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Active Structural Acoustic Control of Sound Radiation from Flat Plates

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ABSTRACT

The control of radiated sound is important for many engineering structures. This paper investigates the active control of sound radiation from flat plates through modelling. Two control systems to attenuate sound radiation from the plates are considered. Firstly, a feedforward control system is studied which can be applied for the case of tonal excitation where a reference signal is available. This control system may be realized by using a feedforward controller with appropriate transducers (actuators and sensors). The control actuators considered provide either a central point force or four corner point forces, and could be piezoelectric or inertial actuators. The error sensor is either a volume velocity sensor using PVDF film bonded on the back panel, or the total sound power sensor. Secondly, feedback control systems are investigated which can be used in the case of random excitation where a reference signal is not available. This analysis is focused on systems using simple single-channel feedback controllers, so that self-contained, compact and light sensor-controller-actuator devices can be built. Up to sixteen point force actuators with collocated point velocity sensors are controlled in a decentralized fashion by a single-channel fixed gain feedback control system for each unit. The control effectiveness, stability and robustness of each control configuration are discussed.

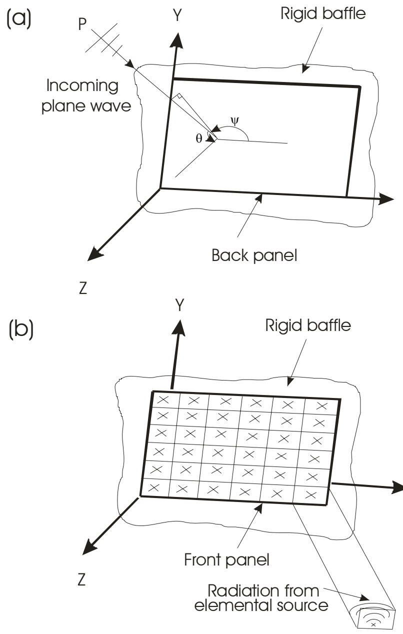
The study has indicated that both control approaches have shown significant reductions in sound radiation. However, each feedback system is unconditionally stable, and the magnitude of the sum of the control forces required by the systems is less than the primary force. Thus, decentralized feedback control could be a feasible way for implementation in practice.

INTRODUCTION

Active control of sound radiation/transmission from flat plates is a topic which has received considerable attention in the past and will continue to do so in the future, as flat plates are the basic building blocks of many engineering structures, including the hull of a marine vessel. The objective of this introduction is to highlight how and why researchers have developed two different control approaches for controlling sound radiation from plates. In the first approach, the aim is to rearrange the vibration field of the plate in order to reduce the sound radiation at specific narrow frequency bands using feedforward control systems. The aim of the second approach is to damp the vibration of the plates at resonant frequencies using feedback control systems so that the sound radiation due to random disturbance can be controlled at low frequencies where the sound radiation is dominated by the resonances of the plates.

Early work on feedforward control of vibration transmission along a semi-infinite plate, excited by primary forces located close to the free end of the plate, was investigated by Pan and Hansen [1-3], who used a wave control method, where the structural vibration is described in terms of waves travelling in various directions, and vibration control requires control of the propagation of these waves by controlling the wave amplitude. They applied an array of independently driven control forces placed in a row across the plate to minimize acceleration averaged over the width of the plate further down from the control forces, which results in reduced vibration transmission downstream.

Johnson and Elliott [4-7] applied the feedforward approach for structural acoustic control of sound radiation from a smart panel. The smart panel is light, flexible and small so that distributed actuators and sensors can be embedded on it. They formulated sound radiation from a panel by using an elemental radiator method, and approximated the vibration and acoustic radiation from a surface by a number of elemental sources which are all oscillating harmonically (Figure 1). They used a distributed actuator (uniform-force actuator) and a distributed sensor (volume velocity) as a matched actuator/sensor pair to reduce control spillover (unwanted modes excited by the control actuator). Then, they demonstrated the cancellation of volume velocity is an effective way to reduce sound radiation at low frequencies.

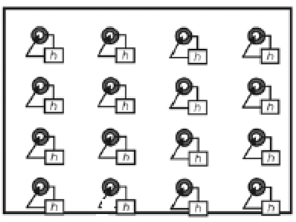


Source: (Pan *et al.* 1998)

Figure 1. (a) Coordinates of back panel which is excited by an incident acoustic plane wave; (b) coordinates of front panel and sound radiation from the panel using a number of elemental radiators.

Pan *et al.* [8] extended the work given by Johnson and Elliott [4-7] to double panel partitions, which are often used in noise control engineering when high sound transmission loss has to be achieved with lightweight structures. An example is an aircraft fuselage shell. Pan *et al.* reported that sound power through the double panel partitions can also be reduced significantly by the cancellation of volume velocity using distributed actuators and sensors.

Recently, control systems have increasingly used a feedback rather than feedforward arrangement, because of its ability to deal with broadband random vibration without an external reference signal. Gardonio and Elliott [9] reviewed the decentralized feedback control approach for a smart panel with different types of transducers. The transducers were either distributed type, such as a uniform-force actuator and volume velocity sensor, or sixteen collocated point forces and velocity sensors which were evenly distributed on the panel (Figure 2). They showed that the use of the sixteen control units provided a reduction of 12 dB in averaged sound radiation compared with a reduction of 16 dB by using the distributed transducers at low frequencies. Also, they indicated the even distribution of the sixteen control units produced slightly better results than the control units moved slightly from their original positions.



Source: (Gardonio and Elliott 2004)

Figure 2. A panel with sixteen decentralized direct velocity feedback control units.

Gardonio *et al*. [10] carried out a preliminary theoretical study for the decentralized feedback control system. Then, they conducted experimental verification [11-12]. They measured, in an anechoic chamber, sound radiation from the smart panel which was mounted on a Perspex box with very thick and rigid walls. The smart panel was excited either by the acoustic field produced by a loud speaker placed in the Perspex box or directly by a point force generated with a shaker. Each control unit consisted of a piezoceramic patch and an accelerometer. They showed that the even distribution of the sixteen control units could produce good reductions of radiated sound power and averaged vibratory field over the panel surface.

The work described here is an extension of the work done on control of sound radiation from a smart panel [4-7, 9-12]. The purpose of the current work is to apply the feedforward and feedback approaches for any size of flat plate, such as hull-like flat plates. The flat plates can be stiffer than the flexible smart panel so that distributed transducers (e.g. PVDF film with quadratically shaped electrodes [9]) are no longer suitable. The size of the flat plates can be large so that it is expected there will be many more structural modes appearing in a certain low frequency range (e.g. below 1000Hz) compared with the smaller smart panel. Thus, the even distribution of the sixteen control units on the flat plate would be more likely to fall on nodal lines of the structural modes to be controlled, which would result in the failure of the decentralized feedback control.

The four key issues in the current paper are:

(1) study of the practical implementation of actuators and sensors;

(2) discussion of optimal locations and amplitudes of the actuators for maximum reduction of sound radiation;

(3) investigation of minimum number of transducers required for controlling sound radiation; and

(4) analysis of the control effectiveness for each control configuration.

THEORY

A model of a simply supported, baffled plate is used for the reason of simplicity. However, the radiation characteristics of all plate structures conform to a similar pattern [13].

Structural and acoustic response

For a harmonically excited simply supported plate, the displacement at a location (x, y) of the plate given by Fuller *et a*l. [14] is

$$w(x, y) = \sum_{m=1}^{\infty}\sum_{n=1}^{\infty} K_{mn} \sin\left(\frac{m\pi x}{l_x}\right)\sin\left(\frac{n\pi y}{l_y}\right), \qquad (1)$$

where K_{mn} is the amplitude of the mnth mode and l_x and l_y are the dimensions of the plate in the x and y directions. The modal amplitude K_{mn} can be written as

$$K_{mn} = A_{mn} F_{mn}. \qquad (2)$$

A_{mn} is the complex resonance term which is shown as

$$A_{mn} = \frac{1}{\rho h(\omega_{mn}^2 - \omega^2 + j\omega D_{mn})}, \qquad (3)$$

where ω_{mn} is the natural frequency of the mnth mode, ρ is the density of the plate material, h is the thickness of the plate and D_{mn} is the damping of the *mn*th mode and is given by $D_{mn} = 2\xi\omega_{mn}$ (where ξ is the damping ratio).

F_{mn} is the modal force. If a point force at a position (x_0, y_0) is applied on the plate, the modal force F_{mn} for the simply supported plate is

$$F_{mn} = \frac{4F_c}{l_x l_y}\sin\left(\frac{m\pi x_0}{l_x}\right)\sin\left(\frac{n\pi y_0}{l_y}\right), \qquad (4)$$

where F_c is the complex force amplitude.

If an incident wave is applied on the panel, the modal force is given by

$$F_{mn} = 8P_i I_m I_n, \qquad (5)$$

where P_i is the amplitude of the incident wave and I_m, I_n are due to geometric coupling between the plate wave and the *m* and *n* modes. The expressions for I_m and I_n are shown in reference [14].

Acoustic power radiation from *r* elements (Figure 1) for a single frequency is purely a function of velocity of the plate [4], which is given by

$$W_r = V^H RV, \qquad (6)$$

where *R* is a $(r \times r)$ radiation matrix and *V* is a $(r \times 1)$ vector of velocities at the centres of the radiation elements.

The sound power incident on the plate can be written as [7]

$$W_i = |P_i^2| \, l_x l_y \cos\theta / 2\rho c, \qquad (7)$$

where *c* is the speed of sound in air and θ is the angle of the incident acoustic plane wave (Figure 1).

Feedforward active control

If the plate is excited by an incident acoustic plane wave and controlled by *s* independent point forces, the vector of the total velocities at the centres of the radiation elements for a single frequency may be written as [1]:

$$V = P_i V_p + F_{c1}V_{c1} + \ldots F_{cs}V_{cs}, \qquad (8)$$

where V_p is a $(r \times 1)$ velocity vector due to a unit primary excitation, F_{ci} (*i=1,…s*) is the *i*th control force complex amplitude and V_{ci} (*i=1,…s*) is a $(r \times 1)$ velocity vector due to the *i*th unit control force excitation. The optimal control forces for minimizing volume velocity may be found by taking the mean square of the velocity vector (Equation (8)) multiplied by the element area, and evaluating the derivative with respect to the control forces and setting the results to zero. The results for an optimal set of control forces are as follows:

$$\begin{bmatrix} F_{c1} \\ F_{c2} \\ . \\ . \\ . \\ F_{cs} \end{bmatrix} = \begin{bmatrix} V_{c1}{}^H V_{c1} & V_{c1}{}^H V_{c2} & \ldots V_{c1}{}^H V_{cs} \\ V_{c2}{}^H V_{c1} & \ldots & \ldots \; \ldots \\ \ldots & \ldots & \ldots \; \ldots \\ V_{cs}{}^H V_{c1} & & V_{cs}{}^H V_{cs} \end{bmatrix}^{-1} \begin{bmatrix} V_{c1}{}^H V_p \\ V_{c2}{}^H V_p \\ . \\ . \\ . \\ V_{cs}{}^H V_p \end{bmatrix} P_i. \quad (9)$$

If the control system is set to minimize the total sound power, then Equation (8) is substituted into Equation (6) and the derivative of Equation (6) with respect to the control forces is set to zero. The results are

$$\begin{bmatrix} F_{c1} \\ F_{c2} \\ . \\ . \\ . \\ F_{cs} \end{bmatrix} = \begin{bmatrix} V_{c1}{}^H RV_{c1} & V_{c1}{}^H RV_{c2} & \ldots V_{c1}{}^H RV_{cs} \\ V_{c2}{}^H RV_{c1} & \ldots & \ldots \; \ldots \\ \ldots & \ldots & \ldots \; \ldots \\ V_{cs}{}^H RV_{c1} & & V_{cs}{}^H RV_{cs} \end{bmatrix}^{-1} \begin{bmatrix} V_{c1}{}^H RV_p \\ V_{c2}{}^H RV_p \\ . \\ . \\ . \\ V_{cs}{}^H RV_p \end{bmatrix} P_i.$$

(10)

Direct velocity feedback control

Considering a feedback control system, the flexural vibration of the plate is also given by the superposition of the acoustic primary excitation and the structural secondary excitation(s) generated by *s* control actuator(s). The velocities at the centres of the *r* elements for a single frequency, *V*, can be derived with following matrix relation

$$V = V_p P_i + V_c u_c, \qquad (11)$$

where V_c is a $(r \times s)$ matrix of velocities at the *r* radiation elements corresponding to the *s* control forces which all have a unit amplitude, and u_c is a $(s \times 1)$ vector of complex input velocity signals to the *s* control actuators.

The $(s \times 1)$ vector of the total velocities, V_{es}, at the *s* error sensors is

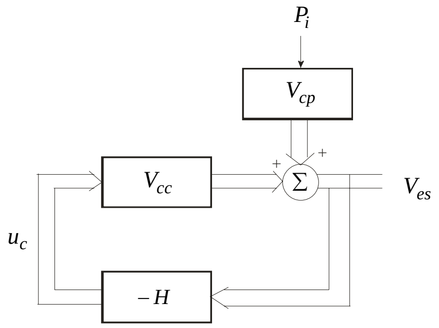
$$V_{es} = V_{cp} P_i + V_{cc} u_c, \qquad (12)$$

where V_{cp} is a $(s \times 1)$ vector of the velocities at the *s* error sensors due to a unit primary excitation, and V_{cc} is a $(s \times s)$ matrix of the velocities at the *s* error sensors corresponding to the *s* control forces which all have unit amplitude. The general block diagram of a multi-channel velocity feedback control system is shown in Figure 3. The output signal(s), V_{es}, can be expressed related to an incident plane wave, P_i, by expression [13]

$$V_{es} = [I + V_{cc}H]^{-1} V_{cp} P_i. \qquad (13)$$

Similarly, the vector of control inputs to the *s* control actuators, u_c, is given by

$$u_c = -H[I + V_{cc}H]^{-1} V_{cp} P_i. \qquad (14)$$



Source: (Gardonio and Elliott 2004)

Figure 3. A multi-channel feedback control system for a plant response, V_{cc}, and a controller, *H*.

If only one control unit is applied on the plate, the vectors and matrices in Equations (12) to (14) reduce to scalars, but in general V_{cc} is a fully populated matrix of input and transfer response between the actuators and sensors on the plate and *H* is normally a diagonal matrix for local control, which we assume to have constant gains on each channel so that $H = hI$, where *h* is the feedback gain and *I* is a $(s \times s)$ unit matrix. If collocated and compatible transducers are used, the control system is unconditionally stable [9]. By substituting Equation (14) into Equation (11), the vector of the total velocities at the radiation elements due to the primary incident acoustic plane wave excitation and direct velocity feedback control becomes

$$V = V_p P_i - V_c H[I + V_{cc} H]^{-1} V_{cp} P_i . \qquad (15)$$

The corresponding total sound power radiation can be obtained by substituting Equation (15) into Equation (6).

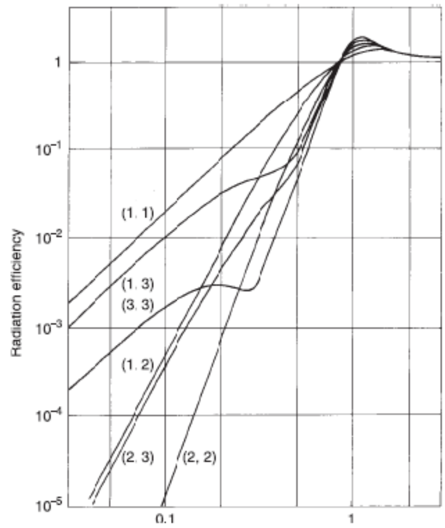
NUMERICAL RESULTS

The numerical results presented here have been calculated for flat plates that are 3 mm thick steel with dimensions 1440 mm by 710 mm (area 1 m^2) and damping ratio of 0.002. The primary source is a plane wave incident at $\theta = 45^0$ and $\psi = 45^0$ (see Figure 1), and the amplitude of the incident wave is one (e.g. $P_i = 1$ in Equation (5)). An alternate primary source of a point force will be discussed in the end of the results section. The first ten modal terms in both *x* and *y* directions are used in the modal summation of Equation (1). The structural natural frequencies of the plate are shown in Table 1.

Table 1. Structural resonances of the plate

Natural freq. (Hz)			*m*		
n	1	2	3	4	5
1	18	63	135	238	370
2	29	73	146	249	381
3	47	91	164	267	399
4	72	116	189	292	424
5	103	148	221	324	459

Figure 4 illustrates some examples of radiation efficiency curves given by Wallace [15]. This shows the significant different form of the radiation efficiency curves for the different modes at low frequencies can take. Figure 4 indicates that below critical frequency, modes with both mode numbers odd have significantly higher radiation efficiencies, where the (1,1) mode has the highest radiation efficiency.



Non-dimensional frequency
Source: (Wallace, 1972)

Figure 4. Radiation efficiency curves for a number of modes of a square plate.

In order to understand the vibration distribution at selected modes, Figure 5 shows modal shapes and regions of uncancelled velocity distribution at those modes. At the (1,1) mode, the highest velocity appears at the centre of the plate. At the (1,3), (3,1) and (3,3) modes, interior regions of positive and negative velocity cancel one another, such that uncancelled cells only appear at the edges and corners of the plate which suggest there are four corner areas to share the common regions at those three modes.

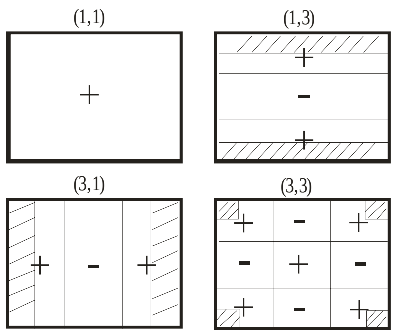


Figure 5. Modal shapes and regions of uncancelled velocity at selected modes.

FEEDFORWARD CONTROL

In the feedforward control system, point control force(s) representing piezoelectric or inertial actuators will be considered to be located at either the centre or near to the four corners of the plate. Two cost functions to be minimized are investigated for each control force configuration. The cost function is either volume velocity or total sound power from the plate. The results presented in this section represent the control possible using a feedforward control system with a perfectly correlated reference signal.

One central control force

In this case, one control force is applied at the centre of the plate. Figure 6 shows the sound radiation from the plate before control, after minimization of radiated power and after cancelling the volume velocity. It can be seen that the strategy of velocity cancellation achieves good reductions in sound radiation at low frequencies below 200 Hz and these reductions are very similar to those achieved using the optimal control strategy of sound power minimization. At 200 Hz onwards the velocity cancellation with the point control force is poorer due to excitation of high-order modes on the plate (control spillover) [7]. However, if sound power cancellation is used, the spillover problem disappears.

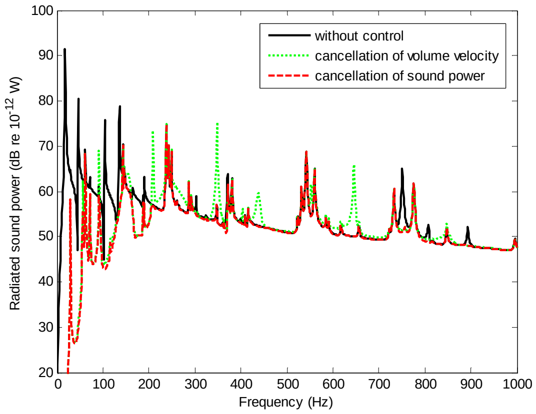


Figure 6. Feed-forward control of sound radiation from the plate excited by a plane acoustic wave with one central control force.

Four corner point control forces

Figure 7 repeats the same calculation as shown in Figure 6, but using four point forces located close to the four corners of the plate. It can be seen that with the four corner forces, significant reductions in radiated sound are achieved at all natural frequencies and there is no control spillover. In this case, the cancellation of sound power produces better reduction in sound radiation below 400 Hz. Both cancellation strategies produce similar results from 400 Hz onwards.

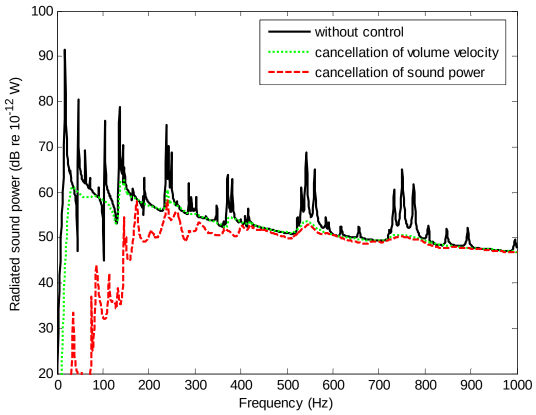


Figure 7. Feed-forward control of sound radiation from the plate excited by a plane acoustic wave with four control forces located close to the four corners of the plate.

DIRECT VELOCITY FEEDBACK CONTROL

This direct velocity feedback control system can be used in the case of random excitation where a reference signal is not available. Up to sixteen point force actuators with collocated point velocity sensors will be controlled in a decentralized fashion by a single-channel fixed gain feedback control system for each unit (refer to Figure 2). Three values of gains ($h = 10, 100, 1000$) are examined. Note that only a limited number of error sensors (up to sixteen) will be used instead of the volume velocity used in the feedforward system.

One control unit

Figure 8 shows controlled (with various values of gains in a feedback loop) and uncontrolled sound radiation when only one control unit is applied on the centre of the plate. Two frequency ranges 0-300 Hz (Figure 8(a)) and 0-1000 Hz (Figure 8(b)) are presented. Figure 8(a) shows even when a single control unit is used, the first four natural frequencies of the plate at the (1,1), (3,1), (5,1) and (1,3) modes are damped. Figure 8(b) shows that there is no reduction from 135 Hz ((3,1) mode) onwards.

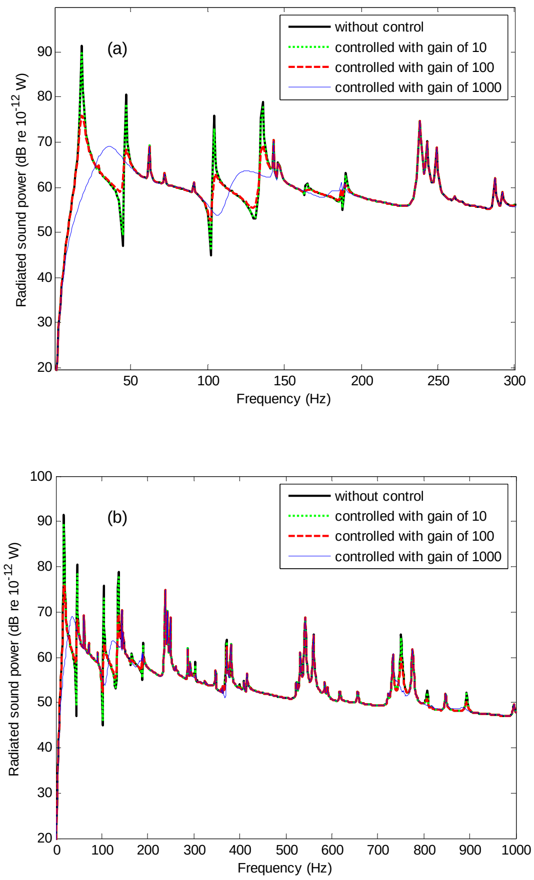


Figure 8. Feedback control of sound radiation from the plate excited by a plane acoustic wave with one control unit. In this and following figures, results are plotted in the frequency ranges 0-300 Hz (a) and 1-1000Hz (b).

The effect of feedback gain is analysed. Below 30 Hz, it can be seen the greater the feedback gain the greater the reduction in the sound radiation. At and above 30 Hz, the control system with a high gain of 1000 produces new modes (control spillover). Using the best gain examined of 100, an average of maximum reductions in sound radiation is achieved.

Four control units

Figure 9 shows the result of repeating the calculation displayed in Figure 8 but with the even distribution of four control units. In this case, the peaks at 18 Hz ((1,1) mode) and 103 Hz ((5,1) mode) are reduced. However, there is no reduction at peaks 47 Hz ((3,1) mode) and 135 Hz ((1,3) mode). This is because the even distribution of the four control forces are located at nodal lines of the (3,1) and (1,3) modes.

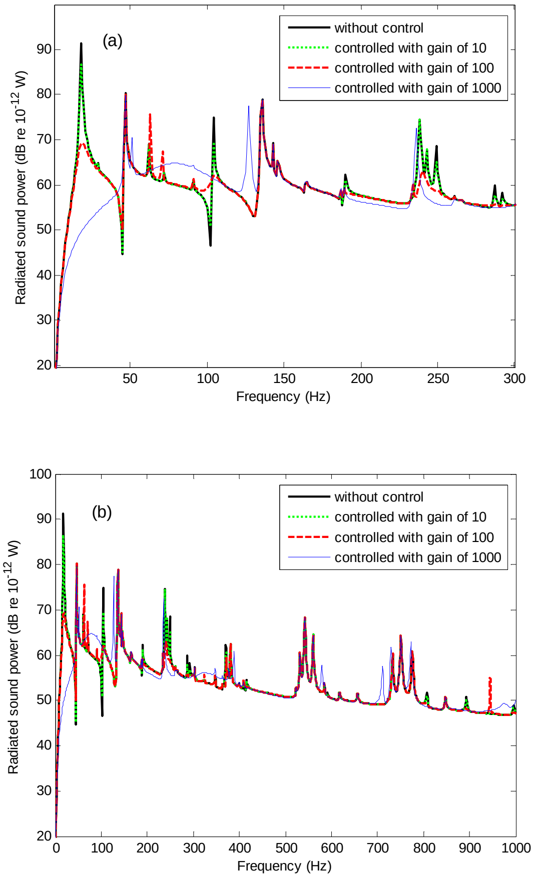


Figure 9. Feedback control of sound radiation from the plate excited by a plane acoustic wave with the even distribution of the four control units.

In order to move away from the nodal lines, the four control units are randomly distributed on the plate as shown in Table 2 for all the excitation frequencies and positions are shown in Figure 10. Figure 11 presents the results with a random distribution of the four control units. Comparing Figure 11 with Figure 9, it can be seen that significant improvement in reductions of sound radiation is achieved with the random distribution of the four control units. Using the best gain of 100, all the resonances in the frequency range 0-1000Hz are damped which results an overall reduction of sound radiation of about 8 dB.

Table 2. Control force locations for the random distribution of the four control units

Force number	x	y
1	0.752	0.371
2	0.894	0.441
3	1.144	0.564
4	1.377	0.679

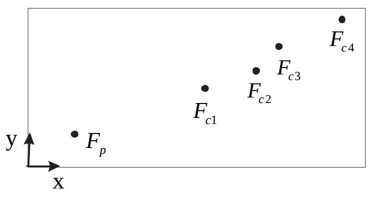


Figure 10. Position of primary force (when the primary force is applied), F_p , and four control units, F_{ci} (i=1,…4).

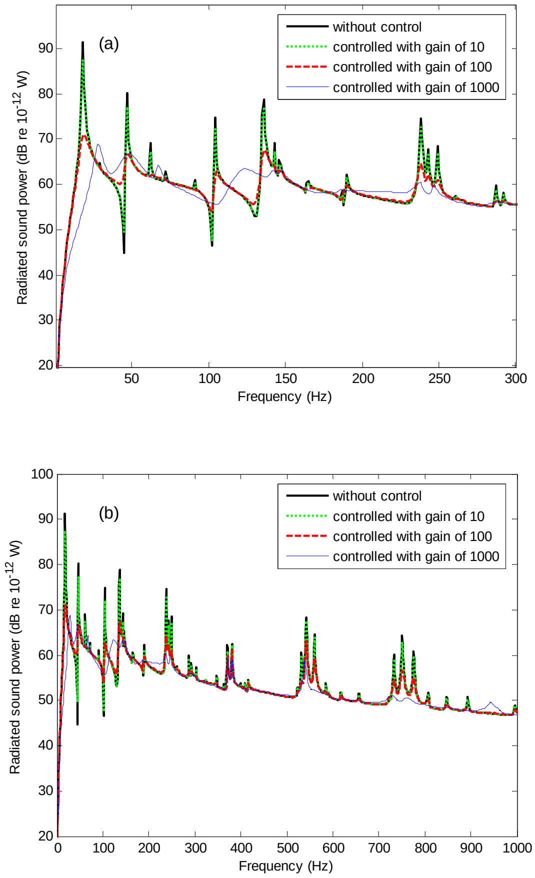


Figure 11. Feedback control of sound radiation from the plate excited by a plane acoustic wave with the random distribution of the four control units.

Sixteen control units

Figures 12 and 13 show respectively the sound radiation displayed in Figures 9 and 11 but with sixteen control units. Comparing Figures 12 and 13 with Figures 9 and 11 respectively, it was found that the control phenomena follow a similar trend. A better overall reduction of sound radiation of about 9.2 dB is achieved with the random distribution of the sixteen control units. This is because that increasing number of control units increases the number of local minima.

EFFECT OF PRIMARY FORCE EXCITATION

In this section, an example is given for the plate excited by a point force rather than an incident plane wave excitation as shown in the previous sections. To conserve space, only four control units with the random distribution are considered. The primary point force is located at x=0.22 m and y=0.19 m on the plate. The control force locations are shown in Table 2. The positions of the primary force and the four control forces are shown in Figure 10.

Figure 14 presents the results of repeating the calculations shown in Figure 11 but with the primary force excitation. The modal response of the plate is quite different to that shown in Figure 11 since the point force excites most, if not all, the modes of the plate. The sound radiation from the point force excitation is characterized by a larger number of resonances when compared to that from an incident acoustic plane wave excitation. Most of the resonances in the frequency range 0-1000Hz are damped when the best gain of 100 is used.

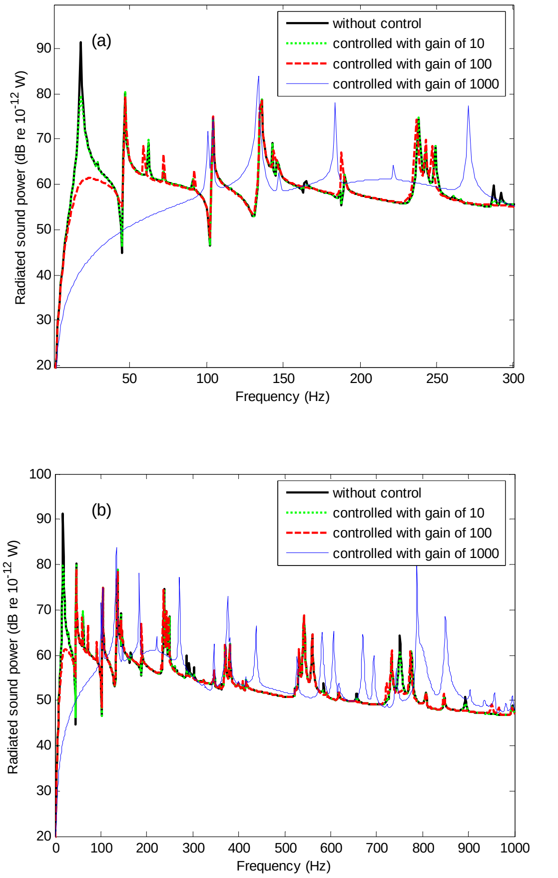


Figure 12. Feedback control of sound radiation from the plate excited by a plane acoustic wave with the even distribution of the sixteen control units.

The maximum achievable reductions of sound radiation at the first five resonances of the plate corresponding to the control force amplitudes are presented in Table 3. Table 3 shows that the required control forces vary at different resonances. At the first resonance (18Hz), the sum of the four control force amplitudes is about 0.38 times the primary force amplitude and slightly higher at the other resonances. Significant reductions in sound radiation up to 20 dB are achieved at those resonances. This appears to be a reasonable prospect of achieving a useful experimental result with this plate and source arrangement.

Table 3 Control force amplitudes and reductions of sound radiation from the plate excited by a point force and with the random distribution of the four control units

Natural frequency (Hz)	$\frac{\lvert F_{c1}\rvert}{\lvert F_p\rvert}$	$\frac{\lvert F_{c2}\rvert}{\lvert F_p\rvert}$	$\frac{\lvert F_{c3}\rvert}{\lvert F_p\rvert}$	$\frac{\lvert F_{c4}\rvert}{\lvert F_p\rvert}$	Red. of sound (dB)
18	0.15	0.06	0.00	0.17	20
29	0.47	0.43	0.02	0.18	10
47	0.19	0.27	0.02	0.48	17
63	0.29	0.31	0.02	0.08	12
73	0.56	0.52	0.03	0.09	13

* F_{ci} $((i=1,...4)$ is the *i*th control force and F_p is the primary force.

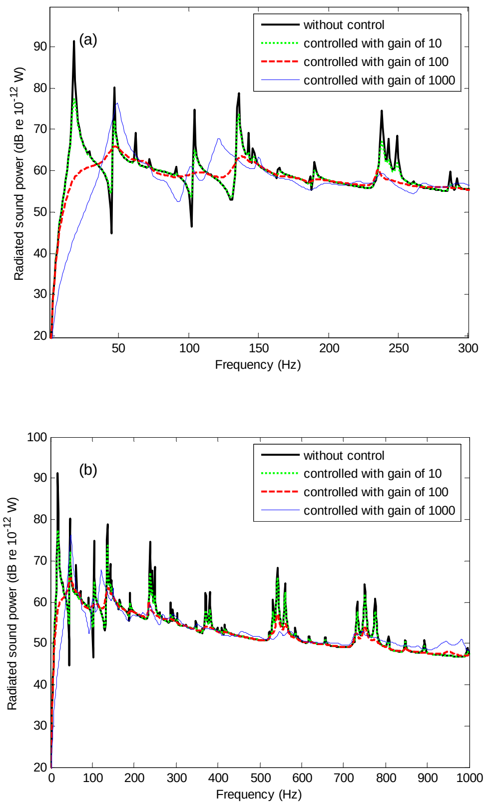


Figure 13. Feedback control of sound radiation from the plate excited by a plane acoustic wave with the random distribution of the sixteen control units.

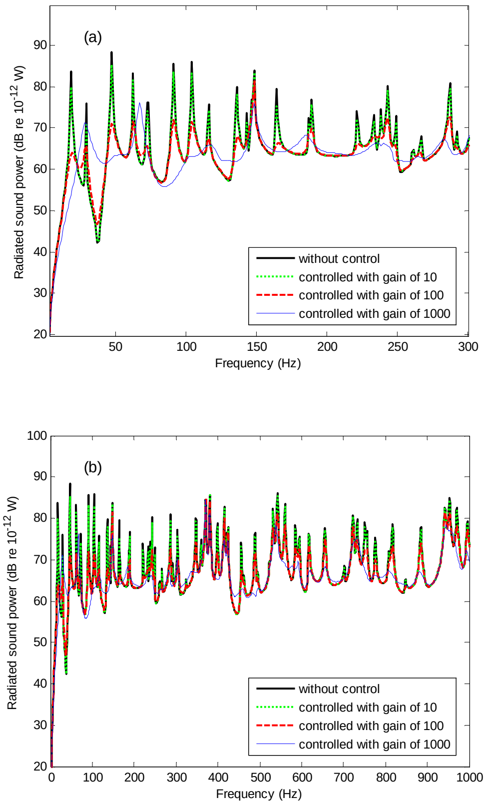


Figure 14. Feedback control of sound radiation from the plate excited by a point force with random distribution of the four control units.

DISCUSSIONS

The major issues in the feedback control system will be discussed in this section. The locations of the control units should be positioned away from the nodal lines of the modes to be controlled. For the current plate ($1\,m^2$), it was found that the random distribution of four or sixteen control units produces better reduction in sound radiation than the even distribution. When the size of a plate reduces to very small (e.g. a smart panel), the force locations may not be critical as the control units are not likely to be located on the nodal lines of the modes to be controlled (refer to the examples for the even distribution of the sixteen control units given by Gardonio and Elliott [9]).

For the current plate, four control units would be essential for controlling sound radiation from the plate, provided that the control units are randomly located. With the random distribution, increasing the number of control units increases the control performance but the improvement becomes less significant for sixteen or more control units.

It is noticed that as the feedback gain increases, the active damping effect to the structural modes increases and consequently the total sound radiation of the plate averaged over a certain frequency band decreases. However, it is also found that this behaviour is only valid up to the best feedback gain of 100 for the current plate, above which the damping effect drops and so sound radiation by the plate increases again and can even become larger than before control. This is due to the feedback controller acting as a pin jointed mounting location on the plate at the error sensor positions [10] for large control gains, so that the vibration of the plate becomes a lightly damped structure with extra pinning points. Therefore, a set of new lightly damped structural modes are created which could be excited at new resonance frequencies and radiate sound even more effectively than the original modes. Thus, care must be taken not to use an overly high gain.

CONCLUSIONS

Sound radiation from a flat plate can be significantly reduced by active control using point forces, such as piezoelectric or inertial actuators. Two different control approaches have been analysed for reducing sound radiation.

For the feedforward approach, it was found that one point force located on the centre of the plate to minimize volume velocity can reduce sound radiation below 135 Hz. Four point forces located near the four corners of the plate to minimize volume velocity provide significant reductions in sound radiations at resonance frequencies up to 1000Hz.

For the decentralized feedback approach, up to sixteen control units were examined. It was found that four control units would be essential for controlling sound radiation from the plate, provided that the four control units are away from the nodal lines of the modes to be controlled. In fact, a random distribution of the control units provides better reductions in sound radiation than that with the even distribution of the control units. Increasing the number of the control units increases the control performance but improvement is less significant for sixteen or more control units. As the feedback gain increases, the active damping effect to the structural modes increases and consequently the sound radiation decreases. However, the behaviour is only valid up to the best feedback gain.

Two primary excitations are examined. They are either an acoustic incident excitation or a point force excitation. It was found that the sound radiation from the point force excitation was characterized by a larger number of resonances compared with an incident acoustic plane wave excitation. For either excitation, the random distribution of four or more control units has damped the modes up to 1000 Hz which results in reductions in sound radiation of 10-20 dB at the modes. The results have demonstrated that the control performance is not sensitive to variation in control unit location and the magnitude of the sum of the control forces required by the feedback systems is less than the primary force at most natural frequencies. As the collocated and compatible transducers are used, each feedback control system is unconditionally stable. Thus, decentralized feedback control could be a feasible way for implementation in practice. This study will provide a guideline for further studies of active control measures for hull radiation.

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