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Active Control of Interior Noise in Model Aircraft Fuselages Using Piezoceramic Actuators

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Active control of interior noise in model aircraft fuselages using piezoceramic actuators is experimentally studied. The actuators are bonded directly to the structure and error information is taken from up to two microphones located in the interior acoustic field. The results demonstrate that global attenuation of the order of 10–15 dB of interior noise can be achieved with piezoceramic actuators, irrespective of whether the shell system is vibrating at an acoustic or structural resonant frequency. For the case of an acoustic resonance, reduction of the interior field was accompanied by an increase in shell response. The work also proves that active control using vibration (moment) inputs works well when a floor simulating that of an aircraft is installed in the model. This result suggests that the technique will be successful in controlling interior noise in realistic aircraft structures, although further work is needed to ensure that the approach works satisfactorily under realistic situations.

Introduction

PREVIOUS work[1-3] has demonstrated the potential of using active forces to control sound transmission into the interior of aircraft fuselages. However, active forces have a number of disadvantages. As they are usually applied at a point, they are "spectrally white" in a spatial sense, leading to undesired spillover and degradation in control performance. Similarly, the forces are usually applied by electromechanical means and thus usually require bulky back reaction mounts.

Recently, the use of piezoceramic elements as actuators has been investigated in an attempt to overcome some of these disadvantages. The piezoceramic elements are light, can be bonded directly to the structural surface, and are cheap to purchase. Analytical work[4] has also demonstrated that tailoring the location, number, and sizes of these patches can lead to selective excitation of modes in structures. Thus, they show much potential for use in a distributed controller configuration with reduced spillover characteristics when correctly designed. Experiments[5] have also confirmed that when piezoceramic elements are bonded to a panel they can be used as part of an active control system to give high global reduction of radiated sound from the panel.

Piezoceramic elements thus show much potential for use as attached or embedded actuators to actively control airborne and/or structure-borne interior noise in aircraft. As in previous tests, the basic concept is to attach the element directly to the structure (fuselage) while obtaining error information from the interior acoustic field.[2] In this paper, the use of piezoceramic elements in such a scenario is studied using an aircraft model fuselage. This model fuselage has been used in previous tests[6] and its acoustic and structural response characteristics are well known. The major attribute of the test rig is that it has a removable internal floor thus enabling the study of interior noise mechanisms in simple geometries (floor removed) and more complex geometries representative of aircraft interiors (floor installed).

Experimental Apparatus

Figure 1 is a photograph while Fig. 2 is a schematic of the test arrangement. The aircraft fuselage was modeled as an aluminum cylinder 0.508 m in diameter, 1.245 m long, and 1.63 mm thick. The removable floor was 0.381 m wide and consisted of thin aluminum skin attached to a lattice structure. Figures 3a and 3b show the internal configuration of the bare cylinder and with the floor installed, respectively. Full details of the rig are given in Ref. 6. Propeller noise was modeled by a 60-W horn driver attached to a tapered horn whose outlet was positioned 76 mm from the exterior of the cylinder. All tests were performed at single pure-tone frequencies. The inte-

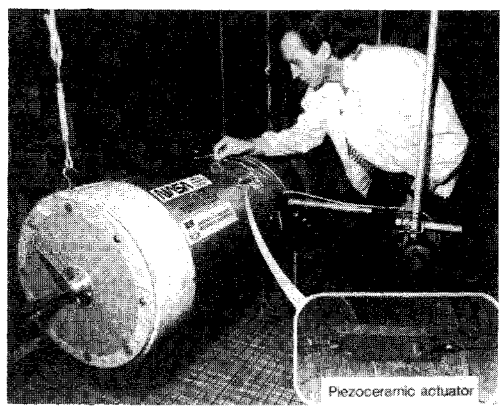

Fig. 1 Photograph of experimental rig and piezoceramic actuator.

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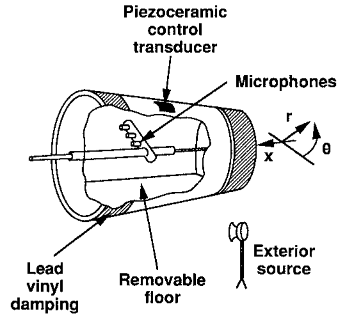


Fig. 2 Schematic of cylinder test apparatus.

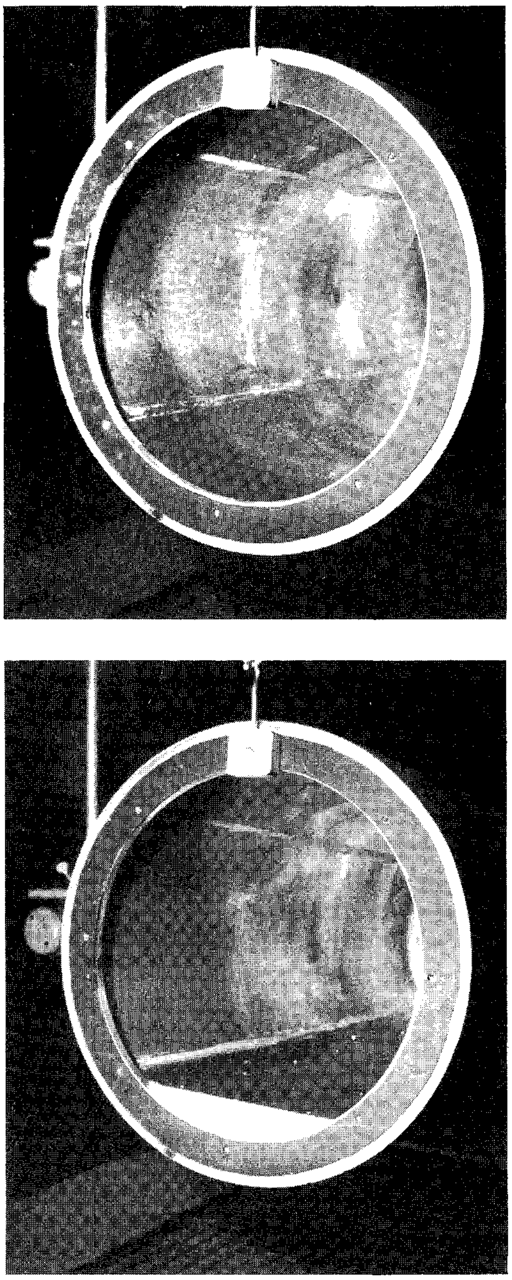
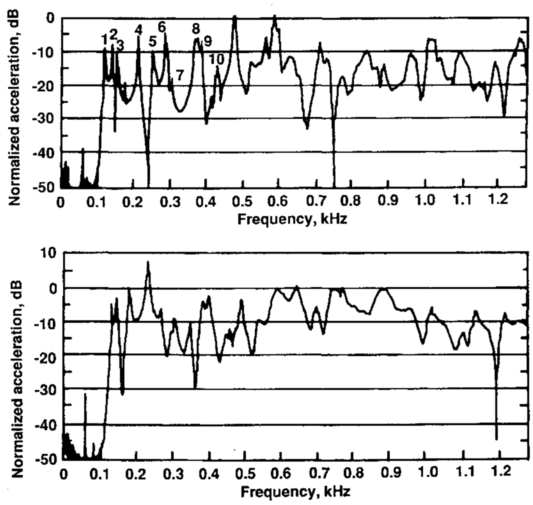

a)
b)

Fig. 3 Photograph of the interior of the cylinder: a) bare and b) floor installed.

rior pressure field was measured by three 1/2-in. microphones mounted on a movable traverse. The following results will be presented as sound pressure level contour plots located in the source plane. Additionally, the structural response was measured by an array of 24 uniformly spaced mini-accelerometers attached to the cylinder in the source plane. To stimulate a free-field environment, the experiments were performed in an anechoic chamber at NASA Langley Research Center.

To achieve active inputs, two bimorph piezoceramic actuators (see inset of photo) of dimensions 50.8 × 12.7 × 0.51 mm were bonded to the exterior of the cylinder in the source plane at 180 and 45 deg (0 corresponds to the acoustic source location). A bimorph actuator has two collocated piezoceramic elements driven 180 deg out-of-phase to increase the surface bending. A reference signal was used to drive the acoustic noise source and the same signal was passed through a two-channel manually operated phase shifter. The control signals were then amplified, passed through transformers with a voltage gain of 7:1, and connected to each bimorph element. The experimental procedure was as follows. The noise source was driven at the required frequency and level. The amplitude and phase of each control signal (for some tests only one channel was used) were adjusted to minimize the interior sound levels at up to two error microphones located at fixed positions in the interior cavity. In practice, a computer-based adaptive controller as used in Ref. 7 could have been used. However, the manual system used here was more convenient for the purpose of these tests—to demonstrate the potential of piezoceramic actuators. Once the error signals were minimized, the interior field was mapped by using the traversing microphone array. The control signal(s) were then turned off and the interior noise or primary field mapped. For both of these conditions, the signals from the accelerometer array were simultaneously acquired. The preceding tests were performed for the two test configurations without the floor installed (bare cylinder) and cylinder with the floor fitted. When the floor was installed, the subcavity was filled with acoustic foam to inhibit acoustic resonances in this space which might complicate the system response. In general, the actuators were located at regions of greatest strain while the error microphones were located in regions of high sound pressure for the uncontrolled response.

Before carrying out the preceding tests, it was necessary to characterize the structural and acoustic frequency response functions (FRF) of the cylinder and interior acoustic space. For the cylinder, the FRF was measured for the two conditions of with and without the internal floor by driving the structure with a minishaker attached to the cylinder at 180 deg in the source plane and excited with broadband noise. The FRF of the output of the nearest accelerometer, referenced to the input excitation, was then obtained for each case using a spectral analyzer. Similar results were obtained for the interior cavity, except in this case the excitation source was a small loudspeaker located at a corner of the cylinder and output was



a)
b)

Fig. 4 Structural frequency response function of cylinder: a) bare and b) floor installed.

taken from a microphone located at $\Theta = 0$ deg in the source plane and radial distance of $r/a = 0.925$, where a is the radius of the cylinder. Again, results were obtained with and without the internal floor.

Results

The frequency response functions of the bare cylinder and with the floor installed are presented in Figs. 4a and 4b, respectively, while Figs. 5a and 5b are for the corresponding acoustic cavity cases. Various experimentally determined structural and cavity resonant frequencies correlated with their associated theoretical mode number distribution are given in Tables 1 and 2 for the bare cylinder. The mode numbers were identified using relations which predict the resonance frequencies of these systems.[8] The indices (m,n,ℓ) correspond to axial, azimuthal, and radial acoustic mode numbers, respectively, while (m,n) corresponds to structural mode numbers as outlined in Ref. 8. It can be seen from the lack of sharpness of the peaks in the FRFs of Figs. 4b and 5b that when the floor is installed the response exhibits increased damping due to the acoustic foam located in the subspace beneath the floor, particularly at higher frequencies.

It was decided to choose two frequencies, one corresponding to a structural resonance and the other a cavity (or acoustic) resonance for the tests. Accordingly for the bare cylinder, test frequencies of 260 and 666 Hz were chosen. As can be seen from the corresponding FRFs, these frequencies provide dominantly structural and acoustic response of the system in the (1,2) and (1,2,1) modes, respectively. The test frequencies are also slightly different than those indicated by Tables 1 and 2 for the particular modes of interest. This behavior occurred because the piezoceramics exhibited little mass loading on the cylinder structure as contrasted to the minishaker. Thus, for each test, the cylinder or acoustic response was tuned slightly in frequency to ensure the cylinder system was on resonance. When the floor was installed, these resonance frequencies were observed to have lowered to 240 Hz for the structural mode, most likely due to the increased mass of the system, whereas for the acoustic mode, the frequency increased to 687 Hz due to the smaller cavity size. Hence these frequencies were chosen for the tests when the floor was installed. The resonance frequencies of the system were tracked by comparing the FRFs, interior pressure distributions, and shell response in the source plane with and without the floor installed. For the case of the structural resonance, the similarity of the interior pressure distributions and shell response identified the modes. For the acoustic resonance, the nearest peak in the FRF of the acoustic space with floor installed to 666 Hz (the bare resonance) was chosen.

Table 1 Structural resonance frequencies of cylinder (see Fig. 4a)

Peak	Measured frequency, Hz	Theoretical mode $(m,n)^a$
1	125	(1,4)
2	146	(1,3)
3	158	(1,5)
4	217	(1,6)
5	258	(1,2)
6	293	(1,7)
7	305	——
8	373	(1,8)
9	390	(3,7)
10	418	——

[a]m: axial; n: azimuthal.

Table 2 Acoustic resonance frequencies of cylindrical cavity (see Fig. 5a)

Peak	Measured frequency, Hz	Theoretical mode $(m,n,\ell)^a$
1	142	(1,0,1)
2	284	(2,0,1)
3	390	(0,1,1)
4	414	(1,1,1)
5	430	(3,0,1)
6	481	(2,1,1)
7	566	(4,0,1)
8	637	(0,2,1)
9	670	(1,2,1)
10	691	(4,1,1)

[a]m: axial; n: azimuthal; ℓ: radial.

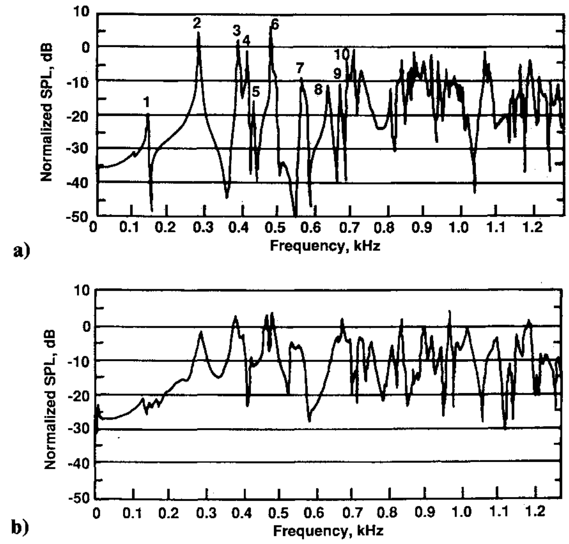


Fig. 5 Acoustic frequency response function of interior cavity: a) bare and b) floor installed.

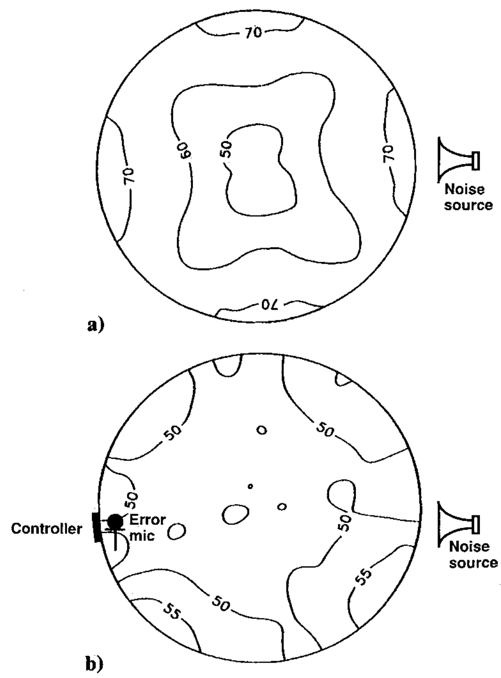


Fig. 6 Interior sound pressure levels (bare cylinder, $f = 260$ Hz): a) primary and b) controlled.

Bare Cylinder

Figures 6a and 6b present sound pressure level contour plots (dB relative to 2×10^{-5} N/m^2) in the source plane for the primary (noise) and controlled fields of the bare cylinder at a frequency of 260 Hz. As already discussed, this frequency corresponds to a structural resonance of the (1,2) mode. In

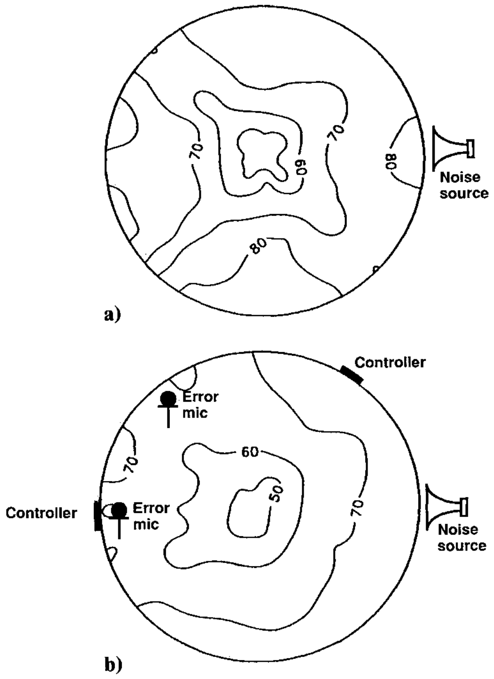


Fig. 7 Interior sound pressure levels (bare cylinder, $f = 666$ Hz): a) primary and b) controlled.

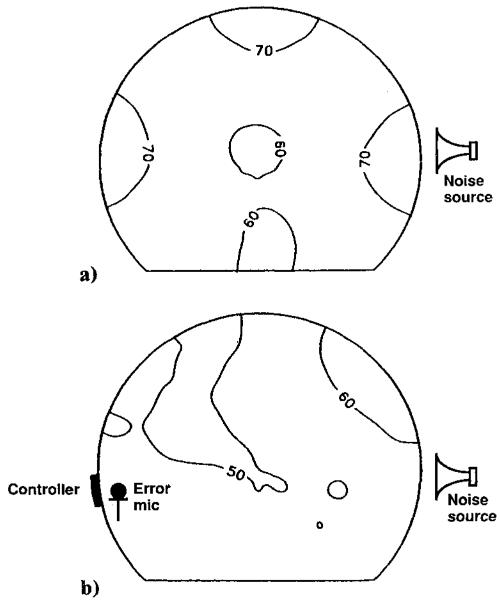


Fig. 8 Interior sound pressure levels (floor installed, $f = 240$ Hz): a) primary and b) controlled.

this case, only one piezoceramic control actuator at 180 deg was used and the sound field was minimized at a single microphone error sensor located at 180 deg, $r/a = 0.925$. The primary field can be seen from Fig. 6a to exhibit the four nodal behavior associated with the azimuthal modal order of two. Comparisons of Figs. 6a to 6b show global attenuations of up to 15 dB. It can be seen that attenuations are achieved where they are needed—in the regions of high sound levels.

For the next test, the frequency of excitation was increased to 666 Hz which was seen to correspond to a resonance of the (1,2,1) acoustic mode. The primary field of Fig. 7a is seen to again exhibit behavior associated with an azimuthal mode of order two. However, it is also apparent that other modes are present, showing up as asymmetry in the pressure field. Figure 7b presents the controlled case in which two piezoceramic actuators were employed in conjunction with two error microphones at 130 and 180 deg, $r/a = 0.925$. It is again evident that global attenuations of the order of 10 dB were achieved for this frequency. In this second case, it was necessary to use two channels of control because of the higher modal density of the acoustic cavity. Although the single control actuator could exert control over the (1,2,1) mode that was on resonance, additional modes were present, perhaps due to mode coupling. As seen in previous work,[2] this situation dictates the use of more than one control channel, in order to exert controllability over a number of modes. Sound pressure level measurements made outside of the source plane also confirm the global nature of the controlled fields discussed in the preceding two cases, attenuations being close to the same values as obtained in the source plane.

Cylinder with Floor

In the next series of tests, the internal floor was installed. Figure 8a shows the primary field for the structural resonance now at a frequency of 240 Hz. The primary field can be seen to be a distortion of the cos (2Θ) distribution observed in the bare cylinder. As the floor had an inclusion angle of 45 deg, its presence has little effect on the $n = 2$ azimuthal mode being near its nodal lines. Since the structure is on resonance and the acoustic field is being "forced" into response, the interior pressure field is little changed even with the noncircular cavity shape caused by the pressure of the floor. Control was applied in this case again using a single actuator at 180 deg in conjunction with a single error microphone at 180 deg, $r/a = 0.925$. The controlled field, given in Fig. 8b, again demonstrates global attenuations of the order of 10 dB.

The normalized shell acceleration levels in the source plane corresponding to Figs. 8a and 8b are given in Fig. 9. It appears from the noise field that the shell is responding in primarily a distorted $\cos(2\Theta)$ mode with some content from other lower order modes. Figure 9 shows the cylinder response is relatively lower over the angles $215 < \Theta < 310$ deg, which was thought to be due to the presence of the tightly packed foam located in the cavity space under the floor damping the response in this region. Application of control leads to a significant drop in the acceleration levels and the residual response now appears to be of much higher azimuthal order. It appears that the controller has forced a reduction in level of the structural mode which is on resonance. This has produced a corresponding reduction in the forced acoustic response in the shell interior space.

The test frequency was then increased to 687 Hz corresponding to the shifted acoustic resonance with the floor installed. The primary acoustic field in the source plane is given in Fig. 10a. In this case, the primary field is significantly different from the corresponding bare cylinder case of Fig. 7a. This behavior is not surprising as the acoustic space is significantly different with the floor installed and thus leads to a different mode shape on resonance.

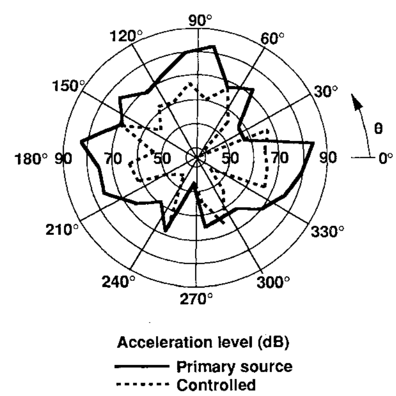


Fig. 9 Shell radial acceleration (floor installed, $f = 240$ Hz).

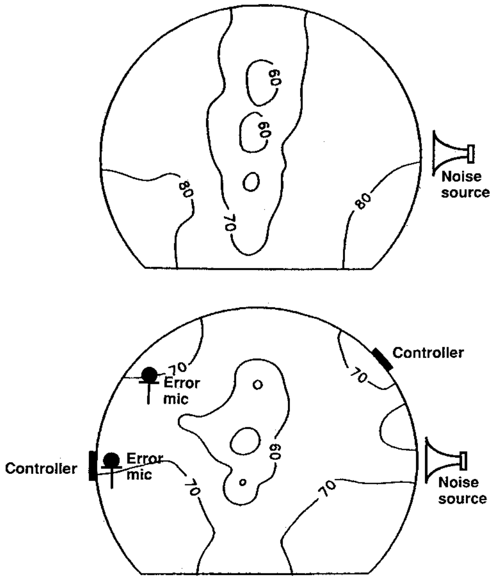


Fig. 10 Interior sound pressure levels (floor installed, $f = 687$ Hz): a) primary and b) controlled.

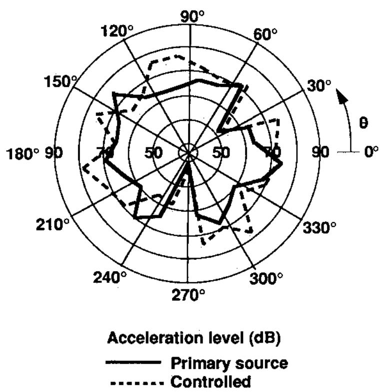


Fig. 11 Shell radial acceleration (floor installed, $f = 687$ Hz).

Control was applied in this case using both piezoceramic actuators in conjunction with two error microphones at 135 and 180 deg; $r/a = 0.925$. The controlled field is given in Fig. 10b and again exhibits peak reductions on the order of 10dB. Smaller reductions were obtained for $0 < \Theta < 180$ deg, but in general, reductions were obtained everywhere throughout the source plane. Results obtained out of the source plane again confirm that attenuations in sound pressure level of the same order are also achieved throughout the cylinder. It is interesting to see that the residual pressure response of Fig. 10b appears to exhibit a shape corresponding to a $\sin(2\Theta)$ mode.

The normalized shell radial acceleration levels in the source plane corresponding to Figs. 10a and 10b are given in Fig. 11. When control is applied, the results show that in general the acceleration levels have increased for this case. Figure 11 illustrates an important concept associated with this control approach: To achieve global attenuations of the interior acoustic field, it is not necessary to reduce the shell response. In fact, there are two main mechanisms of control.[9] In the first, called *modal suppression*, only the dominantly radiating modes in the structure are reduced in amplitude. Other modes, which may be of larger amplitude but do not couple with the interior field, are left untouched. In the second, called *modal restructuring*, the overall modal response distribution of the cylinder is changed, both in amplitude and temporal phase, to provide a lower level of coupling with the interior acoustic field. Note, as shown in Fig. 10, that modal restructuring sometimes leads to an increase in shell levels while the interior acoustic sound levels fall. This increase in shell response could be a disadvantage if control spillover is too great. It is in these cases that distributed arrays of actuators and sensors will be needed to keep the shell response bounded.

Conclusions

The use of piezoceramic actuators to control interior noise in model aircraft fuselages has been experimentally studied. The results show that elementary arrays of piezoceramic actuators bonded to the wall of the cylinder can provide global attenuations of the order of 10 dB of interior noise transmitted through the wall. Piezoceramic actuators, either attached or embedded in the transmitting structure, thus have much potential of controlling low-frequency interior noise in a variety of aerospace structures.

This simplified configuration was also shown to work effectively in more complex systems representative of real aircraft. This conclusion is due to the nature of the low-frequency structural-acoustic coupling. Independent of the structural or interior cavity shape, in the low-frequency region there will be an interface modal filtering effect where only a limited number of structural modes will strongly couple to acoustic modes.[1] Thus global control can be achieved, largely independent of geometry, with a low number of control inputs applied directly to the structure while error information is taken from the interior acoustic field. The work reported in this paper, however, is preliminary in nature. Further work studying the optimal location of actuators and sensors on realistic structures is presently in progress at NASA Langley Research Center and Virginia Polytechnic Institute and State University.

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