Influence of a Floor on Sound Transmission into an Aircraft Fuselage Model

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A simplified cylindrical model of an aircraft fuselage is used to study experimentally the effect of an internal floor on low-frequency sound transmission into aircraft cabins. A geometrically scaled lattice floor support and floor skin are designed based on selected characteristics of a business aircraft. The pressure amplitude at various interior locations is presented along with a modal decomposition of the associated shell response. Results indicate that the main effect of the floor on interior pressure levels is due to a modification of the interior acoustic mode shape and not due to the structural modification of the fuselage model caused by the lattice floor support. Thus, the model provides a simplified procedure for studying the influence of various structural modifications as well as other important effects.

Nomenclature

a	= radius of test cylinder, 0.254 m
A_n	= complex modal amplitude coefficients, Eq. (1a)
B_n	= complex modal amplitude coefficients, Eq. (1b)
f	= frequency (Hz)
m	= number of axial half waves
n	= circumferential mode number
N_p	= number of measuring positions
p	= $1, 2, 3, \ldots, N_p$
r, θ, x	= cylindrical coordinates
w	= radial displacement
ϵ_n	= Neumann factor: $\epsilon_o = 1$; $\epsilon_n = 2$, $n > 0$
$\Delta\theta_p$	= $2\pi / N_p$

Introduction

INTEREST has steadily increased over the use of advanced turboprop (ATP) engines in commercial aircraft due to their potential 25% reduction in fuel consumption over today's state-of-the-art turbofan engines. However, associated with the use of ATP engines are unacceptably high acoustic levels near the propeller plane due to higher tip speeds and large amplitude structural vibrations induced on the wings by engine harmonics and propeller wakes.[1] Previous investigations have established that low-frequency sound transmission into aircraft cabins is directly coupled to the modal vibration of the fuselage.[2,3] Therefore, any structural modification to the fuselage is expected to have a significant impact on sound transmission into the aircraft interior. Since the cabin floor represents one of the major structural modifications to the fuselage, an experimental investigation was designed to study the influence of an internal floor on low-frequency sound transmission into aircraft cabins.

A simplified model of an aircraft fuselage was used in a controlled environment to study the effect of an internal floor. A geometrically scaled lattice floor support and floor skin were designed and built based on selected fuselage characteristics of a business aircraft.[4] Scaling of the mass, damping, and stiffness properties of the floor was not considered because the cylindrical test model was not designed to reproduce exactly the characteristics of an actual aircraft fuselage. Rather, the test model was intended as a simple approximation of an aircraft fuselage for the purpose of identifying and studying the fundamental influence of the floor on sound transmission into the enclosed acoustic cavity. The mechanisms behind the results are discussed and related to a companion theoretical investigation.[5] Thus, this study has led to an enhanced understanding of the transmission phenomena in more realistic configurations than previously considered.

Experimental Setup and Procedure

A schematic diagram of the experimental setup showing model details and microphone locations is presented in Fig. 1. The aircraft fuselage was modeled as a finite unstiffened aluminum cylinder 0.508 m in diameter and 1.245 m long. The cylinder was formed from a 1.63 mm thick aluminum sheet and has an epoxy-bonded butt-joint seam with a 5 mm wide exterior strap. The cylinder was sealed at both ends with 1.9 cm thick wooden end caps and was freely supported at the ends. The ends of the cylinder were damped with 15.3 cm wide strips of Soundcoat vibration damping material around the circumference of the cylinder. This reduced the axial standing wave of the radial vibration of the cylinder and helped simulate the decay in structural vibration with increasing distance from the source plane found in actual aircraft fuselages. The interior sound field was acoustically damped by attaching 52 cm × 35.6 cm × 2.54 cm thick foam to each end of the upper cavity of the cylinder. The foam was used to study the influence of acoustic absorption on sound transmission into the model. Acoustic damping is similarly found in actual aircraft in the form of seating and interior trim. A single noise disturbance was used in this investigation as shown in Fig. 1 to eliminate additional effects due to source phasing (i.e., propeller synchrophasing). The noise disturbance, which modeled the propeller acoustic source, was com-

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posed of a pair of 60-W University Sound driver units and had radiation properties similar to a monopole source. The separation distance from the shell corresponded to the actual scaled propeller distance from the fuselage and provides a realistic pressure distribution on the exterior of the cylinder. The experiments were performed in an anechoic environment to represent the free-field conditions of flight.

Three 6-mm-diam condenser microphones were mounted on an interior traverse mechanism at radial positions $r/a = 0.150$, 0.513, and 0.925. The microphone cables were passed through a hole in one of the wooden end plates, which was subsequently sealed with modeling clay. These three microphones were used to evaluate the axial, radial, and circumferential pressure distributions inside the cylinder. Another 6-mm-diam microphone was used to measure the exterior noise incident upon the cylinder. Nine small accelerometers were used to monitor the response of the shell due to the source excitation at 19 equally spaced base positions (i.e., accelerometer measurement points) around the circumference of the shell in the source plane. This allowed evaluation of the modal decomposition of the cylinder response. All microphone and accelerometer signals were in-turn conditioned and then processed using a fast Fourier transform analyzer. The cutoff frequency was set to 1500 Hz giving a frequency resolution of 4.8 Hz. A phase meter and oscilloscope were used to monitor all transducer signals.

Figures 2a and 2b are photographs of the test cylinder with the lattice floor support and the complete floor installed, respectively. The floor was constructed so that individual components could be separately installed. Thus, with just the lattice in place, the effect of the "floor" was purely a structural influence on cylinder response, being acoustically transparent. With the complete floor installed, the effect of interior cavity modification on system response could be included. When the complete floor was installed, a layer of 2.54 cm thick acoustical foam, 1.2 m × 0.25 m, was then placed on the interior of the cylinder below the floor to damp acoustically the lower cavity.

Figure 3 shows the coordinate system used in this investigation. The interior microphones were initially positioned in the source plane ($x/a = 0.0$) at $\theta = -38$ deg. This angular position corresponds to a location just above the floor. While in the source plane, interior pressure measurements were recorded at seven circumferential positions, $\theta = -38$, 0, 45, 90, 135, 180, and 218 deg. Interior pressure measurements were also recorded along $\theta = 0$ deg at axial positions of $x/a = 0.0$, ± 0.2, ± 0.4, ± 0.8, ± 1.6 and ± 2.0. Exterior pressure measurements were recorded at identical axial positions just outside the cylinder along $\theta = 0$ deg at a distance of approximately 6 mm (1/4 in.) from the shell wall. All microphone pressure measurements and cylinder response measurements were recorded for the three main model stages of the unmodified cylinder, the cylinder with the lattice floor support only, and the cylinder with the lattice floor support and the skin (i.e., the complete floor). Pressure measurements were also recorded for two additional model stages with Soundcoat vibration damping material attached to the floor and with a layer of 2.54 cm thick acoustic foam laid on the floor over the damping material. The cylinder response was recorded only for the later of these two additional stages. The experimental investigation was repeated for frequencies of 576 and 680 Hz. These frequencies were chosen because they were within a range of typical scaled fundamentals of the propeller noise.

Modal Decomposition of Shell Vibration

The radial vibration response of the cylinder was measured for the decomposition algorithm to determine the modal structure of the cylinder response. The relative amplitude and phase of the radial shell vibration were measured at 19 equally spaced base positions around the cylinder in the source plane ($x/a = 0.0$). Results from the decomposition algorithm define the relative modal structure of the cylinder vibration, thereby enabling changes in the cylinder response to be determined for the model with various stages of the floor installed. A knowledge of the modal structure of the cylinder response is essential in undestanding the effects of structural modifications on low-frequency sound transmission into the fuselage model.

The decomposition technique used in this investigation is the same as used by Jones and Fuller.[2] The relative modal

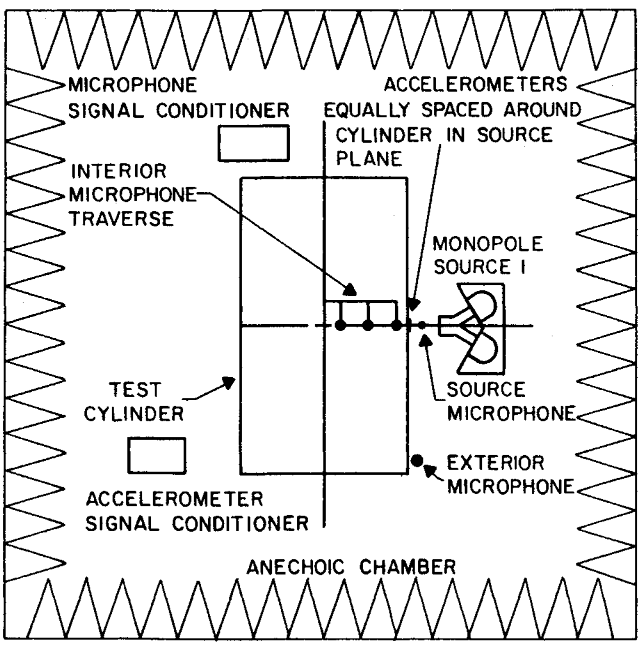

Fig. 1 Schematic diagram of experimental setup.

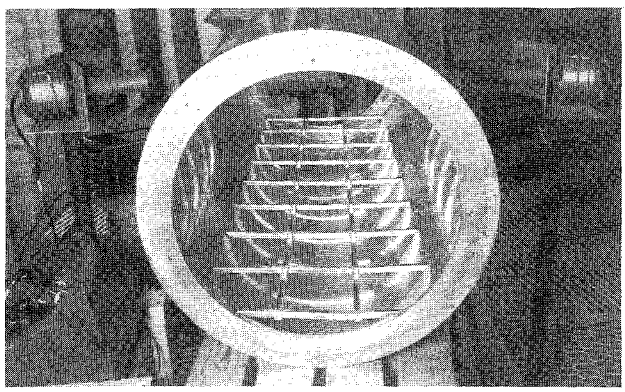

Fig. 2a Cylinder with support lattice.

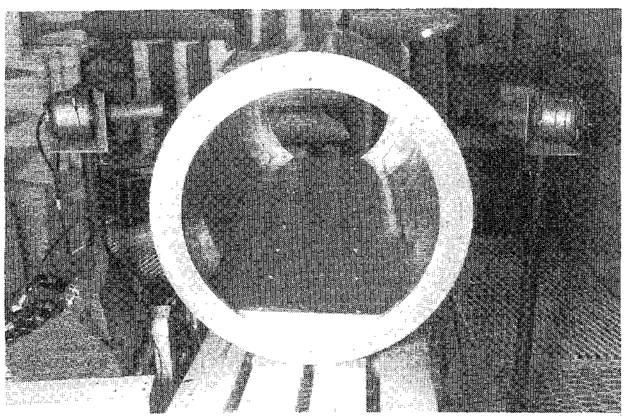

Fig. 2b Cylinder with complete floor.

amplitudes are represented as summations of the form

$$A_n = \frac{1}{\epsilon_n \pi} \sum_{p=1}^{N_p} w(\theta_p) \cos(n\theta_p)\Delta\theta_p \tag{1a}$$

$$B_n = \frac{1}{\epsilon_n \pi} \sum_{p=1}^{N_p} w(\theta_p) \sin(n\theta_p)\Delta\theta_p \tag{1b}$$

where N_p is the number of circumferential base positions and $\Delta\theta_p = 2\pi/N_p$ for equally spaced base positions.

With the lattice floor support installed, the stiffened test cylinder is structurally modified; thus, the modes of vibration of the stiffened cylinder will vary from those of the unstiffened cylinder. Nevertheless, the assumed $\cos(n\theta)$ and $\sin(n\theta)$ distributions are still used in the decomposition model in order to identify clearly how the stiffened shell responds in comparison to the unstiffened shell. Hence, the modal decomposition results of the stiffened cylinder are representative of the amplitudes of the $\cos(n\theta)$ and $\sin(n\theta)$ distribution functions and not of the circumferential modes of vibration of the stiffened shell.

In practice, the assumed mode shapes are fitted to the measured data; thus, any measurement error or contributions from modes excluded from the decomposition model could cause serious errors in the results of the decomposition. The decomposition algorithm will always reproduce the measured data within the constraints of the system, however, exclusion of significant higher-order modes results in a folding back of these modes known as spatial aliasing. How the aliasing occurs (i.e., which higher-order modes fold back into which low-order modes) is dictated by the number of base points (i.e., accelerometer measurement positions).

In this investigation, the highest-order mode obtained from the decomposition model was the $n=9$ mode. Results indicate that the higher-order modes ($n>9$) were small enough to eliminate aliasing problems for most cases. Potential problems for some cases, however, are discussed.

Results and Discussion

The results presented here are for harmonic source conditions of 576 and 680 Hz. These two frequencies show contrasting results which indicate the major effect of the floor on cylinder response and corresponding interior pressure distribution.

Driving Frequency of 576 Hz

Figure 4 shows the axial pressure response on the exterior of the cylinder along $\theta = 0$ deg. The axial pressure distribution is similar to that measured on the exterior of an actual twin-engine turboprop aircraft fuselage.[6] The axial pressure response is symmetric about the source plane and decays about 18 dB by two cylinder radii. A similar distribution was measured in the circumferential direction but is not shown.

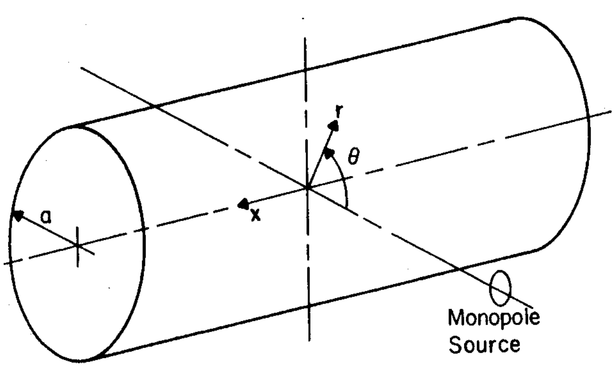

Fig. 3 Coordinate system.

Figure 5 shows the relative modal amplitudes of the cylinder in the source plane for the various model stages. Inspection of the structural characteristics of the shell[7] indicates the driving frequency is near the (3,4) structural resonance of the shell [i.e., the three axial "half waves" ($m=3$) and four circumferential waves ($n=4$)]. Hence, the strongly excited $n=4$ structural mode is not surprising. The presence of a significant B_4 mode for the unmodified stage is not predominantly due to modal coupling associated with the discontinuity of the rigid butt joint as discussed in a previous investigation.[8] If this were the case, suppression of the A_4 mode in the lattice stage would have resulted in reduction of the B_4 mode. However, the B_4 mode is most strongly influenced by insertion of the acoustically nontransparent skin, which, as is shown later, substantially modifies the contained acoustic sound field. As illustrated in a previous investigation,[8] the contained acoustic field can play a part in coupling or exciting structural modes in the shell. Thus the B_4 mode appears to be predominantly coupled by the interior acoustic field.

Previous work[9,10] has established that the lattice floor support causes a stiffening of the cylinder by essentially forcing nodal points at the floor attachments. Thus, insertion of the acoustically transparent lattice dramatically modifies the modal structure of the cylinder. The previously dominant A_4

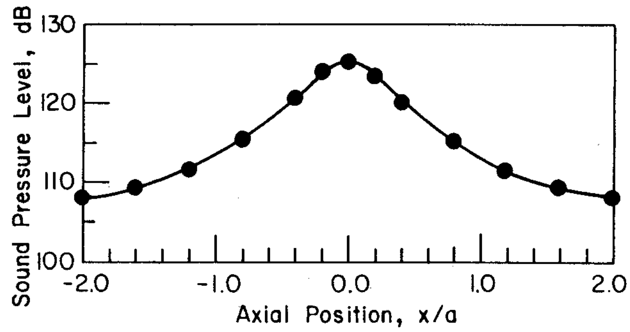

Fig. 4 Axial pressure distribution on exterior of the cylinder at $\theta = 0$ deg and $f = 576$ Hz.

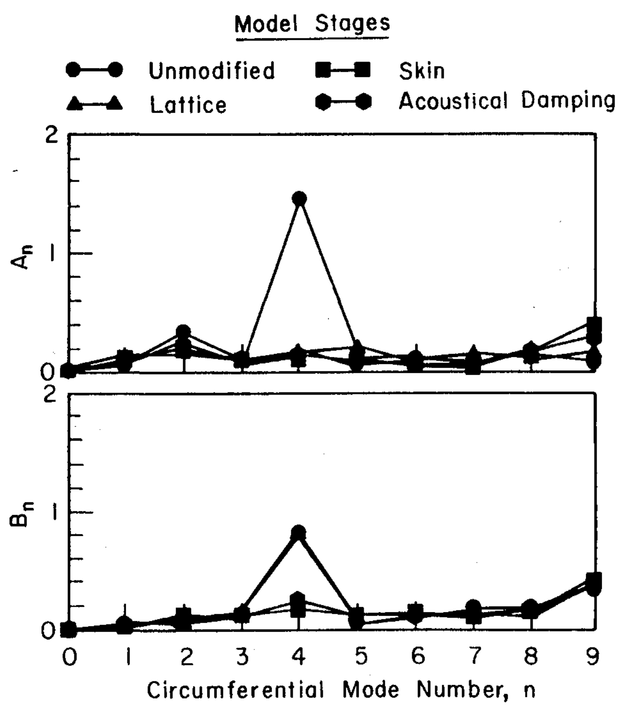

Fig. 5 Relative modal amplitudes of the cylinder at $x/a = 0.0$ and $f = 576$ Hz.

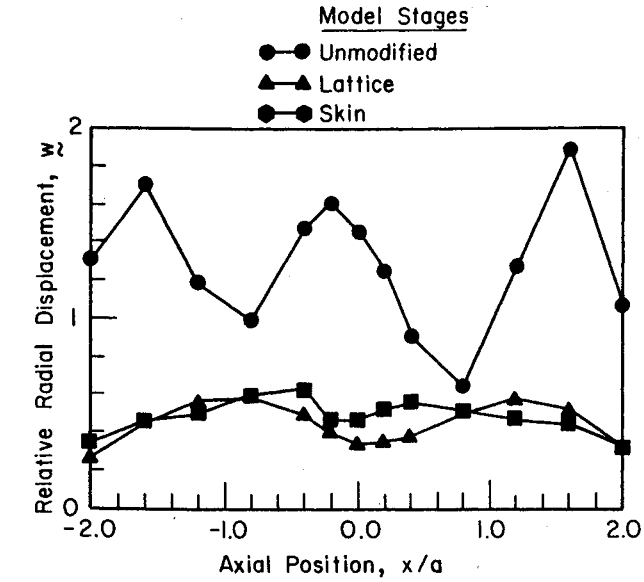

Fig. 6 Axial distribution of the radial response of the cylinder at $\theta = 0$ deg and $f = 576$ Hz.

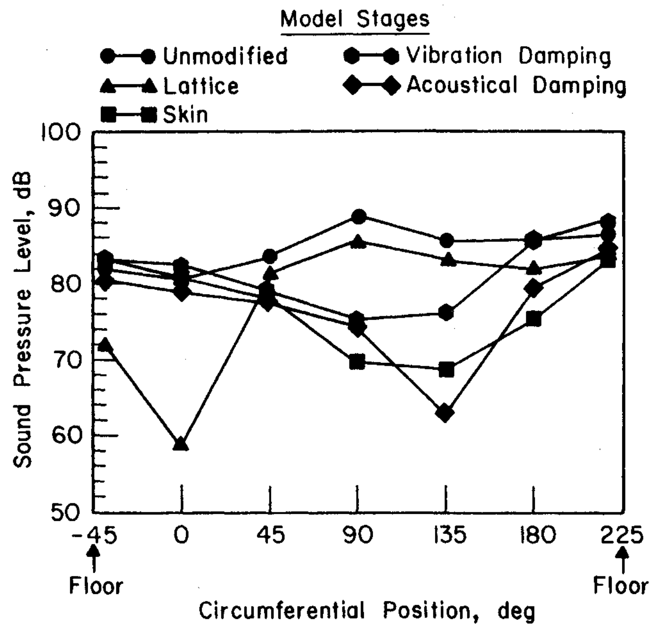

Fig. 7 Interior pressure response at $r/a = 0.513$, $x/a = 0.0$, and $f = 576$ Hz.

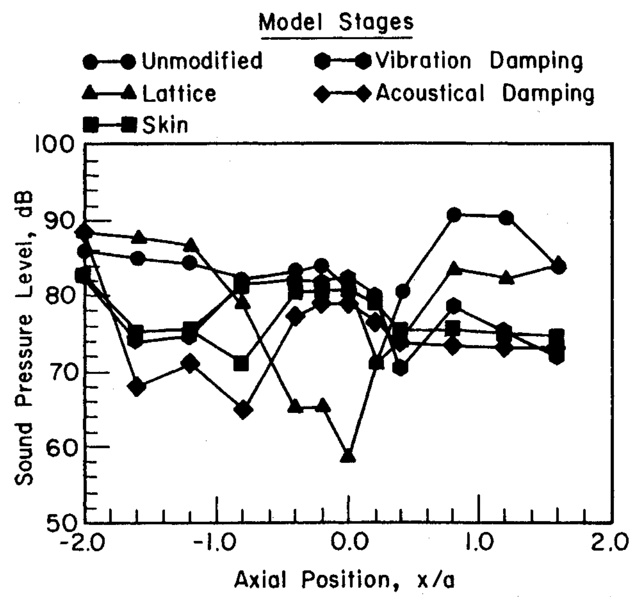

Fig. 8 Interior pressure response at $r/a = 0.513$, $\theta = 0$ deg and $f = 576$ Hz.

mode is almost completely suppressed by the insertion of the lattice. This result is not surprising considering that the angular location for the lattice attachments ($\theta = 45$ and 225 deg) corresponds to the antinodes to the A_4 mode. Insertion of the lattice causes a reduction in the input mobility of the A_4 shell mode. Near resonance, the input mobility is very sensitive to structural variations in the characteristics of the shell. Hence, insertion of the lattice results in a substantial reduction in the shell response. It is important to note that suppression of the dominant A_4 mode represents a global reduction in the vibrational energy in the shell and not just a redistribution of this energy. The B_4 mode remains unchanged for this stage since the position of the lattice attachments correspond to nodal points of the B_4 mode. Considering the dramatic reduction in the dominant A_4 mode, significant global reductions in the corresponding interior pressure response are expected.

Insertion of the acoustically nontransparent skin causes a modification of the interior cavity. This modification changes the structural-acoustic coupling between the cylinder and the contained sound field since the acoustic modes can no longer set up in a purely cylindrical coordinate system. The modified sound field changes the input impedance of the acoustic cavity resulting in the reduction of the controlling B_4 structural mode. Hence, the B_4 mode is no longer coupled via the contained acoustic field. With damping and acoustical foam in place, only minimal changes in the modal structure were recorded and thus have little influence on the structural-acoustic coupling phenomena.

As discussed above, insertion of the floor causes nodal points at the floor attachments. To satisfy the boundary conditions at the floor attachments, two levels of modal restructuring occur. First, the dominant level of modal restructuring is the direct suppression of the A_4 mode caused by the presence of the lattice as discussed above. This results in a significant reduction in the response of the shell at the floor attachment locations. However, contributions from the remaining modes still cause some vibration at the attachment points. To reduce the vibration levels even further, a second minor level of modal restructuring occurs. This level involves the generation higher-order circumferential modes in such a manner as to reduce the shell response to a minimum at the floor attachments. The generation of higher-order modes is minimal for this case since the dominant A_4 modes is suppressed by the insertion of the lattice. However, a noticeable increase in the $n = 9$ mode can be seen. Similar modal restructuring is observed in a companion theoretical investigation.[5] Since the generation of higher-order modes is minimal for this frequency, potential aliasing problems are insignificant.

Figure 6 shows the axial distribution of the radial response of the cylinder at $\theta = 0$ deg for the three main model stages. The axial distribution of the unmodified cylinder clearly has three axial half-waves in the standing-wave pattern, which is consistent with the assumed (3,4) resonant frequency in the shell. The response of the unmodified cylinder is as much as three times that of the cylinder with the lattice or the complete floor installed. This dramatic reduction in the cylinder response due to the lattice is expected to result in substantial global reduction in the corresponding interior pressure response. The variation in the cylinder response between the lattice and skin stages is negligible; however, this result is somewhat misleading. As dominant B_4 mode of the lattice

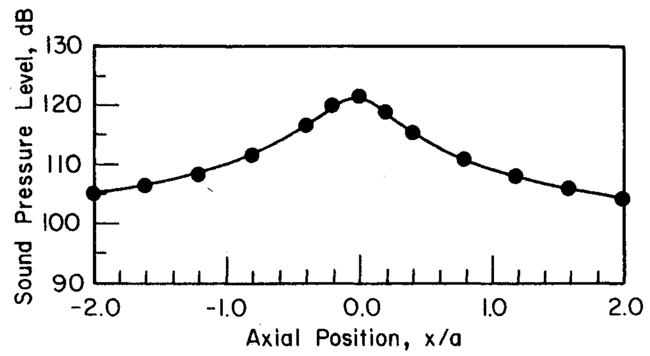

Fig. 9. Axial pressure distribution on exterior of the cylinder at $\theta = 0$ deg and $f = 680$ Hz.

stage has a node at $\theta = 0$ deg, the axial distribution of the radial response of the cylinder is somewhat higher than shown at $\theta = 0$ deg for most angular positions of lattice stage due to contributions from the B_4 mode. Regardless of this fact, one can deduce that the structural modification associated with the lattice stage of the floor has the largest influence on the shell response.

Figure 7 shows the interior pressure response in the source plane at radial station $r/a = 0.513$ over a range of circumferential positions for the various stages of the model. The change in the interior pressure levels between the unmodified cylinder stage and the lattice stage is less than 4 dB at angles away from the floor attachments. This result is surprising considering the dramatic reduction in cylinder response with the lattice in place as shown in Fig. 5. Analytical results from the companion investigation[5] support the idea of interface modal filtering (IMF), which is a characteristic of the structural-acoustic coupling between the shell and contained acoustic space. Although the vibrational response of the shell is often governed by a number of structural modes, the IMF effect essentially filters out the coupled pressure contributions from most of the higher-order structural modes. Hence, the interior pressure response is typically controlled by only one or two low-order acoustic modes. In other cases, when a single structural resonance governs the shell response, the coupled pressure field is frequently governed by a differing acoustic mode because of the IMF effect. Hence, the IMF principle is crucial in determining sound transmission characteristics into flexible cylinders. The dominant A_4 mode, which is suppressed by the insertion of the lattice, apparently does not effectively couple to the interior acoustic field. Thus, only minor changes in the corresponding interior pressure response result. This gives rise to uncertainty over the possibilities for significant noise reduction by structural modifications (i.e., stiffening the fuselage). This result is consistent with the conclusions of Pope et al.[11] and Prydz et al.[12], which indicate that stiffening the structure will not substantially influence the coupled acoustic response in the cavity. That is, structural modifications of the fuselage may produce dramatic reductions in the fuselage response under certain conditions but as demonstrated here, this reduction in fuselage response does not guarantee an equally dramatic reduction in the corresponding interior acoustic response.

With the skin inserted, noise reductions appear to be due to the modification of the interior acoustic cavity by the nontransparent skin. This modification changes the nature of the coupling between the shell wall and the acoustic space. In other words, the Bessel function coupling no longer governs the modal transmission of sound into the cavity because the interior acoustic space is no longer cylindrical. Hence, this cavity modification has a significant influence on the input mobilities of the coupled acoustic modes.

When structural damping material is attached to the floor the interior pressure levels increase by as much as 10 dB over

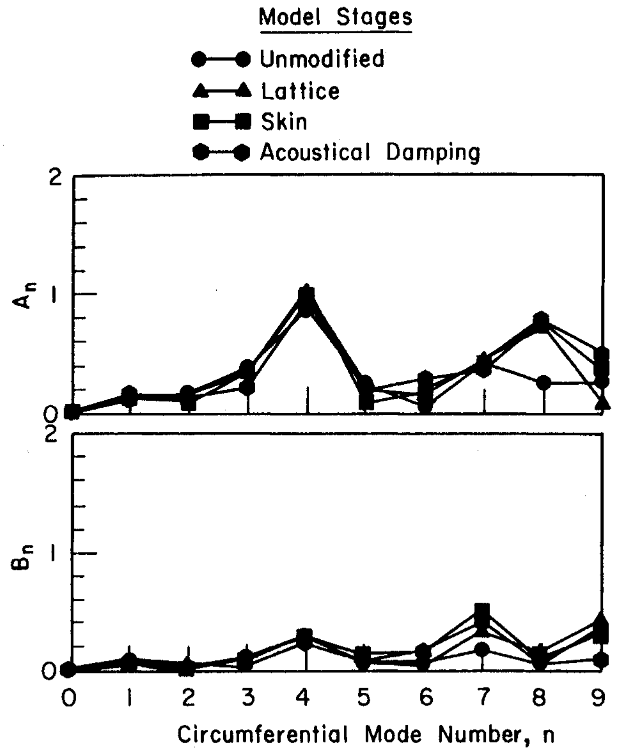

Fig. 10 Relative modal amplitudes of the cylinder at $x/a = 0.0$ and $f = 680$ Hz.

the skin stage at microphone location $x/a = 0.0$ and $r/a = 0.513$. This result is noticeable at all angular microphone locations in the source plane; however, the increase in the pressure response was generally closer to 1–2 dB at other locations. This result is surprising, considering that a measurement of the floor response revealed that the floor vibrational levels are comparable to those of the cylinder walls. This leads to the conclusion that, for this case, the floor is not a significant radiator to the interior cavity. In fact, results for this case indicate that the floor acts somewhat as an acoustical sink to the interior sound field.

A small reduction in the pressure response is observed with the acoustic foam placed over the damping floor. This result is, of course, due to an increase in the acoustic absorption due to the presence of the foam. However, as the reduction is not large, the results indicate that the interior acoustic field is not at resonance but being driven by the cylinder walls.

Figure 8 shows the internal pressure response along $\theta = 0$ deg over a range of axial positions for radial station $r/a = 0.513$. The pressure response of the unmodified cylinder generally does not change significantly with the lattice in place except near the source plane where a dramatic reduction is observed. The reduction, which is also present in Fig. 7, is comparable to the expected noise reduction, considering the dramatic global reduction in the cylinder response presented in Fig. 8. However, this dramatic reduction is very localized in both the axial and circumferential directions. Thus, the majority of the contained acoustic field is not significantly altered by the insertion of the lattice. The trends of the remaining stages of the model in the axial direction are generally similar to those in the circumferential direction. Hence, the results of Fig. 8 support the conclusions discussed earlier for Fig. 7.

Driving Frequency of 680 Hz

The remaining results presented are for a harmonic driving frequency of 680 Hz. Figure 9 shows the axial pressure distribution on the exterior of the cylinder along $\theta = 0$ deg. As in Fig. 4, the pressure distribution is representative of those measured on the exterior of an actual turboprop aircraft.

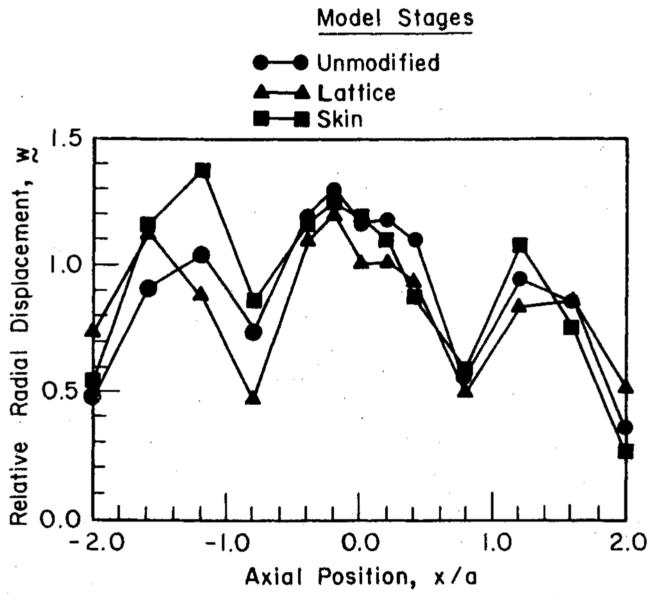

Fig. 11 Axial distribution of the radial response of the cylinder at $\theta = 0$ deg and $f = 680$ Hz.

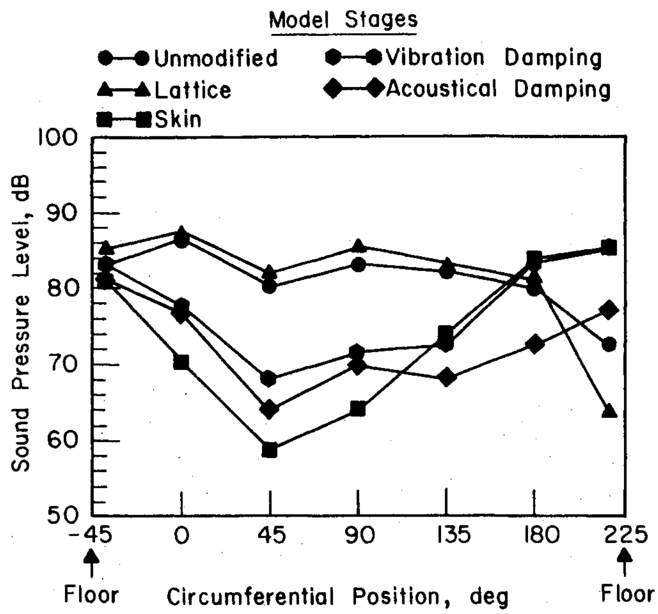

Fig. 12 Interior pressure response at $r/a = 0.513$, $x/a = 0.0$, and $f = 680$ Hz.

Figure 10 shows the relative modal amplitudes of the cylinder in the source plane for the various stages of the model. The modal response of the unmodified cylinder is again dominated by the $n = 4$ mode with subordinate contributions from the $n = 7$, 8, and 9 modes. In contrast to the 576 Hz case, driving frequence of 680 Hz represents a forced response in the shell. From Ref. 7, the unmodified shell response appears to be governed by the (3,4), (5,7), (5,8), and (5,9) structural modes, all of which have resonant frequencies below the driving frequency of 680 Hz. Results by Pope et al.[11] indicate that off-resonance response of the shell can be governed by a number of stuctural modes whose resonance frequencies are above and/or below the driving frequency. Hence, the results presented here are not surprising. The strong response in the (3,4) structural mode is probably due to distributed loading on the shell by the acoustic sources tending to excite lower-order structural modes more effectively. The presence of the B_4 mode for this case is due to the shell seam, as discussed previously.

With the lattice installed, the dominant A_4 mode remains unchanged; however, there is a substantial increase in the $n = 7$, 8, and 9 modal distributions. This result is distinctly different from the results of $f = 576$ Hz and merits further discussion. The lack of change in the A_4 mode is probably due to the insensitivity of its modal response to structural modifications under forced conditions. Boundary conditions due to the floor require that the response of the cylinder approaches a nodal value near the floor attachments. This was checked by measuring the radial response at the floor location points and found to be small. This does not necessarily require that each mode generated has a node at the floor attachments, but that the sum of all of the modes generated result in a node in the total shell response at the floor attachment points. Thus, rather than suppressing the dominant A_4 modes as in the 576 Hz case, the lattice causes the generation of significant higher-order modes in such a manner as to reduce the shell response at the floor attachments. That is, the relative phasing of the increased higher-order structural modes oppose that of the $n = 4$ mode thus reducing the shell response at the floor attachments based on superposition of the modal amplitudes. This results in an increase in the A_8 and B_7 structural modes, which are more easily excited by the localized influence of the rigid floor attachments. Interestingly, the enhanced excitation of these modes results in an actual increase in the vibrational response of the shell as the previously existing modes were essentially unaffected. Similar results were observed in the companion analytical investigation.[5] Generation of significant higher-order modes may lead to spatial aliasing problems that could contaminate the decomposition results. However, since the change in the lower-order modes is minimal for the various stages of the model, this contamination appears to be negligible.

As previously discussed, insertion of the skin causes a modification in the acoustic mode shape. However, only minimal change in the corresponding modal character of the shell is observed when the complete floor was installed at this frequency. This implies that off-resonant acoustic response in the cavity has little influence on the coupled shell response because of the inherently low input impedance of the cavity. Thus, the main effect of the floor on cylinder response is again due to the structural modification of the shell associated with insertion of the lattice. In contrast to the 576 Hz case, results of the modal decomposition at 680 Hz indicate that stiffening the fuselage through structural modifications does not guarantee a reduction in the vibrational response of the shell. In fact, at some frequencies, structural modifications may even enhance the shell response.

Figure 11 shows the axial distribution of the radial response of the shell at $\theta = 0$ deg for the three main stages of the model. In contrast to the dramatic reduction of the 576 Hz case, the variation in the radial shell response at 680 Hz is minimal for the three main stages. Thus, the floor has little influence on the radial shell response along $\theta = 0$ due to the "forced" response of the shell.

Figure 12 shows the interior pressure response in the source plane over a range of circumferential positions at radial station $r/a = 0.513$. As shown in Ref. 7, this frequency corresponds to an A_2 acoustic resonance within the cavity. However, the acoustic damping within the shell dramatically reduces the dominant A_2 acoustic resonance and hence makes the angular pressure distribution more uniform for all the stages of the model. The pressure response with the lattice in place varies only about 1 dB from the unmodified stage response at all circumferential positions except one. This result is unexpected, considering the generation of significant higher-order modes as presented in Fig. 10. Apparently, the higher-order modes do not couple well to the acoustic space due to IMFD characteristic of the structural-acoustic coupling as discussed previously. With the acoustically nontransparent skin in place, a reduction in the pressure response of about 10 dB occurs at most circumferential positions. Similar to the results of the 576 Hz case, the significant reduction occurs even though there is only a negligible change in the modal character of the cylinder. This indicates that the main effect of the floor on interior pressure response at this frequency is again due to a modification of the acoustic mode shape of the cavity.

Adding structural damping material layered over the floor, the interior pressure response increases slightly as in the previous case. Conclusions for this effect are the same as in the 576 Hz case (i.e., the floor appears to be acting as an acoustic sink). With acoustical foam over the floor, a moderate reduction in the pressure response is again recorded, due to the increase in acoustical absorption in the cavity.

Figure 12 shows the interior pressure response vs axial position along $\theta = 0$ deg at $r/a = 0.513$. The results of this figure are generally similar to those of Fig. 8, which indicate that the same general trends are present globally throughout the cylinder.

Figure 13 shows the noise reduction vs axial location for the various model stages. The results of this figure are generated by calculating the differences between the exterior pressure response (assumed constant for all stages) at a distance of 6 mm (¼ in.) from the shell wall and the interior pressure response at $r/a = 0.925$ for the various model stages. The exterior pressure field stays essentially constant for all model stages because the fraction of acoustical energy entering the shell is generally 20–30 dB lower than the acoustical energy reflected off the shell. Hence, small changes in the reflection coefficient of the shell will have an insignificant effect on the exterior pressure response. In other words, the shell response has little effect on the exterior pressure response.

The noise reduction of the shell for the unmodified and lattice stages is 20–30 dB, which is typical for thin flexible shells. As expected, the skin stage has the most significant impact on the noise reduction, increasing it by as much as 15–20 dB near the source plane. Similar results were obtained by Pope et al.[11] Reduction in the interior pressure response with the skin in place supports the IMF theory. That is, a change in the cavity shape relative to the forcing shape reduces the degree of structural-acoustic coupling and hence sound transmission into the acoustic field. Vibration and acoustical damping have little influence on the noise reduction but generally tend to decrease it slightly.

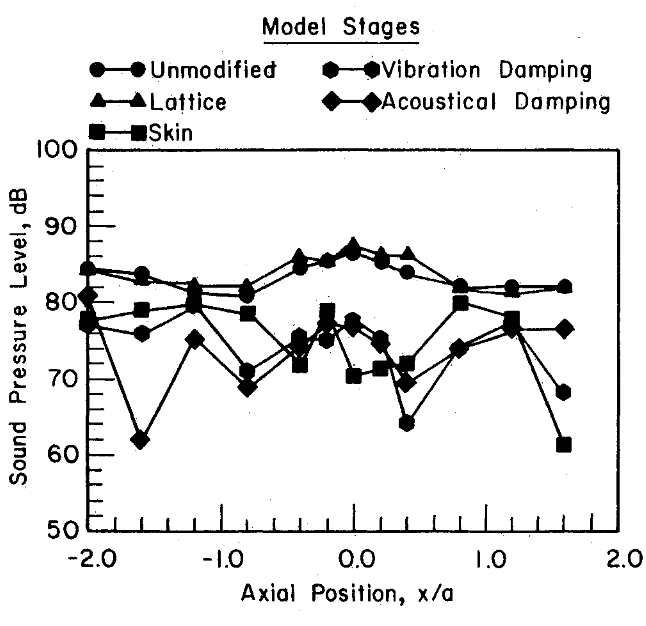

Fig. 13 Interior pressure response at $r/a = 0.513$, $\theta = 0$ deg, and $f = 680$ Hz.

The interior pressure response at radial stations $r/a = 0.150$ and 0.924 generally show similar trends as presented here for $r/a = 0.513$ for both driving frequencies. Hence, these results are not shown.

Concluding Remarks

A geometrically scaled lattice floor support and floor skin were designed and implemented in a simple aircraft fuselage model to study experimentally the effects of complex geometries on low-frequency sound transmission into aircraft cabins. The effects of the simulated cabin floor were shown to be frequency dependent, thus the results of two contrasting frequencies were presented that show the main influence of the floor on sound transmission into the model interior. Several important conclusions were identified.

1) The main effect of the floor on cylinder response was due to the structural modification associated with the lattice floor support. Near a structural resonance, the input mobility of the shell at the floor attachments was sensitive to structural modifications. Hence, substantial reductions in shell response were possible by stiffening the shell. In the case of "forced" response, the input mobility of the shell at the floor attachments was relatively insensitive to structural modifications. Hence, to satisfy the semirigid boundary conditions associated with the floor attachments, higher-order modes were generated in such a manner as to satisfy the modal boundary conditions. This higher-order excitation represented an increase in the structural response of the shell and not just a redistribution of the vibrational energy as the lower-order modes were essentially unaffected.

2) In contrast to intuition, significant reductions in the shell response did not guarantee an equally significant reduction in the global interior pressure response. In fact, this result will seldom be the case due to the IMF effect of the structural-acoustic coupling.

3) The floor skin was shown not to be a significant radiator to the interior acoustic field even though the vibrational levels are comparable to those of the shell walls. For the frequencies investigated, the floor appeared to behave as an acoustic sink, drawing acoustical energy from the enclosed cavity.

4) The main effect of the floor on interior pressure response was due to a modification of the interior acoustic mode shape rather than structural modification of the lattice floor support. The reduction in the acoustical response associated with the acoustic mode shape modification was a result of changing the input admittance of the interior cavity.

5) Interior pressure response of the two cases presented indicate that not all modes generated in the shell couple effectively to the interior acoustic field. Similar results from the companion investigation[5] suggest that, in general, only one or two modes couple effectively, thus dictating the contained acoustic field. Only when the well-coupled modes are modified will a significant change in the pressure field occur. It appears difficult to achieve this by structural means.

This experimental investigation defines the major effects of an internal floor on low-frequency sound transmission into aircraft cabins. The results also tend to indicate the potential influence of other such structural modifications (such as wing attachments, ribs, stringers, etc.). Thus, this investigation leads to a better understanding of sound transmission into complex geometries, such as aircraft, as well as an indication of the influence of more realistic geometries than previously considered.

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