Abstract

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 and differences are identified.

Introduction

Previous studies have shown that there is a need for accurate methods of predicting sound transmission into an airplane cabin. This is especially evident for propeller-driven aircraft where maximum noise intensity occurs at low frequencies. 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 aircraft through various paths under a prescribed noise source.

This paper presents analytical methods to predict 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 which are stiffened with frames and stringers and of windows which are usually of a Plexiglass 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 on sound transmission have involved single panels, monocoque shells (where the effect of stiffeners is "smeared" into an equivalent skin), periodically stiffened infinite panels, discretely stiffened finite panels, and numerical solutions using finite element methods. "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 excludes 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. The single panels are modeled by simple plate theory, discretely stiffened panels by the transfer matrix method, and windows by a double wall theory. 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 define the feasibility of such tests for noise transmission control by the stiffness treatment.

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. The solution method has been improved by transforming the time dependent and the absorbing boundary conditions into the governing equation using Green's theorem. The solution of the acoustic equation is then coupled to the vibration of the elastic panels.

Acoustic Model

The requirements of an analytical model to predict transmission of airborne noise into an enclosure such as the one shown in Figs. 1 and 2 can be satisfied by solving the linearized acoustic wave equation for the perturbation pressure \( p \)

\[
\nabla^2 p - \beta \frac{p}{c^2} = \beta/c^2
\]

(1)

where \( \nabla^2 \) is the Laplacian operator, \( \beta \) and \( c \) are the acoustic damping and speed of sound inside the enclosure, respectively. In order to develop meaningful solutions to equation (1), idealized models to describe the cabin geometry need to be selected. The geometry for which the analytical solutions can be readily developed include...
To develop a solution for equation (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:

(a) At a rigid boundary
\[ \frac{\partial p}{\partial n} = 0 \]  
where \( n \) is outward normal to the boundary.

(b) At a flexible boundary
\[ \frac{\partial p}{\partial n} = -\rho \ddot{w} \]  
where \( \rho \) is the air density in the enclosure and \( w \) is the motion of the flexible wall.

(c) At an absorptive boundary
\[ \frac{\partial p}{\partial n} = -\rho \ddot{w} / Z(\omega) \]  
where \( Z(\omega) \) is the specific acoustic impedance.

A general model to characterize boundary conditions where all the walls, including the vibrating surfaces are absorptive, 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 equation (1) are
\[ -\frac{\partial p}{\partial z} = -\rho \left[ \frac{\ddot{p} + B(\omega) \left( \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) \ddot{p} }{Z(\omega)} \right] \]  
\[ -\rho \ddot{w} \quad \text{at} \quad z = 0 \]  
and
\[ \frac{\partial p}{\partial n} = -\rho \left[ \frac{\ddot{p} + B(\omega) \ddot{w} }{Z(\omega)} \right] \]  
otherwise

where \( B(\omega) \) and \( v_s^2 \) are 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
\[ p(x,y,z,t) = q(x,y,z,t) + \rho \ddot{w}(x,y,t) G(z) \]  
where \( q \) are the solutions to the associated homogeneous problem and \( G(z) \) is chosen to satisfy the given boundary conditions. Furthermore, by utilizing Green's theory, the effect of absorption introduced through \( Z(\omega) \) and \( B(\omega) \) can be transformed into the governing equation. 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. However, in most cases the acoustic modes are calculated under the assumption that the cabin walls are rigid by using equation (2). Furthermore, if the rigid walls are allowed to have curvature or the acoustic shapes are irregular, calculation of acoustic eigenvalues and eigenfunctions is an involved task and numerical procedures such as perturbation techniques, finite difference methods, or finite element methods need to be used.

The solution for the perturbation pressure \( \ddot{p} \) can be written as:
\[ \ddot{p}(x,y,z,\omega) = \frac{8}{a b d} \sum_{i=0}^{\infty} \sum_{j=0}^{\infty} \sum_{k=0}^{\infty} Q_{ijk}(\omega) Y_{ijk}(x,y,z) \]  
\[ -\omega^2 \rho \ddot{w}(x,y,\omega) G(z) \]  
where \( a, b, \) and \( d \) are the dimensions of the rectangular enclosure, \( Y_{ijk} \) are the acoustic modes, \( Q_{ijk} \) are the solutions for the generalized acoustic pressure, and \( \ddot{w} \) 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 equation (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
\[ \text{SPL}(x,y,z,\omega) = 10 \log \left[ \frac{S_p(x,y,z,\omega)}{p_0^2} \right] \]  
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
\[ NR(x^*,y^*,z,\omega) = 10 \log \left( \frac{S_e(x^*,y^*,\omega)}{S_p(x,y,z,\omega)} \right) \]  
where \( x^*, y^* \) are selected spatial points at which the input surface pressure is estimated or measured. It is convenient to define the noise reduction on a one-third octave frequency scale by setting.
where \( \omega_e \) and \( \omega_u \) are the lower and the upper bounds limiting frequencies of each one-third octave band. The solutions for the sound pressure levels and noise reduction given in these equations are functions of the flexible wall motion \( \bar{W}(x,y,w) \). The response of simple panels, discretely stiffened finite panels, and double wall window units is considered next.

### Structural Model

The sidewalls of aircraft such as the one 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 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 

\[
\text{Equation of motion for the flexural response can be written as}
\]

\[
Dv^4w + \gamma w + Mw = p'(x,y,t) \tag{12}
\]

where \( v = 4/ax^4 + 2a^4/ax^2ay^2 + a^4/ay^4 \), \( w \) is the panel displacement, \( \gamma \) indicates differentiation with respect to \( D, \gamma, M, \) and \( p' \) are stiffness, damping coefficient, mass, and external random surface pressure, respectively. Equation (12) can easily be modified to include curvature and cabin pressurization effects,

\[
\text{Response of the panel can be written as the superposition of normal modes}
\]

\[
w(x,y,t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} q_{mn}(t) \chi_{mn}(x,y) \tag{13}
\]

where \( q_{mn} \) are the generalized coordinates and \( \chi_{mn} \) are the mode shapes. After the mode shapes are defined, the solution for \( q_{mn} \) can be obtained by substituting equation (13) into equation (12), 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 and the finite element-strip method 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 is 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 \( y = L_w \). Then, the normal modes corresponding to the \( y \) coordinate are \( \sin(m\pi y/L_w) \). Substitution of this relation into equation (12) and setting \( p' = 0 \) results in a fourth order homogeneous equation for each \( n \). The solution to this equation can be written in a state vector form: \( \{W_n\} = \{q_{x}, q_{\theta}, M_n, V_n\} \) where \( q_{x}, q_{\theta}, M_n\) and \( V_n \) are components of deflection, slope, moment, and shear, respectively. A transfer matrix \( [R]_{N}^{L} \) 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).

\[
\{W_n\}_{N} = [R]_{N}^{L} \{W_n\}_{0} \tag{14}
\]

where

\[
[R]_{N}^{L} = [G]_{L} \cdot [F]_{N} \cdot [G]_{N-1} \cdots [F_1] \cdot [G]_{0} \tag{15}
\]

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 equation (14) 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.\(^1\) These modes are then used in equation (13) for the solution of panel motions.

Double Wall Windows. The double wall aircraft windows are composed of curved external and flat internal Plexiglass panels. The air space between the two panels is approximated by a uniformly distributed air spring. 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 Plexiglass 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. The governing equations of motion of the two panels coupled through the linear spring can be written as

$$m_T \ddot{w}_T + c_T \dot{w}_T + D_T \nabla^4 w_T + K_S [w_T - w_B]$$

$$+ (1/3)m_s \ddot{w}_s + (1/6)m_s \ddot{w}_b = p^T(x, y, t)$$

$$m_B \ddot{w}_B + c_B \dot{w}_B + D_B \nabla^4 w_B + K_S [w_B - w_T]$$

$$+ (1/3)m_s \ddot{w}_s + (1/6)m_s \ddot{w}_T = 0$$

where

$$m_T = \rho_T h_T$$

$$m_B = \rho_B h_B$$

$$m_s = \rho_s h_s$$

$$D_T = E_T h_T^3/[12(1 - \nu_T^2)]$$

$$D_B = E_B h_B^3/[12(1 - \nu_B^2)]$$

Several linear models exist to characterize the behavior of the core, depending on the type of material used.\(^3\) However, for a core filled with air, we set $K_S[w_s] = E_S w_b/h_s$ where $h_s$ is the average distance between the top and the bottom plates. For an air spring model, $E_S = \rho_S c_S^2$, where $\rho_S$ and $c_S$ are the air density and speed of sound.

The solution to equations (16) and (17) is expanded in terms of simply supported plate modes

$$w_T(x, y, t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} A_{mn}^T(t) X_{mn}(x, y)$$

$$w_B(x, y, t) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} A_{mn}^B(t) X_{mn}(x, y)$$

where $A_{mn}^T$ and $A_{mn}^B$ are the generalized coordinates of the top and bottom plates. Substituting equations (20) and (21) into equations (16) and (17) and utilizing the orthogonality principle yields a set of coupled differential equations in $A_{mn}^T$ and $A_{mn}^B$. Taking the Fourier transformation of these equations gives

$$\tilde{A}_{mn}^T(\omega) = H_{mn}^T(\omega) \left\{ \tilde{p}_{mn}^T(\omega) + \tilde{A}_{mn}^B(\omega) \left[ E_S(\omega)/h_S + \omega^2 b_S/m_B \right] \right\}$$

$$\tilde{A}_{mn}^B(\omega) = H_{mn}^B(\omega) \left\{ \tilde{A}_{mn}^T(\omega) [E_S(\omega)/h_S + \omega^2 b_S/m_B] \right\}$$

$$H_{mn}^T(\omega) = \left[ (\omega_{mn}^T)^2 - (a_T/m_T)\omega^2 + 2i(\omega_{mn}^T)\omega \left\{ \tilde{A}_{mn}^T(\omega) + C_{mn}^T \right\} \right]^{-1}$$

$$H_{mn}^B(\omega) = \left[ (\omega_{mn}^B)^2 - (a_B/m_B)\omega^2 + 2i(\omega_{mn}^B)\omega \left\{ \tilde{A}_{mn}^B(\omega) + C_{mn}^B \right\} \right]^{-1}$$

in which $a_T = m_T + m_S/3$, $a_B = m_B + m_S/3$, $b_S = m_S/6$, $\tilde{A}_{mn}^T$ and $\tilde{A}_{mn}^B$ are the transformed modal amplitudes of $A_{mn}^T$ and $A_{mn}^B$, $\tilde{p}_{mn}^T$ are the generalized random forces

$$\tilde{p}_{mn}^T(\omega) = \frac{1}{M_{mn}^T} \int_0^L \int_0^L \tilde{p}^T(x, y, t) X_{mn}(x, y) dx dy$$

214
where $M_{mn}$ and $M_B^D$ are the generalized masses of the top and bottom plates. From equations (22)-(25), the frequency response functions of the double wall system are

$$
\theta_{mn}^T = \frac{H_{mn}^T}{1 - (E_S/h_S + \omega^2 b_S)^2(H_{mn}^T/m_T)(H_{mn}^T/m_B)}
$$

$$
\theta_{mn}^B = \theta_{mn}^T(E_S/h_S + \omega^2 b_S)(H_{mn}^T/m_B)
$$

After performing Fourier transformation, equation (21) is substituted into equation (8) from which the solution for the perturbation pressure $F$ and the spectral density $S_p$ are determined.

The natural frequencies of the coupled system can be determined by setting $\varepsilon_{mn} = \varepsilon_B^D = 0$ and maximizing the frequency response functions of equations (27) and (28). For each set of modal indices $(m,n)$, the natural frequencies of the coupled system can be calculated from

$$
\omega_{mn} = \left\{ \frac{B_{mn} + (B_{mn}^2 - 4C_{mn})^{1/2}}{2\alpha} \right\}^{1/2}
$$

where

$$
A = a_T a_B - b_S^2
$$

$$
B_{mn} = (m_T \omega_{mn}^2 + E_S/h_S) a_B + (m_B \omega_{mn}^2 + E_S/h_S) a_T
$$

$$
+ 2b_S E_S/h_S
$$

$$
C_{mn} = (m_T \omega_{mn}^2 + E_S/h_S)(m_B \omega_{mn}^2 + E_S/h_S)
$$

$$
- (E_S/h_S)^2
$$

where $\omega_{mn}$ and $B_{mn}$ are the uncoupled modal frequencies of the top and bottom plates. Equation (29) 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

$$
(\omega_{mn})_{\text{curved}} = (\omega_{mn})_{\text{flat}} + \frac{E_S}{\rho_f R^2} \left[ \frac{m^2}{m_T} + \frac{L_x}{L_Y} \right] \left( \frac{n^2}{n_T} \right)
$$

where $R$ is the average radius of the curvature.

Pressurization of the cabin and/or depressurization of the air space between the two window sheets increases the stiffness of the Plexiglas plates. Such an effect can be included through the average in the plane loads $N_x = \Delta p R^2/2$ and $N_y = \Delta p R$ corresponding to the axial and circumferential directions, respectively. The natural frequencies are then calculated from

$$
(\omega_{mn})_{\text{flat}} = \sqrt{\frac{m^2}{m_T}(m^2/L_x^2 + n^2/L_Y^2) + (R_x m^2/L_x^2 + R_y n^2/L_Y^2)/\rho_f h}^{1/2}
$$

It should be noted that the average value of the radius $R$ is different for the interior and exterior window sheets.

External Noise Pressure Field

The near field surface sound pressures $p^f$ 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 $p^f$ can be written as

$$
S_{p_{mn}}(\varepsilon,\sigma,\omega) = S_p(\varepsilon,\sigma,\omega) R_x(x_1,\ldots,x_n) R_y(y_1,\ldots,y_n)
$$

where the integration in equation (35) spans the surface area of the panel. 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.,

$$
S_{p_{mn}}(\varepsilon,\sigma,\omega) = S^F(\omega) R_x(x_1,\ldots,x_n) R_y(y_1,\ldots,y_n),
$$

where $S^F(\omega)$ is the spectral density of the random pressure. $R_x$ 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. However, tests indicate 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

$$S_p(\xi, n, \omega) = S^E(\omega) \exp[i\omega t V_x] \exp[i\omega n V_y]$$

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 generalized random forces corresponding to propeller noise can be calculated using equations (35) and (36).

Boundary Layer Turbulence. The surface pressures to convecting turbulent boundary layer flow are described by the semi-empirical cross-spectral density forms. 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

$$S_p(\xi, n, \omega) = \begin{cases} K & 0 < \omega < \omega_u \\ 0, \text{ otherwise} \end{cases}$$

where $\omega_u$ is the upper cut-off frequency and $K$ is a constant which measures the noise intensity.

Reverberant Field. The spatial correlation coefficients are taken to be

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

where $k = \omega/c$. Substituting these relations into equation (35) and assuming panel modes $X_{mn}(x, y) = \sin(m\pi x/L_x) \sin(n\pi y/L_y)$, the cross-spectral density of the generalized random forces for $m = r$ and $n = s$ is

$$S_{rr}(\omega) = S^E(\omega) \cdot I_r(\omega) \cdot I_n(\omega)/M_{rr}^2$$

where

$$I_r(\omega) = (I_{1m} + I_{2m} + I_{3m}) L_x^2$$
$$I_{1m} = \frac{1}{2\pi k N} \left[ \sinh(k L_x) \sinh(m L_x) - \cosh(m L_x) \sinh(k L_x) \right]$$
$$I_{2m} = \frac{1}{2\pi k N} \left[ \sinh(m L_x) \sinh(k L_x) - \cosh(k L_x) \sinh(m L_x) \right]$$
$$I_{3m} = \frac{1}{4\pi^2 k N} \left[ (m^2 - k^2 L_x^2) \right]$$

where $\sinh$ and $\cosh$ are the sine and cosine integrals. The expression for $I_r(\omega)$ is 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 using an acoustic guide (Fig. 1) 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. x 20 in. or 30 in. x 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. 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 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 enclosures 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 Plexiglass with panel thicknesses of 0.14 in. The noise transmission through these panels is calculated for a spatially uniform Gaussian white noise input described by equation (37). 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 \( \zeta_{\text{mn}} = \zeta_0 (\omega_{\text{mn}} / \omega_{\text{mn}}) \) and \( \zeta_{\text{ijk}} = \zeta_0 (\omega / \omega_{\text{wijk}}) \), respectively. The \( \zeta_0 \) and \( \zeta_0 \) are the damping coefficients for the fundamental modes, \( \omega_{\text{mn}} \) and \( \omega_{\text{wijk}} \) are the structural and acoustical modal frequencies, and \( \omega \) is the lowest acoustic modal frequency in the enclosure. For the present analytical study, it was assumed that \( \zeta_0 \) equals 0.01 for single panels, 0.02 for discretely stiffened panels, 0.03 for honeycomb treated panels, and 0.05 for the Plexiglass windows. The damping coefficient \( \zeta_0 \) 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.

In calculating the 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

\[
Z(\omega) = \rho_f c_f \left[ 1 + 0.0571 \frac{\rho_f \omega^2}{2 R_f} \right]^{0.754} \left[ 1 - 0.087 \frac{\rho_f \omega^2}{2 R_f} \right]^{0.732}
\]

where \( \rho_f \), \( c_f \), and \( R_f \) are the density, speed of sound and flow resistivity of the wall material, respectively. For an acoustically transparent condition \( \text{no wall} \), the point impedance reduces to the characteristic impedance \( \rho_f c_f \), while for a rigid wall condition \( Z(\omega) \rightarrow \infty \). To simulate the conditions of the lightly treated aircraft wall and the nearly rigid surfaces of the receiving room of the noise transmission loss apparatus, the values of \( R_f = 4 \times 10^4 \text{ mks raysl/m} \) and \( R_f = 4 \times 10^5 \text{ mks raysl/m} \) 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 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 honeycomb panels were added to the aircraft sidewall. The combined stiffness (panel + honeycomb) can be approximated by replacing the flexural rigidity of an untreated panel \( D = Eh^3/12(1 - v^2) \) by

\[
D = \frac{E}{(1 - v^2)} \left[ \frac{h_1^3}{12} + \frac{h_2^2}{12} + \frac{h_1 h_2}{12} \left( \frac{h_1}{2} + \frac{h_2}{2} + \frac{h_c}{2} \right)^2 \right]
\]
where $E$, $\nu$, $h_1$, $h_2$, and $h_c$ are the stiffness modulus, Poisson's ratio, thicknesses of skin panel, face plate, and honeycomb core, respectively. However, for discretely stiffened panels with honeycomb 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 with $h_c = 0.125$ in. and $h_2 = 0.02$ 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. $\times$ 12 in. $\times$ 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). Similar results are presented in Fig. 10 for the same panel but treated with a honeycomb add-on design for which $h_c = 0.125$ in. and $h_2 = 0.032$ in. 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 honeycomb treated panel. Thus, no gains in noise reduction are achieved with honeycomb treatment in the stiffness controlled region (below the panel fundamental frequency (235 Hz). For frequencies at and above the panel fundamental frequency, fair agreement between theory and experiment is obtained. However, even these results might be influenced by the noise transmitted by the entire wall.

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 provides additional noise attenuation. However, large gains of noise reduction measured for individual panels might not be obtained when the entire sidewall is considered.

References


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

Figure 2. Top view of the noise transmission loss apparatus.

Figure 3. Panel identification for port side.

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

Figure 5.- Noise reduction of a stiffened aircraft panel: noise input from acoustic guide.

Figure 6.- Noise reduction of an aircraft sidewall for a diffuse noise input.
Figure 7.- Noise reduction of a double wall aircraft window.

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

Figure 9.- Noise reduction of an untreated panel for transmission loss apparatus.

Figure 10.- Noise reduction of a panel stiffened with honeycomb for transmission loss apparatus.