsymbol{r})$ is the acoustic pressure p at some location $\boldsymbol{r}$ in space, with the location defined by the spherical coordinates $\boldsymbol{r} = (r, \theta, \phi)$. The terms ρ_0 and c_0 are the density of air and speed of sound in air, respectively.

The expression in (2) is in the desired form, but would require an infinite, or at least large, number if pressure measurements. However, as described in the companion paper [18], it is possible to re-express pressure as a function of multipole radiation patterns, similar in concept to expressing vibration as a function of modal components. Doing this, radiated acoustic power can be approximated and written in matrix form as

$$W \approx \boldsymbol{a}^H \boldsymbol{A}_a \boldsymbol{a} \quad (3)$$

where $\boldsymbol{a}$ is the vector of acoustic multipole amplitudes and $\boldsymbol{A}_a$ is a square weighting matrix, the (i, j)th term of which is defined by

$$\boldsymbol{A}_a(i,j) = \int_0^{2\pi} \int_0^{\pi/2} \frac{\psi_i^H(\boldsymbol{r})\psi_j(\boldsymbol{r})}{2\rho_0 c_0} |\boldsymbol{r}|^2 \sin\theta d\theta d\varphi \quad (4)$$

where ψ is the radiated transfer function between the acoustic multipole volume velocity and acoustic pressure at a point in space. The vector of acoustic multipole amplitudes can be derived from a set of acoustic pressure measurements using a modal filtering operation

$$\boldsymbol{a} = \boldsymbol{\Psi}^{-1}\boldsymbol{p} \quad (5)$$

where $\boldsymbol{\Psi}^{-1}$ is the inverse matrix of transfer functions between the multipoles and measurement locations and $\boldsymbol{p}$ is the vector of complex pressure at the measurement points.

In practice, further work is required before the modal filtering system can be implemented [18]. First, the weighting matrix $\boldsymbol{A}$ in (3) must be diagonalized to ensure that each of the multipoles is an independent component of acoustic power. This is accomplished via an orthonormal transform [18]. Second, the terms in the modal filter matrix $\boldsymbol{\Psi}^{-1}$ in (5) are often frequency-dependent [5], [9]. For practical reasons, these should be frequency independent; what is desired is to multiply each raw acoustic pressure signal by some scalar quantity and sum the results to derive a given multipole amplitude, something that is only possible if the modal filter weights are frequency independent. It is possible to separate the modal filter weights into a frequency dependent component and one approximately frequency independent [18], and move the frequency dependence entirely into the weighting matrix. Performing these operations, acoustic power can be restated as [18]

$$W \approx \boldsymbol{p}^H \left\{\boldsymbol{\Psi}_n^{-1}\right\}^H \boldsymbol{\Lambda}_N \boldsymbol{\Psi}_n^{-1}\boldsymbol{p}. \quad (6)$$

This is in the form of (1) where $\boldsymbol{q} = \boldsymbol{\Psi}_n^{-1}\boldsymbol{p}$ and $\boldsymbol{A}_a = \boldsymbol{\Lambda}_N$.

The expression for acoustic power in (6) is the ideal starting point for control system design. The system states $\boldsymbol{q}$ are now explicitly the product of a set of acoustic pressure measurements in $\boldsymbol{p}$ and a frequency-independent vector of modal filter weights $\boldsymbol{\Psi}_n^{-1}$. This is the sensing system part of the problem. The diagonal, frequency-dependent weighting matrix $\boldsymbol{\Lambda}_N$ is ideal for a range of control system design approaches. It is also ideal for a control reduction exercise, as it will show that some of the system states are far more important at a given frequency than others [18]. Physically, the terms in $\boldsymbol{\Lambda}_N$ are equivalent to radiation efficiencies for the multipoles used to decompose the pressure measurements. It is intuitive that some multipoles, such as

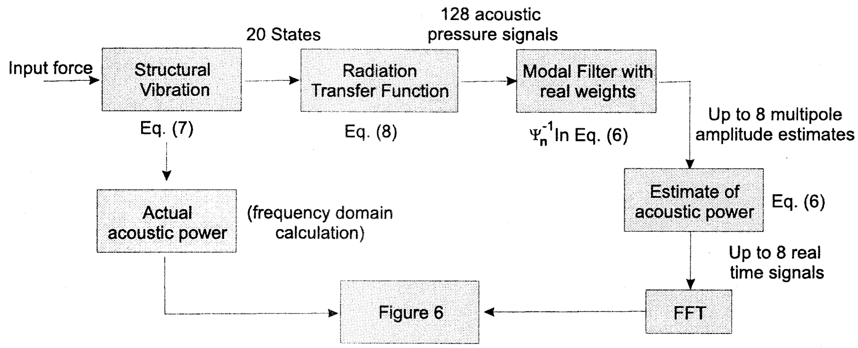


Fig. 2. Simulation procedure.

where the components are all in phase, have a high radiation efficiency, while others, such as where half of the monopole components are in phase and the other half out of phase, have lower radiation efficiencies. More detail is provided elsewhere [17], [18].

The remainder of this paper will focus on development, implementation, and quality assessment of a practical sensing system to provide the state measurements $\boldsymbol{q} = \boldsymbol{\Psi}_n^{-1}\boldsymbol{p}$. The development will be unique in its application to an acoustic sensor array.

III. Sensing System Simulation

The technique described in Section II implicitly assumes that complex (number) pressure measurements are available for the decomposition process. If this approach was to use vibration measurements, then, provided that the structure was lightly damped, this requirement could be simplified to positive or negative real values. However, in the acoustic space, the phase difference between sensing points is not due to combinations of modes vibrating in or out of phase, but rather propagation delays between source(s) and sensor(s). Implementation in the frequency domain to obtain complex numbers is not computationally viable with tens or hundreds of sensors, and simplification in the time domain to remove the complex number constraint requires examination.

Simulating the acoustic radiation from the structure involves two steps: simulation of the structural vibration in response to the force input, followed by simulation of radiation into free space. Shown in Fig. 2 is an overview of the simulation procedure. In the time domain, vibration of a single structural mode is governed by the state equation

$$\begin{aligned}\dot{\boldsymbol{x}} &= \boldsymbol{A}\boldsymbol{x} + \boldsymbol{B}\boldsymbol{u}\\ \boldsymbol{y} &= \boldsymbol{C}\boldsymbol{x} + \boldsymbol{D}\boldsymbol{u}\end{aligned}$$

where $\boldsymbol{u}$ is the input force

$$\boldsymbol{u} = \begin{bmatrix} 0 \\ f(t) \end{bmatrix}$$

$\boldsymbol{x}_n$ is the system states

$$\boldsymbol{x}_n = \begin{bmatrix} x_n \\ \dot{x}_n \end{bmatrix} \quad (n = 1, 2, \ldots)$$

$\boldsymbol{y}_n$ is the system output vector

$$\boldsymbol{y}_n = \begin{bmatrix} y_n \\ \dot{y}_n \end{bmatrix} \quad (n = 1, 2, \ldots)$$

and matrices $\boldsymbol{A}_n$, $\boldsymbol{B}_n$, $\boldsymbol{C}_n$, and $\boldsymbol{D}_n$ defined as

$$\begin{aligned}\boldsymbol{A}_n &= \begin{bmatrix} 0 & 1 \\ -\omega_n^2 & -2\zeta\omega_n \end{bmatrix}, \quad n = 1, 2, \ldots\\ \boldsymbol{B}_n &= \begin{bmatrix} 0 \\ \frac{1}{m}\Psi_n \end{bmatrix}, \quad n = 1, 2, \ldots\\ \boldsymbol{C}_{d,n} &= [1 \quad 0] \quad \boldsymbol{C}_{v,n} = [0 \quad 1], \quad n = 1, 2, \ldots\\ \boldsymbol{D}_n &= 0\end{aligned} \tag{7}$$

where n is the modal index and $\boldsymbol{C}_{v,n}$ denotes the $\boldsymbol{C}$ matrix for velocity as an output and $\boldsymbol{C}_{d,n}$ for displacement, m is the modal mass of the structure, ω_n is the natural frequency of the nth mode, ζ is the damping ratio, and Ψ_n is the mode shape function value for the nth mode at the input ($\boldsymbol{B}$ matrix) and measurement location ($\boldsymbol{C}$ matrix). The total structure vibration model is built up from multiple modes in parallel.

Radiation from a planar, baffled structure, with the geometry shown in Fig. 2 of the companion paper, is governed by the Raleigh integral. If a far-field measurement point is considered, the radiation transfer function between pressure and velocity can be expressed as a time-domain transfer function

$$Z_{\text{rad}}(n, \mathbf{r}) = \frac{s\rho_0 L_x L_y}{2\pi r M_n N_n \pi^3} e^{-sT} \left[\frac{(-1)^{M_n} e^{-sT_\alpha} - 1}{\left(\frac{\alpha}{M_n \pi}\right)^2 - 1} \right] \times \left[\frac{(-1)^{N_n} e^{-sT_\beta} - 1}{\left(\frac{\beta}{N_n \pi}\right)^2 - 1} \right] \tag{8}$$

where r is the distance, ρ_0 is the density of air, L_x and L_y are the dimensions of the panel, and M_n and N_n are the nth modal indices in the x and y directions, respectively. $T = r/c_0$ is the acoustic time delay between the structure and the point of interest in the acoustic space, T_α is the additional delay term due to the expanse of the panel in the x direction

$$T_\alpha = \frac{L_x \sin\theta \cos\phi}{c_0} \tag{9}$$

TABLE I
STRUCTURAL RESONANCES BELOW 250 HZ FOR THE TEST PANEL

Mode	Resonance Frequency (Hz)
1 , 1	33.0
2 , 1	53.1
3 , 1	86.6
1 , 2	111.8
2 , 2	131.9
4 , 1	133.5
3 , 2	165.4
5 , 1	193.8
4 , 2	212.3
1 , 3	243.2

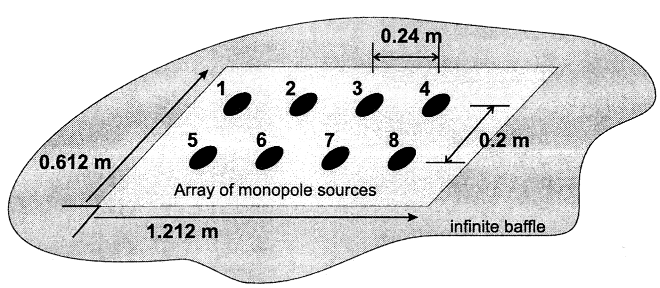


Fig. 3. Monopole arrangement.

and T_β is the additional delay term due to the expanse of the panel in the y direction

$$T_\beta = \frac{L_y \sin\theta \sin\phi}{c_0}. \tag{10}$$

The exponential terms in (8) can be efficiently approximated using a Padé approximation. The order of the approximation is important for an accurate estimate.

As outlined in the introduction, the aim of the methodology developed here is to produce a sensing system that balances the design ideals of global error criterion measurement with a minimum of input signals to the control law and/or tuning algorithm, without knowledge of the mode shape functions or other structural information (if, indeed, the noise source *is* a structure). To assess, in simulation, the quality of the approach described here, the sound power radiated by a simple structure will be compared to the estimates of this global quantity from using the modal filtering arrangement. These comparisons will provide several insights into where the sensing system design approach will be applicable and how the range of operation can be extended.

The simple structure used in the simulation is a rectangular steel panel, dimensions $1.212\ \text{m} \times 0.612\ \text{m} \times 0.004\ \text{m}$. The panel is lightly damped, with a damping coefficient $\zeta = 0.005$ and excited at (0.256, 0.356). The frequency range of interest is up to 250 Hz. There are 10 modal resonances in this range, as listed in Table I.

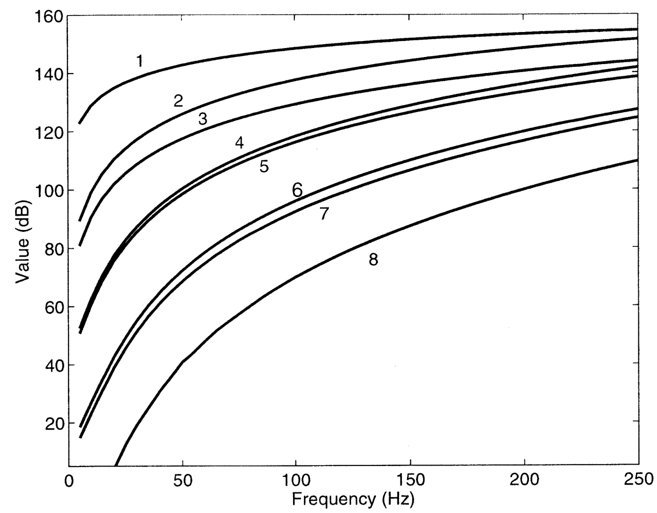


Fig. 4. Eigenvalues of the weighting matrix $\boldsymbol{A}_a$ plotted as a function of frequency.

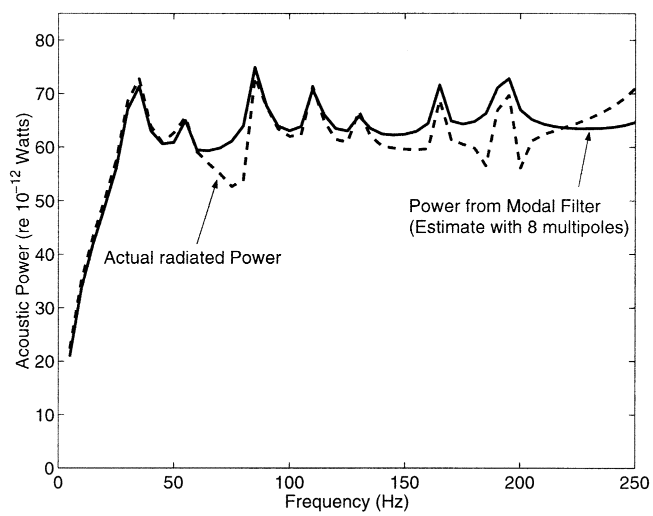


Fig. 5. Sound power radiation from the plate in simulation.

The modal filtering parameters for the simulation problem are derived on the basis of radiation patterns from multipole combinations of eight monopole sources, arranged similar to [18]. This various combinations are described in detail in [18], however, there is a slight difference in this case where the multipoles positions have been chosen to be evenly distributed across the plate, as illustrated in Fig.3.

Referring back to Fig. 1, the simulated sensing system contains 128 microphones evenly spread over the range (2π) of ϕ at each of $\theta = 10°$, $20°$, $30°$, and $40°$. The sensing grid location is chosen to be in the far field at a radius of 10 m.

The most straightforward approach to overcoming the requirement of complex number acoustic pressure measurements is to place all acoustic sensors on the same radius from the noise source and then assume that the pressure signal at all sensing locations is either in or out of phase (i.e., force the phase to be either "positive" or "negative"), ignoring smaller variations between individual sensors. Working through the problem, this is not exactly correct. For example, the acoustic pressure signal between two "positive" sensors might differ by

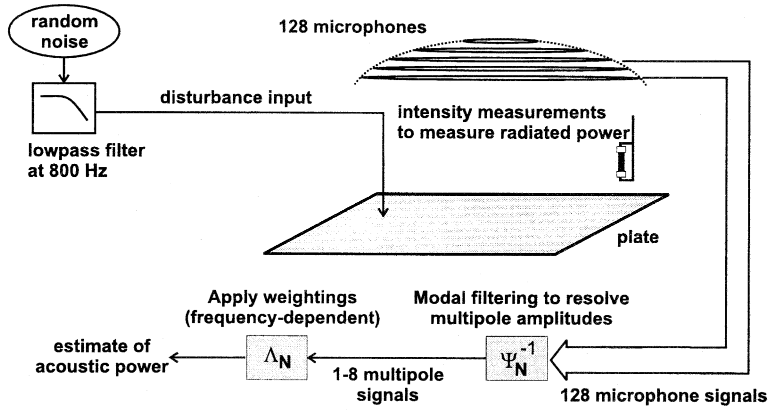


Fig. 6. Experimental setup.

a few degrees. It is, however, reasonably close. If the modal filter weight phases are all 0° or 180° different, then the system is purely real and the complex number requirement can be dropped. The aim of the simulations is to see how accurate the acoustic power estimate from the modal filtering arrangement will be with this assumption.

The measured sound pressure level at each microphone location will be filtered into the eight multipoles, with the sound power estimated using (6).

As mentioned, the weights in $\mathbf{\Lambda}_N$ in (6) provide an indication of the radiation efficiency of the various multipoles, the amplitudes of which will be output from the modal filters. These numbers can be used in a truncation exercise, where only the most efficient radiators are used as inputs to the control law and tuning algorithm. The eigenvalues for the case being simulated are plotted over the target frequency range in Fig. 4. Observe that at in this frequency range the radiation efficiencies of five of the modal filter quantities (the multipoles) greatly exceed the other three. This suggests that a sensing system that derives only five multipole quantities will provide a reasonably accurate estimate of radiated acoustic power at low frequencies.

Illustrated in Fig. 5 is a plot of radiated sound power estimated by using the resolved modal quantities, all eight multipoles, in (6), averaged over time. For reference, the actual acoustic power radiated from the plate, calculated using the method outlined by Hansen and Snyder [4], is also shown. The plots compare well, especially at the spectral peaks, which would be of most concern in the control exercise. The poor prediction at the troughs in the spectrum are likely due to neglecting small phase differences in the modal filter weights in the time-domain implementation. The troughs, or transmission zeros, typically involve some form of phase cancellation, and neglecting phase differences of a few degrees would impede this prediction. This figure also illustrates a performance drop off in the multipole prediction technique at higher frequencies, as would be expected considering the number of monopoles used in the decomposition.

It must be emphasized that the estimate of acoustic power from the modal filtering arrangement comes from only eight resolved quantities, a number of signals that many controllers could easily work with, and the modal filter weights in $\mathbf{\Psi}_n^{-1}$ were fixed to be the exact values at 100 Hz.

IV. Sensing System Experiments

The aim of the work to be presented here is experimental implementation of a modal filtering arrangement based upon the theoretical development outlined in Section III. The acoustic sensing system is built on an eight-armed parabolic frame and was located 2 m above a simply supported plate with the same dimensions as Section III. The sensing system has 128 microphones, divided up and distributed evenly on four rings. The outer and largest ring contains 48 microphones with the remaining rings containing 40, 24, and 16 microphones. The distance between the rings is 250 mm with the first ring located 350 mm from the center. For the experiments, the array was opened to an angle of 15 degrees. The experiment arrangement is sketched in Fig. 6.

The 128 microphone signals interface to a custom-built I/O and modal filtering system. The modal filtering system was built using (low-cost) Analog Devices AD1845JST 16-b stereo codecs, interfacing to an Analog Devices ADSP21062 EZKIT. The system is also capable of 128 outputs, although the eigenvalue plots in Fig. 4 suggest that only three to five outputs, equating to the amplitudes of three to five multipoles in the acoustic field decomposition, will actually be required. With the modal filtering weights being fixed values, as described in Section III, the single low-cost DSP can perform the required calculations at a sample rate of 5.5 kHz.

The sound power radiated by the panel was estimated by measuring the sound intensity on a 100 mm $\times$ 100 mm grid of points, or 15 points in the x direction by nine points in the y direction, on a plane 200 mm above the surface of the panel. The sound intensity measurements were made as a spectrum in 0.25-Hz increments. The measurement process was automated using a computer-controlled traverse integrated with a Bruel & Kjaer PULSE system. Note that this method only approximately measures the sound power radiated by the panel as some of the sound radiated by the panel would have not have been radiated perpendicular to the panel surface and so would have not been recorded on the measurement plane. The result of taking a fast Fourier transform (FFT) on the output from the modal filtering system to gain an estimate of radiated power is illustrated in Fig. 7 along with an estimate from the scanning intensity probe.

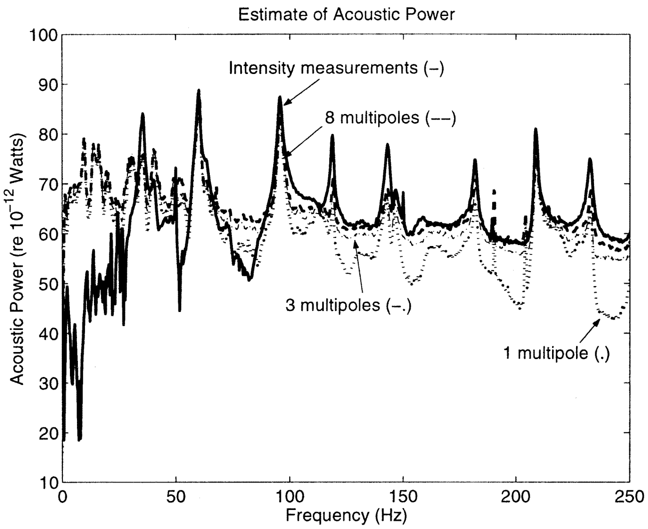


Fig. 7. Estimate of sound power from the modal filter.

Observe that the data correlates well above 50 Hz; below this frequency the performance of the acoustic sensing system decreases due to its proximity (2 m) to the plate. It is worth reiterating a few points concerning this result.

- The modal filter weights are a fixed set of numbers, applied to the 128 input signals. The results are summed to produce the output corresponding to a given multipole.
- The relatively few outputs from the modal filtering process provide an almost perfect measurement of acoustic power. This was the aim of the exercise: to have a small number of outputs that provide a high-fidelity measurement of a global error criterion.
- The modal filter weights have been derived with only the most passing reference to the structural system, that reference being the general size (order of magnitude) of the structure.

While the work described here specifically targets sensing system design, the next step would be implementation of a control system using the (small set of) multipole signals.

V. Conclusion

A method of modal filtering using radiation patterns produced by acoustic multipoles has been presented. The acoustic field produced by a radiating structure is decomposed using the multipole radiation patterns as basis functions and so requires no knowledge of structure modes shapes. This opens the application to more complex structural radiation problems, something not possible with previous techniques. Simulations of the sensing system in both the frequency and time domains illustrate that a minimum number of signals (resolved multipoles) are required to provide a good approximation of radiated power. The modal filter and acoustic sensing system has been shown to work over a wide frequency band with experiments showing that, in practice, a large number of sensors signals can be condensed into only a few inputs which give a good estimate of a global error criterion. The technique requires no knowledge of structural resonances or mode shapes, a leap over current state of the art sensing systems.

References

[1] S. D. Snyder and N. Tanaka, “On feedforward active control of sound and vibration using error signals,” *J. Acoust. Soc. Amer.*, vol. 94, no. 4, pp. 2181–2193, 1993.

[2] S. D. Snyder, “Microprocessors for active control: Bigger is not always enough,” *Noise Control Eng.*, vol. 49, pp. 21–29, 2001.

[3] M. J. Balas, “Active control of flexible systems,” *J. Optim. Theory Applicat.*, vol. 25, no. 3, pp. 415–436, 1978.

[4] C. H. Hansen and S. D. Snyder, *Active Control of Noise and Vibration*. London, U.K.: E&FN Spon, 1997.

[5] C. K. Lee and F. C. Moon, “Modal sensors/actuators,” *J. Appl. Mech.*, vol. 57, pp. 434–441, 1990.

[6] L. Meirovitch and H. Baruh, “Control of self-adjoint distributed-parameter systems,” *AIAA J.*, vol. 5, no. 1, pp. 60–66, 1982.

[7] ——, “The implementation of modal filters for control of structures,” *AIAA J. Guidance, Contr. Dynam.*, vol. 8, no. 6, pp. 707–719, 1985.

[8] L. Meirovitch and J. K. Bennighof, “Modal control of traveling waves in flexible structures,” *J. Sound Vib.*, vol. 111, no. 1, pp. 131–144, 1986.

[9] D. R. Morgan, “An adaptive modal-based active control system,” *J. Acoust. Soc. Amer.*, vol. 89, no. 1, pp. 248–256, 1991.

[10] A. Berry, “Advanced sensing strategies for the active control of vibration and structural radiation,” in *Proc. Active 99*, 1999, pp. 73–90.

[11] S. D. Snyder, N. Tanaka, and Y. Kikushima, “The use of optimally shaped piezo-electric film sensors in the active control of free field structural radiation, Part 1: Feedforward control,” *J. Vib. Acoust.*, vol. 117, pp. 311–322, 1995.

[12] A. P. Berkhoff, “Sensor scheme design for active structural acoustic control,” *J. Acoust. Soc. Amer.*, vol. 108, no. 3, pp. 1037–1045, Sep 2000.

[13] W. Baumann, F.-S. Ho, and H. Robertshaw, “Active structural acoustic control of broadband disturbances,” *J. Acoust. Soc. Amer.*, vol. 92, pp. 1998–2005, 1992.

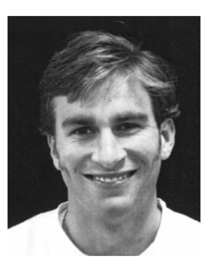
[14] C. R. Fuller, C. H. Hansen, and S. D. Snyder, “Active control of sound radiation from a vibrating rectangular panel by sound sources and vibration inputs: An experimental comparison,” *J. Sound Vib.*, vol. 145, no. 2, pp. 195–215, 1991.

[15] C. R. Fuller and R. A. Burdisso, “A wavenumber domain approach to the active control of structure-borne sound,” *J. Sound Vib.*, vol. 148, no. 2, pp. 355–360, 1991.

[16] S. J. Elliott and M. E. Johnson, “Radiation modes and the active control of sound power,” *J. Acoust. Soc. Amer.*, vol. 94, no. 4, pp. 2194–2204, 1993.

[17] S. D. Snyder, N. C. Burgan, and N. Tanaka, “An acoustic-based modal filtering approach to sensing system design for active control of structural acoustic radiation: Theoretical development,” *Mech. Syst. Signal Process.*, vol. 16, no. 1, pp. 123–139, 2002.

[18] N. C. Burgan, S. D. Snyder, N. Tanaka, and A. C. Zander, “A generalized approach to modal filtering for active noise control—Part I: Vibration sensing,” *IEEE Sensors J.*, vol. 2, Dec. 2002.



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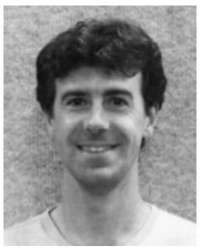

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