Noise Transmission Through Aircraft Panels

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This paper describes analytical and experimental studies of noise transmission through aircraft panels. The theoretical solutions of the governing acoustic-structural equations are developed utilizing modal decomposition and a Galerkin-type procedure. Single, discretely stiffened, and double wall panels are considered. Theoretical predictions are compared with experimental measurements.

Introduction

PREVIOUS studies have shown that there is a need for accurate methods of predicting sound transmission into an airplane cabin.[1-9] This is especially evident for propeller-driven aircraft where maximum noise intensity occurs at low frequencies.[2-4] To better understand the complex noise transmission process, information about the dynamics of a fuselage structure and the properties of the interior acoustic space is needed. In order to design for reduced noise levels with minimum effect on aircraft performance and weight, it is important to develop models capable of predicting noise transmission into an aircraft through various paths for a prescribed noise source.

This paper presents analytical methods for the prediction of sound transmission into enclosures through localized vibrating elastic surfaces. Experimental data are also presented to verify these predictions. The emphasis is placed on light propeller-driven aircraft. However, the methods presented could be generalized and extended to other types of aircraft. The sidewalls of the aircraft are composed of panels that are stiffened with frames and stringers and of windows that are usually of a Plexiglas double-wall construction. The basic structural features of a typical light twin-engine aircraft are shown in Fig. 1. The exact dynamic analysis of such a structure is too complicated, and simplified models need to be constructed. Analytical studies of sound transmission have involved single panels,[2,10-12] monocoque shells (where the effect of stiffeners is "smeared" into an equivalent skin),[1,3,13-15] periodically stiffened infinite panels,[16,17] discretely stiffened finite panels,[18-22] and numerical solutions using finite-element methods.[23-27] The "smeared" models are only valid for the cases where the wavelengths of circumferential and longitudinal shell motions are much longer than the distances between stringers and frames. Noise transmission through single panels does not include the effect of discrete stiffeners and in many cases is limited to higher frequencies. The periodically stiffened infinite panels and discretely stiffened finite panels provide means to include the effect of stiffeners but are restricted to a one-dimensional array of panels. Even though there are limitations of using these panel models to characterize the noise transmission into aircraft, they provide tractable analytical means for calculating noise transmission through localized regions of the fuselage.

The noise transmission through the localized regions is obtained by solving the linearized acoustic wave equation for the interior noise field and the vibration equations for the sidewall panels. The solution to this system of equations is obtained by using modal expansions and a Galerkin-type procedure.[11,12,18-22] The single panels are modeled by simple plate theory,[10,12] discretely stiffened panels by the transfer matrix methods,[18,28] and windows by a double-wall theory.[29,30] To reduce vibration levels and increase noise attenuation, theoretical and experimental feasibility studies of stiffening the metallic panels with lightweight honeycomb construction are undertaken. The laboratory study described herein was performed to provide data for verification of analytical predictions and to study the feasibility of noise control by increased stiffness.

Analytical Models

The basic concept of the analytical procedure used to calculate noise transmission through localized regions into acoustic enclosures is that of modal analysis.[5,6,19-22] The solution method has been improved by transforming the time-dependent and the absorbing boundary conditions into the governing equation using Green's theorem.[29,30] The solution of the acoustic equation is then coupled to the vibration of the elastic panels.

Acoustic Model

An analytical model to predict transmission of airborne noise into an enclosure as shown in Figs. 1 and 2 can be developed by solving the linearized acoustic wave equation for the perturbation pressure p

$$\nabla^2 p - \beta \dot{p} = \ddot{p}/c^2 \tag{1}$$

where ∇^2 is the Laplacian operator, and β and c are the acoustic damping and speed of sound inside the enclosure, respectively. In order to develop meaningful solutions to Eq. (1), idealized models to describe the cabin geometry need to be selected. The geometry for which the analytical solutions can be readily developed include rectangular parallelepiped,[2,12,18] deformed parallelepiped,[7,25,27] semicylinder,[19,20] cylinder,[1,3,4,13-15] and partial cylindrical shapes.[7-9] In the present approach the aircraft cabin is approximated by a rectangular enclosure for which the acoustic modes are well known.[12,18]

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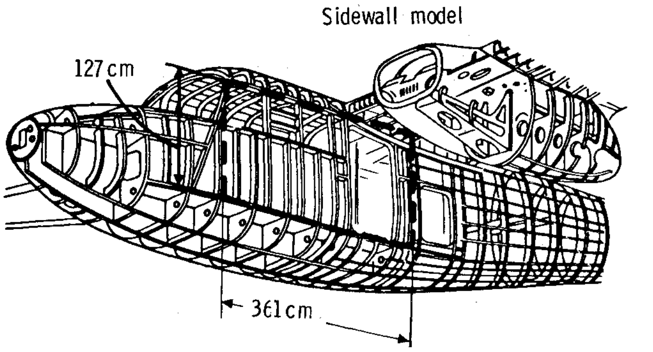

Fig. 1 Structural features of a twin-engine aircraft used for interior noise study.

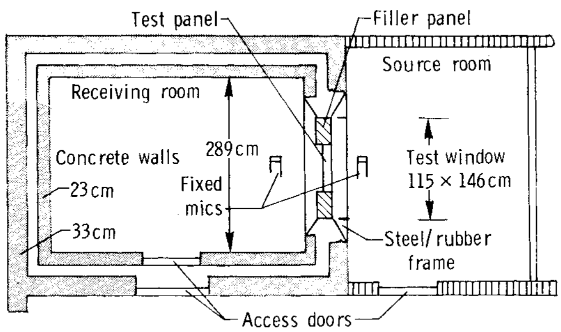

Fig. 2 Top view of the noise-transmission loss apparatus.

To develop a solution for Eq. (1), the boundary conditions at the interior surfaces of the cabin need to be prescribed. Depending on the interior treatment, these can be selected from the following:

1) At a rigid boundary

$$\partial p/\partial n = 0 \tag{2}$$

where n is outward normal to the boundary.

2) At a flexible boundary

$$\partial p/\partial n = -\rho \ddot{w} \tag{3}$$

where ρ is the air density in the enclosure and w is the motion of the flexible wall.

3) At an absorptive boundary

$$\partial p/\partial n = -\rho \dot{p}/Z(\omega) \tag{4}$$

where $Z(\omega)$ is the specific acoustic impedence. A general model to characterize boundary conditions where all the walls, including the vibrating surfaces, are absorbent has been proposed in Ref. 31. If the vibrating surface is located on the wall at $z=0$, then the general boundary conditions to be satisfied by Eq. (1) are[31]

$$\partial p/\partial z = \rho \ddot{w} + \rho [\dot{p} + B(\omega) \nabla_s^2 \dot{p}]/Z(\omega) \text{ at } z=0 \tag{5}$$

and

$$\partial p/\partial n = -\rho [\dot{p} + B(\omega)\nabla_s^2 \dot{p}]/Z(\omega), \text{ otherwise} \tag{6}$$

where $B(\omega)$ and ∇_s^2 are, respectively, the bulk reaction coefficient and the Laplacian at the surface of the enclosure.

The solution to the system of equations with nonhomogeneous time-dependent boundary conditions can be achieved by first transforming the inhomogeneous term $\rho\ddot{w}$ into the governing equation by setting

$$p(x,y,z,t) = q(x,y,z,t) + \rho \ddot{w}(x,y,t) H(z) \tag{7}$$

where q are the solutions to the associated homogeneous problem and $H(z)$ is chosen to satisfy the given boundary conditions.[12,29] Furthermore, by utilizing Green's theorem, the effect of absorption introduced through $Z(\omega)$ can be transformed into the governing equation.[11,29] The result is a set of inhomogeneous equations with homogeneous boundary conditions.

When the boundary conditions include absorption, the resulting cavity eigenvalues and eigenfunctions are complex.[32] In most cases, however, the acoustic modes are calculated by using Eq. 2 under the assumption that the cabin walls are rigid. Furthermore, if the rigid walls are allowed to have curvature or if the acoustic shapes are irregular, calculation of acoustic eigenvalues and eigenfunctions is an involved task, and numerical procedures such as perturbation techniques,[33] finite-difference methods,[8,9] or finite-element methods[23-27] need to be used.

The solution for the perturbation pressure $\bar{p}$ can be written as:

$$\bar{p}(x,y,z,\omega) = \frac{8}{abd} \sum_{i=0}^{\infty} \sum_{j=0}^{\infty} \sum_{k=0}^{\infty} Q_{ijk}(\omega) \times Y_{ijk}(x,y,z) - \omega^2 \rho \bar{w}(x,y,\omega)\, H(z) \tag{8}$$

where $a, b,$ and d are the dimensions of the rectangular enclosure, Y_{ijk} the acoustic modes, Q_{ijk} the solutions for the generalized acoustic pressure,[29] and a bar indicates a Fourier-transformed quantity.

The spectral density of the acoustic pressure, $S_p(x,y,z,\omega)$, can be obtained by taking mathematical expectation of Eq. (8) and then using spectral decomposition as presented in Ref. 28. Then, the sound pressure levels inside the enclosure measured in decibels, relative to a reference pressure p_0, are determined by

$$SPL(x,y,z,\omega) = 10 \log \{S_p(x,y,z,\omega)\, \Delta\omega / p_0^2\} \tag{9}$$

where $\Delta\omega$ is the selected frequency bandwidth at which the spectral density is estimated and $p_0 = 2.9 \times 10^{-9}$ psi ($20 \mu N/m^2$). A quantity relating the spectral density of the acoustic pressure to the spectral density of the external pressure $S_e(x^*, y^*, \omega)$ is the noise reduction NR, which is defined as

$$NR(x,x^*;y,y^*;z,\omega) = 10 \log \{S_e(x^*,y^*,\omega)/S_p(x,y,z,\omega)\} \tag{10}$$

where x^*, y^* are selected spatial points at which the input surface pressure is estimated or measured. The solutions for the sound pressure are functions of the flexible wall motion $\bar{w}(x, y, \omega)$. The response of simple panels, discretely stiffened finite panels, and double-wall windows is considered next.

Structural Model

The sidewalls of aircraft shown in Fig. 1 are composed of the load-bearing external skin and several window units. Thermal insulation and acoustic trim are usually intact at the interior sides of the sidewall. However, for the present analysis only the untreated case is considered. Various analytical representations of the fuselage structure have been used by different investigators. Some of those models are discussed in the review articles in Refs. 6 and 7. In the present paper, simple panels, discretely stiffened panels, and double-wall windows are considered as possible structural models for noise transmission estimation through localized regions of the sidewall.

Simple Panels

The simplest representation of a localized vibrating surface is a flat rectangular panel which is either simply supported or clamped on all four sides. Such a model has been used for numerous laboratory[11,12,34-37] and theoretical studies.[2,10-12,38]

The equation of motion of the flexural panel response can be written as

$$D\nabla^4 w + \gamma \dot{w} + M\ddot{w} = p^r(x,y,t) \tag{11}$$

where $\nabla^4 \equiv \partial^4/\partial x^4 + 2\partial^4/\partial x^2 \partial y^2 + \partial^4/\partial y^4$; w is the panel displacement; dot indicates differentiation with respect to time; D, γ, M, and p^r are stiffness, damping coefficient, mass and external random surface pressure, respectively. The effect of the acoustic radiation pressure is not included directly in Eq. (11) but accounted for through addition of acoustic radiation damping to the γ damping coefficient.[4] Equation (11) can easily be modified to include curvature and cabin pressurization effects.[22,30] The response can be written as the superposition of normal modes

$$w(x,y,t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} q_{mn}(t) X_{mn}(x,y) \tag{12}$$

where q_{mn} are the generalized coordinates and X_{mn} are the mode shapes. After the mode shapes are defined, the solution for q_{mn} can be obtained by substituting Eq. (12) into Eq. (11), utilizing the orthogonality principle and taking Fourier transformation. In this procedure it is assumed that the effect of the acoustic back-up pressure (cavity pressure) on panel response is negligible. Such an assumption is justified for relatively deep enclosures considered in this paper.

For simple support boundary conditions, the modes X_{mn} are sine functions, while for clamped supports the beam eigenfunctions can be utilized. When the boundary conditions are more complicated, the modes can be calculated utilizing numerical procedures such as the finite-element method.

Discretely Stiffened Panels

The structural response of discretely stiffened finite panels can be obtained by following procedures similar to the ones presented for simple panels in the previous section. However, the natural frequencies and the mode shapes of these stiffened panels are complicated functions involving bending, torsion, and warping interactions of the stiffeners. The transfer matrix technique[18,28] and the finite-element strip method[20] have been used to calculate the natural frequencies and normal modes of flat and curved stiffened panels. A brief description of the transfer matrix method is given.

Consider the sidewall of the aircraft shown in Fig. 1 modeled by several discretely stiffened panel units. These panels, shown in Fig. 3, are assumed to be simply supported along the boundaries perpendicular to the stiffeners. For example, panel unit number 1 is simply supported at $y=0$ and $y=L_y$. Then, the normal modes corresponding to the y coordinate are sin $(n\pi y/L_y)$. Substitution of this relation into Eq. (11) and setting $p_r = 0$ results in a fourth-order homogeneous equation for each n. The solution to this equation can be written in a state vector form $\{w_n\} = \{\delta_n, \theta_n, M_n, V_n\}$, where δ_n, θ_n, M_n, and V_n are components of deflection, slope, moment, and shear, respectively. A transfer matrix ${}_N^r(\Gamma)_0^\ell$ is then constructed which transfers the state vector from the left of station 0 to the right of station N ($N=3$ for panel No. 1)

$$\{w_n\}_N^r = {}_N^r[\Gamma]_0^\ell \{w_n\}_0^\ell \tag{13}$$

where

$${}_N^r[\Gamma]_0^\ell = [G]_N [F]_N [G]_{N-1} \cdots [F_1][G_0] \tag{14}$$

The point matrix $[G]$ transfers the state vector across a stiffener and the field transfer matrix $[F]$ transfers the state vector across a panel. The detailed expressions of these transfer matrices are given in Ref. 28. Utilizing the natural boundary conditions at $x=0$ and $x=L_x$ in Eq. (13) gives the transcendental frequency equation which then can be solved for the natural frequencies of the stiffened panel system. Similarly, by defining a local coordinate at arbitrary points on the panel, normal modes corresponding to the x direction can be calculated.[19] These modes are then used in Eq. (12) for the solution of panel motions.

Double-Wall Windows

The double-wall aircraft windows are composed of curved external and flat internal Plexiglas panels. The air space between the two panels is approximated by a uniformly distributed air spring.[45] A linear spring-dashpot model is used to characterize the behavior of the air spring. Then, a simple double-wall structural model is constructed where both Plexiglas panels are taken to be flat and simply supported on all four edges. To account for the effect of the curvature of the outside panel, the stiffness of this panel is increased accordingly. A detailed treatment of the response of the double-wall aircraft windows is given in Ref. 30, It is shown that the solutions for the deflection of the top (exterior) and the bottom (interior) windowpanes are

$$\bar{w}_T(x,y,\omega) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} A^T_{mn}(\omega) X_{mn}(x,y) \tag{15}$$

$$\bar{w}_B(x,y,\omega) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} A^B_{mn}(\omega) X_{mn}(x,y) \tag{16}$$

where A^T_{mn} and A^B_{mn} are the generalized coordinates of the top and bottom plates

$$A^T_{mn} = H^T_{mn} \{ P^T_{mn} + A^B_{mn} [E_s/h_s + \omega^2 b_s] / m_T \} \tag{17}$$

$$A^B_{mn} = H^B_{mn} \{ A^T_{mn} [E_s/h_s + \omega^2 b_s] / m_B \} \tag{18}$$

P^T_{mn} are the generalized random forces acting on the top plate and

$$H^T_{mn} = [(\omega^T_{mn})^2 - (a_T/m_T)\omega^2 + 2i\omega\omega^T_{mn}\zeta^T_{mn} + E_S/(h_S m_T)]^{-1} \tag{19}$$

$$H^B_{mn} = [(\omega^T_{mn})^2 - (a_B/m_B)\omega^2 + 2i\omega\omega^B_{mn}\zeta^B_{mn} + E_S/(h_S m_B)]^{-1} \tag{20}$$

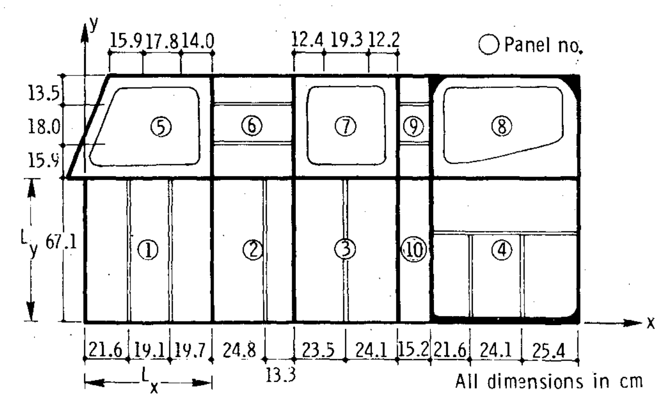

Fig. 3 Panel identification for port side.

in which $a_T = m_T + m_S/3$, $a_B = m_B + m_S/3$, $b_S = m_S/6$, $m_T = \rho_T h_T$, $m_B = \rho_B h_B$, $m_S = \rho_S h_S$, ω_{mn} and ζ_{mn} are modal frequencies and modal damping coefficients, and h and ρ are thickness and material density. The subscripts T, B, S and the superscripts T and B denote the top plate, the bottom plate and core, respectively. In obtaining these equations, it was assumed that the inertia force varies linearly across the thickness of the soft core. Thus a consistent mass distribution is used with the terms $m_S/3$ and $m_S/6$ representing the apportioned contributions of the mass of the core to the two plates.

Several linear models exist to characterize the behavior of the core, depending on the type of material used.[30,45] However, for a core filled with air, we set $K_S[w_S] = E_S w_S/h_S$, where h_S is the average distance between the top and the bottom plates, $w_S = w_T - w_B$, and K_S is the constitutive law operator representing the forces exerted on the plates by the core. For an air-spring model, $E_S = \rho_S c_S^2$, where ρ_S and c_S are the air density and speed of sound.

The natural frequencies of the coupled system can be determined by setting $\zeta_{mn}^T = \zeta_{mn}^B = 0$ and maximizing the frequency response function. For each set of modal indices, (m, n), the natural frequencies of the coupled system can be calculated from

$$\omega_{mn} = \{ [B_{mn} \pm (B_{mn}^2 - 4AC_{mn})^{1/2}]/2A \}^{1/2} \tag{21}$$

$$A = a_T a_B - b_S^2 \tag{22}$$

$$B_{mn} = (m_T \omega^{T^2}_{mn} + E_S/h_S) a_B + (m_B \omega^{B^2}_{mn} + E_S/h_S) a_T + 2b_S E_S/h_S \tag{23}$$

$$C_{mn} = (m_T \omega^{T^2}_{mn} + E_S/h_S)(m_B \omega^{B^2}_{mn} + E_S/h_S) - (E_S/h_S)^2 \tag{24}$$

where ω_{mn}^T and ω_{mn}^B are the uncoupled modal frequencies of the top and bottom plates. Equation (21) gives two real characteristic values for each set of modal indices (m, n). These roots are associated with in-phase flexural and out-of-phase dilatational vibration frequencies of the double-wall system. The dilatational vibration frequencies are strongly dependent on the core stiffness represented by E_S/h_S.

To account for the curvature effect of the exterior panel, the uncoupled modal frequencies of flat panels are modified according to a procedure suggested in Ref. 39. Then the uncoupled modal frequencies of the exterior window are calculated from

$$(\omega_{mn}^T)^2_{\text{curved}} = (\omega_{mn}^T)^2_{\text{flat}} + \frac{E_T m}{\rho_T R^2 [m^2 + (L_x/L_y)^2 n^2]^2} \tag{25}$$

where R and E_T are the average radius of the curvature and elastic modulus of the exterior windowpane.

Pressurization of the cabin and/or depressurization of the air space between the two window sheets increase the stiffness of the Plexiglas plates. Such an effect can be included through the average in the plane loads $\bar{N}_x = \Delta pR/2$ and $\bar{N}_y = \Delta pR$ corresponding to the axial and circumferential directions where Δp is the pressure differential. The natural frequencies are then calculated from

$$(\omega_{mn})_{\text{flat}} = \pi \left[\frac{D\pi^2}{\bar{\rho}h} (m^2/L_x^2 + n^2/L_y^2)^2 + (\bar{N}_x m^2/L_x^2 + \bar{N}_y n^2/L_y^2)/\bar{\rho}h \right]^{1/2} \tag{26}$$

where D and $\bar{\rho}$ are plate stiffness and material density.

External Noise Pressure Field

The near-field surface sound pressures p^r are required as noise inputs to the panels for the analytical noise transmission calculations. These pressures are characterized by the pressure amplitude or sound level, the spatial distribution of the coherence function and convection or phase velocity. In general, the cross-spectral density of the random input pressure p^r can be written as $S_{p^r}(\xi,\eta,\omega)$, where $\xi = x_2 - x_1$ and $\eta = y_2 - y_1$ are the spatial separations corresponding to panel coordinates x and y. Then, the cross-spectral density of the generalized random force acting on the external surface of a panel in question can be written as

$$S_{mnrs}(\omega) = \frac{1}{M_{mn}^2} \iiiint S_{p^r}(x_1,x_2,y_1,y_2,\omega) X_{mn}(x_1,y_1) \times X_{rs}(x_2,y_2) \mathrm{d}x_1 \mathrm{d}x_2 \mathrm{d}y_1 \mathrm{d}y_2 \tag{27}$$

where the integration in Eq. (27) spans the surface area of the panel and M_{mn} is the generalized mass of the plate. To evaluate the above relation, it is necessary to have representations for the excitation field. It has been an acceptable practice to separate the longitudinal and transverse correlations as a product, i.e., $S_{p^r}(\xi,\eta,\omega) = S^E(\omega) R_x(\xi,\omega) \cdot R_y(\eta,\omega)$ where $S^E(\omega)$ is the spectral density of the random pressure and R_x and R_y are the correlation coefficients corresponding to x and y directions. For flight conditions, the surface pressure inputs corresponding to propeller noise and turbulent boundary-layer flow need to be defined. The laboratory tests usually require simulation of uniform or reverberant pressure fields.

Propeller Noise

The simplest representation of propeller noise is a family of sinusoidal acoustic waves incident at some angle of incidence. In this case the correlation coefficients R_x and R_y are unity, and the trace velocity of the pressure field is supersonic. Tests[40] indicate, however, that such an assumption might be too restrictive and that the pressure field in the vicinity of the plane of rotation is aerodynamic rather than acoustic and rotates with the propeller. However, as the distance between the propeller tip and the panel increases, the surface pressure characteristics would change from aerodynamic to acoustic. Since the dimensions of a single panel are relatively small and no appreciable surface pressure variation can be observed within the area of the panel, a useful approximation of propeller noise input can be written in terms of the cross-spectral density function

$$S_{p^r}(\xi,\eta,\omega) = S^E(\omega) \exp[i\omega\xi/V_x] \exp[i\omega\eta/V_y] \tag{28}$$

where V_x and V_y are the pressure field trace velocities in the x and y directions. After the modes X_{mn} are defined, the

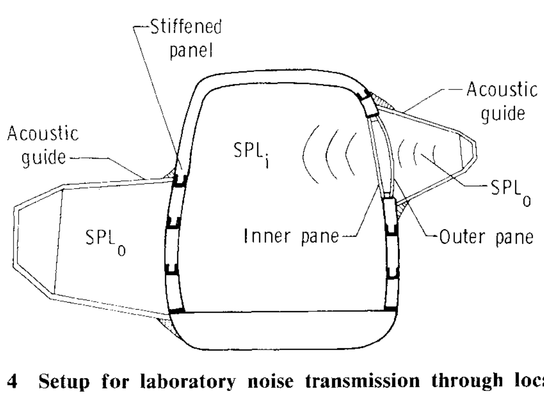

Fig. 4 Setup for laboratory noise transmission through localized regions.

generalized random forces corresponding to propeller noise can be calculated using Eqs. (27) and (28).

Boundary-Layer Turbulence

The surface pressures to convecting turbulent boundary-layer flow are described by the semiempirical cross-spectral density forms.[41] However, for light propeller-driven aircraft, the interior noise is dominated by low-frequency noise (up to about 1000 Hz) due to propeller blade passage harmonics, and the effect of boundary-layer noise is usually neglected.

Uniform Pressure

For spatially uniform pressure, the input cross-spectral density can be taken as band-limited Gaussian white noise

$$S_{p^r}(\xi,\eta,\omega) = \begin{cases} K & 0 \le \omega < \omega_u \\ 0, & \text{otherwise} \end{cases} \tag{29}$$

where ω_u is the upper cutoff frequency and K is a constant which measures the noise intensity.

Reverberant Field

The spatial correlation coefficients are taken to be

$$R_x(\xi,\omega) = \sin(k\xi)/k\xi$$

$$R_y(\eta,\omega) = \sin(k\eta)/k\eta \tag{30}$$

where $k=\omega/c$. Substituting these relations into Eq. (27) and assuming panel modes $X_{mn}(x,y) = \sin(m\pi x/L_x)\sin(n\pi y/L_y)$, the cross-spectal density of the generalized random forces for $m=r$ and $n=s$ is

$$S_{mn}(\omega) = S^E(\omega)\cdot I_m(\omega) I_n(\omega)/M_{mn}^2 \tag{31}$$

where

$$I_m(\omega) = (I_{1m} + I_{2m} + I_{3m})L_x^2 \tag{32}$$

$$I_{1m} = \frac{1}{2\pi m L_x k}[\mathrm{Cin}(kL_x + m\pi) - \mathrm{Cin}(\,|m\pi - kL_x|\,] \tag{33}$$

$$I_{2m} = \frac{1}{2L_x k}[\mathrm{Si}(kL_x + m\pi) - \mathrm{Si}(m\pi - L_x k] \tag{34}$$

$$I_{3m} = \frac{[1-(-1)^m \cos kL_x]}{[(m\pi)^2 - (kL_x)^2]} \tag{35}$$

where Si and Cin are the sine and cosine integrals.[42] The expressions for $I_n(\omega)$ are the same as $I_m(\omega)$, but m is replaced with n and L_x with L_y.

Laboratory Study of Noise Transmission Through Aircraft Panels

Experiments were carried out to measure noise transmission through single panels, discretely stiffened panels, and double-wall windows. The effect of panel stiffening by addition of honeycomb panels was investigated. These tests were performed on fuselage sidewall panels and windows of a twin-engine aircraft shown in Fig. 1 using an acoustic guide and on test specimens installed in the NASA Langley noise transmission loss apparatus illustrated in Fig. 2.

Acoustic Guide

The localized noise inputs to the aircraft sidewall panels and windows were generated by an acoustic guide shown in Fig. 4. The basic features of the acoustic guide design include a high-quality speaker and a slowly diverging rectangular duct. The walls of the duct are constructed from 3/8-in.-thick plywood. To minimize noise leakage from the interior enclosure of the guide, two layers of noise barriers, each with a surface density of 1 lb/ft^2, were added to the exterior surfaces of the guide. Between the duct and the sidewall of the aircraft, soft insulation material (foam), ranging in thickness from about 2 in. to 4 in. was installed around the periphery of the guide. The present design with duct dimensions of either 20 in. × 20 in. or 30 in. × 30 in. can be used to generate acoustic inputs for small panels, windows, and larger discretely stiffened panels. The noise-measuring system includes a microphone inside the guide at about 1 in. from the sidewall surface and a microphone inside the cabin at about 10 in. from the interior wall.

Noise Transmission Loss Apparatus

The noise transmission loss apparatus is designed around two adjacent reverberant rooms, of which the receiving room is acoustically and structurally isolated from the rest of the building.[36,43] The test specimen is mounted on a heavy, stiff particle board frame which is installed as a partition between the two rooms (Fig. 2). In the source room a diffuse noise field is produced by two reference sound power sources. The particle board is accommodated by a steel and rubber mounting frame which is designed for minimum acoustic and structural flanking. Noise measurements were obtained by stationary microphones at a distance of 1 in. from the panel in the source room and 12 in. in the receiving room. Noise reduction is

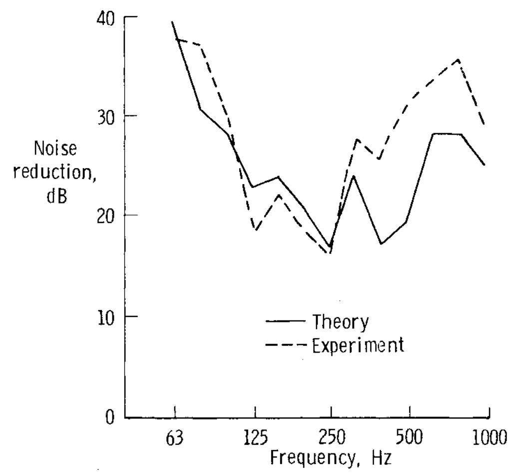

Fig. 5 Noise reduction of a stiffened aircraft panel: noise input from an acoustic guide.

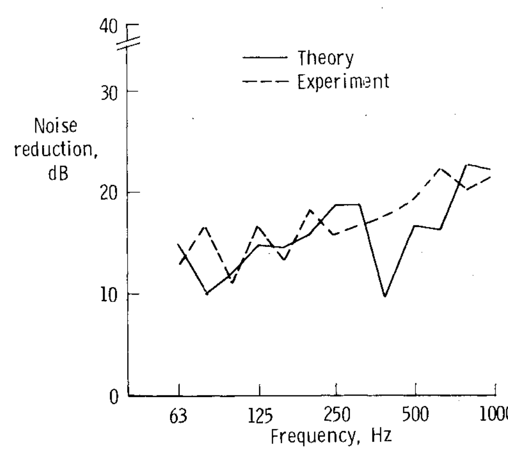

Fig. 6 Noise reduction of an aircraft sidewall: noise input from a diffuse field.

defined as the difference between the measured sound pressure levels of the microphones in the source and receiving rooms.

The aluminum test panels are intended to be representative of single sidewall panels as typical for general aviation aircraft. Noise-reduction measurements and acceleration response characteristics were obtained for an untreated 14 in. × 12 in. × 0.063 in. aluminum panel and for several different honeycomb designs which were bonded to similar size aluminum panels.

Numerical Results

The numerical results presented in this paper are noise transmission through the sidewall panels of the light aircraft shown in Fig. 1 and noise reduction of single panels installed in the noise transmission facility shown in Fig. 2. It is assumed that airborne noise enters the receiving enclosure only through the vibrating panels without any flanking paths. The noise-transmitting sidewall of the aircraft is composed of several stiffened panels which range in dimensions from about 6 in. × 15 in. to 11 in. × 27 in. and thicknesses from 0.032 in. to 0.064 in. The cabin windows are double-wall Plexiglas with panel thicknesses of 0.14 in. The noise transmission through these panels is calculated for a spatrially uniform Gaussian white noise input described by Eq. (29). A reverberant diffuse noise input was assumed for the noise transmission calculation through panels installed in the noise transmission loss apparatus.

The numerical calculations were obtained for structural and acoustical modal damping ratios of $\zeta_{mn} = \zeta_0(\omega_{11}/\omega_{mn})$ and $\xi_{ijk} = \xi_0(\omega^\ell/\omega_{ijk})$, respectively. The ζ_0 and ξ_0 are the damping coefficients for the fundamental modes, ω_{mn} and ω_{ijk} are the structural and acoustic modal frequencies, and ω^ℓ is the lowest acoustic modal frequency in the enclosure. For the present analytical study, it was assumed that ζ_0 equals 0.01 for single panels, 0.02 for discretely stiffened panels, 0.03 for honeycomb-treated panels, and 0.05 for the Plexiglas windows. The damping coefficient ξ_0 was assumed to be equal to 0.03 for a lightly treated aircraft cabin and 0.0025 for the reverberant receiving room in the noise-transmission loss apparatus. The structural damping values were selected utilizing the experimental information given in Ref. 46. The acoustic damping coefficients were based on the best engineering judgment. However, since wall absorption is provided through the point impedance $Z(\omega)$, the numerical results are less sensitive to the variations of acoustic damping when compared to the variations in structural damping.

In calculating noise transmitted into an acoustic enclosure, it is necessary to prescribe the impedance and the bulk reactance at the interior walls. As the interior walls of a typical aircraft are not treated uniformly, the wall impedance needs to be represented in an average sense. Numerical results were obtained for $B(\omega) = 0$ and $Z(\omega)$, as given in Ref. 44. For an acoustically transparent condition (no wall), the point impedance reduces to the characteristic impedence ρc, while for a rigid wall condition $Z(\omega) \rightarrow \infty$. To simulate the conditions of the lightly treated aircraft walls and the nearly rigid surfaces of the receiving room of the noise transmission loss apparatus, the values of the flow resistivity $R_f = 4 \times 10^4$ mks rayls/m and $R_f = 4 \times 10^6$ mks rayls/m were used.

Test Aircraft

The noise transmission through each panel unit indicated in Fig. 3 was calculated. Assuming independence of these noise transmission paths, the total interior sound pressure is determined by the superposition of the contributions of all panels located on the sidewall. The noise reduction for a typical sidewall panel and the entire sidewall is given in Figs. 5 and 6. Similar results are presented in Fig. 7 for a double-wall window. These results correspond to an interior location at ear level in the propeller plane and 10 in. from the sidewall. The inputs for noise transmission measurements through the entire sidewall were generated by a two-speaker setup. As can be observed from these results, the agreement between theory and experiment is relatively good in view of the complexities involved. The theoretical model tends to predict lower noise reduction at the structural modal resonance frequencies. These differences might be attributed to damping and idealistic mode shapes used in the theoretical calculations. Furthermore, the assumed spatially uniform inputs for the theoretical model cannot be accurately simulated with the present experimental setup.

To investigate the effect of skin stiffening on noise transmission, lightweight treatments in the form of

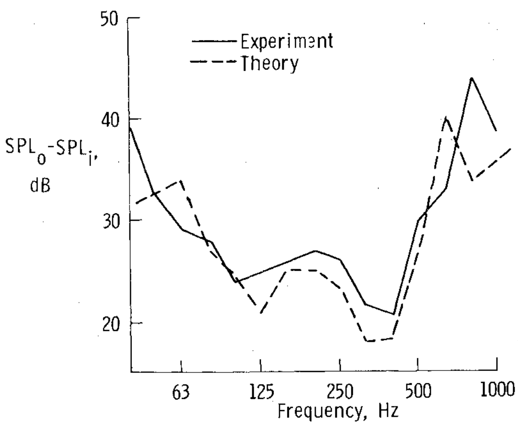

Fig. 7 Noise reduction of a double-wall aircraft window: noise input from an acoustic guide.

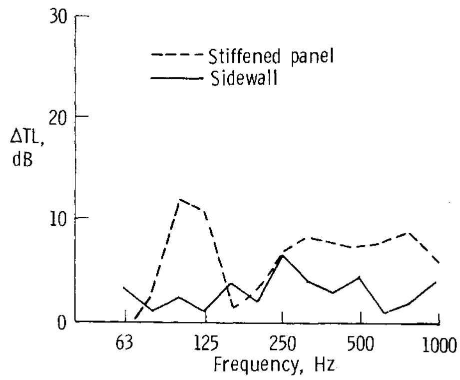

Fig. 8 Measured insertion loss of an aircraft panel and a sidewall due to honeycomb add-on treatment.

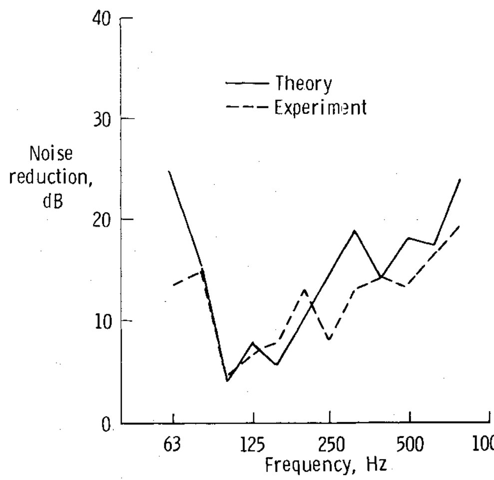

Fig. 9 Noise reduction of an untreated panel measured using transmission loss apparatus.

honeycomb panels were added to the aircraft sidewall. The combined stiffness (panel plus honeycomb) was calculated utilizing the relations given in Refs. 21, 35, and 39. However, for discretely stiffened panels with honeycomb add-on treatment, the estimation of stiffness, modal frequencies, and mode shapes is more complicated.[18,21,22]

The experimental results of noise attenuation due to honeycomb add-on treatment composed of a core $h_c = 0.125$ in. and a face plate $h = 0.02$ in. are shown in Fig. 8. These results tend to indicate that substantial gains of noise reduction can be achieved for individual panel units by honeycomb add-on treatment. However, those gains are only modest when the entire sidewall is considered. These differences might be attributed to the vibrations of the entire sidewall and transmission of noise through the aircraft windows.

Transmission Loss Apparatus

Analytical calculations and experimental measurements of noise transmission through single panels were carried out for the noise-transmission loss apparatus illustrated in Fig. 2. The measured and predicted noise reduction values for an untreated aluminum panel of 19 in. × 12 in. × 0.063 in. are shown in Fig. 9. These results indicate reasonably good agreement between theory and experiment and relatively strong contributions to the transmitted noise by panel resonances (fundamental resonance frequency of 92 Hz). These results clearly show that noise for the experimental model is transmitted by the entire wall composed of the particle board, frame supports and the panel. Additional results of noise transmission through built-up aircraft panels which include frames and stringers are given in Ref. 47.

Concluding Remarks

This paper describes theoretical and experimental studies of sound transmission through aircraft panels and double-wall windows. Tests were carried out using the fuselage of a light twin-engine aircraft and the NASA Langley noise-transmission loss apparatus.

Predicted noise transmission using modal methods is in reasonable agreement with data for discretely stiffened aircraft panels and double-wall windows. Similar level of agreement is obtained for untreated panels tested in the noise-transmission loss apparatus.

Stiffening aircraft panels with honeycomb add-on treatment provide additional noise attenuation. However, large gains of noise reduction measured for individual panels might not be obtained when the entire sidewall is considered.

Acknowledgments

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