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Active Structural Acoustic Control of Noise Transmission Through Double Panel Systems

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A preliminary parametric study of active control of sound transmission through double panel systems has been experimentally performed. The technique used is the active structural acoustic control (ASAC) approach where control inputs, in the form of piezoelectric actuators, were applied to the structure while the radiated pressure field was minimized. Results indicate the application of control inputs to the radiating panel resulted in greater transmission loss due to its direct effect on the nature of the structural-acoustic coupling between the radiating panel and the receiving chamber. Increased control performance was seen in a double panel system consisting of a stiffer radiating panel with a lower modal density. As expected, more effective control of a radiating panel excited on-resonance is achieved over one excited off-resonance. In general, the results validate the ASAC approach for double panel systems and demonstrate that it is possible to take advantage of double panel behavior to enhance control performance, although it is clear that further research must be done to understand the physics involved.

Introduction

RECENT developments in turbofan technology have prompted research into innovative ways of reducing interior noise of aircraft. Ultrahigh-bypass turbofans and unducted fans have increased tip Mach numbers leading to increased low-frequency noise fields impinging on the exterior of the aircraft fuselage. Traditional methods of low-frequency noise reduction require heavy damping material which leads to significant weight penalties, offsetting the performance gains of the turbofans. These factors have prompted the research into applying active control techniques to reduce the interior noise field of a fuselage. Research has progressed two ways, applying active control to a realistic aircraft fuselage structures and to simplified models of the noise transmission path.

Early research on realistic fuselages concentrated on active control of interior sound fields using acoustic control inputs. Although good attenuation can be achieved near the error sensors, spillover often caused an increase in sound fields elsewhere.[1] This result coupled with the undesirability of placing many acoustic sources in aircraft led to the implementation of active vibration control of the fuselage using point sources, or electromechanical shakers.[2] Significant global reductions of the interior noise field were presented, but control spillover in the form of increased noise levels was noticed in some areas of the cabin.[3] This was attributed to the spectrally white excitation provided by point sources. Studies have suggested that a distributed actuator, in the form of piezoelectric (PZT) control inputs, can overcome some of the disadvantages. Early work by Fuller et al.[4] experimentally demonstrated the use of piezoelectric actuators on scale fuselage models. Silcox et al.[5] subsequently demonstrated active control of interior noise using PZT actuators mounted on a large-scale composite fuselage model. Global attenuations of up to 6.6 dB were achieved; however, significantly increased vibrational energy of the fuselage was observed at some locations due to the PZT actuators, leading to a concern of structural fatigue.

In addition to the disadvantage just mentioned, implementing PZTs on the fuselage shows other key disadvantages. Design of a fuselage for dynamics conducive to more effective noise attenuation is limited. Installation and repair of sensors and actuators would be extremely difficult since the structure is not removable. Most of these disadvantages are the result of the fuselage doubling as a pressure vessel and, therefore, is subject to strict FAA regulation. Although these limitations may be overcome with further research, an alternative approach of applying the sensors and actuators to the internal trim is presented.

Additional knowledge has been gained from experiments performed on simple models of the aircraft shell and the interior trim. Several experiments have been performed on reverberant noise transmission through single panels, a simple model for a fuselage.[6,7] Although significant global reductions in the transmitted noise field have been achieved for on-resonance excitation, off-resonance reductions have been poor due to the high modal density of the panel.[7] Experiments have also been performed on reverberant sound transmission through double panel systems, modeling the fuselage of the aircraft and the interior trim. Grosveld and Sheperd[8] applied acoustic control inputs to the cavity in between the double panel system achieving global attenuations of interior noise field. Thomas et al.[9] applied electromagnetic shakers acting between a double panel system, achieving 3–17 dB of reduction depending on the frequency of excitation.

This paper investigates the application of active structural acoustic control (ASAC) to the radiating panel of a double panel system, modeling the interior trim of an aircraft. The application of control inputs to the interior trim (radiating panel) has distinct advantages. There is likely to be no increase in fuselage vibrational energy. Design of the panel can be changed to be more conducive to a higher transmission loss. Panels can be removed allowing facilitated installation and repair of sensors and actuators. In general, advantage can be taken of the double panel behavior. However, it should be noted that application of the control inputs to the interior trim will tend to impart local control of acoustic fields whereas control inputs applied to the fuselage will generally impart global control. Thus, there is a need to study the active control of double panel systems and contrast their performance vs the use of fuselage mounted actuators.

A set of experiments were performed in a noise transmission test facility to test the concept. The control technique implemented was ASAC, where the control inputs, in the form of piezoelectric actuators, were applied to the structure and the radiated pressure field was minimized. The control approach was a feedforward narrowband least mean square (LMS) algorithm. Several test cases were performed to prove the validity of applying ASAC to a double panel system. Specifically, the influence of control system configuration, radiating panel stiffness, excitation frequency, and incident acoustic field on control performance was studied.

First, the specifics on the experimental setup are discussed, including physical dimensions, acoustic measurement techniques, and control system implementation. The experimental procedure is then detailed followed by the characteristics of the reverberant acoustic environments. Preliminary results of panel coupling are discussed.

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Table 1 Summary of panel properties

Properties	Incident panel	Flexible panel	Stiff panel
Material	Aluminum	G10 fiberglass	Sandwich board
Dimensions, m	0.381 × 0.305 × 0.00159	0.381 × 0.305 × 0.00159	0.381 × 0.305 × 0.01022
Density, kg/m^3	2700	179.8	2.51
Modulus of elasticity, GPa	64	18.6	39.3

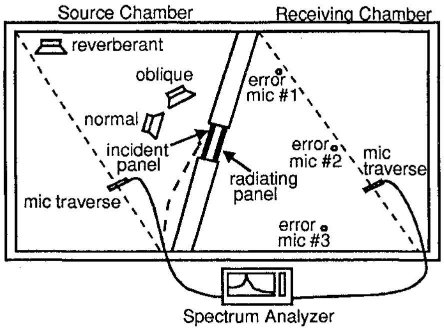

Fig. 1 Transmission loss test facility.

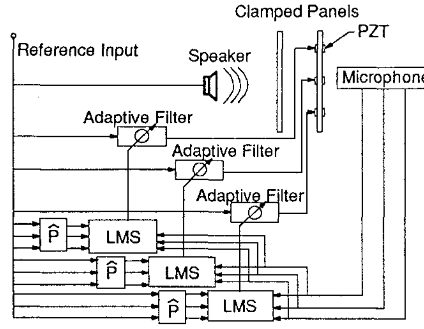

Fig. 2 Control system schematic.

Results from varying control system configuration, radiating panel stiffness, excitation frequency, and incident acoustic field are then presented, followed by concluding remarks.

Experimental Investigation

Experimental Setup

The double panel system, consisting of two panels separated by an air cavity, was mounted in the common wall between two reverberation chambers of a transmission loss test facility. The arrangement of the test chamber, the double panel system, and the location of the three test positions of the acoustical source are shown in Fig. 1. The incident panel was made of aluminum and had dimensions $L_x = 381$ mm, $L_y = 305$ mm and a thickness of 1.6 mm. Two radiating panels of different materials were tested, one made of G10 fiberglass (relatively flexible) and one made of sandwich board construction (relatively stiff). Both radiating panels had dimensions $L_x = 381$ mm, $L_y = 305$ mm, and thicknesses of 1.6 mm and 10.2 mm for the flexible and stiff panels, respectively. A summary of panel properties are presented in Table 1. The panels were mounted in a heavy steel frame which approximates clamped boundary conditions, allowing very little displacement or rotation of the panel edges. The air cavity, which separated the panels by 48 mm, also had dimensions $L_x = 381$ mm and $L_y = 305$ mm. Harmonic excitation for the incident panel was produced by a large speaker placed in the source chamber. The structure borne flanking sound transmission path from the source to the receiving chamber was assumed negligible.[7] Therefore, the sound transmission path can be stated as follows: The incident acoustic wave excites the incident panel which radiates energy into the panel air cavity thereby exciting the radiating panel. The radiating panel then radiates energy to the receiving chamber.

A fundamental question arises from the proposed scope of this work. Is it more effective to apply active control to the radiating panel than the incident panel? Experiments were performed where the controller configuration was varied from controlling the incident panel to controlling the radiating panel. Since the radiating panel is coupled directly into the acoustic field of the receiving chamber, more effective control should be attained with the second configuration.

Two different types of radiating panels were tested to determine the influence of radiating panel stiffness on performance; one flexible panel made of G10 fiberglass, and one relatively stiff panel made of sandwich board construction. Reasoning that the stiff radiating panel has a lower modal density for a particular excitation, more effective control should be achieved with a fixed number of control channels and excitation frequency.

The flexible and stiff panels were excited on-resonance and off-resonance to determine the influence of excitation frequency. As determined by the double panel system frequency response function, the on-resonance frequency was chosen where both panels were close to a resonance, conversely the off-resonance frequency was chosen where both panels were not at a resonance. The choice of test frequencies is detailed later.

To determine the influence of incident acoustic field, three different types of acoustic excitation, normal plane wave, oblique plane wave, and reverberant, were tested as detailed in Fig. 1. The normal plane wave excitation was produced by a speaker placed 0.260 m from the incident panel, at an angle of 0 deg from the panel normal, producing a uniform pressure wave at the incident panel with no phase variation over the panel. The oblique plane wave excitation was placed the same distance from the incident panel, but at 45 deg from the normal, producing a uniform pressure wave at the incident panel with variations in phase. Placing a speaker in the far corner of the room produces a pressure wave at the incident panel with random phase, indicative of reverberant excitation.

Acoustic measurements were performed using a Bruel and Kjaer Type 2032 dual channel signal analyzer. Traversing microphones in the source and receiving chambers were calibrated allowing direct measurements of acoustic pressure. Acoustic pressure measurements for the incident field were taken from a microphone mounted directly in front of the incident plate for the normal and oblique plane wave excitations. For the reverberant case, a microphone was traversed across the chamber, and sound pressure was sampled at 10 discrete locations. The source acoustic power level in the chamber was determined from an average of these measurements. For all excitation cases, radiated field acoustic power level was determined from an average of 10 pressure measurements taken across a traverse of the receiving chamber, see Fig. 1. Note that all acoustic measurements are in decibels referenced to 20 μPa.

The active control system used was a multichannel filtered X LMS algorithm previously implemented on a TMS320C25 digital signal processor (DSP) mounted in a personal computer.[10] The system is capable of reading three error signals plus one reference signal and generating three control signals. A schematic of the control system is presented in Fig. 2. The signal generator in a Bruel and Kjaer Type 2032 dual channel signal analyzer provided the control system reference as well as harmonic excitation for the double panel system. Note that the controller convergence parameter, which determines the rate of convergence, and the sampling frequency were varied to provide a stable system.

Three microphones placed in the receiving chamber provided error signals for the ontroller. Error microphone sensitivities were calibrated using a reference of 114 dB to provide uniform weighting of the error signals to the controller. Control inputs, in the form of three PZT actuators, were placed on the radiating and incident panels as described subsequently. Incident panel PZT locations, shown in Table 2, were placed by Zhou[7] with locations chosen by Clark and Fuller[11] to allow coupling with odd modes in the x and y directions. Radiating panel PZT locations, shown in Table 2, were chosen as

Table 2 PZT central locations, m

PZT	Incident panel	Radiating panel
1	0.305, 0.144	0.127, 0.200
2	0.076, 0.056	0.190, 0.150
3	0.191, 0.240	0.082, 0.060

follows: PZT 1 was located over the node lines for the higher order odd modes and, therefore, unable to couple into these modes. PZT 2 was positioned orthogonal to higher order even modes, whereas PZT 3 was located to couple into both higher order odd and even modes. At each PZT location, two PZTs were mounted on each side of the panel and wired out of phase to allow control inputs to produce pure bending about the neutral axis of the panel. As described by Dimitriadis and Fuller[12] this configuration produce line moments located at the edges of the actuator.

To perform a detailed vibrational analysis of the radiating panels, a scanning laser vibrometer was used to measure the out-of-plane vibration velocity of the panel in a grid of 32 points along the x axis and 20 points along the y axis. A calibrated accelerometer was positioned at one of the scan points on the panel, which was used as the reference signal for the vibrometer. This arrangement allows the calibration of the vibration measurements in engineering units (m/s^2). These detailed vibrational measurements were also used to perform a modal decomposition of the radiating panel response to determine the effects of active control on the various double panel system configurations.[13]

Experimental Procedure

Experiments were performed on the two different radiating panels, at on- and off-resonance frequencies. Several configurations were tested, in an effort to determine the influence of control system configuration, radiating panel stiffness, excitation frequency, and incident acoustic field. The testing procedures discussed were applied to all test cases, therefore only one will be described in detail.

For all cases, the system was set up in the prescribed configuration, then the microphones were calibrated to a reference of 114 dB. A system identification was performed for use in the filtered-X LMS control algorithm. The speaker in the source chamber was turned on to excite the double panel system, producing a pressure level of approximately 85–95 dB at the receiving chamber traverse microphone, which is approximately 30–40 dB above the background noise level in the receiving chamber.[7] Acoustical measurements of uncontrolled acoustic power level were taken in the source and receiving chamber. In the source chamber, the acoustic power level was determined at the incident panel by a single microphone mounted in the center of the panel for normal plane wave and oblique plane wave excitation.[14] In the case of the reverberant excitation, the source chamber microphone was traversed across the chamber and sampled at 10 evenly spaced points to determine the acoustic power level in the room. In the receiving chamber, the acoustic power level was determined by a microphone traversed across the chamber and sampled at 10 evenly spaced points for all excitation types.

After the uncontrolled acoustical measurements were made, the control system was turned on to minimize the error signals. Once the control system stabilized, the convergence parameter was set to zero, preventing any further adaptation due to uncorrelated influences such as the movement of the microphone traverse. Acoustical measurements for the controlled case were taken as described for the uncontrolled case. Additional acoustical measurements, provided by the error microphones, were also recorded for the uncontrolled and controlled cases to determine the amount of reduction at these points.

Vibrational measurements using the laser vibrometer were taken after the acoustical measurements were completed. While the adaptive filters were locked at the optimal values, the laser vibrometer was placed in the receiving chamber perpendicular to the radiating panel and the scanning mechanism was centered on the panel. As explained earlier, an accelerometer was positioned at one of the scan points and fed into the vibrometer as a reference signal. With the acoustic source and controller still operational, the vibrometer scanned the radiating panel for the controlled case. The control system was then turned off and the uncontrolled vibration data was taken.

Table 3 Reverberant chamber characteristics

Parameters	Source chamber	Receiving chamber
Volume, m^3	47.76	49.67
Reverb time at 500 Hz, s	6.11	7.63
Reverb radius, m	0.16	0.14
Cutoff frequency, Hz	588	648

Reverberant Chamber Characteristics

The reverberation chamber has certain characteristics which must be taken into account due to the reflection of sound sources off of the walls of the room. The total pressure measured at a point in the reverberant environment can be approximated as the sum of the direct sound field radiated by the source and the effective pressure of the reverberant field, given by

$$P^2 = \rho_0 c \Pi \left(\frac{1}{4\pi r^2} + \frac{4}{A} \right) \tag{1}$$

where $\rho_0 c$ is the impedance of air, Π the acoustic output of the source in watts, r the radial distance from the source, and A the total sound absorption of the room.[14]

An important parameter in the choice of placing the source to achieve plane wave excitation is the radius of reverberation, defined as the radius where the direct field and the reverberant field contribute equally to the total pressure. Points located very close to the source, $4\pi r^2 \ll (A/4)$, are dominated by the direct field, whereas points located very far from the source, $4\pi r^2 \gg (A/4)$, are dominated by the reverberant field. By setting the contribution of the direct field equal to the reverberant field and substituting the definition of reverberation time ($T = 0.161V/A$, where V is the volume of the reverberation chamber) the radius of reverberation can be found to be

$$R_0 \approx 0.1\sqrt{V/T\pi} \tag{2}$$

A summary of example reverberation chamber characteristics at 500 Hz is shown in Table 3.

An important consideration in performing the measurement of acoustic power level in a reverberant environment is the Schroeder cutoff frequency, defined as the frequency above which the sound field is theoretically diffuse, where sound travels uniformly in all directions. Below this frequency, standing wave patterns can exist in the room. The Schroeder cutoff frequency is defined as[15]

$$f_s = 2000\sqrt{T/V} \tag{3}$$

The Schroeder cutoff frequencies of both the source and receiving chambers are presented in Table 3. Since the excitation frequencies are below the cutoff frequency of the receiving chamber (648 Hz), an average of sound pressure across the chamber must be used to get a reasonable estimate of the total radiated power.

Results and Discussion

Although a full matrix of test cases were experimentally performed, only illustrative results will be presented due to the extensive amount of information collected.

Preliminary Results: Double Panel Behavior

Preliminary frequency response testing of the panels was performed to determine the on- and off-resonance frequencies of excitation, mentioned earlier. Tests were performed on the panels clamped in the steel frame of the transmission loss test facility. Spectrally white excitation approximated by an impulse from a modal hammer disturbed each panel. Panel response was measured with an accelerometer, while the signals were processed by a Bruel and Kjaer Type 2032 dual channel signal analyzer. Frequency response

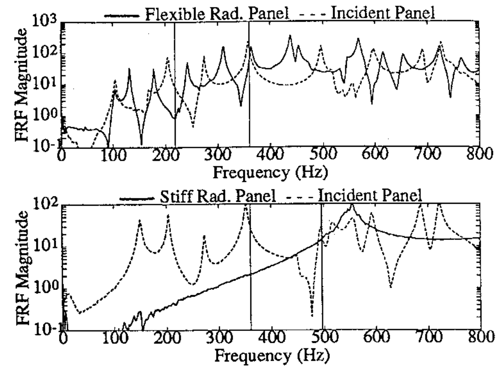

Fig. 3 Flexible and stiff double panel system response.

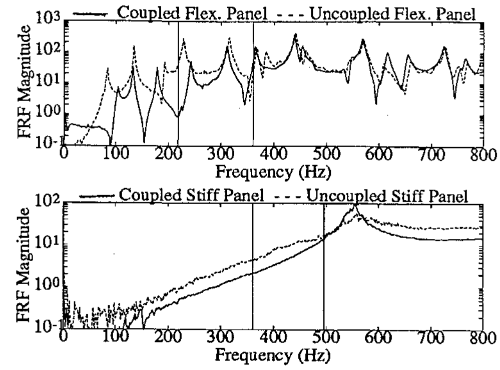

Fig. 4 Flexible and stiff panel coupling.

functions were taken for two double panel systems, consisting of the incident panel (aluminum panel), and either the flexible radiating panel (G10 fiberglass) or the stiff radiating panel (sandwich board), which is shown in Fig. 3. Frequency response functions were also taken for a single panel system, consisting of the flexible panel (G10 fiberglass), or the stiff panel (sandwich board). Comparisons of the coupled and uncoupled radiating panels are presented in Fig. 4. A summary of panel properties was previously presented in Table 1.

The two panels were excited on-resonance and off-resonance, as determined by the double panel system frequency response function, shown in Fig. 3. The flexible double panel system was excited on-resonance at 361 Hz, corresponding to a region where both the incident panel and the radiating panel show resonance behavior. Off-resonance excitation was chosen to be 218 Hz, which does not correspond to a resonance for both panels. Because of the high stiffness of the sandwich board radiating panel, both on-resonance and off-resonance conditions were chosen below the fundamental frequency of the radiating panel. The on-resonance case, 496 Hz, is close to the fundamental frequency of the stiff radiating panel, and is at a resonance of the incident panel. Off-resonance excitation was chosen to be 361 Hz, which is well below the fundamental of the radiating panel, and off a resonance peak for the incident panel as well.

Another issue was the amount of panel-cavity-panel coupling the individual panels exhibited when placed in a double panel system. As can be seen in Fig. 4, there is a significant difference in the individual panel response and the coupled panel response of the flexible panel. Below 300 Hz, the panel fundamental resonance frequency (FRF) exhibits a shift in resonance frequencies and the density of these frequencies. In particular, the fundamental resonance frequency was shifted from 85 Hz to 102 Hz. The incident panel experiences the same type of frequency shift for the fundamental resonance, decreasing from 118 Hz to 102 Hz. From this behavior, it is evident that the incident panel and the flexible panel are highly coupled. The response for the stiff panel, also shown in Fig. 4, does not display a shift in frequency as seen for the flexible panel, but displays a difference in the sharpness of the fundamental resonance peak. The uncoupled stiff panel response displays a

Table 4 Incident vs radiating panel control performance, 218 Hz, flexible panel, reverberant

Controlled panel	Incident	Radiating
Increase in TL, dB	0.3	5.4

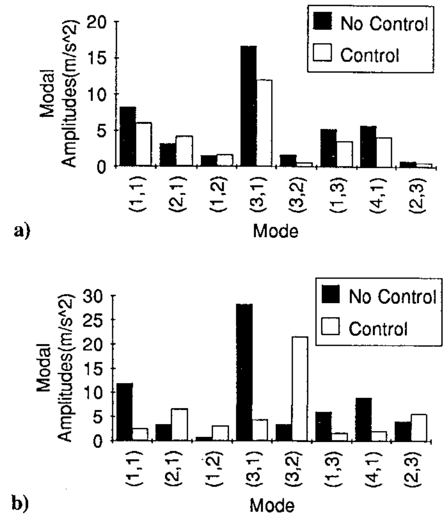

Fig. 5 Modal decomposition of flexible radiating panel, excited reverberantly off-resonance, 218 Hz: a) control applied to incident panel and b) control applied to radiating panel.

significant amount of structural damping, evident from the flatness of the uncoupled response. With the addition of the incident panel, the stiff panel response exhibits a decrease in the system damping. Overall, the incident panel has little effect on the stiff panel.

Control System Configuration

To study the influence of the control system configuration, the control inputs were applied to the incident panel and then the radiating panel, while holding all other parameters constant; the double panel system (with the flexible radiating panel) was excited reverberantly and off-resonance at 218 Hz. Acoustical results indicating the increase in transmission loss (TL) due to the implementation of control are presented in Table 4. Modal decompositions of the radiating panel response for the control inputs applied to the incident panel and the radiating panel are presented in Figs. 5a and 5b, respectively.

Control of the radiating panel exhibits better overall increase in transmission loss by 5 dB over control of the incident panel, shown in Table 4. Figure 5a demonstrates that when the control inputs are applied to the incident panel, the radiating panel response shows a reduction in the amplitude of the efficient acoustic radiating modes [(1,1) and (3,1)], as well as most of the other modes. This type of change in the modal structure of the panel is referred to as modal suppression.[13] In effect the controlled incident panel is decreasing the source strength transmitted across the air cavity to the radiating panel.

Conversely, Fig. 5b shows that when control inputs are applied to the radiating panel, the reduction of the (1,1) and (3,1) modes (the most efficient acoustic radiators) is more extreme, and there is a sharp increase in the level of the (3,2) mode. This type of change in the modal structure of the panel is referred to as modal restructuring, where the controller restructures the panel modal response into less efficient acoustic radiators.[13]

The differences in acoustic performance stem from the ability of the control system to affect the structural-acoustic coupling between the radiating panel and the receiving chamber. Control inputs that can directly affect the structural-acoustic coupling (applied to the radiating panel) can control just the efficient acoustic radiators (modal restructuring) whereas control inputs that cannot directly affect the structural-acoustic coupling (applied to the incident panel) must control all of the excited modes (modal suppression). There-

Table 5 Flexible vs stiff radiating panel control performance, plane wave

Panel	Flexible	Flexible	Stiff	Stiff
Frequency, Hz	218	361	361	496
Resonance	no	yes	no	yes
Dominant mode	(3,1)	(1,3)	(1,1)	(1,1)
Uncontrolled TL, dB	30.8	30.5	43.6	32.1
Controller increase in TL, dB	10.1	11.1	4.6	20.4
Overall TL, dB	40.9	41.6	48.2	52.5

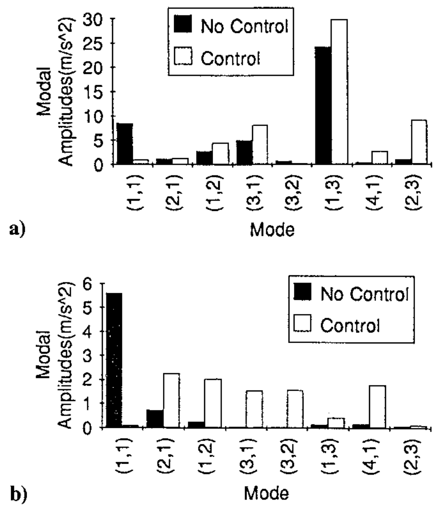

Fig. 6 Modal decomposition of radiating panels, plane wave excitation: a) flexible radiating panel, on-resonance, 361 Hz and b) stiff radiating panel, off-resonance, 361 Hz.

fore, for a control system with a finite number of control channels, modal restructuring is more efficient at reducing sound transmission than modal suppression due to the relatively lower number of system modes that require control (i.e., efficient acoustic radiators vs all modes).

Radiating Panel Stiffness

To study the influence of radiating panel stiffness, the double panel system was tested with a flexible radiating panel then a stiff radiating panel. Other test parameters were held constant; the double panel system (with the control inputs applied to the radiating panel) was excited by a normal plane wave acoustic field. Both radiating panels were tested at double panel system resonance and off-resonance frequencies. Acoustical results are presented in Table 5, which contains TL data for the flexible and stiff radiating panels. Modal decompositions of the flexible and stiff radiating panels excited at a frequency of 361 Hz are presented in Figs. 6a and 6b, respectively.

Figure 6a demonstrates that the uncontrolled flexible radiating panel has a high modal density, which is indicated by several modes contributing to the panel response, specifically the resonant (1,3), (1,1), and the (3,1) modes. Conversely, Fig. 6b demonstrates that the uncontrolled response of the stiff panel has one significant mode, the (1,1), which indicates a panel with a low modal density.

A panel of low modal density has a fewer number of significant degrees of freedom (in terms of response) and, therefore, better acoustic attenuation is attained with three channels of control. This is indicated in Table 5, which shows that control of a stiff vs a flexible radiating panel excited on-resonance results in an increased overall TL due to better control performance. In this test case, increased overall TL was 10.9 dB.

However, a panel of low modal density also has a greater stiffness to weight ratio than a panel of high modal density, which increases uncontrolled TL. As shown in Table 5, a comparison of flexible and stiff panels excited off-resonance reveals that uncontrolled TL for the stiff radiating panel is 12.8 dB higher than the flexible radiating panel. However, the increased stiffness reduces the effect the control actuators have on panel response and, although overall transmission loss is higher, the amount due to active control is reduced.

Overall, a double panel system with a stiff radiating panel shows greater overall TL compared to one with a flexible radiating panel.

Table 6 Excitation frequency control performance, stiff panel, plane wave

Frequency, Hz	361	496
Resonance	no	yes
Increase in TL with control, dB	4.6	20.4

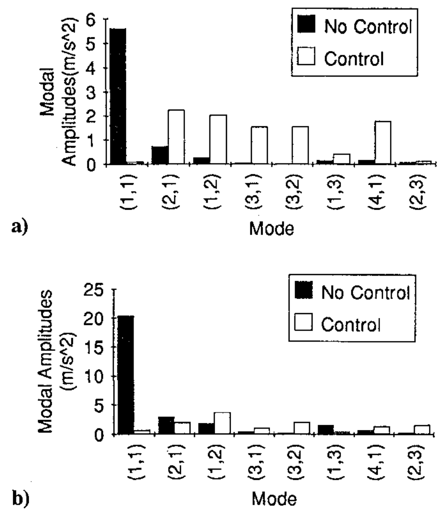

Fig. 7 Modal decomposition of stiff radiating panel, plane wave: a) double panel system off-resonance, 361 Hz and b) double panel system on-resonance, 496 Hz.

Excitation Frequency

A comparison of the controlled double panel system response excited off-resonance and on-resonance is presented. For this test, the double panel system (with a stiff radiating panel) was excited by a plane wave acoustic field at 361 Hz (off-resonance) and 496 Hz (resonance). Acoustical results indicating the increase in TL due to the implementation of control are presented in Table 6. Modal decompositions of the stiff radiating panel excited off-resonance and on-resonance are presented in Figs. 7a and 7b, respectively.

Control of the double panel system excited on-resonance was more effective than one excited off-resonance by 15.8 dB, as seen in Table 6. The modal decomposition of the stiff radiating panel, shown in Figs. 7a and 7b, indicates that the uncontrolled response of the (1,1) mode is significantly higher when excited on-resonance compared to off-resonance.

Controlled response of the double panel system excited off-resonance, shown in Fig. 7a, shows a significant reduction in the (1,1) mode along with significant increases in most of the other modes. However, controlled response of the system excited at resonance, shown in Fig. 7b, shows little or no increase in the other modes with a significant reduction in the (1,1) mode.

It is evident that the dominance of the (1,1) mode when excited on-resonance allows the reduction of this mode without significantly increasing the response of the other modes.

Incident Acoustic Field

To study the influence of the incident acoustic field, the speaker was placed in three different locations indicative of normal plane wave, oblique plane wave, and reverberant excitation, holding all other parameters constant; the double panel system with the control inputs applied to the flexible radiating panel was excited off-resonance at 218 Hz. Acoustical results indicating the increase in TL due to the implementation of control are presented in Table 7. Modal

Table 7 Excitation field control performance, 218 Hz, flexible panel

Excitation	Plane wave	Oblique plane wave	Reverberant
Increase in TL with control, dB	10.4	8.2	5.4

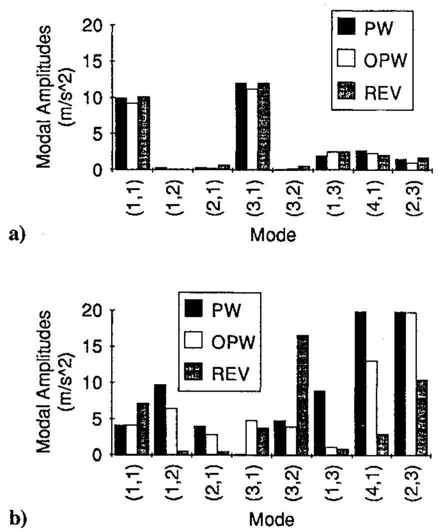

Fig. 8 Modal decomposition of flexible radiating panel, off-resonance, 218 Hz: a) uncontrolled and b) controlled.

decompositions of the uncontrolled and controlled flexible radiating panel response are presented in Figs. 8a and 8b, respectively. Note that PW denotes normal plane wave, OPW denotes oblique plane wave, and REV denotes reverberant excitation in Figs. 8a and 8b.

Control performance for a double panel system excited by a plane wave acoustic field is better by 2.2 dB over an oblique plane wave acoustic field and 5.0 dB over a reverberant acoustic field, as seen in Table 7.

As seen in Fig. 8a, all three acoustic excitation fields, PW, OPW, and REV, exhibit similar uncontrolled modal response, accounting for variations in source strength. Note that the (1,1) and (3,1) modes were primarily excited.

However, controlled modal response shows significant differences due to excitation field type, as seen in Fig. 8b. Reductions in the (1,1) mode for normal plane wave and oblique plane wave incident fields are similar, whereas reductions in the (1,1) mode for reverberant incident field was less. Reductions in the (3,1) mode for plane wave incident field was good, whereas reductions in the (3,1) mode for and oblique plane wave and reverberant incident fields was less. It is evident that the good control performance of the double panel system excited by a normal plane wave acoustic field is due to the reduction of the two most efficient acoustic radiators, the (1,1) and (3,1) modes. Oblique plane wave acoustic field control performance was slightly less due to the poor reduction of the (3,1) mode. Reverberant acoustic field control performance was less due to the poor reduction of the (1,1) and (3,1) modes (compared to the normal plane wave case).

Although the results present the reason for the poor control performance in terms of the radiating panel response, it is evident that further research must be done to investigate the incident panel response to determine the influence of the incident acoustic field on double panel system behavior.

Conclusions

In this experimental study of the behavior of active control of sound transmission through double panel systems, several important conclusions have been reached.

1) The application of control inputs to the radiating panel resulted in greater transmission loss due to its direct effect on the nature of the structural-acoustic coupling between the radiating panel and the receiving chamber.

2) Use of a stiff radiating panel in a double panel system resulted in increased overall attenuation due to both passive and active effects.

3) As expected, control of the radiating panel excited on-resonance was more effective than one excited off-resonance.

4) Further research must be done to determine the influence of the incident acoustic field on double panel system behavior.

This research has shown the potential for applying the control inputs to the internal trim of the fuselage as well as the advantage of a double panel system with a low modal density.

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