Active vibroacoustic control with multiple local feedback loops

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When multiple actuators and sensors are used to control the vibration of a panel, or its sound radiation, they are usually positioned so that they couple into specific modes and are all connected together with a centralized control system. This paper investigates the physical effects of having a regular array of actuator and sensor pairs that are connected only by local feedback loops. An array of 4×4 force actuators and velocity sensors is first simulated, for which such a decentralized controller can be shown to be unconditionally stable. Significant reductions in both the kinetic energy of the panel and in its radiated sound power can be obtained for an optimal value of feedback gain, although higher values of feedback gain can induce extra resonances in the system and degrade the performance. A more practical transducer pair, consisting of a piezoelectric actuator and velocity sensor, is also investigated and the simulations suggest that a decentralized controller with this arrangement is also stable over a wide range of feedback gains. The resulting reductions in kinetic energy and sound power are not as great as with the force actuators, due to the extra resonances being more prominent and at lower frequencies, but are still worthwhile. This suggests that an array of independent modular systems, each of which included an actuator, a sensor, and a local feedback control loop, could be a simple and robust method of controlling broadband sound transmission when integrated into a panel. [DOI: 10.1121/1.1433810]

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I. INTRODUCTION

The vibration of a structure can be actively controlled by feeding back the signals measured by sensors on the structure to integrated actuators.[1,2] Sound radiation from structures can also be actively controlled,[3] although in this case it is most important to control the components of vibration which radiate sound most efficiently, which has been termed active structural acoustic control.

Active control systems are usually designed by selecting the number and position of the actuators, the number and position of the sensors, and the controller response. The positions of the actuators and sensors used in active vibroacoustic control systems are often chosen so that they can couple into the structural modes that dominate the vibration or the sound radiation.[1–3] As the frequency of excitation increases, however, the detailed shape of these structural modes become increasingly sensitive to the boundary conditions and external loads on the structure and hence become more uncertain. It may thus be preferable to use a larger number of actuators and sensors than are strictly required, arranged in a regular array so that the structural modes are controlled whatever their shape.

There are considerable advantages in collocating the actuators and the sensors in such a feedback control system. When the actuator and sensor are also dual, in the sense that the product of the actuator input and the sensor response is proportional to the power supplied to the structure,[4] the plant response, from actuator input to sensor output, will have a positive real part, since the uncontrolled structure is passive. If a collocated force actuator and velocity sensor were used, for example, the plant response would be proportional to the input, or point, mobility of the structure, which must have a positive real part. The bandwidth over which this passivity property holds will, in practice, be limited by the dynamics of the transducers used. Provided the frequency response of the feedback controller also has a positive real part, the polar plot of the open loop frequency response function, i.e., the Nyquist plot, must stay in the right-hand half of the complex plane and so the system is unconditionally stable, since the polar plot cannot encircle the Nyquist point. The generalization of this simple passivity property to multichannel systems is discussed in the following, where it is shown that if the collocated actuators and sensors are coupled only by local feedback control loops with positive feedback gains, then the controller is passive and stability is assured for a passive plant. Such an array of locally acting feedback loops is referred to as a decentralized control system. It may also be possible to economically implement such an array of integrated transducers using micro-electromechanical systems technology.

In this paper, an initial version of which was published in Ref. 5, we investigate the consequences of a decentralized feedback control strategy, which uses a set of 16 collocated actuators and sensors on a panel, when the panel is subject to an incident acoustic excitation. Each actuator is driven individually by the output of the corresponding sensor so that only local feedback control is implemented, with each actuator, sensor, and controller operating independently.

The objective is to investigate the effect on both the vibration of the panel, as quantified by its kinetic energy, and the sound radiated by the panel, as quantified by the sound power it radiates. It will be assumed that each of the sensors

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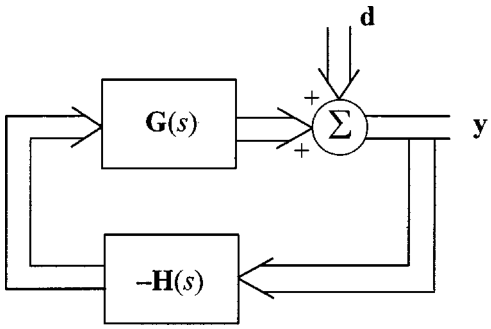

FIG. 1. Multichannel feedback control system, which for a passive plant response, $\mathbf{G}(s)$, and a passive controller $\mathbf{H}(s)$, is unconditionally stable.

measures the panel velocity at the corresponding point, which could be achieved in practice by integrating the output of a small accelerometer for example. Initially, the actuators will be assumed to be collocated devices which generate out of plane forces, as discussed previously. Although such actuators have useful theoretical properties, they require an inertial base to react off. It would be more useful in practice to have actuators that are fully integrated with the panel. In the latter parts of the paper the corresponding results will be described with velocity sensors and small strain actuators directly underneath them, such as could be implemented using piezoelectric devices. The similarity between the behavior of the ''ideal'' force actuator and velocity sensor control system will be compared with that of the ''practical'' control system with piezoelectric actuators and velocity sensors.

In Sec. II the performance of such a multichannel decentralized system is investigated, together with the conditions for stability. Section III describes the theoretical model used to calculate the panel vibration and sound radiation. Sections IV and V discuss the results of feedback control using force actuators and piezoelectric actuators, respectively, and the overall conclusions are summarized in Sec. VI.

II. MULTICHANNEL FEEDBACK CONTROLLERS

In this section we consider multichannel feedback control systems with equal numbers of collocated actuators and sensors.[4,6,7] In this case, the plant and controller responses are square matrices, $\mathbf{G}(s)$ and $\mathbf{H}(s)$, and the control objective is disturbance rejection, as illustrated by the block diagram in Fig. 1. Provided the control system is stable, the vector of spectra for the residual signals at the sensor outputs, $\mathbf{y}(j\omega)$, is related to that of the sensor outputs before control, $\mathbf{d}(j\omega)$, by

$$\mathbf{y}(j\omega)=[\mathbf{I}+\mathbf{G}(j\omega)\mathbf{H}(j\omega)]^{-1}\mathbf{d}(j\omega). \tag{1}$$

Similarly the vector of control inputs to the actuators, $\mathbf{u}(j\omega)$, is given by

$$\mathbf{u}(j\omega)=\mathbf{H}(j\omega)[\mathbf{I}+\mathbf{G}(j\omega)\mathbf{H}(j\omega)]^{-1}\mathbf{d}(j\omega). \tag{2}$$

In the case under consideration here, $\mathbf{G}(j\omega)$ is the fully populated matrix of input and transfer responses between the actuators and sensors on the panel and $\mathbf{H}(j\omega)$ is a diagonal matrix, which we will assume to have constant gains on each channel so that $\mathbf{H}(j\omega)=h\mathbf{I}$, where h is the feedback gain. Thus, given a set of panel responses, $\mathbf{G}(j\omega)$, and a feedback gain, h, the actuator signals can be calculated and hence the total response of the panel to both the primary and secondary excitations can be found. These results rely on the control system being stable, and the conditions for stability in this application are discussed in the following.

If collocated and dual transducers are used, then the real part of $\mathbf{G}(j\omega)$ must be positive definite, since the total power supplied to the uncontrolled system by all the actuators must be positive. If force actuators and velocity sensors are used, for example, then $\mathbf{u}(j\omega)=\mathbf{f}(j\omega)$, where $\mathbf{f}(j\omega)$ is the vector of applied forces and $\mathbf{y}(j\omega)=\mathbf{v}(j\omega)$, where $\mathbf{v}(j\omega)$ is the vector of measured velocities, and the power supplied by the actuators to the system at a frequency ω can be written as

$$\Pi(\omega)=\tfrac{1}{2}\,\mathrm{Re}[\mathbf{f}^H(j\omega)\mathbf{v}(j\omega)], \tag{3}$$

where Re denotes the real part and H denotes the Hermitian (conjugate) transpose. Since $\mathbf{v}(j\omega)=\mathbf{G}(j\omega)\mathbf{f}(j\omega)$ in this case, then Eq. (3) can be written as

$$\Pi(\omega)=\tfrac{1}{2}\,\mathrm{Re}[\mathbf{f}^H(j\omega)\mathbf{G}(j\omega)\mathbf{f}(j\omega)]. \tag{4}$$

Assuming reciprocity, $\mathbf{G}(j\omega)$ is also symmetric, so that

$$\Pi(\omega)=\tfrac{1}{2}\mathbf{f}^H(j\omega)\mathrm{Re}[\mathbf{G}(j\omega]\mathbf{f}(j\omega), \tag{5}$$

and it is clear that $\mathrm{Re}[\mathbf{G}(j\omega]$ must be positive definite if $\Pi(\omega)$ is to remain positive for all combinations of applied force, and the real parts of all the eigenvalues of $\mathbf{G}(j\omega)$ must be positive for all ω, so that the system is passive.[7,8] We assume that there is always some level of damping in the structure, so that $\Pi(\omega)$ can never be exactly zero unless $\mathbf{f}(\omega)$ is identically zero.

We also assume that the controller is designed so that it too has a positive definite real part at all frequencies, as would be the case if $\mathbf{H}(j\omega)=h\mathbf{I}$ and $h>0$. The plant and controller are thus both passive and the feedback control system illustrated in Fig. 1 must be unconditionally stable.[7,8] Thus if multiple local feedback loops are implemented with fixed gains then the system is stable provided each of the individual feedback gains is positive. Under these conditions the feedback gains can, in principle, be increased without limit and the signals from the control sensors can be driven to zero.

It should also be noted in passing that if an independent reference signal is available and feedforward control was implemented on such a system with collocated transducers, then independent loops could also be used for the adaptation of each actuator signal, using only the error signal from the corresponding sensor.[9] Because each of the eigenvalues of the matrix of plant responses has a positive real part, the stability of such a decentralized feedforward controller is guaranteed for slow convergence, provided the estimated plant responses used by the individual adaptation loops also had a positive real part.

Although the multichannel plant response is guaranteed to be passive if collocated point force actuators and velocity sensors are used, this property cannot be guaranteed with piezoelectric actuators and velocity sensors. The stability of a general multichannel control system can, however, always be determined[10] by examining whether the locus of the de-

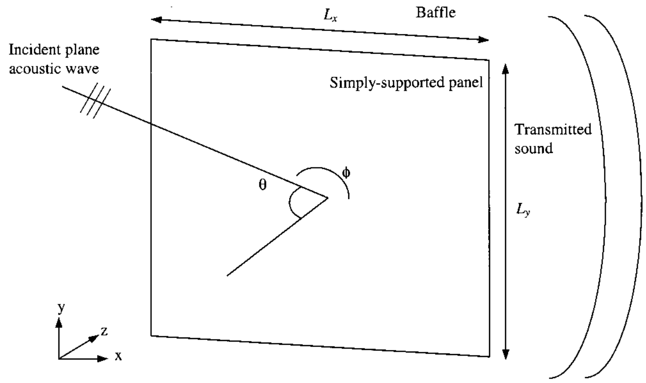

FIG. 2. Physical arrangement for the computer simulations, in which the vibration of a simply supported panel is excited by a plane acoustic wave on one side and radiates sound into an anechoic half space on the other side of the panel.

terminant of $[\mathbf{I}+\mathbf{G}(j\omega)\mathbf{H}(j\omega)]$ encloses the origin as ω varies from $-\infty$ to ∞. Alternatively the fact that the determinant of a matrix is the product of its eigenvalues can be used to derive a series of polar plots, each of which are analogous to the single channel Nyquist criteria. Specifically we note that

$$\det[\mathbf{I}+\mathbf{G}(j\omega)\mathbf{H}(j\omega)]=(1+\lambda_1(j\omega))(1+\lambda_2(j\omega))\cdots, \tag{6}$$

where $\lambda_i(j\omega)$ is the ith eigenvalue of $\mathbf{G}(j\omega)\mathbf{H}(j\omega)$, and so provided the locus of none of the eigenvalues encloses the Nyquist point $(-1,0)$ as ω varies from $-\infty$ to ∞, the system will be stable.

III. SIMULATION STUDY

The arrangement used in this simulation study is shown in Fig. 2, in which a thin aluminum panel, 278 mm $\times$247 mm$\times$1 mm, is subject to an incident plane acoustic wave of unit pressure. The plane wave is assumed to be incidental at azimuthal and lateral angles of 45° and 45° and thus excites all the structural modes of the panel.[11] The panel is assumed to be driven into motion by the incident acoustic wave and then to radiate sound on the other side. A weakly coupled analysis is used, in that the radiated pressure is assumed to have no effect on the panel vibration, which is a reasonable assumption in air for this thickness of panel. The panel is assumed to be simply supported and its velocity distribution is represented by the finite modal series

$$v(x,y,\omega)=\sum_{n=1}^{N} a_n(\omega)\psi_n(x,y), \tag{7}$$

where x and y are the spatial coordinates on the panel, $a_n(\omega)$ is the frequency-dependent amplitude of the nth mode, and $\psi_n(x,y)$ is its mode shape. The mode shape is assumed to be real and normalized so that the surface integral of its square value is equal to the surface area, so that in this simply supported case

$$\psi_n(x,y)=4\sin\left(\frac{n_1\pi x}{L_x}\right)\sin\left(\frac{n_2\pi y}{L_y}\right), \tag{8}$$

where L_x and L_y are the dimensions of the panel and n_1 and n_2 are the two modal integers, which are denoted above by the single index n.

The modal amplitude is given by the product of a second-order resonance term, $A_n(\omega)$, and the modal excitation term, $F_n(\omega)$,

$$a_n(\omega)=A_n(\omega)F_n(\omega), \tag{9}$$

where

$$A_n(\omega)=\frac{j\omega}{m(\omega_n^2-\omega^2+j2\zeta_n\omega\omega_n)}, \tag{10}$$

and m is the total mass of the panel, ζ_n is the modal damping ratio, which was taken to be 0.01 (1%) for all modes in these simulations, and ω_n is the natural frequency of the nth mode which is given by

$$\omega_n=\sqrt{\frac{Eh^2}{12\rho(1-\nu^2)}}\left[\left(\frac{n_1\pi}{L_x}\right)^2+\left(\frac{n_2\pi}{L_y}\right)^2\right], \tag{11}$$

where ρ is the density of the material, E is its Young's modulus of elasticity, ν is the Poisson's ratio, and h is the panel's thickness.

The modal excitation term will have a component due to the incident plane wave, $F_{np}(\omega)$, and a component due to each of the M secondary actuators, $F_{nm}(\omega)$, so that

$$F_n(\omega)=F_{np}(\omega)+\sum_{m=1}^{M} F_{nm}(\omega). \tag{12}$$

For the plane acoustic wave excitation assumed here, $F_{np}(\omega)$ is given by Wang *et al.*,[12] who also analyze the modal excitation terms for the plate when driven by a point force or the line moments generated by a piezoelectric actuator, which were used here to calculate $F_{nm}(\omega)$, as summarized in Ref. 3. The size of each of the piezoelectric actuators is 25$\times$25 mm.

The total kinetic energy of the panel is defined to be

$$E(\omega)=\frac{m}{4S}\int_S |v(z,y,\omega)|^2 dx\,dy, \tag{13}$$

where S is the surface area of the panel. Using the orthonormal properties of the mode shapes the kinetic energy is also equal to

$$E(\omega)=\frac{1}{4m}\sum_{n=0}^{N}|a_n(\omega)|^2. \tag{14}$$

The sound power radiated by the panel is calculated using an elemental approach.[13] The velocities at the center of each of a dense grid of elements is calculated using Eq. (7) to form the vector $\mathbf{v}$, and it is assumed that the vector of pressures in front of each element on the panel, $\mathbf{p}$, is related to $\mathbf{v}$ by the acoustic impedance matrix $\mathbf{Z}$, so that

$$\mathbf{p}=\mathbf{Z}\mathbf{v}. \tag{15}$$

The radiated sound power can then be approximated by

$$W=\frac{\Delta S}{2}\mathrm{Re}[\mathbf{v}^H\mathbf{p}], \tag{16}$$

where ΔS is the area of each element, and this can also be written as

$$W = \frac{\Delta S}{2} \mathrm{Re}[\mathbf{v}^H \mathbf{Z} \mathbf{v}] = \mathbf{v}^H \mathbf{R} \mathbf{v}, \tag{17}$$

where $\mathbf{R} = (\Delta S/2)\mathrm{Re}[\mathbf{Z}]$, which has a particularly simple form for planar radiators.[13] 52×52 elements were used in these simulations so that their spacing is small compared with the acoustic wavelength at the highest frequency of interest.

Although the sound power radiated by the panel usefully quantifies the far-field pressure it generates, high levels of vibration in weakly radiated modes can give rise to significant pressure levels in the near field of the panel. It has been shown that the total kinetic energy of a panel provides a better measure of near-field pressure than radiated sound power,[11] and so if there is any possibility that listeners may be in close proximity to the panel, as well as being further away, then both of these criteria are important for active structural acoustic control.

For practical computations only a finite number of modes can be used in the expansion for the velocity, Eq. (7). The convergence of the modal series can be investigated by calculating the ratio of the velocity computed at a point on the panel with a modal summation using N modes, to that computed with a large number of modes, such as 500, with natural frequencies up to 38 kHz. For excitation at 300 Hz, for example, which is not a resonant frequency of the panel, the results show that for a point force actuator, the velocity at the measured point converges to within 1% of the result with 500 modes when about 100 modes are included in the modal summation, whereas 200 modes are required to reach this level of accuracy with the piezoelectric actuator.[14] This result can be misleading, however, for the active control simulations presented here since very high levels of attenuation are predicted at some frequencies and so the residual components of the vibration may be more sensitive to modal truncation. The results presented here were obtained by taking values of both n_1 and n_2 in Eq. (5) from 1 to 17, i.e., about 300 modes, with natural frequencies up to about 20 kHz. This was chosen since none of the results presented here was significantly altered if the upper limit of the modal summation was increased to n_1 and $n_2 = 25$, i.e., about 600 modes.

A large number of modes is required to accurately model the velocity with a collocated actuator because the velocity is influenced by the near field of the actuator, which is more intense for the piezoelectric actuator than it is for the point force. It should be noted that only the line moment excitation of the piezoelectric actuator[12] has been taken into account in the model, not the local stiffening effect.

A uniform array of 4×4 actuators and sensors was modeled on the panel and control systems were investigated for which each pair of the 16 individual actuators and sensors were connected in 16 control loops, as illustrated in Fig. 3 for the case of piezoelectric actuators. The transducers were uniformly arranged on the panel so that their centers were $\frac{1}{8}$, $\frac{3}{8}$, $\frac{5}{8}$, and $\frac{7}{8}$ of the plate length and breadth away from the edges.

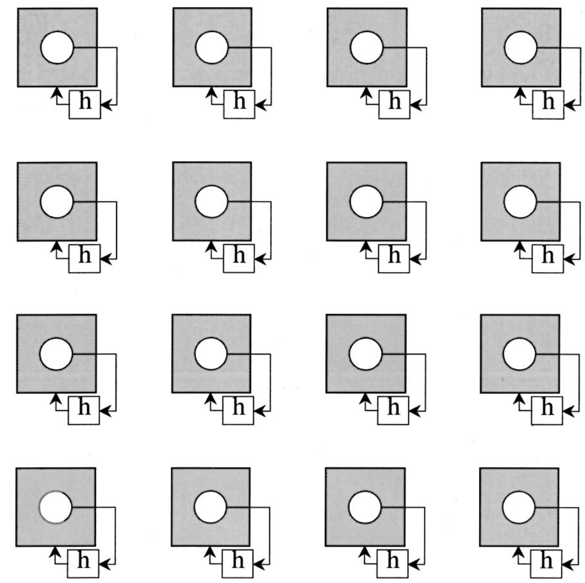

FIG. 3. Arrangement of 16 piezoelectric actuators, as shown by squares, driven locally by the output of 16 velocity sensors, as shown by circles, via individual control loops with a gain of h.

IV. RESULTS WITH POINT FORCE ACTUATORS

Figure 4 shows the total kinetic energy of the panel excited by the plane wave before control and when subject to control with 16 individual single channel control system with various feedback gains, h.

The modal response of the panel is clearly seen in the plot of the kinetic energy against frequency before control, with the resonance associated with the first, (1,1), mode occurring at about 72 Hz. As the gains of the feedback loops are increased, the resonances in the response become more heavily damped, as one would expect with velocity feedback control. If the gains of the feedback loops are increased beyond a certain value, however, the closed loop response displays new peaks, such as that at about 600 Hz, for example, which become more pronounced as the feedback gain is in-

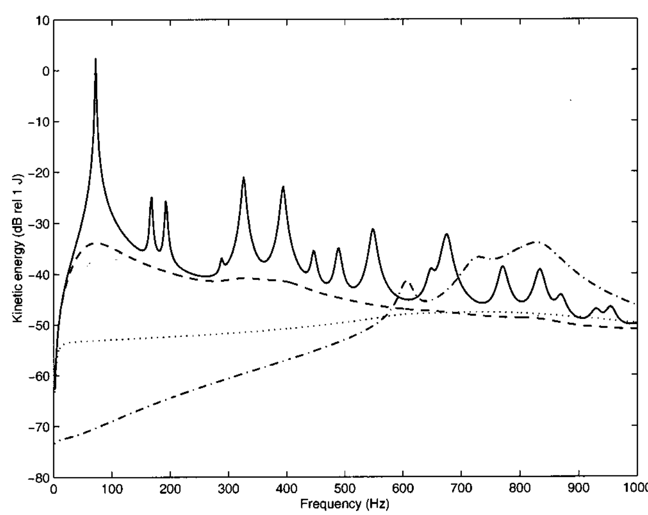

FIG. 4. Kinetic energy of the plane wave-excited panel with no control (solid line) and with a 16 channel decentralized feedback controller using force actuators and having feedback gains of 10 (dashed line), 100 (dotted line), and 1000 (dot-dashed line).

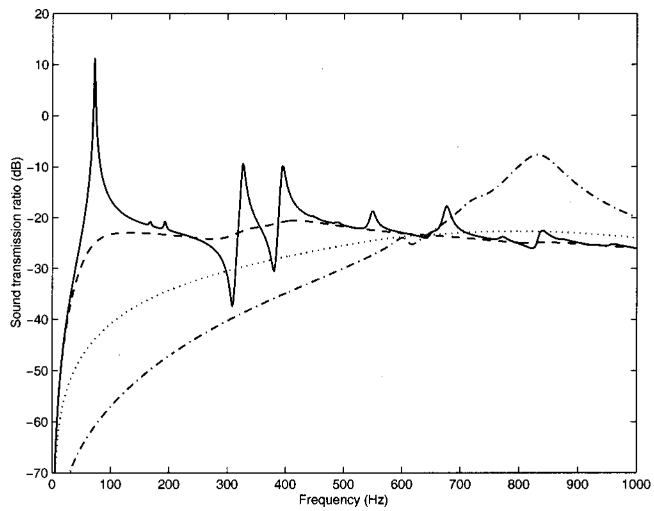

FIG. 5. Sound transmission ratio of the plane wave-excited panel with no control (solid line) and with a 16 channel decentralized feedback controller using force actuators and having feedback gains of 10 (dashed line), 100 (dotted line), and 1000 (dot-dashed line).

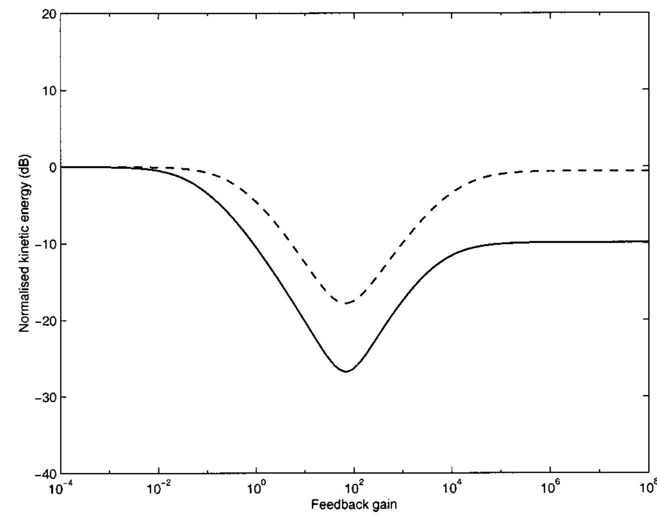

FIG. 6. Normalized kinetic energy level of the panel, integrated from 0 Hz to 1 kHz, plotted against the gain in the decentralized feedback controller, h, for the force actuators (solid line) and the piezoelectric actuators (dashed line).

creased further. These extra peaks are due to the resonances of the controlled dynamic system, which is effectively pinned at the sensor positions with high feedback gain. If feedback controllers having very high gain were used, the velocities at each sensor could be driven to zero and the physical result would be equivalent to that of perfect control of the sensor outputs with a feedforward control system, which could have been implemented if a suitable reference signal were available. With force actuators and velocity sensors the feedback gains have the same units as a mechanical impedance ($\mathrm{N\,s\,m^{-1}}$).

Figure 5 shows the ratio of the sound power radiated on one side of the panel to the incident sound power due to the plane wave excitation on the other side, which is termed the sound transmission ratio, T. Before control only the modes whose modal integers are both odd radiate sound significantly at low frequencies and also antiresonances appear, due to destructive interference between the sound pressures radiated by adjacent odd–odd modes. As the feedback gains are increased, similar trends are observed in the reduction of the sound transmission ratio as in the reduction of the panel's kinetic energy, except that the new resonance at about 830 Hz has the greatest prominence, since its velocity distribution has the greatest net volume velocity.

The panel's kinetic energy, integrated across the bandwidth shown in Fig. 4 (up to 1 kHz), is plotted against feedback gain as the solid line in Fig. 6, and a clear minimum is observed, for a gain of about 100. It should be emphasized that the individual outputs of the velocity sensors monotonically decrease as the feedback gains are increased, and that this is the only information available to the control system about the panel's vibration. These velocities give rise to a poor estimate of the panel's response when the feedback gains are high enough for the new resonances to become significant. Figure 7 shows the result for the sound transmission ratio integrated across this bandwidth, which corresponds to the total radiated sound power if the plate is subject to broadband excitation by a plane wave up to a frequency of 1 kHz, and which also has a minimum value for a feedback gain of about 100. Note also that at high feedback gains the overall sound power radiated after control is some 8 dB higher than it was with no control, because of the effect of the new resonance at about 830 Hz. The optimum feedback gain to minimize both kinetic energy or sound transmission is thus about 100 $\mathrm{N\,s\,m^{-1}}$, and the control system essentially synthesises an array of 16 mechanical dampers with this damping coefficient. It is interesting to note that the input or point impedance of an infinite 1 mm aluminum panel is real, frequency independent, and has a value of about 34 $\mathrm{N\,s\,m^{-1}}$.

V. RESULTS WITH PIEZOELECTRIC ACTUATORS

Although we can theoretically guarantee the unconditional stability of the 16 channel decentralized feedback control system in the case of point force actuators and velocity

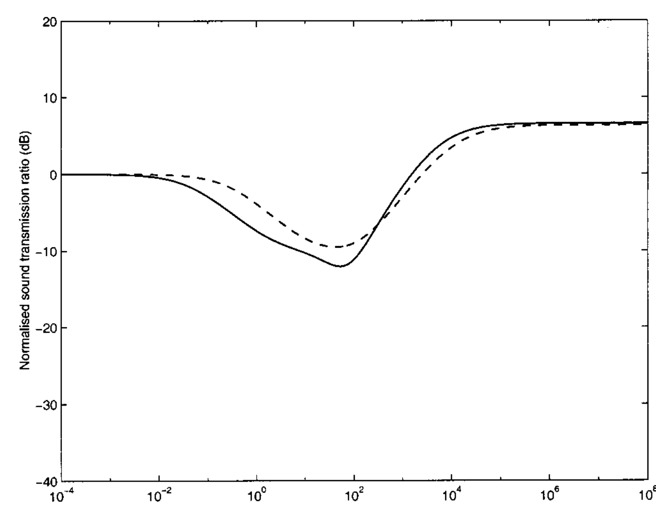

FIG. 7. Normalized sound transmission ratio level, integrated from 0 Hz to 1 kHz, plotted against the gain in the decentralized feedback controller, h, for the force actuators (solid line) and the piezoelectric actuators (dashed line).

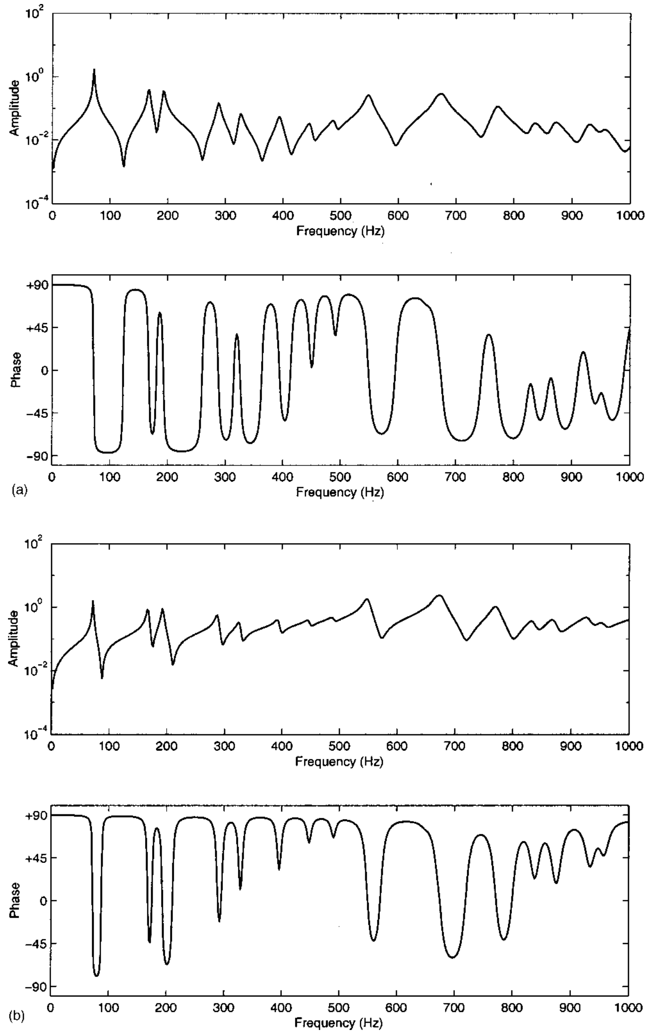

FIG. 8. Frequency responses between an individual force actuator and velocity sensor (a) and piezoelectric actuator and velocity sensor (b).

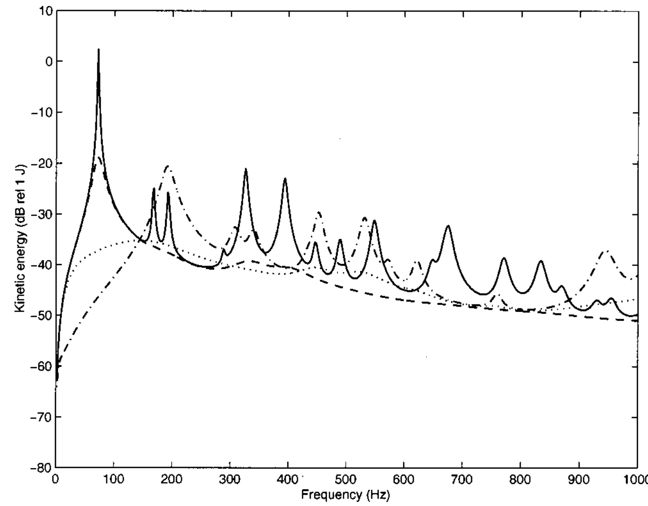

FIG. 9. Kinetic energy of the plane wave-excited panel with no control (solid line) and with a 16 channel decentralized feedback controller using piezoelectric actuators and with feedback gains of 1 (dashed line), 10 (dotted line), and 100 (dot-dashed line).

sensors, since this system is passive, the stability range of this controller must be determined when piezoelectric actuators are used before the control performance can be calculated. The stability of the decentralized feedback control system using piezoelectric actuators has been determined by examining the loci of the 16 eigenvalues of $\mathbf{G}(j\omega)$. This investigation was carried out on the simulated frequency responses up to 10 kHz and showed that within the accuracy of the simulation, all eigenvalues had positive real parts and so, in principle, the control system is again unconditionally stable. A practical investigation of this result is currently under way[15] in order to establish the upper limit imposed on the gain due to unmodeled effects in the interaction between the actuator, sensor, and structure. Rather than present the rather complicated eigenvalue loci referred to previously, the frequency response of a single diagonal element of the plant response up to 1 kHz is illustrated in Fig. 8 for both a force actuator (a) and a piezoelectric actuator (b), to illustrate the features of one of the individual control loops. The phase of both responses is confined to between $-90°$ and $+90°$, so that the real part of both responses has a positive real part. The absolute magnitude of the response between the piezoelectric actuator and the velocity sensor will depend on the piezoelectric constants of the actuator, but the response in Fig. 8(b) has been normalized to have a similar value of that with the force actuator at the first resonance peak, at about 72 Hz.

Note that although the high frequency response using the piezo actuator is then about 20 dB greater than that using the force actuator, the shapes of the two frequency responses are surprisingly similar. When the piezoceramic actuator is small compared with the flexural wavelength in the panel, then at a given frequency and with these boundary conditions the moment excitation at the edges of the actuator thus has a similar effect to that of a point force. The theoretically dual sensor for a piezoceramic actuator would be a strain-measuring device of the same size and shape as the actuator. In order to collocate such a sensor it would typically be positioned on the other side of the panel, immediately opposite the actuator.[16] Unfortunately, such an actuator-sensor pair is then coupled by the in-plane motion of the panel as well as the flexural motion we are hoping to control. The in-plane motion can become dominant at high frequencies and destroy the passive property of the frequency response between such an actuator and sensor.[17,18] Another advantage of using the velocity derived from inertial accelerometers, rather than using strain sensors, is that the former is sensitive to any rigid body motion of the panel, whereas the latter is not.

The kinetic energy of the panel when using 16 individual single channel control systems with 16 piezoelectric actuators having velocity sensors at their centers are shown for various feedback gains in Fig. 9. Compared with the results using point force actuators, Fig. 4, the levels of reduction are somewhat smaller, although still very worthwhile below about 100 Hz. The frequencies at which extra resonances are induced at higher feedback gains are also significantly lower with the piezoelectric actuators than with the point forces, and the peak value of the first extra resonance, at about 200 Hz, is higher.

Figure 10 shows the sound transmission ratio when the

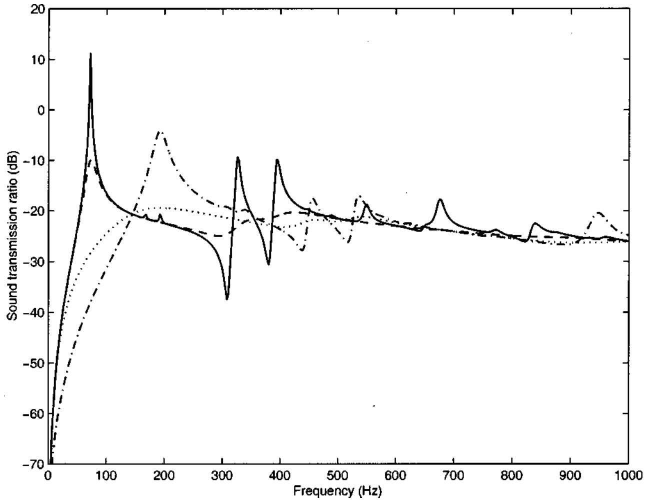

FIG. 10. Sound transmission ratio of the plane wave-excited panel with no control (solid line) and with a 16 channel decentralized feedback controller using piezoelectric actuators and with feedback gains of 1 (dashed line), 10 (dotted line), and 100 (dot-dashed line).

velocities are controlled by the piezoelectric actuators using 16 individual control systems with various feedback gains. Once again the performance is good for moderate feedback gains below about 100 Hz, but not as good as with force actuators, and the additional resonance at about 200 Hz degrades the performance for higher feedback gains.

The variation of the kinetic energy, integrated up to 1 kHz, with the feedback gain for piezoelectric actuators has been plotted in Fig. 6, together with this variation for force actuators. A clear minimum in the integrated kinetic energy is again seen for one value of feedback gain. This value of feedback gain is about the same as it was when using the force actuators, although it should be remembered that the piezoelectric responses have been scaled to be similar to force responses at the first resonance, as described previously. Of more importance is that at high feedback gains the integrated kinetic energy is not decreased when using piezoelectric actuators, due to the extra resonance at 200 Hz, whereas with force actuators reductions of about 10 dB in integrated kinetic energy are achieved even at high gains, because the extra resonances occur at higher frequencies and are not so prominent.

A similar comparison for the integrated sound transmission ratio is shown in Fig. 7, which again shows a clear minimum for a certain feedback gain, although the amplification for high feedback gains is now about the same when using piezoelectric actuators and when using force actuators.

VI. CONCLUSIONS

This paper reports the results of an initial simulation study of active vibroacoustic control using an array of collocated actuators and sensors and local feedback. It is shown that if perfect point force actuators and velocity sensors are used, such a feedback system is unconditionally stable.

The physical consequences of this control strategy are then investigated in a simulation of a panel excited by a plane acoustic wave and having a 4×4 array of force actuators and collocated velocity sensors. It is shown that both the kinetic energy of the panel and its transmitted sound power can be significantly reduced in the bandwidth up to 1 kHz provided an appropriate feedback gain is chosen. If the feedback gain is too large, the control systems will tend to pin the panel at the sensor locations, generating new resonance frequencies which can increase the response of the panel at higher frequencies.

Although an array of point force actuators has attractive theoretical properties, it cannot be implemented in practice without reacting the forces off a separate structure. Piezoelectric strain actuators, on the other hand, which generate line moments at their edges, can easily be integrated into a structure. Such an actuator cannot, strictly speaking, be collocated with a velocity sensor, and the stability of a control system using piezoelectric actuators and velocity sensors cannot be guaranteed in the same way as with force actuators and velocity sensors. The computer simulation has been used to calculate the responses from each of 16 small piezoelectric actuators on the panel to each of 16 velocity sensors at the center of these actuators, which form the matrix of plant responses for the feedback loop in this case. These responses have then been used to calculate the range of gains for which the 16 single channel feedback controllers will be stable, which was found to be unlimited in these idealized simulations.

Having established the stability of this control strategy with piezoelectric actuators, the performance has been calculated for various feedback gains. Once again significant reduction can be achieved with appropriate feedback gains in both the vibration of the panel, as quantified by the kinetic energy, and in its sound radiation, as quantified by the sound transmission ratio. With higher values of feedback gain not only is there a danger of instability due to unmodeled dynamics, but the feedback control systems again tend to pin the panel causing additional resonances, which now occur at much lower frequencies than was the case with point force actuators.

The overall performance of the various control strategies can be estimated by integrating the kinetic energy and the sound transmission ratio over the bandwidth considered, up to 1 kHz. Using this criterion, the maximum reduction in vibration which can be obtained with point force actuators is about 28 dB, whereas the maximum attenuation with piezoelectric actuators is about 18 dB. The maximum attenuation in the sound power transmission integrated over this bandwidth is about 12 dB with point force actuators and about 9 dB with piezoelectric actuators. It is emphasized that the reductions in the panel's kinetic energy may give a better indication of the change in sound pressure close to the panel than the sound power radiated into the far field.

This initial investigation thus suggests that useful reductions can be obtained in both the near-field and the far-field sound radiated from a panel using this decentralized feedback control approach, which is able to control broadband random primary fields as well as impulsive or tonal disturbances. Since each pair of actuators and sensors is only connected by local, constant-gain, feedback controllers, no coupling is necessary between each actuator and sensor pair, which could potentially be manufactured as identical modular units with simple integrated electronics.

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