Adaptive feedforward and feedback methods for active/passive sound radiation control using smart foam

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This work investigates and compares the potential of adaptive feedforward and feedback methods for a hybrid active/passive radiation control using smart foam. The radiating structure is a vibrating plate mounted in a rigid baffle in an anechoic chamber. The smart foam, designed to reduce sound by the action of the passive absorption of the foam (which is effective at higher frequencies) and the active input of an embedded PVDF element driven by an oscillating electrical input (which is effective at lower frequencies), is positioned on the plate. The first test consists of using a single-input single-output (SISO) adaptive feedforward LMS controller to minimize the error sensor signal provided by a microphone in the close proximity of the active element under narrow-band excitation and broadband random excitation. For feedforward control, two different reference signals are considered: the voltage sent to the piezoceramic actuator driving the plate (disturbance) and the signal from an accelerometer directly mounted on the plate (more realistic in practice). In the latter case, the effect of the smart foam on the reference signal (or acceleration level) can be taken into account (feedback removal). An adaptive feedback controller is also implemented to avoid the use of a reference signal. In this case, a reference signal is obtained from the error signal using the internal model approach. The results from these three different control methods are compared in terms of the sound attenuation achieved. For broadband excitation, a feedforward adaptive control with an external reference is shown to be more efficient for this arrangement than a feedback adaptive control. © *1998 Acoustical Society of America.* [S0001-4966(98)00207-0]

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INTRODUCTION

In the last few years, there has been an increased interest in the reduction of sound and/or vibrations by use of hybrid active–passive control techniques.[1] The passive device usually carries the primary noise attenuation function, while the active component is used to enhance the passive system performance or overcome the limitations of the passive system. The smart foam design integrates a lightweight distributed piezoelectric actuator (the active component) between individual layers of sound-absorbing foam (the passive component) such that the control can efficiently operate over a broad range of frequencies. Porous sound-absorbing materials are commonly implemented for a large range of applications (such as buildings, machinery enclosures and aircrafts) to reduce sound propagation. The use of a polyimide foam layer has been investigated for reducing sound transmission.[2] The oscillations of the air molecules in the interstices of the porous material at the frequency of the exciting sound wave result in frictional losses. Changes in flow direction and expansions and contractions of the flow throughout irregular pores result in a loss of momentum in the direction of wave propagation. These two phenomena account for most of the energy losses in the high-frequency range.[3] The piezoelectric PVDF element serves as the active input to contribute to the low-frequency sound attenuation. It is cylindrically curved to couple the in-plane strain associated with the piezoelectric effect with the vertical motion needed to radiate sound from the foam surface. The PVDF active layer is bonded between the two foam halves forming a very thin, compact arrangement. The smart foam can be bonded directly to a vibrating structure and acts as an active surface coating that provides reduction of structural sound radiation. The goal of implementing the smart foam is to modify the acoustic impedance seen by the structure in order to yield a net decrease in the far-field sound power radiated by the vibrating noise source. The potential of the smart foam to simultaneously control low- and high- frequency sound was demonstrated on a simple radiating source (piston).[4] Global cancellation of harmonic and broadband noise induced by a vibrating piston was successfully achieved by the smart foam. The passive–active device was able to modify the radiation impedance observed by the piston over a wide range of frequencies. The potential of smart foam to globally reduce low and high frequency sound radiating from a simple acoustic source was demonstrated. Recently, multiple smart foam modules [with a multiple-input multiple-output (MIMO) control system] were also successfully implemented to reduce the sound radiation from a complex radiating source (plate).[5] Such smart foam systems were also mounted on several fuselage crown panels of a mid-size business jet Cessna Citation III to reduce interior noise at the pilot's ear level.[6] Its potential for reducing aircraft interior noise was demonstrated under band-limited random excitation.

These works, providing the basis of the present study,

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have been concerned with an adaptive feedforward controller based on the filtered- x LMS (FXLMS) algorithm.[7,8] Such a controller requires the use of a coherent reference signal in order to generate the control signal. A nonadaptive feedback usually consists of a large gain controller over the frequency band of interest. The advantages are the relative simplicity of the control algorithm and the lack of a reference signal. However, this approach suffers from potential instability when the phase shift of the secondary path approaches 180°; to ensure stability at these frequencies, the open-loop gain then has to be less than unity (''rolloff''). Therefore, it is commonly reported that it provides only limited cancellation over limited frequency bandwidth.[8] To overcome the limitations of a nonadaptive feedback controller and to avoid the use of a reference sensor required by a feedforward controller, an adaptive feedback controller is investigated in this work. An adaptive feedback controller can be viewed as an adaptive feedforward controller that generates its own reference signal based on the adaptive filter output (control signal) and the error sensor signal.[8] This type of control architecture is also known as internal model control (IMC) and has been previously studied by Elliott *et al.*[9] and Morari and Zafiriou.[10]

In this paper, the performance of the smart foam to reduce sound radiation from a vibrating plate is investigated with the use of adaptive feedforward and feedback controllers for different disturbance frequency bandwidths. An error microphone located in close proximity (for reason of compactness) of the smart foam provides the error signal, which has to be minimized by both adaptive controllers. For the feedforward controller, two different reference signals are implemented. First, the signal sent to the piezoceramic actuator exciting the plate (i.e., disturbance) is used as reference signal. The second reference signal is provided by an accelerometer located on the plate, which is more realistic in practice. The results of these three different control systems are compared in terms of the sound power attenuation achieved.

I. CONTROL ALGORITHMS

For the sake of completeness, the control algorithms and their corresponding block diagram for a single-input single-output (SISO) configuration are briefly presented in this section. More detailed discussion can be found in Refs. 7–10 The real-time implementation of these two SISO adaptive feedforward and feedback algorithms was performed on the TMS320 C30 DSP board resident in a personal computer.

A. Adaptive feedforward FXLMS algorithm

The block diagram of the feedforward FXLMS algorithm is shown in Fig. 1(a). The secondary path transfer function $S(z)$ (i.e., the transfer function between the error sensor and the control actuator) is assumed to be modeled off-line and its estimate is denoted $\hat{S}(z)$. As shown in Fig. 1(a), such a controller requires the use of a coherent reference signal $x(n)$ in order to generate the control signal $y(n)$. The undesired noise from the primary disturbance is measured by a reference sensor, which is filtered through an

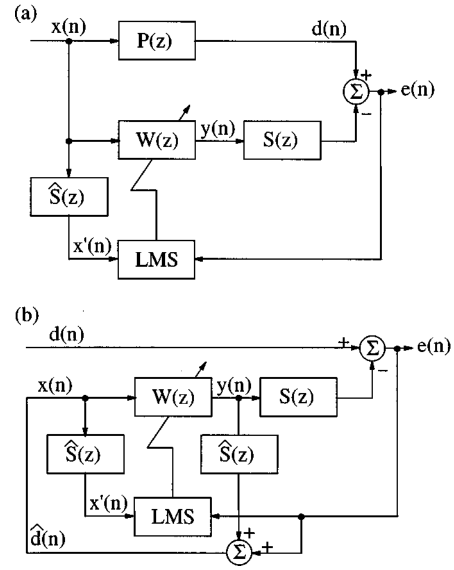

FIG. 1. Block diagram of an adaptive (a) feedforward and (b) feedback control system.

adaptive filter $W(z)$, and used to drive the secondary or control source to minimize the error sensor signal $e(n)$. The reference signal is also filtered by the estimate of the secondary path $\hat{S}(z)$ to obtain the filtered reference used to update the coefficients of the adaptive filter $W(z)$. The reference sensor signal should be highly correlated with the primary disturbance. For broadband random disturbances, the reference sensor should provide advance information about the primary disturbance before it reaches the secondary control source, which is the requirement for a causal controller. The signal from the error sensor is used to update the coefficients of the adaptive filter $W(z)$ in order to minimize the error sensor signal. The adaptive filter $W(z)$ as well as the estimate of the secondary plant transfer function $\hat{S}(z)$ are realized as finite impulse response (FIR) filters. The error signal $e(n)$ at time n is expressed as

$$e(n)=d(n)-\sum_{m=0}^{M-1}\hat{s}_m y(n-m), \tag{1}$$

where $\hat{s}_m$, for $m=0$ to $M-1$, are the coefficients of the Mth order FIR filter $\hat{S}(z)$ used to estimate the secondary path transfer function. The secondary or control signal is generated as

$$y(n)=\sum_{l=0}^{L-1} w_l(n)x(n-l), \tag{2}$$

where $w_l(n)$, for $l=0$ to $L-1$, are the coefficients of the Lth order adaptive FIR filter $W(z)$ at time n. These coefficients $w_l(n)$ are updated by the FXLMS algorithm as

$$w_l(n+1)=w_l(n)+\mu x'(n-l)e(n), \quad \text{for } l=0,\ldots,L-1, \tag{3}$$

where μ is the convergence parameter (or step size). In Eq. (3), the filtered reference signal $x'(n)$ at time n is defined as

$$x'(n)=\sum_{m=0}^{M-1}\hat{s}_m x(n-m), \tag{4}$$

and corresponds to the reference signal $x(n)$ filtered through the estimate of the secondary path $\hat{S}(z)$.

B. Adaptive feedback FXLMS algorithm

Feedback systems are required for applications in which it is not possible or not practical to sense a coherent reference signal. Such applications could include spatially incoherent noise generated by turbulence. A single-channel adaptive feedback algorithm that uses a reference signal synthesis technique is described in this section. Figure 1(b) depicts the block diagram of such a feedback system. This technique can be viewed as an adaptive feedforward system that synthesizes or generates its own reference signal based only on the adaptive filter output and the error signal. Therefore, the basic idea of the adaptive feedback system used in this work is to estimate the primary disturbance and use it as a reference signal. The reference signal $x(n)$ at time n is synthesized as an estimate of $\hat{d}(n)$, which is expressed as

$$x(n)\equiv\hat{d}(n)=e(n)+\sum_{m=1}^{M}\hat{s}_m(n)y(n-m). \tag{5}$$

This type of control architecture transforms a feedback control problem to a feedforward control. It is also known as internal model control (IMC) and is equivalent to a Youla parametrization since the internal model or estimate of the secondary path $\hat{S}(z)$ is used to give an estimate of the disturbance prior to control.[9,10] Note that IMC or Youla parametrization is originally used to find the set of all closed loop stabilizing controllers. The coefficient of the adaptive filter $W(z)$ are updated by the FXLMS in the same way as described in the previous section [see Eq. (3)]. The adaptive filter $W(z)$ of the feedback system acts as an adaptive predictor of the primary disturbance $d(n)$ to minimize the residual signal $e(n)$. Therefore, the performance of such an adaptive feedback system depends on the predictability of the primary disturbance $d(n)$.

II. EXPERIMENTAL SETUP

The experimental setup is depicted in Fig. 2. A rectangular aluminum plate ($6.75\times 6\times 1/64$ in.^{3}) with fixed boundary conditions is treated with the smart foam [Fig. 2(a)]. Note that the size of this plate is about half that of the Cessna Citation III fuselage crown panels on which the smart foam was previously tested.[6] The plate is mounted in a rigid baffle in an anechoic chamber. A piezoceramic actuator was bonded to the plate and provided the primary disturbance.

A sound-absorbing material, consisting of partially reticulated polyurethane foam, provides the passive element of ''smart foam'' (2 in. thick). Such material dissipates incident acoustic wave energy through friction associated with the coupling of the liquid and solid phases of the foam. Partially reticulated polyurethane foam is an acoustical grade, open cell, flexible ester-based urethane foam designed to give maximum sound absorption per given thickness.[11] Since passive sound control is only significant at high frequencies, a 28-μm Ag metallized PVDF film is embedded in the foam to implement the active control, which is most effective at low frequencies. A silver-electroded PVDF was chosen as it can sustain high-voltage amplitudes required for actuator applications. The main physical characteristic of the PVDF actuator is that it is intentionally curved to couple the predominantly in-plain strain associated with the piezoelectric effect and the vertical motion that is required to accelerate fluid particles and hence radiate sound away from the surface of the foam. The PVDF film is curved into a half-cylinder of 1.5-in. diameter and 2.5-in. length and is embedded in two foam halves with spray glue. This leads an ''active cell'' which is 3 in. long, 2 in. wide and 2 in. thick as depicted in Fig. 2(b). Note that the active cell is encased in a balsa wood frame for improved radiation efficiency.[5] The ''smart foam'' consists of the active cell tightly embedded in the center of a piece of passive foam [see Fig. 2(b)]. It is then mounted on the plate. An error microphone, whose signal has to be minimized in the active control process, is located at 3 in. away from the center of the active cell in the direction normal to the plate [see Fig. 2(a)].

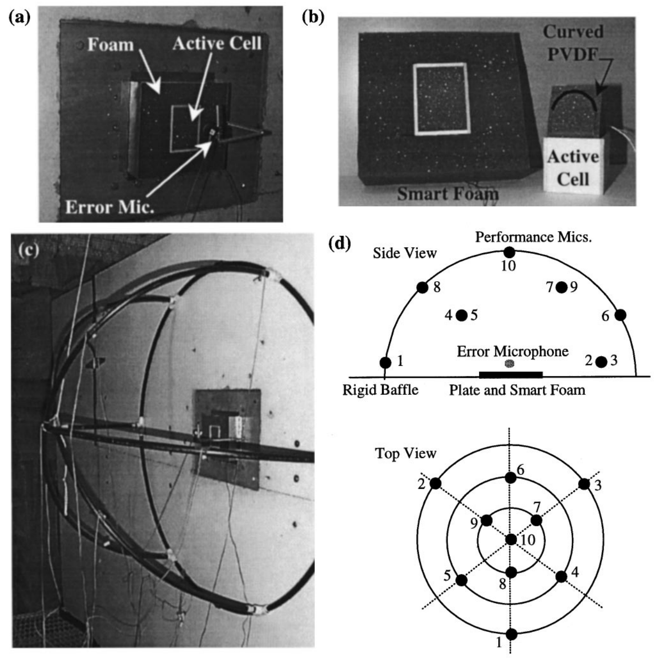

FIG. 2. Experimental setup.

The effect of minimizing the error microphone signal was monitored at ten microphones located on a hemispherical structure of 2-ft. radius. [see Fig. 2(c) and (d)]. The positions of these microphones are such that they measure equal areas on the hemisphere and therefore the acoustic power radiated in free space by the system can be deduced using the expression

$$\Pi=\frac{(S/10)}{2\rho_0 c_0}\sum_{i=1}^{10}|p_i|^2, \tag{6}$$

where S is the surface of the hemisphere and p_i is the sound

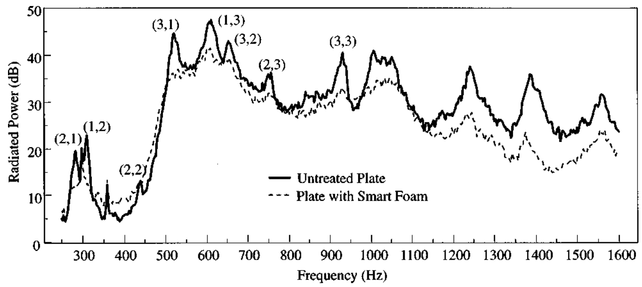

FIG. 3. Passive effect of the smart skin on the acoustic power radiated from the plate.

pressure measured at the ith microphone located on the hemisphere as shown in Fig. 2(d).

The passive effect of the smart foam on the radiated power is presented in Fig. 3. The smart skin provides good passive attenuation at the plate resonance frequencies mainly in the form of additional vibration damping. Its effect off-resonance is negligible, except in the high frequency range, i.e., above 1 kHz. It should also be mentioned that the resonance frequencies of the plate are slightly lowered by the presence of the smart foam, implying a slight added mass effect in addition to damping.

III. CONTROL RESULTS

The following results present a comparison of the effect on the radiated power achieved with the adaptive feedforward and feedback controllers for harmonic and band-limited disturbances. For the feedforward controller, two different reference signals were implemented. First, the signal sent to the piezoceramic actuator exciting the plate (i.e., disturbance) is used as reference signal; this case will be referred to as ''generator as reference.'' The second reference signal is provided by an accelerometer located on the plate; this case will be referred to as ''accelerometer as reference'' and corresponds to a more realistic system in practice. The effect of the active cell on the accelerometer signal was found to be negligible. Indeed, control results compensating for this feedback path on the reference signal (feedback removal) showed no improvement compared to those neglecting this

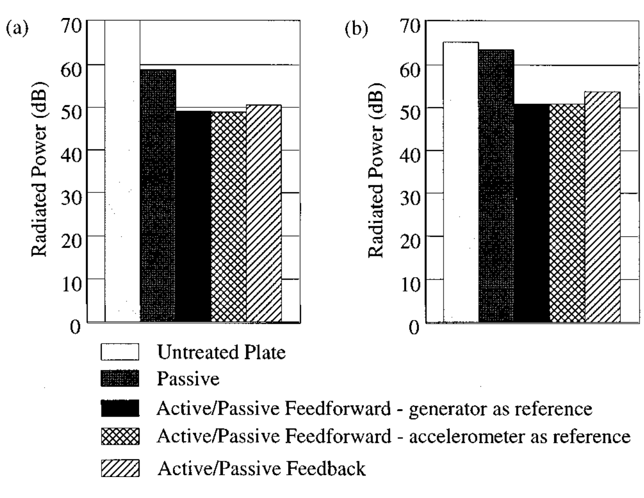

FIG. 4. Harmonic control at (a) 610 Hz (on-resonance) and (b) 560 Hz (off-resonance).

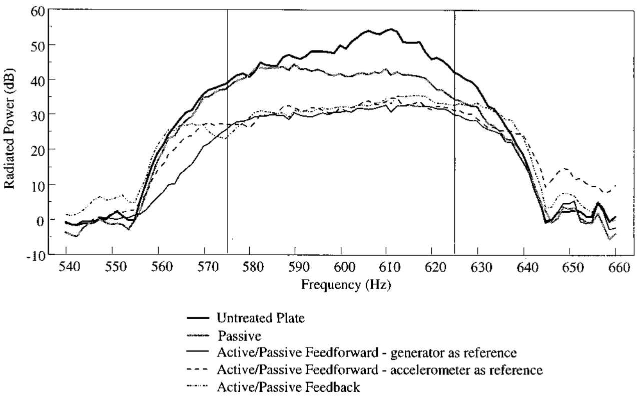

FIG. 5. Narrow-band control 600±25 Hz (50-Hz bandwidth).

feedback path. Therefore, the results are presented only for the accelerometer as a reference signal without feedback removal taken into account. The error microphone was located at 3 in. in front of the active cell (which corresponds to the main dimension of the active cell). Increasing the distance of the error microphone from the foam (up to 9 in.) yielded comparable control results in the frequency range of interest.

A. Harmonic disturbances

The plate was first excited via a piezoceramic actuator with a single frequency disturbance. The first harmonic frequency studied is 610 Hz, which corresponds to the resonance frequency of the (1,3) mode for the plate treated with smart foam (see Fig. 3). The second harmonic disturbance was chosen to be 560 Hz, corresponding to an off-resonance frequency of plate–foam system [between resonance frequencies of modes (3,1) and (1,3) as seen in Fig. 3] at which the passive effect of the smart foam is very limited. The radiated power for the different control cases is shown in Fig. 4. For the harmonic disturbance at 610 Hz, the smart foam provides about 11 dB of passive attenuation; while implementing active control, an extra 10 dB attenuation is achieved [see Fig. 4(a)]. When the plate is excited off-resonance at 560 Hz, the passive effect of the foam is very limited (less than 2 dB). Active control is seen to provide more than 10 dB power attenuation with respect to the untreated plate. For both frequencies, the adaptive feedback system performs as well as the adaptive control system since the disturbance is harmonic, i.e., perfectly predictable.

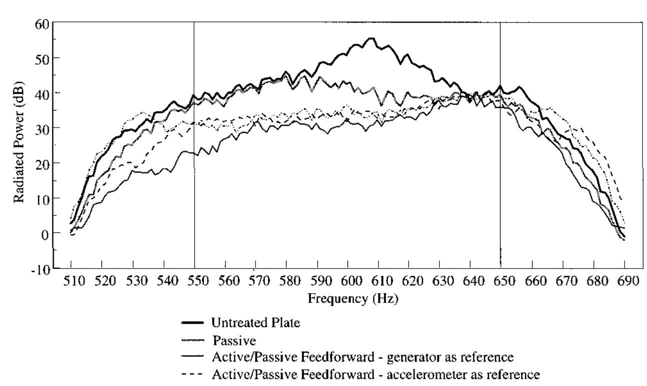

FIG. 6. Narrow-band control 600±50 Hz (100-Hz bandwidth).

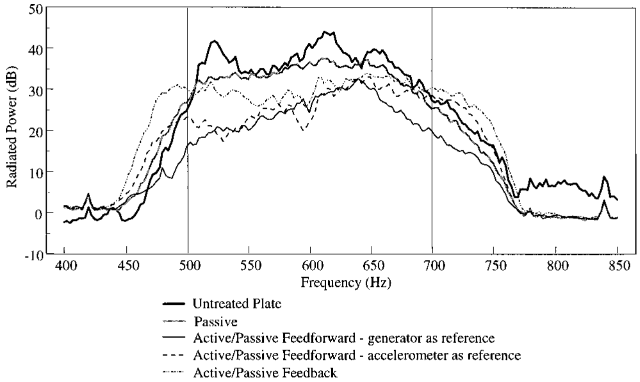

FIG. 7. Narrow-band control 600±100 Hz (200-Hz bandwidth).

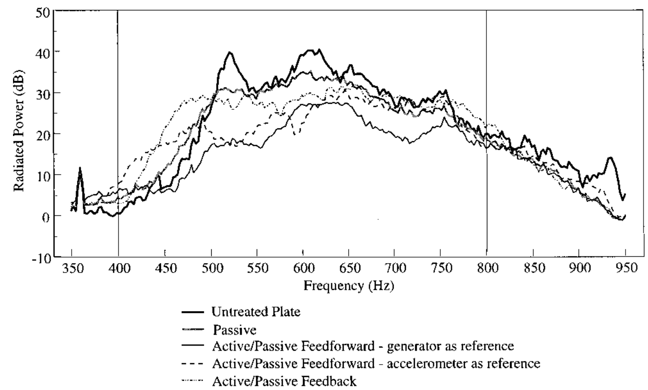

FIG. 8. Narrow-band control 600±200 Hz (400-Hz bandwidth).

B. Band-limited disturbances

The plate was then excited with band-limited random disturbances of increasing bandwidth around the center frequency 600 Hz [close to resonance frequency of (1,3) mode]. The radiated power for the different control cases investigated are presented in Figs. 5–8 for 50-, 100-, 200-, and 400-Hz frequency bandwidth, respectively. The achieved total power attenuation for the different frequency bandwidths and control systems is summarized in Table I. Note that for the different cases presented in this section, the sampling frequency associated with the DSP was fixed at 3.2 kHz and the FIR filters [estimated secondary path $\hat{S}(z)$ and adaptive filter $W(z)$] had 128 coefficients (i.e., $L=M=128$).

The control results obtained when the disturbance is limited to 50- and 100-Hz bandwidth with a center frequency of 600 Hz (see Figs. 5 and 6) are first investigated. The performance of the feedforward system with the disturbance signal as reference is slightly better than that when the reference signal is obtained from the accelerometer directly mounted on the plate. The adaptive feedback control system performs as well as the feedforward control system with the accelerometer signal as reference. The averaged power attenuation associated with these two control systems is similar. An active/passive power reduction of about 16 and 10.5 dB for the 50- and 100-Hz frequency bandwidths, respectively, is obtained with a realistic control system as seen in Table I. For these two frequency bandwidths, the plate response is associated with a single mode corresponding to a resonance frequency of 610 Hz. Due to the presence of this plate resonance frequency in the control bandwidth, the reference signal is more predictable, leading to good control results associated with the adaptive feedback control system.

The frequency bandwidth of the disturbance was then increased to 200 and 400 Hz (with a center frequency of 600 Hz). In this case, several modes of vibration are associated with the plate response [mainly modes (3,1), (1,3) and (3,2) are present in the control bandwidth]. The feedforward control with the disturbance generator signal as reference performs best, as expected. When the accelerometer is used to provide the reference signal to the controller, the control performance is decreased (see Table I). The difference in control performance between the two different reference signals for the feedforward control system is essentially related to the fact that the optimal compensating filters are acausal since the propagation time through the control path is longer than the propagation time through the disturbance path. The controller must be able to receive the reference signal and produce the appropriate control signal to the active cell of the smart skin before the structural disturbance from the piezoceramic on the plate propagates through the foam. In this situation, the time delay for the controller to react is shorter when the reference signal is from the accelerometer directly mounted on the plate than when it is the disturbance signal driving the piezoceramic actuator. This effect is increased when the disturbance frequency bandwidth is increased (see Table I). The adaptive feedback control results in even lower performance than the feedforward control with the accelerometer signal as reference. This could be expected; as for the adaptive feedback control, the reference signal has to be synthesized, i.e., predicted. Since the disturbance is random and

TABLE I. Total radiated power attenuation.

	Radiated power attenuation			
		Active passive feedforward		
Frequency bandwidth (Hz) (center frequency 600 Hz)	Passive (dB)	Ref. gener. (dB)	Ref. accel. (dB)	Active passive feedback (dB)
50	7.5	19.2	17.9	17.6
100	7.5	15.8	13.7	13.4
200	4.0	11.8	10.3	7.5
400	4.0	11.2	8.0	5.2

of increasing frequency bandwidth, it becomes more difficult to predict. These results demonstrate that, for broadband excitation of this particular test arrangement, an adaptive feedforward controller (with reference signal obtained from an accelerometer on the vibrating structure) is more efficient in reducing the radiated power than an adaptive feedback controller. However, the adaptive feedback controller provides similar power attenuation between 550 and 750 Hz as the feedforward controller with structural reference sensor. This is probably due to the presence of the resonance frequency of the dominant (1,3) mode in this frequency range; this indeed makes the reference signal more predictable. The major drawback of the adaptive feedback controller is the important increase in radiated power (control spillover) below 500 Hz, where no structural modes are present (off-resonance).

IV. CONCLUSIONS

The performance of SISO adaptive feedback and feedforward controllers to reduce the radiated power from a plate (similar to a small fuselage panel) treated with smart foam was discussed. The adaptive feedback controller was found to be as efficient as the feedforward controller with a structural reference sensor (accelerometer on the plate) when the disturbance was either harmonic or limited up to 100 Hz in bandwidth. This is related to the fact that the adaptive feedback controller generates its own reference signal based on the adaptive filter output (control signal) and the error sensor signal; therefore, it can be viewed as a predictor. When the frequency bandwidth was increased to 400 Hz, the adaptive feedback system was associated with control spillover in the low-frequency range where no structural mode resonance frequency is present. However, close to the system resonance frequencies (i.e., where reference signal is then more predictable), the efficiency of the adaptive feedback controller was found to be similar to that of the feedforward controller with a structural reference sensor (accelerometer).

Further work will involve the investigation of a multiple input multiple output (MIMO) feedback and feedforward control system on a plate located in a wind tunnel (excitation simulating turbulent boundary layer noise).

ACKNOWLEDGMENTS

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