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Cabin Noise Control
For Twin Engine
General Aviation Aircraft

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Modern Analysis Inc.
Ridgewood, New Jersey

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ENGINE GENERAL AVIATION AIRCRAFT

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1. Summary

An analytical model has been developed to predict the noise transmission into the cabin of a twin-engine G/A aircraft. This model is then used to optimize the interior A-weighted noise to an average level of 85 dBA. The basic concept of the analytical model is that of modal analysis wherein the acoustic modes in the cabin and the structural modes of the sidewalls are accounted for.

The noise input pressure due to propeller blade passage harmonics is expressed in the form of a propagating pressure field wherein noise spectral levels measured under static test conditions are used. The cabin interior is treated as a rectangular enclosure. The sidewalls of the aircraft are modeled by several discretely stiffened panel units. Transfer matrix techniques are used to calculate the natural frequencies and normal modes of the skin-stringer panels. The additional noise losses due to cabin sidewall treatments which do not have a direct effect on the structural dynamic characteristics of the skin-stringer panels are estimated by the impedance transfer method.

To reduce the average noise levels in the cabin from about 105 dBA (baseline) to 85 dBA (optimized), add-on treatments which do not involve changes in the fuselage primary structure are used. The add-on treatments considered in this optimization study include lightweight aluminum honeycomb panels, constrained layer damping tapes, porous acoustic blankets, septum barriers and limp trim panels. The added weight of the noise control treatment is about 1.1% of the total gross take-off weight of the aircraft.

2. Introduction

Significant advances in the theoretical formulation and analysis of noise transmission into aircraft have been made in the recent past [1 - 15]. Some of these analytical models have been validated by experiments ranging in complexity from a simple box with one elastic side [9, 12] to full-scale tests measuring the noise transmission through one sidewall of the G/A aircraft under a white noise input [12]. Preliminary noise transmission studies [4, 15] have shown that a satisfactory agreement between theory and experiment can be achieved for built-up sidewalls consisting of a stiffened elastic panel, a-coustic blankets, septum barriers and trim panel. In many of these studies, however, the main emphasis was placed on establishing correct and valid mathematical models of noise transmission, and not enough effort was devoted to deriving practical recommendations for the detailed and systematic evaluations of various methods of add-on treatment for reducing cabin noise to acceptable levels while adding the least amount of weight to the aircraft. In the present paper, a detailed analytical study of noise control with add-on treatments for a twin engine G/A aircraft is presented.

The basic concept of the analytical model is that of modal analysis wherein the acoustic modes in the cabin and the structural modes of the sidewall are accounted for [10 - 14]. The resonant and non-resonant response characteristics of the acoustic cavity and the sidewall panels are estimated by performing a narrow band analysis for frequencies up to 1122 Hz. In order to provide an analytical model which does not have very extensive computational requirements, various assumptions and simplifications are incorporated. The nearly flat floor, sidewalls and ceiling of the aircraft considered in this study of a 680 AeroCommander suggest that the cabin interior may be treated

as a rectangular enclosure. The rectangular shape allows for a simple representation of the acoustic modes rather than the more unwieldy finite element representations of the actual cabin shape. The sidewalls of the aircraft are modeled by several discretely stiffened panel units. The stiffeners and frames are included in the structural model either as discrete elements for the skin-stringer panels or as flexible boundaries for the selected stiffened panel unit. Transfer matrix techniques [12, 16, 17] are used to calculate the natural frequencies and normal modes of the skin-stringer panels. The dynamic characteristics of the double and single wall curved plexiglass windows are determined by performing an analysis on the windows with a dynamically "equivalent" single and flat plexiglass panel. The results from theoretical studies of double wall window constructions [18] are utilized for this purpose.

To estimate the noise losses due to add-on treatments such as acoustic blankets, septum barriers and trim panels, an analytical procedure based on the impedance transfer method is used [19, 20]. A reasonable agreement between these analytical predictions and experimental measurements has been achieved [15]. Differing add-on treatments lend differing amounts of noise reduction and differing amounts of weight to the aircraft. To optimize cabin noise for the least amount of added weight, a sensitivity analysis of the interior noise with respect to the different add-on treatments is performed. The add-on treatments considered in this noise optimization study include damping tape, acoustic blankets, septum barriers, honeycomb panels and trim panels.

To predict the amount of noise transmitted into the aircraft, the external excitation pressure field has to be defined. The noise input due to propeller blade passage harmonics can be expressed in terms of noise spectra levels, correlations and spatial pressure distributions. The surface pressure data corresponding to ground test conditions [21 - 23] are used for the analytical

study.

3. Analytical Model

The basic concept of the analytical model is that of modal analysis. This approach has been used for many noise transmission related problems [1, 6, 9-15]. For resonant acoustic cavities, lightly damped structural components and narrow band inputs such as propeller noise due to blade passage harmonics, modal analysis seems to be an attractive and efficient method to use.

3.1 Acoustic Model

Consider that the interior space of the aircraft shown in Figs. 1 and 2 can be approximated by a rectangular enclosure occupying a volume $V = abd$ as shown in Fig. 3. It is assumed that the main contribution to the interior noise is transmitted by the sidewalls at $z = o,d$ (shown by a dashed line in Fig. 2), and that the remaining surfaces are acoustically rigid. Figure 2 suggests that the floor members are of a heavy construction, which together with the greater distance from the propeller tips (Fig. 1) suggests that the floor transmits a relatively small amount of noise when compared to the airborne noise transmitted by the sidewalls. A similar assumption is invoked for the ceiling. The interior walls at $z = o,d$ are taken to be absorbent while the remaining walls are assumed to be acoustically hard. The contribution to noise losses due to absorption at these walls is included in the expression for the "equivalent" acoustic damping. Taking the perturbation pressure p to be at rest prior to the motions of the flexible sidewalls, the pressure inside the enclosure satisfies the acoustic wave equation

$$\nabla^2 p + \beta \frac{\partial p}{\partial t} = \frac{1}{c^2} \partial^2 p / \partial t^2 \tag{1}$$

where

$$\nabla^2 = \partial^2/\partial x^2 + \partial^2/\partial y^2 + \partial^2/\partial z^2 \qquad (2)$$

and β is the acoustic damping coefficient. The boundary conditions to be satisfied are

$$\partial p/\partial z = - \rho \ \partial p/\partial t / Z_A \qquad \text{at } z = d \qquad (3a)$$

$$\partial p/\partial z = - \rho \ \partial p/\partial t / Z_A - \rho \partial^2 w/\partial t^2 \big|_{S_F} \qquad \text{at } z = 0 \qquad (3b)$$

$$\partial p/\partial n = 0 \qquad \text{otherwise} \qquad (3c)$$

where $\partial p/\partial n$ is the normal derivative at the wall surface, Z_A is the absorbent wall impedance assumed to be uniformly distributed over the wall surface and w is the displacement of the flexible wall S_F.

Taking the Fourier transformation of Eqs. 1 - 3 and writing the solution for the acoustic pressure in terms of orthogonal acoustic modes corresponding to hard walls at x = o,a and y = o,b, yields

$$\overline{p}(x,y,z,\omega) = \sum_{i=0}^{\infty} \sum_{j=0}^{\infty} \phi_{ij}(z,\omega) \ X_{ij}(x,y) \qquad (4)$$

where ϕ_{ij} are the modal coefficients and X_{ij} are the acoustic modes of the rectangular enclosure

$$X_{ij}(x,y) = \cos \frac{i\pi x}{a} \cos \frac{j\pi y}{b} \qquad (5)$$

where a bar indicates a transformed quantity. By expanding the transformed flexible wall motions, $\overline{w}$, in terms of the acoustic modes and utilizing the

transformed boundary conditions given by Eq. 3, it can be shown that

$$\phi_{ij}(z,\omega) = \{G_{ij}(\omega)/(\lambda_{ij} - \rho^2\omega^2/Z_A^2\lambda_{ij}) \sin \lambda_{ij}d\}$$

$$\cdot \{\cos \lambda_{ij}(d - z) + (i\rho\omega/Z_A\lambda_{ij}) \sin \lambda_{ij}(d - z)\} \qquad (6)$$

where

$$\lambda_{ij} = \{\omega^2 - \omega_{ij}^2 - 2i\xi_{ij}\omega_{ij}\omega\}^{\frac{1}{2}}/c \qquad (7)$$

in which $\omega_{ij} = c\{(i\pi/a)^2 + (j\pi/b)^2\}^{\frac{1}{2}}$ and $\xi_{ij} = \beta c^2/2\omega_{ij}$. In the present

study, the quantity βc^2 is taken to be equal to $2\omega_{10}\xi_0$ where ξ_0 is the

"equivalent" acoustic damping coefficient and ω_{10} is the lowest modal fre-

quency in the enclosure. The acoustic-structural coupling is reflected in

the term G_{ij} where

$$G_{ij}(\omega) = \frac{\rho\omega^2 e_{ij}}{ab} \int_{a_0}^{a_0+L_x} \int_{b_0}^{b_0+L_y} \overline{w}(x,y,\omega)X_{ij}(x,y)dxdy \qquad (8)$$

and

$$e_{ij} = \begin{cases} 1 & i = 0, j = 0 \\ 2 & \text{either } i \neq 0 \text{ or } j \neq 0 \\ 4 & i \neq 0, j \neq 0 \end{cases} \qquad (9)$$

The solution for the perturbation pressure given in Eqs. 4 - 9 is in terms of

the flexible wall motions $\overline{w}(x,y,\omega)$. Thus, the response characteristics of the

flexible elastic panels must be determined next.

3.2 Response of Sidewall Panels

The flexible portion of the aircraft sidewalls shown in Figs. 1 and 2 is composed of an external skin which is stiffened by stringers and frames, thermal and acoustic insulation, acoustic barriers, trim panels and several single- and double-wall window units. While it is possible to model a sidewall by a single panel unit, it is much more feasible to break the sidewall down into smaller units. Such a segmentation offers significant advantages for noise transmission path identification and interior noise optimization. Due to the rapid spatial decay of the amplitude of the external noise pressure and the very stiff boundary conditions of some of the panel units, such an approximation seems to be justified for this type of fuselage construction. The details of a segmented sidewall are shown in Fig. 4.

The governing equation of motion for a single elastic panel located at $z = 0$, $a_0 \leq x \leq a_0 + L_x$, $b_0 \leq y \leq b_0 + L_y$, can be written in the frequency domain as

$$D\nabla^4\overline{w} + i\omega\zeta\overline{w} - m_s\omega^2\overline{w} = \overline{p}^r(x,y,\omega) - \overline{p}(x,y,0,\omega) \tag{10}$$

where D is the plate stiffness, ζ is the structural damping coefficient, m_s is the panel mass per unit area, $\overline{p}^r$ is the random external noise pressure, $\overline{p}(x,y,0,\omega)$ is the cavity back pressure acting on the interior surface of the panel, and ∇^4 is the biharmonic operator. In the present analysis it is assumed that the viscous damping coefficient can be expressed as a linear combination of the mass and stiffness so that the resulting modal equations will not be coupled through the structural damping term. Furthermore, it is assumed that the acoustic blankets, septum barriers and trim panels have no direct effect on the response of the elastic panels. The noise reduction due to these treatments is estimated separately.

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The solution for the panel deflection $\bar{w}$ is expressed in terms of the panel modes:

$$\bar{w}(x,y,\omega) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \bar{q}_{mn}(\omega) Y_{mn}(x,y) \tag{11}$$

where $\bar{q}_{mn}$ are generalized coordinates and Y_{mn} are orthogonal panel modes. In the remainder of this work, the indices m and n will be used to denote quantities related to the plate motion, and i,j will refer to the fluid motion. Substitution of Eq. 11 into Eq. 10 and utilization of the orthogonality principle yields

$$\bar{q}_{mn} = H_{mn}[P_{mn}^r - \frac{\rho\omega^2}{abM_{mn}} \sum_{i=0}^{\infty} \sum_{j=0}^{\infty} Z_{ij}(0,\omega) \sum_{r=1}^{\infty} \sum_{s=1}^{\infty} \bar{q}_{rs} L_{ijrs} L_{ijmn}] \tag{12}$$

where the frequency response function of the panel is

$$H_{mn} = [\omega_{mn}^2 - \omega^2 + 2i\zeta_{mn}\omega_{mn}\omega]^{-1} \tag{13}$$

and

$$Z_{ij}(z,\omega) = \{e_{ij}/(\lambda_{ij} - \rho^2\omega^2/Z_A^2\lambda_{ij}) \sin \lambda_{ij}d\}\{\cos \lambda_{ij}(d - z)$$

$$+ (i\rho\omega/Z_A\lambda_{ij}) \sin \lambda_{ij}(d - z) \tag{14}$$

in which $\zeta_{mn} = \zeta_0(\omega_{11}/\omega_{mn})$ where ζ_0 is the damping coefficient and ω_{mn} are modal frequencies. The generalized random forces due to external noise pressure are

$$P_{mn}^r = \frac{1}{M_{mn}} \int_{a_0}^{a_0+L_x} \int_{b_0}^{b_0+L_y} \bar{p}^r(x,y,\omega) Y_{mn}(x,y) dxdy \tag{15}$$

The response of a stiffened panel can be obtained from Eqs. 11-15 by replacing the modes Y_{mn} and the frequencies ω_{mn} by those corresponding to a stiffened panel. Then, the generalized mass M_{mn} can be written as

$$M_{mn} = m_s \int_{a_0}^{a_0+L_x} \int_{b_0}^{b_0+L_y} Y_{mn}^2(x,y)dxdy + \sum_{k=1}^{N} m_k \int_{a_0}^{a_0+L_x} Y_{mn}^2(x,y_k)dx \quad (16)$$

in which m_k is the mass per unit length of the k-th stringer, and $y_k = \sum_{i=1}^{k} b_i$ where b_i is the distance between the i-1 and i-th stringers. The quantities L_{ijmn} of Eq. 12 are parameters which couple the panel vibration modes to the acoustic cavity modes and are defined as

$$L_{ijmn} = \int_{a_0}^{a_0+L_x} \int_{b_0}^{b_0+L_y} Y_{mn}(x,y)X_{ij}(x,y)dxdy \quad (17)$$

In order to completely determine the panel motion and the acoustic pressure inside the cabin, the coupled system of Eqs. 12 must be solved for $\overline{q}_{mn}$. It should be noted that Eq. 12 is general and accounts for all the coupling effects between the panel and the cavity. However, some simplifying assumptions, suggested by previous work [24 - 29] can circumvent the lengthy numerical solution of the coupled system for $\overline{q}_{mn}$. For cases where the cavity is sufficiently deep and the panel is sufficiently stiff, the coupling terms in Eq. 12 due to back-up acoustic pressure can be neglected and the structural modal coefficients $\overline{q}_{mn}$ can be computed explicitly in terms of the generalized random forces P_{mn}^r. Then, from Eqs. 11 and 12, the panel response can be expressed as

$$\overline{w}(x,y,\omega) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} H_{mn}(\omega) \, P_{mn}^r(\omega) \, Y_{mn}(x,y) \quad (18)$$

The cross-spectral density of the panel deflection response $S_w(x_1, x_2; y_1, y_2; \omega)$ can be obtained by taking the mathematical expectation of Eq. 18 and then using the spectral decomposition presented in Ref. 17. By setting $x_1 = x_2 = x$, $y_1 = y_2 = y$, the spectral density of the panel deflection response is

$$S_w(x,y,\omega) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \sum_{r=1}^{\infty} \sum_{s=1}^{\infty} H_{mn}(\omega) \, H_{rs}^*(\omega) \, S_{mnrs}^r(\omega)$$

$$\cdot \, Y_{mn}(x,y) \cdot Y_{rs}(x,y) \tag{19}$$

where the cross-spectral density of the generalized random input forces is

$$S_{mnrs}(\omega) = \frac{1}{M_{mn}M_{rs}} \int_{a_0}^{a_0+L_x} \int_{a_0}^{a_0+L_x} \int_{b_0}^{b_0+L_y} \int_{b_0}^{b_0+L_y} S^e(\bar{x},\bar{y},\omega) \cdot Y_{mn}(x_1,y_1)$$

$$\cdot \, Y_{rs}(x_2,y_2) dx_1 dx_2 dy_1 dy_2 \tag{20}$$

in which S^e is the cross-spectral density of the random input noise pressure p^r and $\bar{x} = x_2 - x_1$, $\bar{y} = y_2 - y_1$. The asterisk in Eq. 19 denotes a complex conjugate. To complete the solution for panel deflections and subsequently for the interior acoustic pressure, the natural frequencies and normal modes of the elastic panels are needed.

3.3 Natural Frequencies and Normal Modes of Sidewall Panels

In the present study, we seek the natural frequencies and normal mode shapes of the two-dimensional stiffened panels shown in Fig. 5 and double wall curved window constructions shown in Fig. 6. These results are used in Eqs. 11 - 17 to calculate the deflection response. The natural frequencies, ω_{mn}, and normal modes, Y_{mn}, of the stiffened panel unit shown in Fig.

5 are determined by using transfer matrix methods [12, 16, 17]. A brief description of this procedure will now be given. The stiffened panel No. 6 shown in Figs. 4 and 5 is assumed to be simply supported along the frames at $x = a_0$ and $x = a_0 + L_x$. Then, the normal modes corresponding to the direction along the stringers are $\sin\{(n\pi/L_x)(x - a_0)\}$. Substitution of this relation into Eq. 10 and setting $\overline{p}^r$ and $\overline{p}(x,y,0,\omega) = 0$ results in a fourth order homogeneous differential equation for each n. Utilizing the characteristic roots, the solution for 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 the components of deflection, slope, moment and shear, respectively. A transfer matrix $_N^r[T]_0^\ell$ can then be constructed which transfers the state vector from the left side of station 0 to the right side of station N (see Fig. 5). Then, we can write

$$\{W_n\}_N^r = {}_N^r[T]_0^\ell \{W_n\}_0^\ell \tag{21}$$

where

$$_N^r[T]_0^\ell = [G]_N \, [F]_N \, [G]_{N-1} \, \cdots \, [F_1][G]_0 \tag{22}$$

The point matrix [G] transfers the state vector across a stringer and the field matrix [F] transfers the state vector across a panel. The detailed expressions for these transfer matrices can be found in Appendix A. Utilizing the natural boundary conditions at $y = b_0$ and $y = b_0 + L_y$ in Eq. 21 yields an equation which can then be solved for the natural frequencies ω_{mn}. For the purpose of illustration, assume simply supported boundaries at the end stations 0 and N for which $\{W_n\}_0^\ell = \{0, \theta_n, 0, V_n\}_0^\ell$ and $\{W_n\}_N^r = \{0, \theta_n, 0, V_n\}_N^r$. Then, from Eq. 21, we obtain

$$\begin{bmatrix} 0 \\ 0 \end{bmatrix}^r_N = {}^r_N\begin{bmatrix} t_{12} & t_{14} \\ t_{32} & t_{34} \end{bmatrix}^{\ell}_0 \begin{bmatrix} \theta_n \\ V_n \end{bmatrix}^{\ell}_0 \tag{23}$$

where t_{ij} are the elements of the transfer matrix ${}^r_N[T]^{\ell}_0$. For a non-trivial solution, the determinant of the coefficient matrix in Eq. 23 vanishes, resulting in a transcendental frequency equation which is solved numerically. Due to ill-conditioning of the field transfer matrix [F] (hyperbolic functions with large arguments) and large stiffness from the stringers, numerical difficulties can arise in calculating the natural frequencies. These difficulties can be circumvented by dividing each panel between two stringers into several segments. Now the field transfer matrix can be written as [F] = [F(ℓ_1)][F(ℓ_2)] .... [F(ℓ_M)] where ℓ_i is the length of a segment and M is the total number of selected segments. By calculating the products of these matrices, the large arguments of the hyperbolic functions are avoided.

The mode shapes, Y_{mn}, are obtained from

$$Y_{mn}(x,y_q) = {}^r_q[t_{11} \quad t_{12} \quad t_{13} \quad t_{14}]^{\ell}_0 \begin{bmatrix} 0 \\ \theta_n \\ 0 \\ V_n \end{bmatrix}^{\ell}_0 \cdot \sin \frac{m\pi(x-a_0)}{L_x} \tag{24}$$

where y_q is the local co-ordinate locating an arbitrary point on the stiffened panel. From Eq. 23, the slopes θ_n can be expressed in terms of the shears V_n at the natural frequencies ω_{mn}.

In the present study, the natural frequencies and normal modes of the stiffened panels shown in Fig. 4 were estimated for flexible supports at the stringer boundaries, i.e., at stations 0 and N. For panel No. 6, these sup-

ports are along the x-boundary, while for panel Nos. 1-4, the flexible supports are along the y-boundary. Such a condition is believed to be a close approximation of an actual boundary of a continuously stiffened elastic panel.

In addition to the stiffened panel units, each aircraft sidewall contains three plexiglass windows. The port side pilot window is a single sheet curved panel while all the other windows are double wall constructions as shown in Fig. 6. Due to the very complex geometry of these windows, the natural frequencies and normal modes are estimated using more tractable analytical models. In this procedure, the double wall constructions are replaced with "equivalent" single sheet curved panels which are simply supported on all four edges. Furthermore, the irregular shape of window unit No. 8 is approximated by a rectangular panel. Then, the natural frequencies and normal modes are calculated using the expressions given in Ref. 30. However, to establish the equivalence criteria between the double wall and the single wall constructions, the natural frequencies and normal modes of the single sheet panels are modified utilizing the results from an analytical study of double wall windows [18]. Since the numerical results of Ref. 18 were available only for a few lower modes and frequencies, the equivalence criteria are established only in approximation.

3.4 Acoustic-Structural Model

The equations developed in previous sections can be combined to construct a noise transmission model. Then, from Eqs. 4 - 8 and 11 - 15, the acoustic pressure inside the cabin is

$$\bar{p}(x,y,z,\omega) = \frac{\rho\omega^2}{ab} \sum_{i=0}^{\infty} \sum_{j=0}^{\infty} Z_{ij}(z,\omega)X_{ij}(x,y) \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} H_{mn}(\omega)L_{ijmn}P_{mn}^{r}(\omega) \qquad (25)$$

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The cross-spectral density of the interior acoustic pressure $S_p(x_1, x_2; y_1, y_2; z_1, z_2; \omega)$ can be obtained by taking the mathematical expectation of Eq. 25 and using the procedures given in Ref. 17. By setting $x_1 = x_2 = x$, $y_1 = y_2 = y$, $z_1 = z_2 = z$, the spectral density of the interior acoustic pressure is

$$S_p(x,y,z,\omega) = (\rho\omega^2/ab)^2 \sum_{i,j,k,\ell=0}^{\infty} Z_{ij}(z,\omega) Z_{k\ell}^*(z,\omega) X_{ij}(x,y) X_{k\ell}(x,y)$$

$$\cdot \sum_{m,n,r,s=1}^{\infty} H_{mn}(\omega) H_{rs}^*(\omega) L_{ijmn} L_{k\ell rs} S_{mnrs}(\omega) \qquad (26)$$

where the cross-spectral density of the generalized random forces, S_{mnrs}, is given in Eq. 20. The sound pressure levels in the cabin are obtained from

$$SPL(x,y,z,\omega) = 10 \log \{S_p(x,y,z,\omega)\Delta\omega/p_0^2\} \qquad (27)$$

where $\Delta\omega$ is the selected bandwidth and p_0 is the reference pressure ($p_0 = 2.9 \times 10^{-9}$ psi, $p_0 = 20 \ \mu N/m^2$).

Noise reduction is defined as the quantity which relates the interior acoustic pressure spectral density S_p to the exterior input pressure spectral density $S_p^e(\omega)$,

$$NR(x,y,z,\omega) = 10 \log \{S_p^e(\omega)/S_p(x,y,z,\omega)\} \qquad (28)$$

The interior noise levels given by Eq. 27 correspond to the noise transmitted by a single stiffened panel or window unit located at z = 0. The total noise transmitted by all the panel units composing the entire sidewall can be determined by superposition of the contributions from each panel unit. In this case, the inputs of each panel unit are taken to be uncorrelated and the motions of each panel unit are assumed to be independent. Then, the in-

terior sound pressure levels transmitted by the entire sidewall can be estimated from

$$SPL^T(x,y,z,\omega) = 10 \log \{ \sum_{i=1}^{M} S_p^i(x,y,z,\omega)\Delta\omega/p_0^2 \} \tag{29}$$

where M is the total number of selected panel units of which the aircraft sidewall is composed.

The solution for the spectral density of the interior noise pressure given in Eq. 26 can be separated into two parts. The first part contains direct terms for which i = k, j = ℓ, m = r, n = s; and the second part contains cross-modal terms for which i ≠ k, j ≠ ℓ, m ≠ r, n ≠ s. A common practice in modal analysis is to neglect the cross-modal terms, since for low damping, well separated modal frequencies and slowly varying input spectral densities, the contribution from these terms to the total response is usually small. However, for rapidly varying inputs such as propeller noise (shown in Figs. 7 - 13) and the anticipated large damping values of the treated structures, the cross-modal terms can have a significant effect on noise reduction at certain frequencies. This is illustrated in Fig. 19 where noise reduction is plotted for a typical stiffened panel unit with and without cross-modal terms. Due to the significance of these results, cross-modal terms are included in the analytical model of the present study. Furthermore, utilizing the computer algorithms available for complex numbers, makes the additional computational time and costs needed to include these terms negligible.

The convection effects for the surface pressures due to propeller rotation are included in the formulation of the generalized random forces. The

A-weighted interior noise levels are shown in Fig. 20 for cases with and without convection. The transmitted noise levels are in general lower when the convection effects of the propeller noise are included. Furthermore, the convection velocities seem to have more effect at higher frequencies. These results correspond to non-decaying spatial correlation functions with a convection velocity of 700 ft/sec along the propeller rotation and sonic convection velocity normal to the propeller rotation plane.

3.5 Acoustic Absorption

When calculating the transmitted noise into the cabin, it is assumed that the interior surfaces are locally reacting such that the interior absorption at the boundary can be represented by a point impedance model $Z_A(\omega)$. To simplify the numerical calculations, it is further assumed that Z_A is uniformly distributed over the interior cabin sidewalls. The point impedance is modeled by

$$Z_A(\omega) = R(\omega) + i \, X(\omega) \tag{30}$$

where the resistance R and the reactance X are given by [19,31]

$$R(\omega) = \rho c [1 + .0571(2\pi R_1/\rho\omega)^{0.752}] \tag{31}$$

$$X(\omega) = \rho c [0.087(2\pi R_1/\rho\omega)^{0.722}] \tag{32}$$

where ρc is the characteristic impedance of the air and R_1 is the flow resistivity of the porous materials. In addition to the acoustic absorption at the sidewalls, the acoustic absorption due to the interior furnishings, carpeting, passengers, etc. need to be accounted for. In the present study, we assume that the acoustic power losses in the cabin due to these treatments

can be represented by a modal acoustic damping coefficient $\xi_{ij} = \xi_0(\omega_{10}/\omega_{ij})$ where ξ_0 is a prescribed damping coefficient of the lowest acoustic mode in the cabin and ω_{ij} are the modal acoustic frequencies. Even though it is possible to obtain a relationship between ξ_{ij} and Z_A [6], it is difficult to prescribe proper values for Z_A to account for the energy losses due to the furnishings, passengers, etc. In the present study, noise transmission was estimated for $\xi_0 = 0.03$ (baseline aircraft), $\xi_0 = 0.09$ (medium acoustic damping) and $\xi_0 = 0.18$ (large acoustic damping). The following values for flow resistivity were chosen:

$$R_1 = 12.5 \text{ lb-sec/in}^4 \qquad\qquad f < 125 \text{ Hz}$$

$$R_1 = 25 \text{ lb-sec/in}^4 \qquad\qquad 125 \text{ Hz} \leq f \leq 250 \text{ Hz}$$

$$R_1 = 62.5 \text{ lb-sec/in}^4 \qquad\qquad f > 250 \text{ Hz}$$

3.6 External Pressure Field

The external pressure acting on the aircraft is propeller noise due to blade passage harmonics. The experimental information on surface pressures [21-23] for ground test conditions is used to select the input pressure levels for each panel unit shown in Fig. 4. These sound pressure levels, digitized at 2 Hz bands, are shown in Figs. 7-13 for the ten panel units of which the sidewall of the aircraft is composed. To include convection effects in the present analytical model, the cross-spectral density of the input pressure is assumed to be separable in the direction of propagation and that perpendicular to it and is given as

$$S_i^e(\bar{x},\bar{y},\omega) = S_i(\omega) \, e^{i\omega\bar{x}/V_x} \, e^{i\omega\bar{y}/V_y} \qquad\qquad (33)$$

where $S_i(\omega)$ is the input power spectral density for the i-th panel unit, and V_x and V_y are the trace velocities corresponding to the x- and y-directions, respectively. Utilizing the information available in Ref. 23, subsonic trace velocities corresponding to the vertical direction y (along the propeller rotation) and sonic trace velocities corresponding to the longitudinal direction x (normal to the propeller rotation plane) were chosen in this study. The values of V_y = 700 ft/sec and V_x = 1120 ft/sec were used for all the numerical computations.

The cross-spectral density of the generalized random forces due to propeller noise input can be evaluated from Eqs. 20 and 33. In this procedure, the pressure levels characterized by the spectral density $S_i(\omega)$ are taken to be uniformly distributed over each panel surface, but varying in a step-wise fashion from one panel unit to another. For stiffened panels, the modes $Y_{mn}(x,y)$ are prescribed numerically. Thus, numerical integration routines need to be utilized to calculate the generalized random forces defined in Eq. 20.

3.7 Total TL Including Add-On Treatment

The analytical model described in Sections 3.1 - 3.7 predicts noise transmission through a fuselage sidewall with or without treatments which are directly attached to the skin of the sidewall. These add-on treatments could include damping tape, honeycomb panels, non-load carrying mass, etc. In addition, the effect of acoustic absorption within the cabin is included according to the procedures described in Section 3.5. To estimate the noise losses due to acoustic blankets, septum barriers and trim panels, an analytical procedure based on the impedance transfer method is used [19,20]. A reasonable agreement between these analytical predictions and experimental measurements has been

achieved for a typical aircraft panel with add-on treatments [4,15]. Then, the interior noise levels in the treated aircraft are estimated by adding the noise losses calculated by the impedance transfer method (acoustic blankets, septum barriers, trim panels) to the noise levels obtained directly by modal analysis from Eqs. 27 and 29. Then, the interior noise transmitted through a panel with acoustic add-on treatments is calculated from

$$SPL(x,y,z,\omega)|_{treated} = SPL(x,y,z,\omega)|_{untreated} + \Delta TL(\omega) \tag{34}$$

where ΔTL is the additional noise loss provided by the add-on acoustic treatments. It should be noted that the effect of treatments which are directly attached to the aircraft skin (damping tape. non-load carrying mass, honeycomb panels) is included in the term $SPL(x,y,z,\omega)|_{untreated}$. Since these treatments have a direct effect on the dynamic characteristics of the load bearing external skin, the ΔTL term, derived from a simplistic impedance transfer method which assumes uniformly treated panels of infinite extent, will not account properly for the additional noise losses from the add-on treatments directly attached to the elastic skin of the sidewall.

The added transmission loss ΔTL at an incident plane wave angle θ_1 (see Fig. 14) is obtained from

$$\Delta TL(\omega,\theta_1) = - 10\log \tau(\omega,\theta_1) \tag{35}$$

where τ is the transmission coefficient of the add-on acoustic treatment defined as [5]

$$\tau(\omega,\theta_1) = \left| \frac{(p_1/p_2)_{untreated}}{(p_1/p_2) \cdots (p_{n-1}/p_n)_{treated}} \right|^2 \tag{36}$$

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where p_{n-1}/p_n are the pressure ratios across the boundaries between adjacent media and the pressure ratios across the media themselves as shown in Fig. 14. Acoustic plane waves are incident on the exterior of the structure with an angle θ_1 and reflected and transmitted according to the various impedances present at the different layers. The interior of the media is assumed to extend to infinity with an acoustic termination impedance ρc. To obtain the pressure ratios p_{n-1}/p_n, the expressions for the characteristic impedances of the various media are utilized.

Following Refs. 5, 19 and 20, and using Fig. 14, the pressure ratio across the untreated elastic panel is

$$(p_1/p_2)_{untreated} = 1 + Z_p/Z_2 \tag{37}$$

where the impedance, Z_p, of an infinite stiffened panel can be written as [5]

$$Z_p = \omega\mu_1[X\eta + \underline{i}(1 - X)] \tag{38}$$

in which

$$X = \omega^2 \sin^4\theta_2\Delta/(\mu_1 c_2^4) \tag{39}$$

$$\theta_2 = \sin^{-1}(c_2 \sin\theta_1/c_1) \tag{40}$$

$$\Delta = D_x \cos^4\phi + 2H \cos^2\phi \sin^2\phi + D_y \sin^4\phi \tag{41}$$

D_x, D_y = bending rigidity of the stiffened panel in sections perpendicular to x-axis and y-axis, respectively

H = cross-rigidity of the panel

ϕ = asimuthal angle relative to x-axis [5]

μ_1 = surface density of the sidewall

η = loss factor of the sidewall

The impedance for the interior acoustic medium is

$$Z_2 = \rho c/\cos \theta_2 \tag{42}$$

The pressure ratios across the different layers of the medium shown in Fig. 14 are estimated in the following fashion. The pressure ratio across the elastic panel bounded by porous acoustic blankets on the interior side (located in space 2) is

$$(p_1/p_2)_t = 1 + Z_p/Z_2' \tag{43}$$

where Z_p is the elastic panel impedance given in Eq. 38 and Z_2' is the impedance of the acoustic blankets

$$Z_2' = Z_{B2} \coth (b_2 d_2 \cos\theta_2 + \psi_2)/\cos\theta_2 \tag{44}$$

where

$$Z_{B2} = - i \, K_2 b_2/\omega Y_2 \tag{45}$$

$$\theta_2 = \sin^{-1} (c_2 \sin\theta_1/c_1) \tag{46}$$

$$\psi_2 = \coth^{-1} (Z_3 \cos\theta_2/Z_{B2}) \tag{47}$$

$$c_2 = \omega/\beta_2; \quad \text{speed of sound in region 2} \tag{48}$$

$$b_2 = \alpha_2 + i\beta_2; \quad \text{propagation constant in region 2} \tag{49}$$

$$b_2 = i\omega(\overline{\rho}_2 Y_2/K_2)^{\frac{1}{2}}(1 - i\overline{R}_2/\overline{\rho}_2\omega)^{\frac{1}{2}} \tag{50}$$

$$\bar{\rho}_2 = \rho_2 \{ \frac{(R_1/\rho_m\omega)^2 (Y_2 + \rho_m/\rho) + 1}{1 + (R_1/\rho_m\omega)^2} \} \qquad (51)$$

$$\bar{R}_2 = R_1/(1 + (R_1/\rho_m\omega)^2) \qquad (52)$$

$$Y_2 = 1 - \rho_m/\rho_f; \quad \text{porosity} \qquad (53)$$

K_2 = compressibility of gas in porous material
($\simeq$ atmospheric pressure)

ρ_m = bulk density of sample

ρ_f = density of fibers

R_1 = flow resistivity of acoustic blankets

The septum impedance, Z_3, is

$$Z_3 = i\omega\mu_3 + Z_4 \qquad (54)$$

where μ_3 is the surface density of the septum material and Z_4 is the impedance of the acoustic blankets in region 4. The pressure ratio across the acoustic blankets of region 2 is

$$(\frac{p_2}{p_3})_t = \frac{\cosh(b_2 d_2 \cos\theta_2 + \psi_2)}{\cosh \psi_2} \qquad (55)$$

The pressure ratio for the septum is

$$(p_3/p_4)_t = Z_3/Z_4 \qquad (56)$$

where

$$Z_4 = Z_{B4} \coth(b_4 d_4 \cos\theta_4 + \psi_4) \tag{57}$$

$$\theta_4 = \sin^{-1}(c_4 \sin\theta_2/c_2) \tag{58}$$

$$c_4 = \omega/\beta_4 \tag{59}$$

The pressure ratio across the acoustic blankets in region 4 is

$$(p_4/p_5)_t = \frac{\cosh(b_4 d_4 \cos\theta_4 + \psi_4)}{\cosh \psi_4} \tag{60}$$

where

$$\psi_4 = \coth^{-1}(Z_5 \cos\theta_4/Z_{B4}) \tag{61}$$

The trim panel impedance, Z_5, is

$$Z_5 = i\omega\mu_5 + Z_6 \tag{62}$$

where μ_5 is the surface density of the trim panel and Z_6 is the impedance of the receiving space

$$Z_6 = \rho_6 c_6/\cos\theta_6 \tag{63}$$

in which

$$\theta_6 = \sin^{-1}(c_6 \sin\theta_4/c_4) \tag{64}$$

c_6 = speed of sound in the receiving space

ρ_6 = air density of the receiving space

The pressure ratio across the trim panel is

$$(p_5/p_6)_t = Z_5/Z_6 \qquad\qquad (65)$$

The acoustic properties of the porous blankets in region 4 can be evaluated from Eqs. 49-53 by substituting the parameters corresponding to those of region 4.

The added transmission loss given in Eq. 35 is calculated using the narrow band approach. It can be shown that the one-third octave transmission loss $\Delta TL|_{1/3}$ can be estimated from

$$\Delta TL(\theta)|_{1/3}^i = 10 \log \frac{\Delta\omega|_{1/3}^i}{\int_{\omega_\ell^i}^{\omega_u^i} (10^{-0.1\Delta TL(\theta,\omega)})d\omega} \qquad\qquad (66)$$

where $\Delta\omega|_{1/3}^i$ is the i-th bandwidth of the one-third octave frequency band and ω_ℓ^i and ω_u^i are the lower and upper frequencies, respectively, of the corresponding one-third octave frequency band.

4. Interior Noise Optimization

The analytical model described in previous sections is applied to optimize cabin noise to an average overall A-weighted level of about 85 dBA. The non-structural add-on treatments applied to the fuselage sidewalls of the aircraft were designed to have low surface density and high transmission loss. To achieve this goal, the following systematic procedure was undertaken.

An interior point in the propeller plane at about ear level and 8 inches from the sidewall was selected. Cabin noise was calculated at this point for all treatment conditions. Numerical computations were performed using a nar-

row band analysis for frequencies ranging from 0 - 1122 Hz and a bandwidth of $\Delta f = 2$ Hz. The one-third octave levels and the overall levels were obtained from the narrow band results and numerical integration. The optimization procedure was based on the A-weighted one-third octave noise levels. Then, the noise transmitted by each panel unit (Fig. 4) was calculated for each add-on treatment and for various combinations of add-on treatments. The amount of treatment was increased until a pre-selected acceptable A-weighted interior noise level was reached. In the present study, a level of 78 dBA for all one-third octave frequencies was selected as the upper bound for the noise transmitted by each individual panel unit. If the selected optimization goal is not satisfied, this noise level might have to be changed. The treatment or combination of treatments which reduces the transmitted noise to this value for the least amount of added weight was taken as the best treatment for that panel unit. The procedure was repeated for all panel units and the results were superimposed (including noise transmitted through windows) to determine the total transmitted noise in the cabin. The add-on treatments include honeycomb panels, damping tapes, non-load carrying mass, porous acoustic blankets, septum barriers and trim panels. A simplified flow chart of the optimization procedure is shown in Fig. 15. The computing prediction system consisted of three basic computer programs. The first program calculated the natural frequencies and mode shapes of the stiffened panels using the transfer matrix approach (Section 3.3), the second program estimated pressure levels in the cabin by the modal approach (Section 3.4), while the third program calculated the additional noise losses ΔTL (Section 3.7) due to add-on treatments not attached directly to the skin.

5.0 Numerical Results

5.1 Baseline Aircraft

The aircraft used in the present study is the Model 680 AeroCommander shown in Figs. 1 and 2. This aircraft has a take-off gross weight of about 7,000 lbs., cruises at an airspeed of 262 ft/sec at 10,000 ft altitude with each engine running at 70% power. Each engine has six cylinders, each engine is rated at 320 hp, and drives a 7.74 ft diameter three-bladed propeller through a gearing system that turns the propeller at about 64% of the engine rpm. The propeller tip clearance from the sidewall is about 5 inches and the propeller plane intersects the fuselage at approximately the middle of the cabin. The original cabin interior was finished in standard trim (acoustic blankets, trim panels) and provided seats for pilot, copilot and four passengers. The flight tests run at an altitude of 7,000 ft and 75% power (both engines running at equal power output) indicate that the A-weighted overall interior noise level varies with the position in the cabin and range from about 96 dBA to 103 dBA [9]. The ground tests under static operations [21 - 23] showed similar values for the interior noise in the cabin. The highest A-weighted noise levels occurred in the frequency range of about 100 - 600 Hz.

The baseline aircraft to which the analytical optimization models were applied was assumed to be an aircraft similar to the one described above, but with all the interior treatments (acoustic blankets, trim panels) and interior furnishings (carpeting, seats) removed. Furthermore, the mathematical model does not provide for noise transmission into the cabin through the rear bulkhead. The inputs to the baseline aircraft corresponding to ground tests under static operations were taken from Refs. 21 and 23.

5.2 Modes and Frequencies

The normal modes and natural frequencies of the stiffened elastic panels
were obtained using the transfer matrix procedures discribed in Section 3.3.
The modes and frequencies of the single and double window constructions were
determined by the procedures given in Refs. 18 and 30. The panel and string-
er geometries are shown in Figs. 4 and 16, respectively. The skin of all the
aluminum panels has a thickness of 0.04 inches except for Panel No. 2 which
has a thickness of 0.08 inches. The windows are made from 0.14 inch thick-
ness plexiglass. The geometric and material properties of the stringers are
given in Table 1 for each stiffened panel unit. Utilizing these data, the
natural frequencies and normal modes were calculated for all the panels shown
in Fig. 4 for a frequency range of 0 - 1122 Hz. The natural frequencies cor-
responding to the odd spanwise modes (n = 1,3,5,...) are given in Table 2.
Due to the vary large number of natural frequencies for the selected frequen-
cy range, only those frequencies below 400 Hz are included in this table. The
modal frequencies obtained by experimental measurements for the same aircraft
are included for comparison [33] . The mode shapes and frequencies were meas-
ured for Panel Nos. 1, 2, 6 and 8, using experimental modal analysis techniques.
For other panels, the modal frequencies were extracted from the frequency res-
ponse functions. The agreement between theory and experiment for estimating
the modal frequencies of the aircraft panels, in general, is satisfactory in
view of the fact that some stringers contained cut-outs and various attach-
ments which are difficult to include in a structural model. The experimental
frequencies given for Panel No. 2 correspond to a skin thickness of 0.04
inches while the theoretical results are for a panel with a thickness of 0.08
inches. The theoretical sidewall model corresponds to the starboard side

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where the thickness of Panel No. 2 is 0.08 inches. Honeycomb panels were attached to this panel. Thus, the frequencies and modes were measured for Panel No. 2 located on the port side which has a thickness of 0.04 inches. The experimental results include those frequencies corresponding to even and odd modes. However, due to the complexity of the modal shape of the stiffened panels and double wall windows, the separation of experimental data into even and odd modes is a difficult task. Several typical measured and calculated modes corresponding to the first frequency band are plotted in Fig. 17 for Panel No. 6.

5.3 Noise Levels in Baseline Aircraft

The interior noise levels in the aircraft due to propeller noise inputs taken from static ground tests were calculated utilizing the analytical models presented in this study. The narrow band ($\Delta f = 2$ Hz) interior sound pressure levels calculated at x = 50 in, y = 40 in, and z = 8 in, are shown in Fig. 18. In order to have a valid comparison between theory and experiment, analytical calculations were performed for interior conditions similar to those of the experimental aircraft. The sidewalls and ceiling of the experimental aircraft were treated with one-inch acoustic blankets and cloth-type interior trim. Furthermore, carpeting and all seats were left intact during the test. The interior noise levels were estimated for the baseline aircraft using modal analysis. (The baseline aircraft is a bare fuselage without treatments or furnishings.) Then, the additional noise losses due to sidewall treatments were added to these levels to arrive at the results shown in Fig. 18. In view of the complexity of the sidewall construction and the uncertainty of the treatment conditions, the agreement between theory and experiment is reasonable. Theory seems to predict higher noise levels

by about 5 - 8 dB for the first four blade passage harmonics. The limitations
of the analytical model with regard to uniform propeller noise pressure dis-
tributions and to the independent responses of each panel unit mean that it
tends to overestimate the transmitted noise levels. Furthermore, due to un-
certainty of the effectiveness of the original add-on treatments (acoustic
blankets, carpeting, cloth trim), the additional noise losses ΔTL calculated
by the impedance transfer method might be underestimated in the low frequency
range. It should be noted that theory was in very good agreement with the
laboratory tests where all the parameters could be carefully controlled or
measured [9, 12, 15].

5.4 Sidewall Treatment With Honeycomb Panels

The effect of the stiffening of the sidewall panels on noise transmission
is investigated. Additional stiffening is achieved by attaching honeycomb
panels to the interior walls of the aircraft as shown in the sketch below:
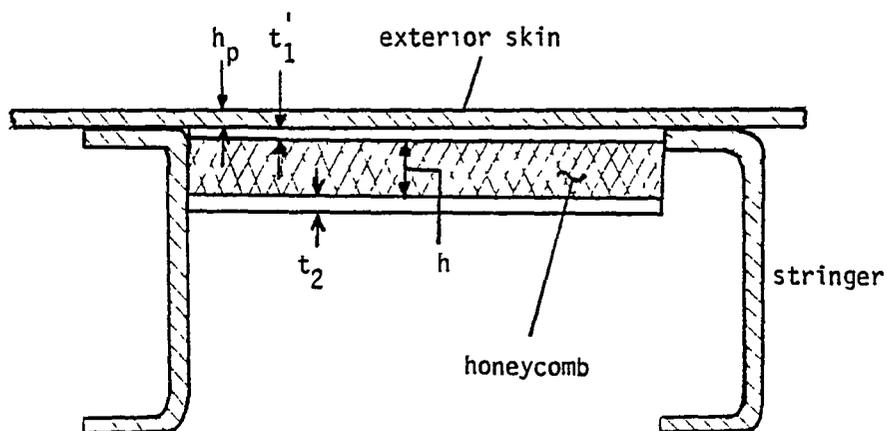
The transverse stiffness per unit width of the treated panel (elastic skin + honeycomb) is obtained from

$$D_t = \frac{E_1 \cdot E_2 t_1 t_2 h^2}{(E_1 t_1 + E_2 t_2)(1 - v^2)} \tag{67}$$

where E_1, E_2 are the effective moduli of the facings, v is Poisson's ratio of the facings ($v_1 = v_2 = v$), h is the distance between the facing centroids and t_1, t_2 are the thicknesses of the facings. In the present study, we take t_1 = $t_1' + h_p$ where t_1' is the thickness of the facing of the honeycomb panel. Numerical results were obtained for E_1 = E_2 = 10.0 x 10^6 psi, v = 0.3, t_1' = 0.0, t_2 = 0.016, h = 0.288, surface density of the honeycomb panel = 0.37 lb/ft^2 and t_1' = 0.032 in, t_2 = 0.032 in, h = 0.429 in, surface density of the honeycomb panel = 1.4 lb/ft^2. The natural frequencies and normal modes of the skin-stringer panels stiffened with the honeycomb construction were calculated using the transfer matrix procedures given in Section 3.3. The natural frequencies corresponding to the first frequency band of both the treated and untreated skin-stringer panels are presented in Table 3. Due to a significant increase in the total panel stiffness (aluminum panel + honeycomb panel), the modal frequencies shift to higher frequency values. The panels with light honeycomb treatment (h = 0.286 in) were assumed to be simply supported at the boundaries perpendicular to the stiffeners and elastically supported at the boundaries parallel to the stiffeners. The panels with heavy honeycomb treatment (h = 0.429 in) were taken to be simply supported on all four edges.

To estimate the effect of sidewall stiffening on noise transmission, the interior noise was calculated for several add-on treatments with honeycomb panels. The one-third octave A-weighted interior noise levels transmitted

by the baseline panels and panels with light honeycomb treatment are given in Figs. 21-26. (Due to the large stiffness of baseline Panel No. 9, no add-on treatment is attached to this panel). The overall noise levels for both treated and untreated panels are also given in these figures. The structural and acoustic damping is taken to be the same for both cases. As can be observed from these results, the effect on the noise transmission due to stiffening the sidewall panels varies from one panel unit to another. This kind of variation can be attributed to structural differences among the stiffened panel units, nonuniformity of surface noise pressure and input spectra composed of sharp peaks at the propeller blade passage harmonics. More noise is transmitted by Panel Nos. 1 and 4 when the stiffness of these panels is increased. However, for Panel Nos. 2, 3 and 6, a significant a-mount of noise reduction is achieved with the stiffening add-on treatment. The noise reduction achieved for Panel No. 2 with this treatment might not be realistic. Experiments [3] indicate a very strong coupling between the vibrations of Panel Nos. 1 and 2 when honeycomb panels are added. For a relatively stiff baseline panel (skin thickness = 0.08 in) stiffened with honeycomb panels, only a few modes are accounted for in the selected fre-quency range 0 - 1122 Hz. A better structural model would be achieved if Panel Nos. 1 and 2 were combined into a single stiffened panel. Such an ap-proach was used in obtaining the results shown in Fig. 28. For Panel No. 10 only, a small amount of noise reduction is obtained with the honeycomb panel treatment. The one-third octave and overall noise levels for the entire side-wall are shown in Fig. 27. With honeycomb treatment, the overall additional noise reduction for the entire sidewall is about 3 dBA when h = 0.286 in. The total added weight to one sidewall is about 7 lbs.

The effect on noise transmission due to heavy honeycomb treatment (h = 0.429 in) is illustrated in Figs. 28-31 for Panel Nos. 1, 2, 3, 4 and 6 and for the entire sidewall in Fig. 32. Since strong coupling was observed between Panel Nos. 1 and 2 [33], for heavy honeycomb treatment, Panel Nos. 1 and 2 were combined into one panel unit. Furthermore, no heavy honeycomb treatment was added to Panel Nos. 9 and 10 and window units Nos. 5, 7 and 8. The results indicate that about 13 dBA noise reduction for the sidewall can be achieved with this honeycomb treatment. Heavy honeycomb treatment was observed to be most effective for Panel Nos. 3 and 6.

5.5 Damping Tape and Mass Addition

The damping tapes chosen in this study are commercially available tapes composed of aluminum foil, synthetic rubber adhesive (or foam adhesive) and liner. These damping tapes show good damping characteristics for frequencies in the range of 100 - 200 Hz and temperatures from (- 65°F) - (+ 250°F). By applying several layers of these damping tapes to the panel surface, an average loss factor on the order of 0.2 (10% of the critical damping) can be achieved. The theoretical loss factor of the aluminum panels with damping tape added is estimated using the procedures of constrained layer damping presented in Ref. 31. It is calculated that with heavy damping tape treatment, the structural damping corresponding to the fundamental panel mode would increase from ζ_0 = 0.03 (baseline aircraft) to ζ_0 = 0.075 (baseline + damping tape). The other modal coefficients are calculated from ζ_{mn} = $\zeta_0 (\omega_{11}/\omega_{mn})$ where ω_{mn} are the natural frequencies of the stiffened panel units.

When damping tape with a thin constraining layer (aluminum foil) is added to the panel surface, the mass of the panel increases, without an

appreciable increase in the panel stiffness. Thus, the modal frequencies of the stiffened panels tend to shift to the lower frequency values. For propeller noise input, such a frequency shift could result in a resonance condition. However, with the large damping provided by the damping tape action, these panel resonances will be suppressed. The increase in panel modal damping due to the addition of damping tape is accounted for by the increase in the structural damping coefficient ζ_0 [14]. Thus, the resonance peaks in Eqs. 12 and 13 are reduced. In Table 4, the fundamental frequencies of the stiffened panels are given for several conditions of damping tape treatment.

The A-weighted interior sound pressure levels transmitted by each panel unit are shown in Figs. 33-38 for a damping tape treatment with a surface density of 0.38 lb/ft^2. The noise transmitted by the entire sidewall (including windows) is plotted in Fig. 39. From these results, it can be seen that approximately 1 - 10 dBA noise reduction can be achieved for different panel units. The amount of noise reduction for each panel unit is influenced by the value of the ratio added mass/panel mass (mass low action) and the location of the panel resonance frequencies (after treatment) with respect to the frequencies of the propeller blade passage harmonics. The overall noise reduction for the entire sidewall is about 6 dBA and the total added weight to the sidewall is 7.05 lbs. For the baseline aircraft, the interior noise is dominated by the first and second blade passage harmonics, while the treated aircraft is dominated by the second, third and fourth blade passage harmonics. This difference is believed to be caused mainly by the shift of the modal frequencies to lower values.

To illustrate the effect of mass and damping tape addition, the noise

transmitted by Panel Nos. 1, 2, 4 and 6 is shown in Figs. 40-45 for several

combinations of damping tape and non-load carrying mass add-on treatments.

It can be observed that when the surface densities of the added mass (non-

load carrying mass + damping tape) reach about 2.3 lbs/ft^2, the noise levels

for Panel Nos. 1 and 2 are close to or below the selected sound pressure le-

vel of 78 dBA. A similar condition is observed for Panel No. 4 when the ad-

ded surface density reaches 1.59 lbs/ft^2. However, for Panel No. 6, this

level is exceeded even when the surface density of the add-on treatment

reaches 3.28 lbs/ft^2. The baseline construction of Panel No. 6 is relatively

stiff when compared to that of the other panel units. The fundamental fre-

quency of this panel is 169 Hz. With added mass, the modal frequencies shift

toward the lower frequency values and the number of modes in the selected

frequency range 0 - 1122 Hz increase significantly. Thus, some blade pas-

sage harmonics could induce resonances which are not observable for the un-

treated (baseline) case.

As can be observed from these results, noise reduction does not follow

a universal rule for the uniform mass/unit area treatment of different pan-

el units. Furthermore, adding mass might even have a negative effect on the

interior noise at some frequencies. The results presented indicate a rela-

tive sensitivity of the various panels to mass and damping tape treatments.

From the results shown in Figs. 37 and 38, it can be observed that for a

treated sidewall, the interior noise is mainly controlled by Panel No. 6.

Panel No. 6 is located in the propeller plane and in the vicinity of where

the interior noise is calculated.

5.6 Honeycomb and Damping Tape Treatment

The results presented indicate that treating the aircraft sidewall with

honeycomb panels or damping tape could have a positive effect on noise reduction. However, in some cases, a large amount of stiffening or heavy mass needs to be added to reduce the transmitted noise to acceptable levels. It was found that better noise reduction can be achieved for less added weight when both of these add-on treatments are combined together. Panel stiffening is achieved by attaching honeycomb panels to the elastic skin and then adding damping tape to the honeycomb construction.

The one-third octave A-weighted interior noise levels for honeycomb-damping tape treatment are given in Figs. 46-52. From these results, it can be seen that the interior noise is dominated by the second blade passage harmonic due to the noise transmitted by Panel No. 3. Even though a significant amount of noise reduction is achieved at most frequencies with honeycomb-damping tape treatment, the noise levels at the second blade passage harmonic are about 10 dBA above the selected optimization level. However, by adding a very stiff honeycomb panel (surface density = 1.4 lb/ft^2) to Panel No. 3 and then applying damping tape, a desirable amount of noise reduction can be achieved as shown in Fig. 53.

5.7 Additional Noise Losses Due to Acoustic Blankets, Septum and Trim Panel

The additional noise losses ΔTL are calculated for various combinations of acoustic blankets, septum and trim panels using the procedures given in Section 3.7. Numerical results were obtained for the following data: η = 0.04 (no damping tape), η = 0.08 (with damping tape), $c_1 = c_6$ = 1128 ft/sec, D_x = 88 lb_m-ft^2/sec^2, D_y = 149,133 lb_m-ft^2/sec^2, H = 3731 lb_m-ft^2/sec^2, ρ_1 = 0.07657 lb/ft^3, $\rho_1 = \rho_2 = \rho_4 = \rho_6$, ρ_m = 0.864 lb_m/ft^3, ρ_f = 21.55 lb_m/ft^3, $K_2 = K_4$ = 14.7 psi, R_1 = 64 lbs-sec/in^4.

The additional transmission losses across a sidewall are given in Figs. 54-58 for acoustic blankets and for semi-rigid materials. A semi-rigid material would correspond to a material which does not have loose fibers and is heavier than an acoustic blanket. The propagation constants of semi-rigid materials were taken from Refs. 5 and 31. The results in Figs. 54-58 show that the general trends of ΔTL are similar for both of these models. However, for large surface densities of trim panel, a stronger double wall resonance condition is observed when the space between the two walls is filled with porous acoustic blankets. Furthermore, with increasing cavity depth, semi-rigid materials show an increasingly larger amount of ΔTL when compared to the results obtained for acoustic blankets. Thus, for distances between the exterior skin and the trim panel on the order of 4 in or more, semi-rigid materials would provide significantly greater noise transmission losses than noise losses from acoustic blankets.

The effect on ΔTL due to variations in the surface density of the trim panel and septum is illustrated in Figs. 59 and 60, respectively. These results indicate that with increasing surface density of the trim panel, a stronger double wall resonance condition is observed. However, the frequency bandwidth of the double wall resonance region decreases with an increasing surface density of the trim panel. For larger values of μ_5 (above 0.3 lb/ft^2) and for frequencies above 250 Hz, ΔTL seems to double for each doubling of the frequency. These results tend to indicate that a heavy trim panel might have a negative effect on noise reduction for this aircraft in the frequency range of about 80 Hz - 200 Hz. The first two propeller blade passage harmonics are within this frequency range. For a septum with a surface density above 0.1 lb/ft^2, a strong double wall resonance is observed

between the elastic panel and the septum. The double wall frequency can be calculated using the formulas given in [5]. The frequency of this resonance is higher than that of the double wall resonance of the elastic panel-trim panel configuration. This is due to the fact that the distance between the elastic panel and septum is one-half the distance between the elastic panel and the trim panel. A heavy septum (0.1 - 1.0 lb/ft^2) might have a negative effect on ΔTL in the frequency range 125 - 250 Hz. The second and third blade passage harmonics are in this frequency range. For frequencies above 300 Hz, a significant amount of ΔTL can be achieved with a heavy septum. From Fig. 63, it can be seen that when the surface densities of both the septum and trim panel are large, two double wall resonance conditions are observed in the frequency range 100 - 300 Hz. Thus, a heavy septum and heavy trim panel might have a negative effect on noise reduction in this frequency range even though a large ΔTL can be achieved for higher frequencies (above 300 Hz).

The additional noise losses corresponding to different cavity depths and a very light septum are shown in Fig. 61. A significant amount of noise loss can be realized for deep cavities and for frequencies above the double wall resonance frequency. In fact, as the distance between the elastic panel and the trim panel increases, the double wall resonance frequency decreases. The effect on ΔTL due to the location of the septum is shown in Figs. 62 and 63 for a light and heavy septum, respectively. For a heavy septum, significant differences in ΔTL were observed for the different distances between the elastic panel and the septum. Two double wall resonance conditions were obtained in the frequency range 100 - 300 Hz. In this frequency range, some advantage in ΔTL can be gained by locating a heavy septum near the trim panel.

5.8 Noise Transmission Through Windows

The noise transmitted through window units (Panel Nos. 5, 7 and 8) is shown in Figs. 64-66. These results are based on simply supported "equivalent" single sheet plexiglass panels as described in Sections 3.3 and 5.1. As can be seen from those results, the noise transmitted by Panel Nos. 5 and 7 satisfy the selected upper noise limit for all frequencies. However, treatments needed to be added to Panel No. 8 in order to meet the upper limit criterion. This was achieved by reducing the window area and adding damping tape to the interior side of the window.

5.9 Noise Optimization in the Aircraft

The interior noise in the cabin was optimized utilizing the computation procedure shown in Fig. 15. In this approach, the noise transmitted by each panel unit was calculated for each add-on treatment and then for a combination of several treatments. The amount of treatment was increased until a selected target noise level at a critical point in the cabin was reached. An interior point in the propeller plane at about ear level and 8 inches from the sidewall was selected as the critical point in the cabin. A level of 78 dBA for all one-third octave frequencies was taken as the upper bound for the noise transmitted by each individual panel unit. The treatment or combination of treatments which reduces the transmitted noise to this value for the least amount of added weight was assumed to be the best treatment for that panel unit. Then, the one-third octave and the overall noise levels were calculated for the entire sidewall for the same add-on treatments considered for individual panel units. An 88 dBA overall noise level was selected as the optimization target for the noise transmitted by the entire sidewall. The one-third octave 78 dBA and the overall 88 dBA levels were

-38-

selected as typical optimization goals for the present aircraft. Depending on the required comfort criteria for a particular aircraft, these levels could be adjusted to meet the prescribed conditions of interior noise. The non-structural add-on treatments applied to the fuselage sidewalls were evaluated for low surface density and high transmission loss. The add-on treatments included honeycomb panels, damping tapes, non-load carrying mass, porous acoustic blankets, septum barriers and trim panels. Acoustic blankets were added to all the panels except the windows.

The results of the optimization study are given in Figs. 67-73 for the individual panels and Fig. 74 for the entire sidewall. Since the amount of add-on treatments varies from one panel unit to another, the surface densities given in Fig. 73 for the sidewall are average values. The added weight was calculated by multiplying the total surface density by the panel area and then adding 5% of the total weight. The optimized interior noise in the cabin at the selected point is 88 dBA. The measured average noise levels in the cabin are typically 3 - 5 dBA less than the highest noise level at the selected critical point [9]. Thus, the optimized average noise level in the cabin would be about 85 dBA. Furthermore, experimental data for this aircraft suggest that no significant increase in interior noise is observed when both engines are running as compared to only one engine running [21]. Therefore, the noise transmitted by one sidewall is taken as the total interior noise in the cabin.

To reach the 85 dBA (average) noise level in the cabin, 30 lbs. of add-on treatments were added to one sidewall (a total of 60 lbs. for both sidewalls). In addition, light treatments (damping tape and acoustic blankets) need to be added to the ceiling area of the aircraft. For a ceiling area

of about 47 ft^2 and surface density of damping tape equal to 0.25 lb/ft^2, the weight of this treatment is about 15 lbs. Interior furnishings such as carpeting and seats are not included with the add-on treatments.

From the results shown in Figs. 67-74, it can be observed that the most critical noise path is through Panel No. 6. The input noise levels prescribed for the sidewall were highest over the region where Panel No. 6 is located. Other panels which seem to transmit high noise levels are Panel Nos. 1, 2 and 3. These panels are also located in the vicinity of the propeller plane where the input noise levels are high. By stiffening these panels, damping out the resonance vibrations and using absorptive materials, it is possible to reduce transmitted noise to acceptable levels. To isolate the vibrations of the skin-stringer panels from those of the limp-mass trim panel, damping tape with a thickness ranging from 0.25 in - 0.5 in is attached to the stringer as shown in Fig. 74. In addition to the vibration isolation, such a construction would increase the distance between the elastic skin and the trim panel, allowing for greater additional noise losses.

The overall A-weighted sound pressure levels are plotted versus the ratio of treated/untreated weight for the sidewall in Fig. 75. The points corresponding to baseline and optimized conditions are connected by a straight line. These noise levels are assumed to be the average sound pressure levels in the cabin. Several points are included in this diagram to illustrate the effect of uniform treatment on all the panel units (except windows). For a light treatment with damping tape, honeycomb panels and honeycomb-damping tape, the points are located near the straight line. However, for a very heavy add-on treatment with damping tape and non-load carrying mass, significant deviations from the straight line relation are observed. These

results tend to suggest that a linear relationship might be established between the noise reduction and added weight, if the add-on treatments are selected according to the optimization procedure used in this study. The final configuration of the aircraft sidewall with add-on treatments is shown in Fig. 76. There are basically three types of add-on treatments used for noise reduction: honeycomb-damping-tape-acoustic-blankets-trim, damping tape-acoustic blankets-trim and damping tape (windows). However, the surface densities of these treatments could vary from one panel unit to another. The distribution of the surface densities (baseline + treated, treated) for the optimized sidewall are given in Fig. 77. These results indicate the relative amount of treatment used for different panel units. The greatest amount of add-on treatment was applied to Panel Nos. 3 and 6.

6.0 Conclusions

An analytical model has been developed to predict the noise transmission into a twin engine G/A aircraft. The model has been used to identify the airborne noise transmission paths and to optimize the interior sound levels due to propeller noise inputs. The average cabin noise levels in the baseline aircraft reach a maximum of about 105 dBA and these levels are about 20 dBA higher than the optimization goal of 85 dBA. The results indicate that the required noise reduction has to be achieved mainly in the low frequency range of 70 - 350 Hz. For the type of aircraft considered, the first four propeller blade passage harmonics are within this frequency range.

The required noise attenuation has been obtained by add-on treatments which do not involve changes in the fuselage primary structure. These add-

on treatments include lightweight aluminum honeycomb panels, constrained layer damping tape, porous acoustic blankets and limp trim panels. Due to the non-uniform input pressure distribution and different structural dynamic characteristics of the sidewall panels, the amount and type of treatment applied to achieve the required noise reduction varies from one panel unit to another. The study indicates that the heaviest amount of treatment needs to be applied to those panels located in the vicinity of the propeller plane. Of the techniques investigated, the combination of honeycomb panels and con-strained layer damping tape applied to the aircraft skin seems to promise the required reductions in noise transmission in the low frequency region (70 - 350 Hz). Noise attenuation for higher frequencies can be achieved with a double wall system composed of porous acoustic blankets and limp trim panels which are isolated from the fuselage vibrations. However, a heavy trim panel might not always be beneficial for noise control since double wall resonances might coincide with one of the low frequency propeller blade passage harmonics. The optimization study indicates that to reduce cabin noise to a satisfactory level for the least amount of added weight, a com-bination of different add-on treatments needs to be used. The total added weight to the aircraft is about 75 lbs which is about 1.1% of the take-off gross weight. It should be noted that in achieving these values the effect of potential flanking paths and noise entering through the front and rear bulkheads have not been included in the analytical model.

The analytical prediction method has been validated experimentally with laboratory tests wherein all parameters could be carefully controlled or measured. A relatively good agreement between theoretical predictions and experimental observations in the field under static operating conditions

has been achieved for the baseline aircraft. Further experimental valida-
tion of the predicted noise reduction for the optimized aircraft is needed,
however.

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Table 1 Material and Geometric Properties of Stiffeners

Panel Unit	$E_s \times 10^{-6}$ psi	$I_z \times 10^2$ in^4	$I_y \times 10^2$ in^4	s_z in	c_z in	c_y in	J_s in^4	$C \times 10^4$ in^4	$C_{ws} \times 10^2$ in^6	A in^2	ℓ in
1	10.5	.347	.736	1.0	0.0	.323	.169	.189	.903	.038	26.4
2	10.5	.810	1.370	1.0	0.0	.319	.369	2.92	2.12	.203	26.4
3	10.5	.651	1.440	1.0	0.0	.320	.328	1.46	1.70	.161	26.4
4	10.5	.651	1.440	1.0	0.0	.320	.328	1.46	1.70	.161	23.5
6	10.5	.483	.926	1.0	0.0	.321	.227	5.42	1.26	.118	19.1
9	10.5	.810	1.370	1.0	0.0	.319	.369	2.92	2.12	.203	5.25
10	10.5	.651	1.160	1.0	0.0	.320	.300	1.46	1.70	.161	23.5

Table 2 Natural Frequencies of Sidewall Panels (n = 1, 2, 3)

Panel Unit	Frequencies, Hz	
	Theoretical	Experimental
1	72,84,111,149,172,178,181,189,224,228,245, 254,259,276,338,345,359	71*',86*',137,146,(157),166,172,(177),183,189, 234,(241),261,(269),282',288',310', 336'
2	104,205,240,247,271,333,390,399	(94*),114*,(161),217,224,241,(269'),276,282', 288',310',336',352
3	62,77,122,127,140,159,194,198,201,248,285, 292,299,348,357,371,392,414	No data below 100 Hz 152,201,228,270 (Crude data)
4	68,80,95,130,138,155,160,173,196,223,230, 237,256,264,285,300,310,340,375,384,406	Insufficient data
5	58	58,70,108,117,143,230,322
6	169,188,239,258,260,344,384,403	170(174),219,232,240,252,264,(278),305
7	122,284+	323,406 Exterior window only
8	72,223+	72*,97,112,(120),(129),146,158,167,185,194,209, 223+,243+,264,273,283+,321
9	276,296,338,377	No data
10	147,206,324,348,434	No data

+ window dilatational mode
* global modes perticipating

( ) two frequencies for similar mode coupling, between panels 1 and 2
' even modes with n (n = 2, 4, ...)

Table 3   Natural Frequencies of Stiffened Panels for Aero-
Commander Sidewall with Honeycomb Add-On Treatment

(First Frequency Band)

Panel Unit	Baseline-Untreated Hz	Honeycomb h = 0.286 in Hz	Honeycomb h = 0.429 in Hz
1	72, 84, 111	161, 194, 262	{ 199*, 232*, 307*
2	104, 205	124, 192	
3	62, 77	184, 233	405, 510
4	68, 80, 96	229, 250, 300	292, 336, 403
6	169, 188, 259	347, 413, 512	519, 617, 765
9	276	379	x
10	147	327	x

Panel Units 5, 7 and 8 are windows
*   Panels 1 and 2 combined into one unit
x   Frequency above 1000 Hz

Table 4 Fundamental Frequencies of the Stiffened Panel Units
with Different Damping Tape Add-On Treatment

Panel Unit	Baseline-Untreated Hz	Damping Tape S.D. = .38 lb/ft² Hz	Damping Tape S.D. = .78 lb/ft² Hz	Damping Tape S.D. = 2.42 lb/ft² Hz
1	72	57	51	33
2	104	92	84	57
3	62	50	45	30
4	68	54	48	32
6	169	131	116	75
9	276	214	189	127
10	147	131	121	84

S.D. = Surface Density
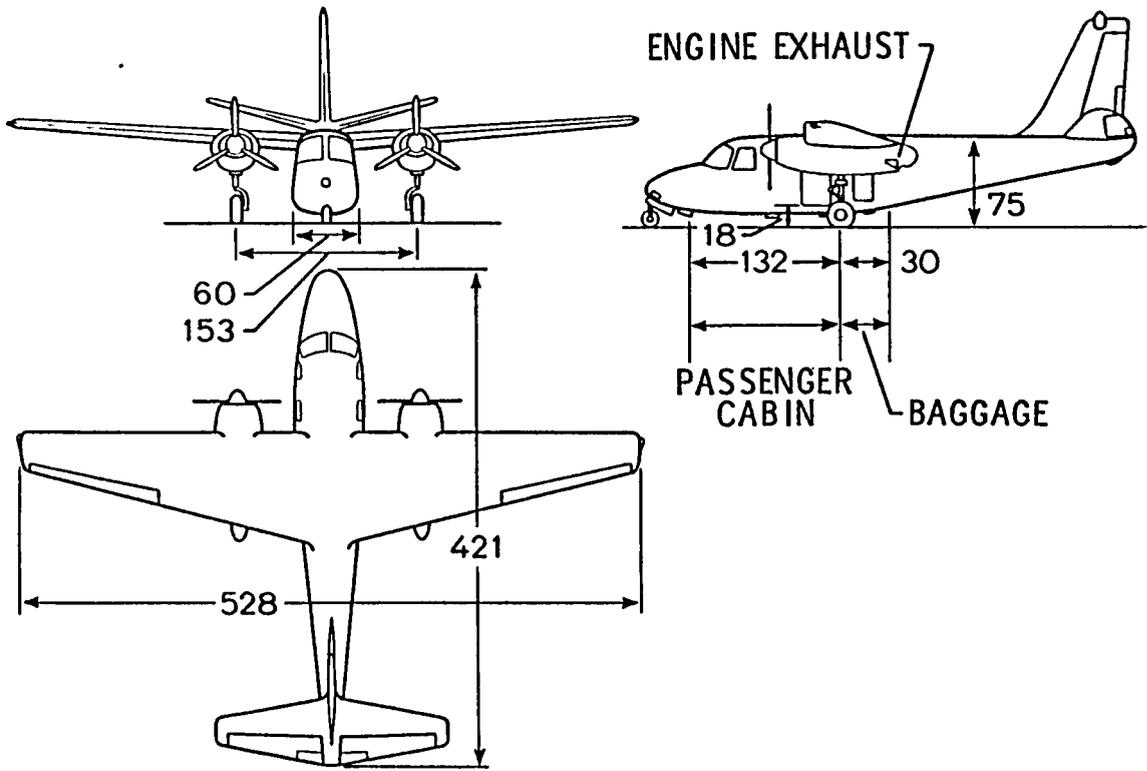
Fig. 1 Twin-engine aircraft used in the noise optimization study
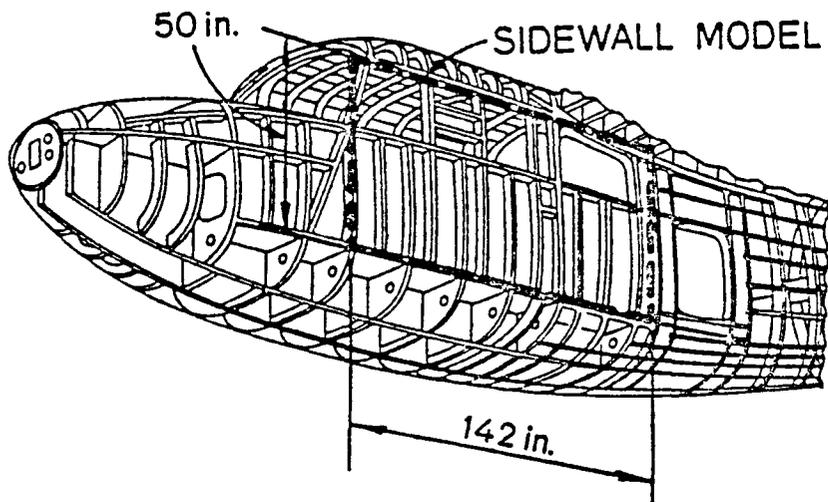
Fig. 2 Structural features of a twin-engine light aircraft
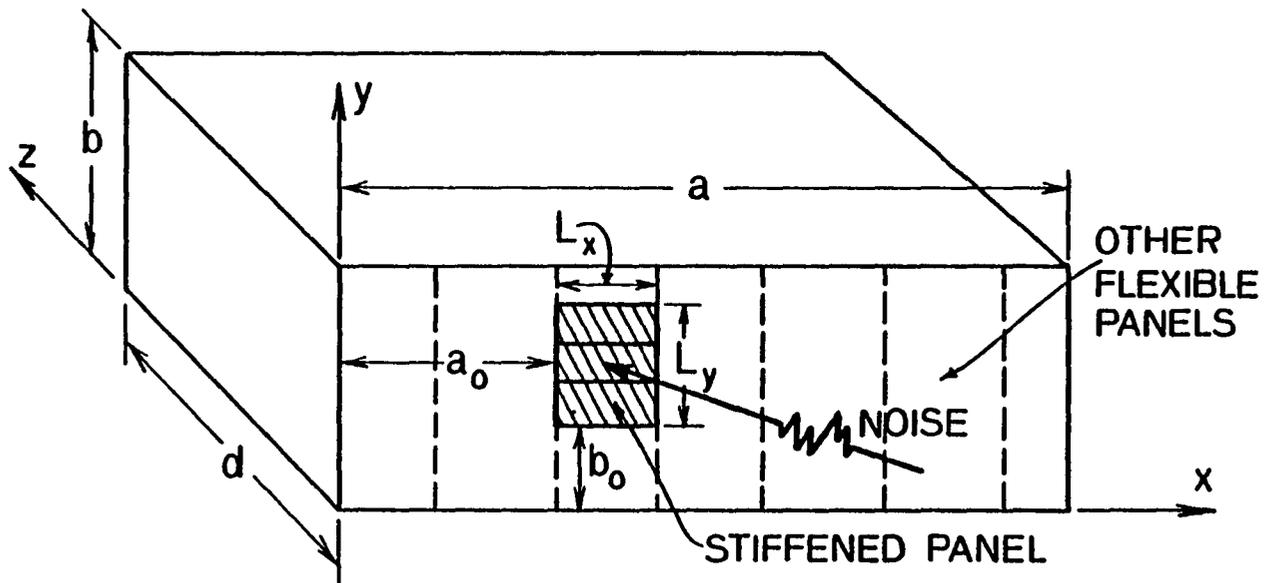
Fig. 3 Simplified geometry of aircraft cabin model
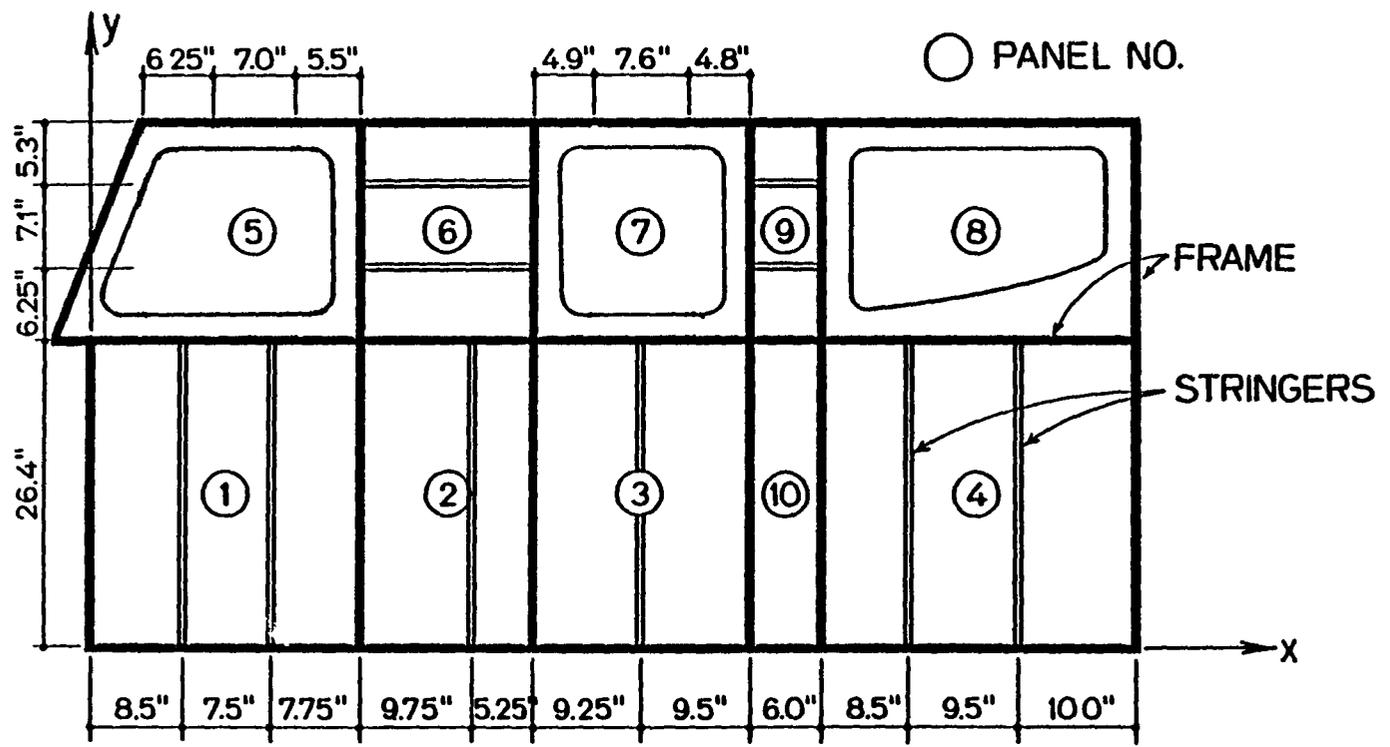
Fig. 4 Aircraft sidewall used for noise transmission study
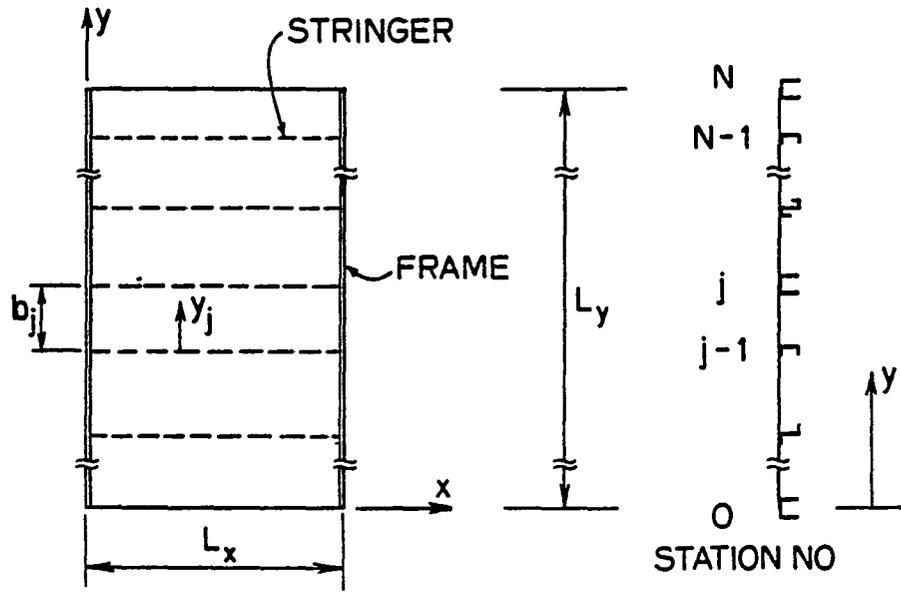
Fig. 5 A multispanned skin-stringer system
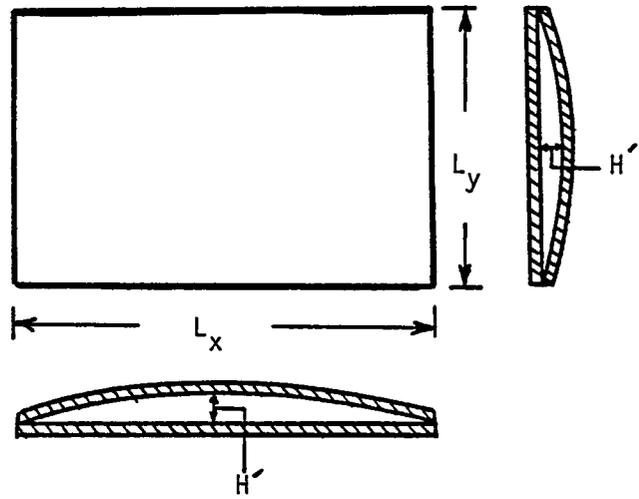
Fig. 6 Geometry of a double wall window construction
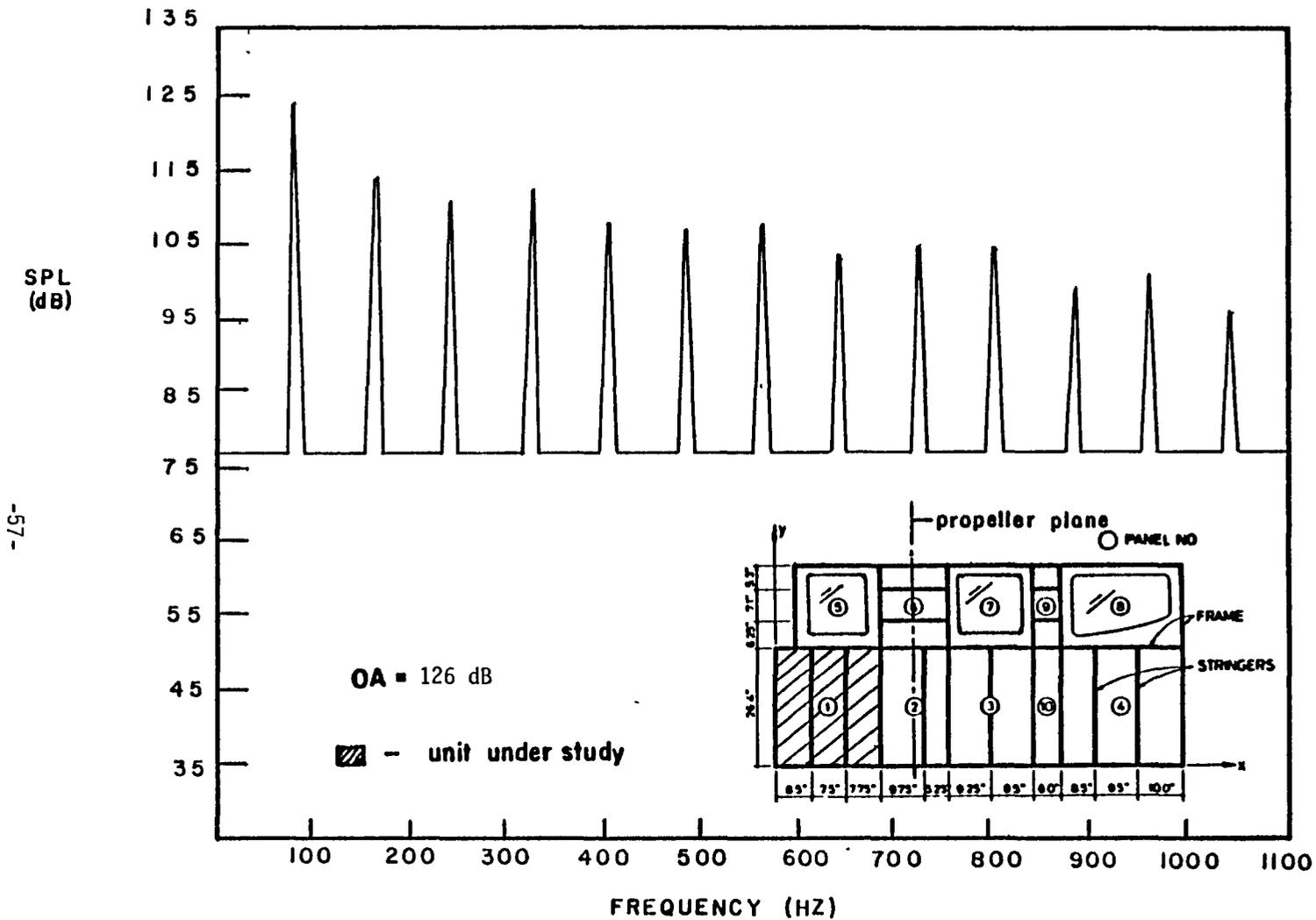
Fig. 7   PROPELLER   NOISE   INPUT   (PANEL UNIT No. 1)
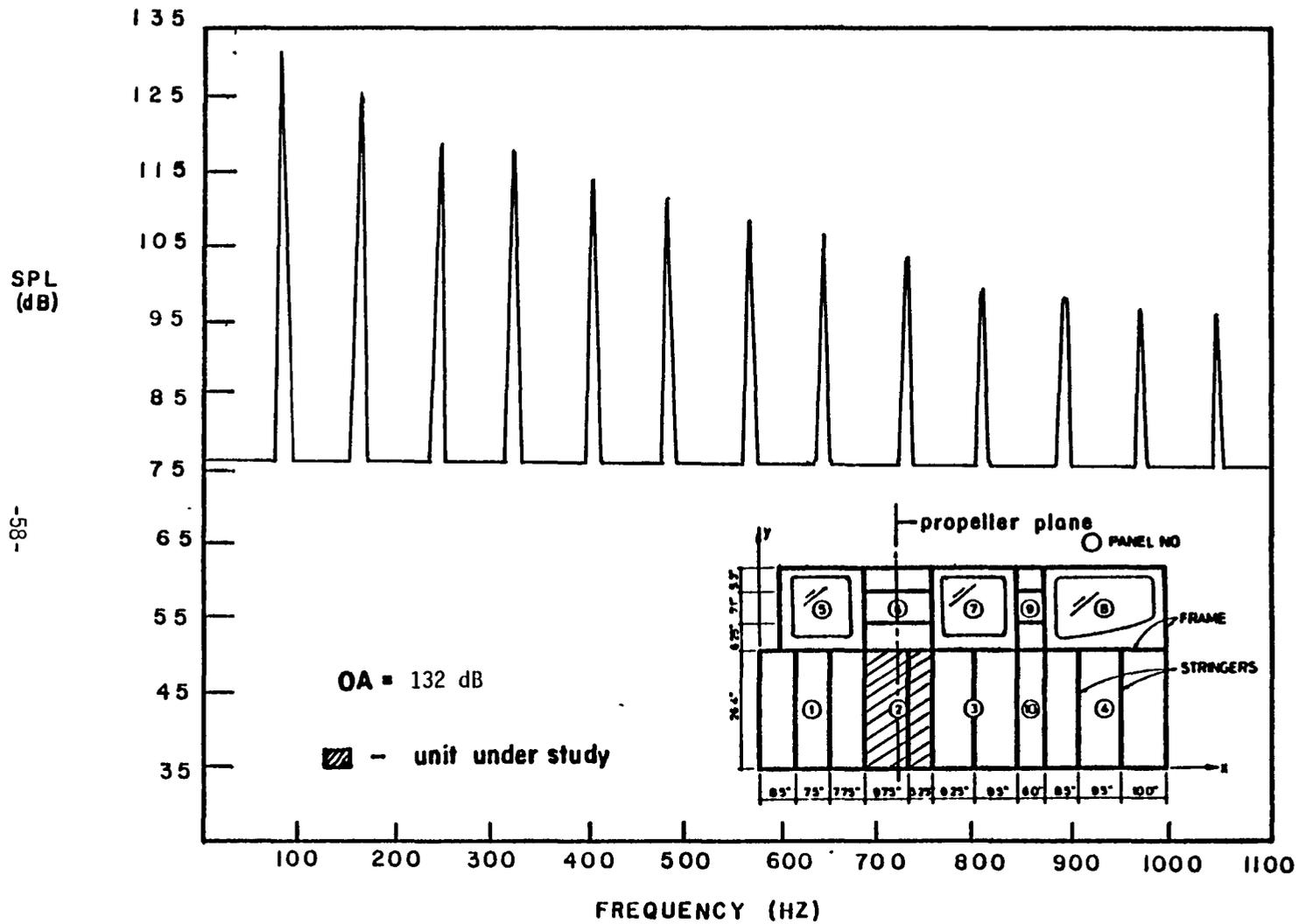
Fig. 8   PROPELLER   NOISE   INPUT   (PANEL UNIT No. 2)
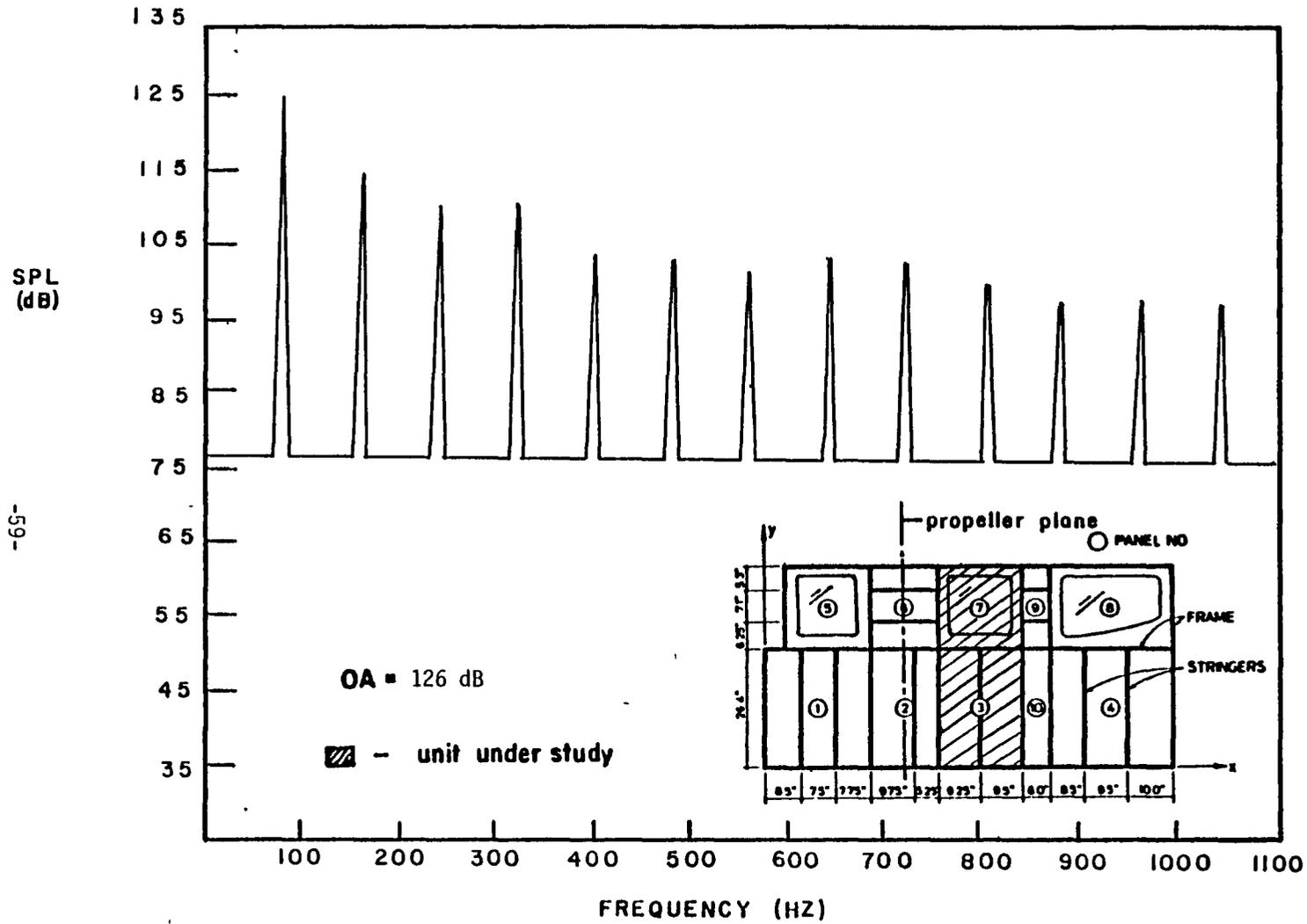
Fig. 9   PROPELLER   NOISE   INPUT   (PANEL UNIT No. 3,7)
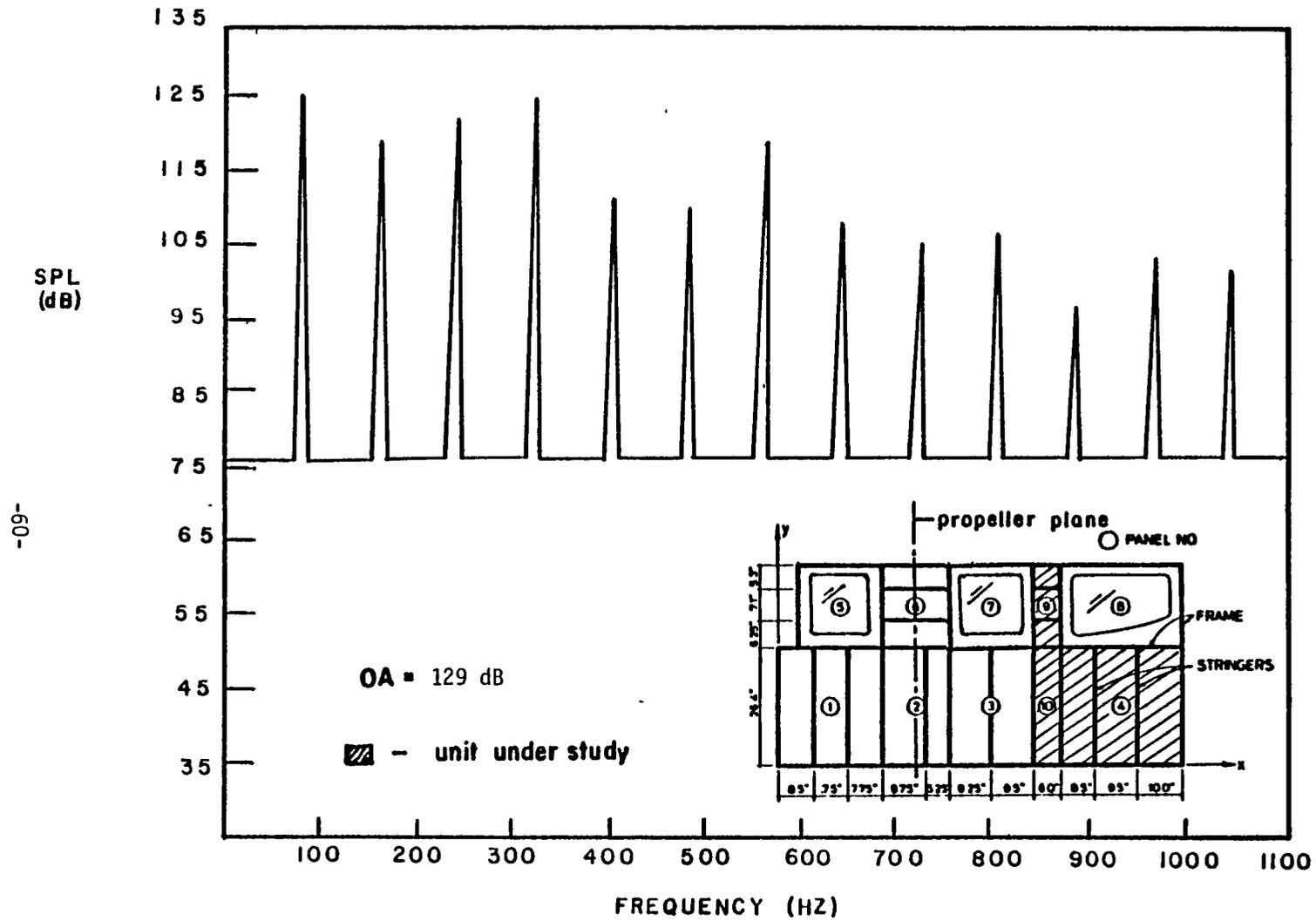
Fig. 10   PROPELLER   NOISE   INPUT   (PANEL UNIT No. 4, 9, 10)
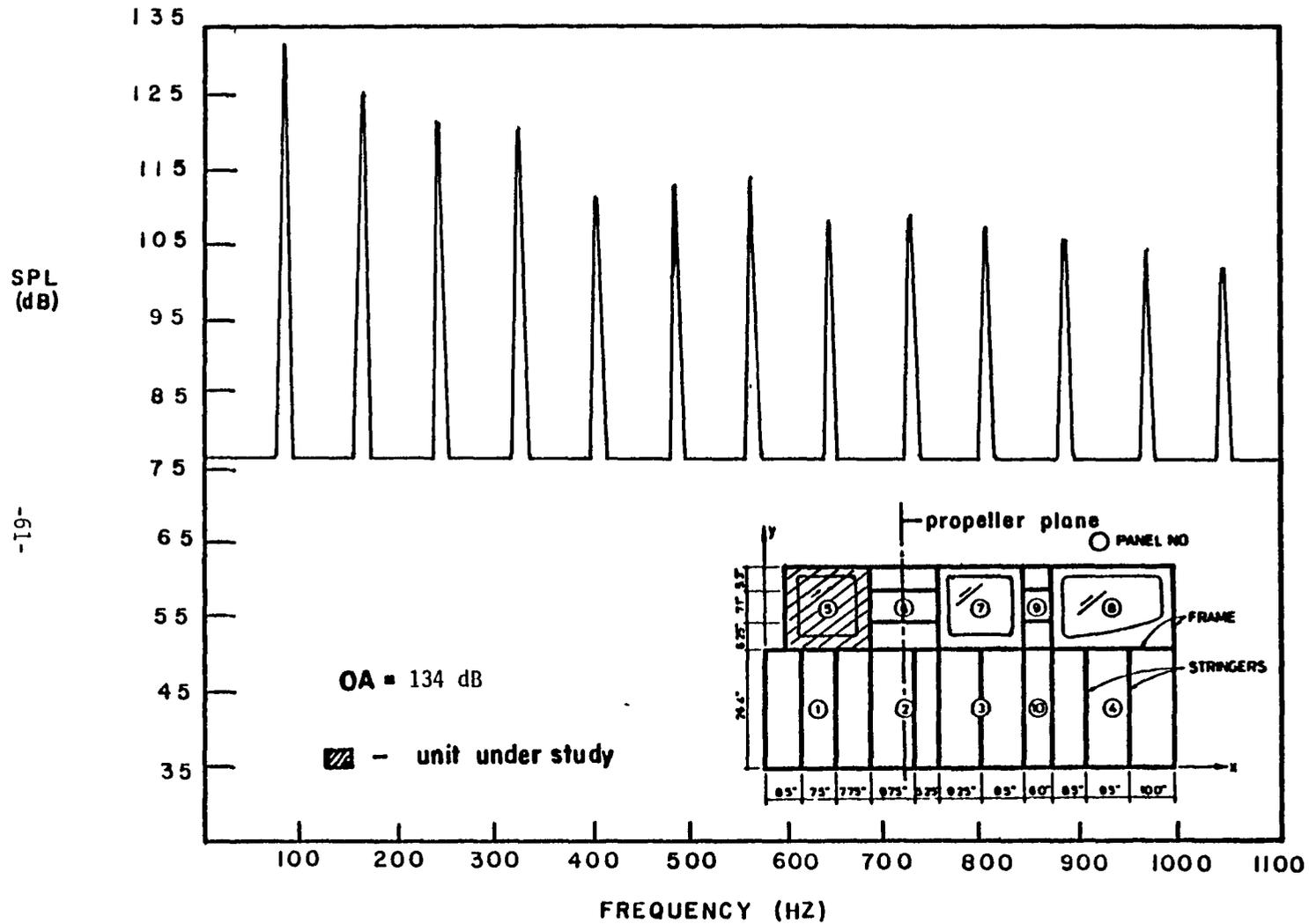
Fig. 11 PROPELLER NOISE INPUT (PANEL UNIT No. 5)

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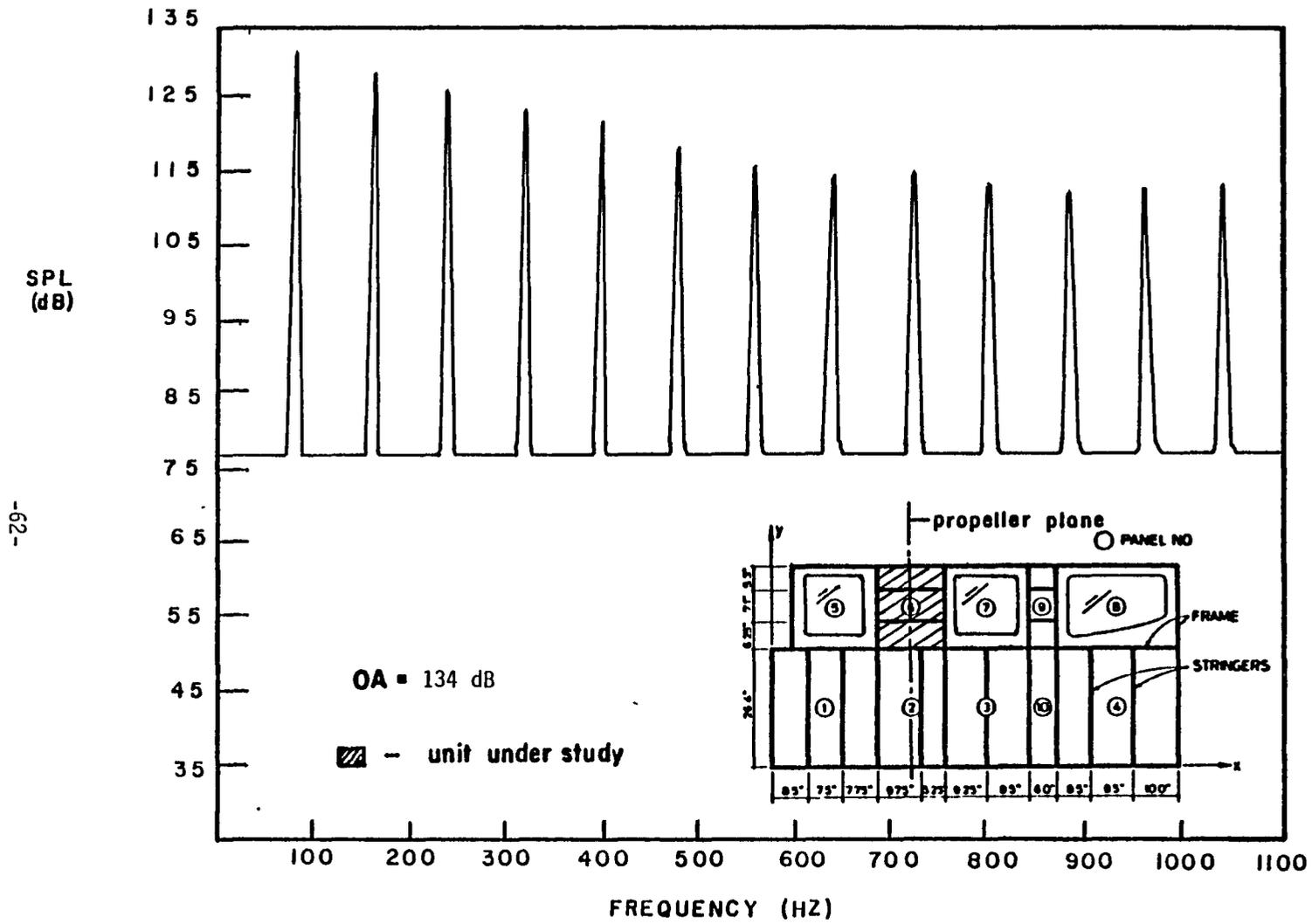
Fig. 12   PROPELLER   NOISE   INPUT   (PANEL UNIT No. 6)
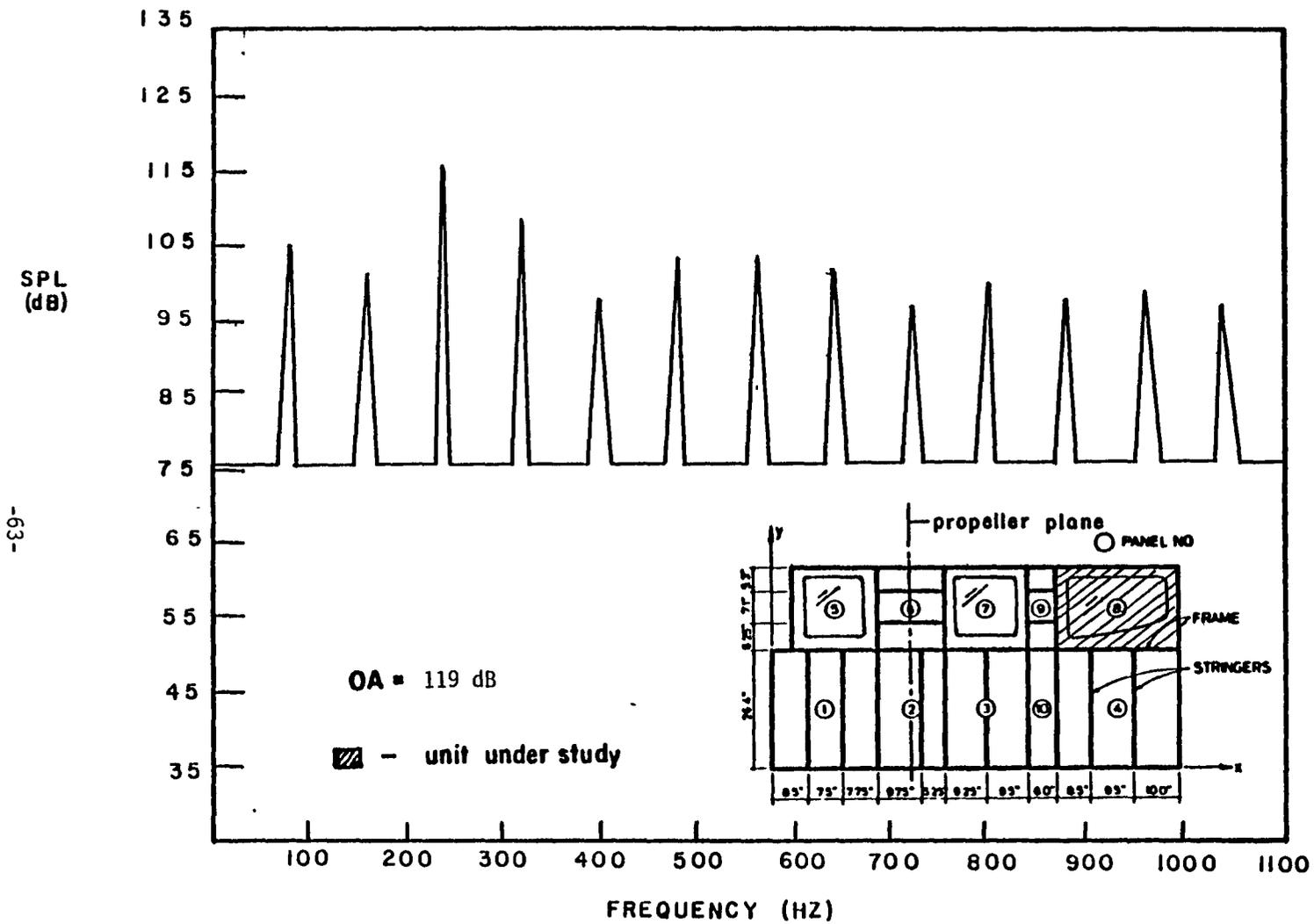
Fig. 13 PROPELLER NOISE INPUT (PANEL UNIT No. 8)

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ACOUSTIC BLANKETS

exterior   interior

P_r

P_t

θ_6

(1) (2)   (3) (4)   (5) (6)

θ_4

θ_2

θ_1

P_i

d_2 d_4

STIFFENED
PANEL
$(\mu_1 = lb/ft^2)$

SEPTUM
$(\mu_3 = lb/ft^2)$

TRIM PANEL
$(\mu_5 = lb/ft^2)$
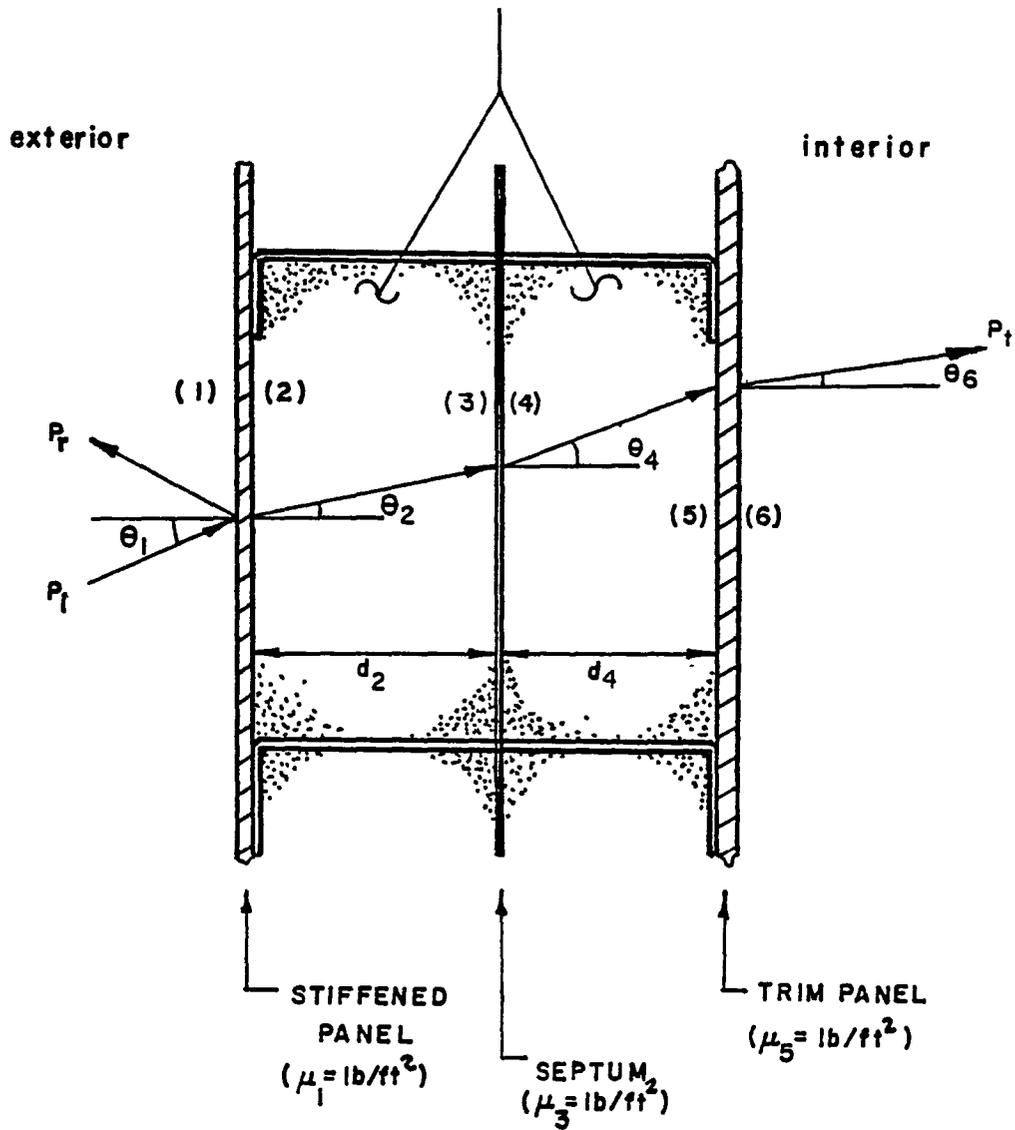
Fig. 14 Geometry of sidewall treatment
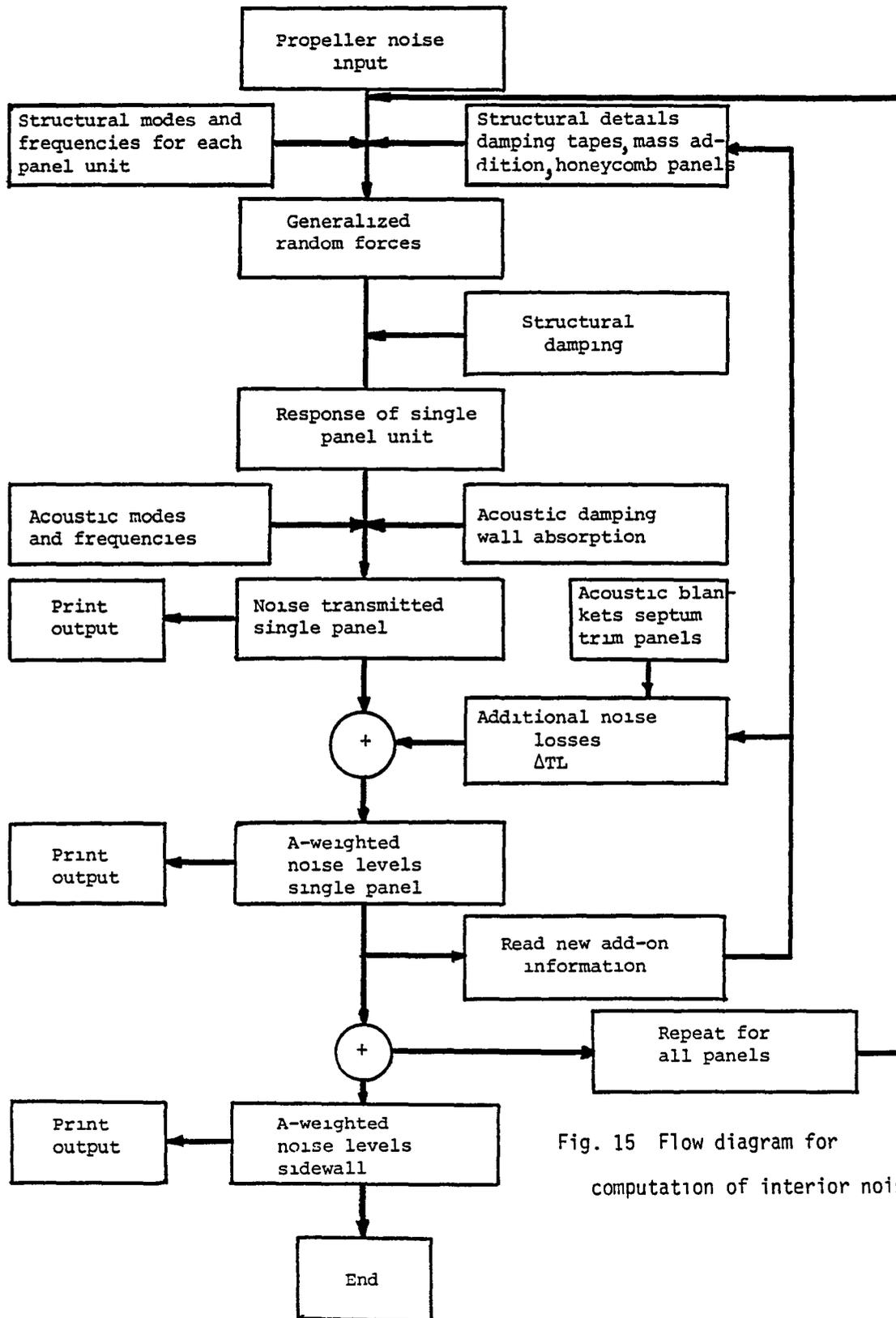
Fig. 15 Flow diagram for computation of interior noise

O - shear center

C' - centroid

S - point of stringer attachment to the skin
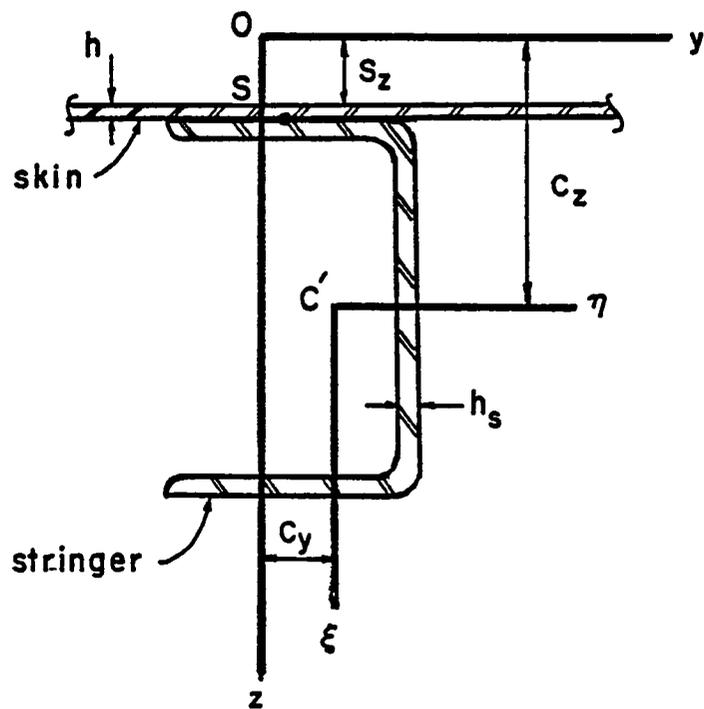
Fig. 16 Details of skin-stringer construction

$f_{11} = 169$ Hz $f_{21} = 188$ Hz

$f_{31} = 259$ Hz

——— Theory
— — — Experiment
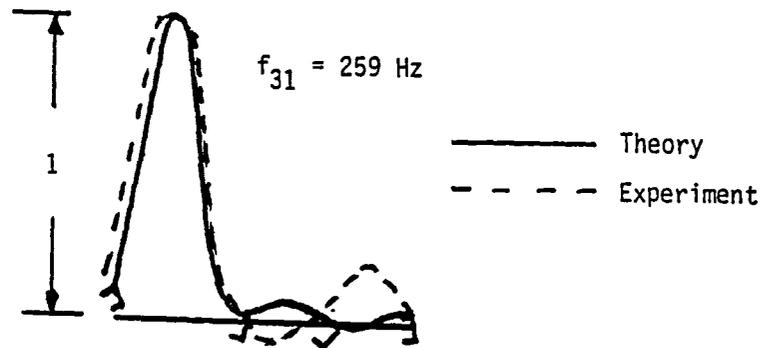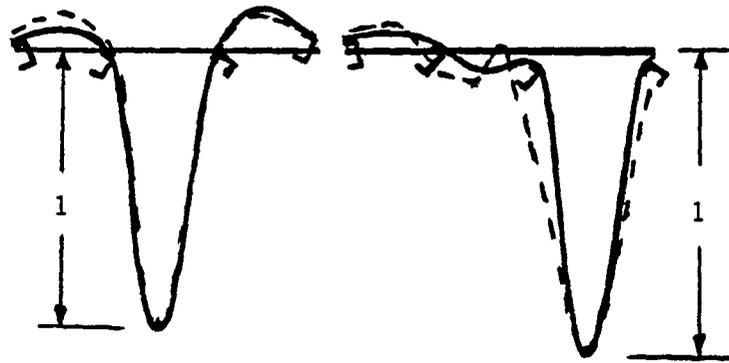
Fig. 17   Mode shapes of a stiffened panel for the
first frequency band (panel No. 6)
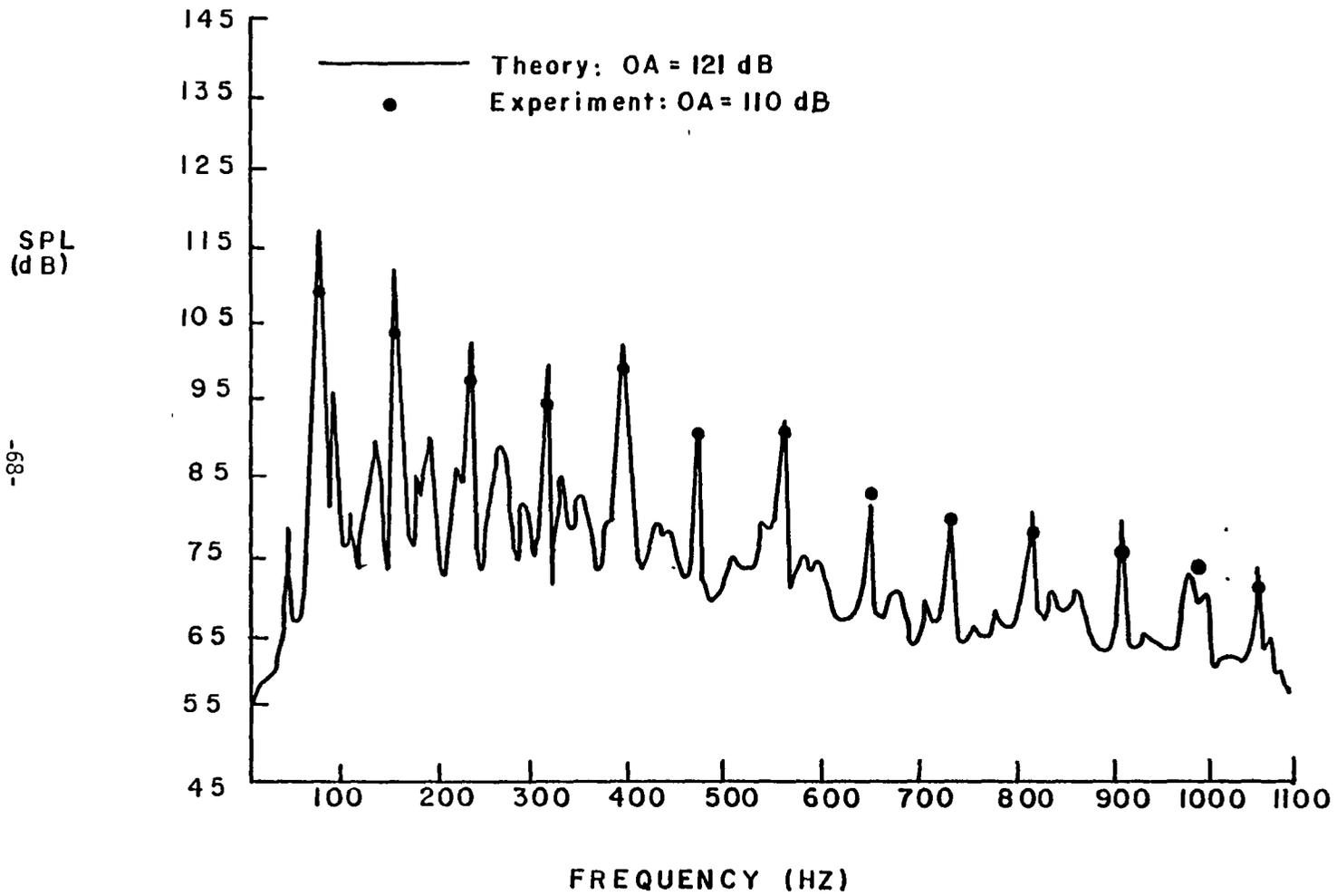
Fig. 18 Sound levels in the aircraft cabin
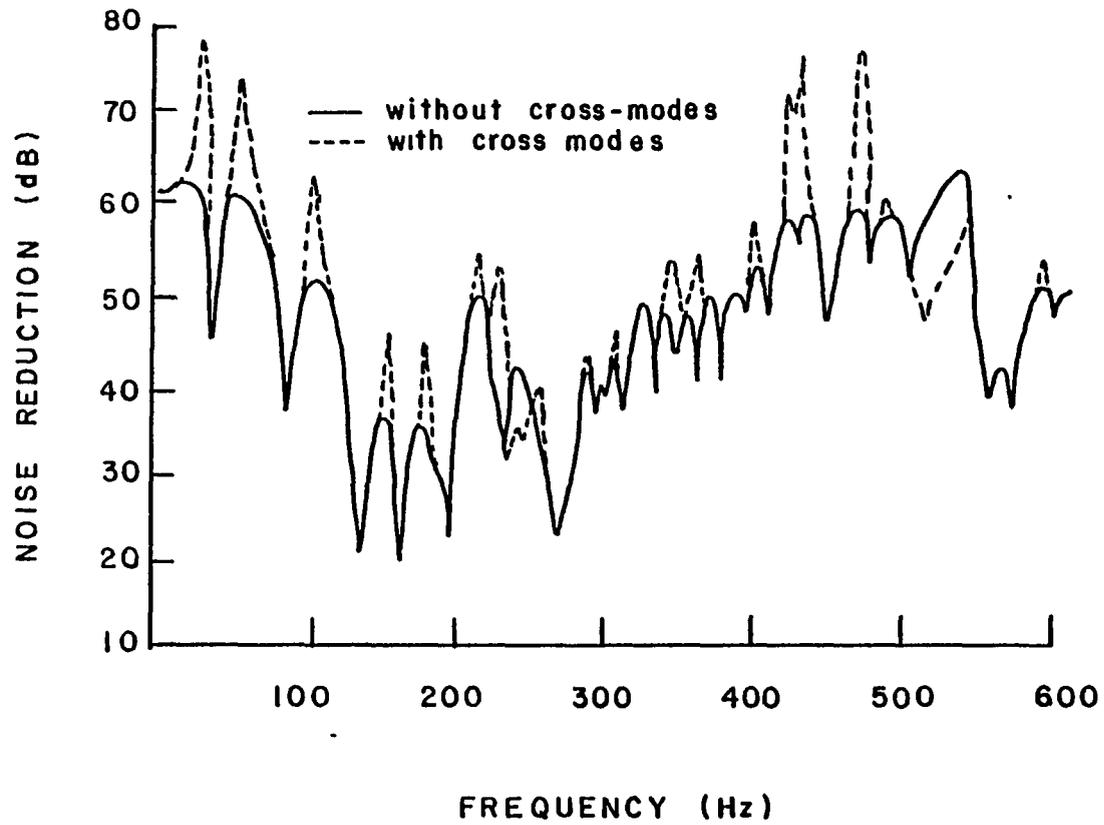
Fig. 19   Noise reduction for a stiffened panel with and without cross-modal terms

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL
(dBA)

100

90

80

70

60

50

40

—70—

—— baseline : OA = 107 dBA
--- no convection : OA = 112 dBA

O PANEL NO

FRAME

STRINGERS

▨ — unit under study

31.5   50   80   125   200   315   500   800   1250
25   40   63   100   160   250   400   630   1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

Fig. 20   Interior noise levels with and without convection
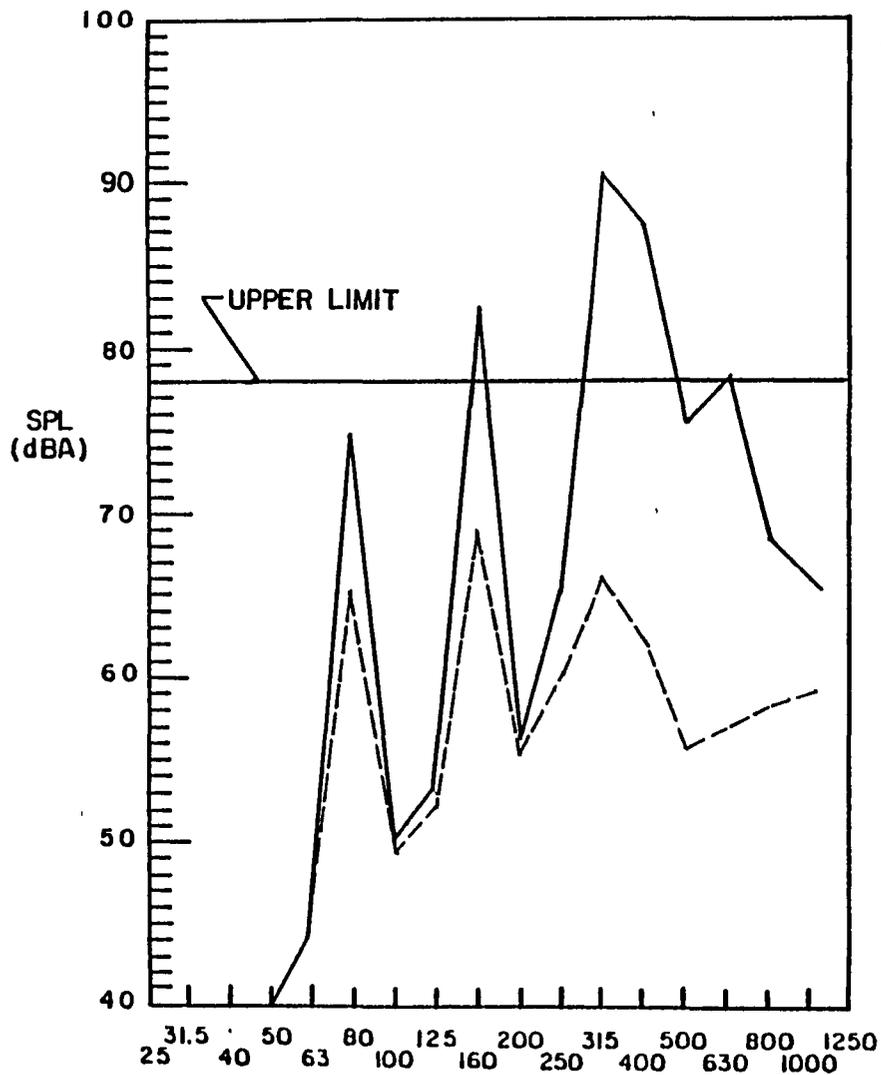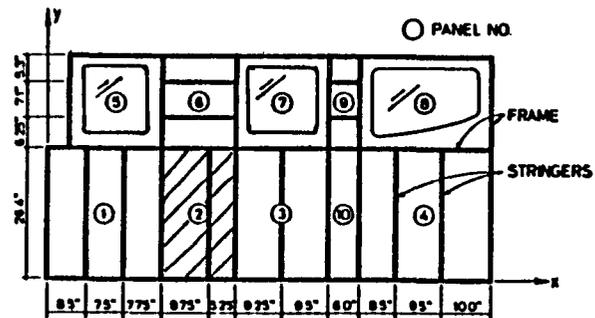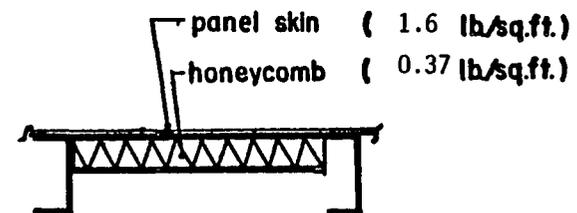
Fig. 21 Interior noise levels with honeycomb treatment (Panel No. 1)

Fig. 22   Interior noise levels with honeycomb treatment (panel No. 2)

Fig. 23   Interior noise levels with honeycomb treatment (Panel No. 3)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

-74-

100

90

SPL
(dBA)

80

UPPER LIMIT

70

60

50

40

31.5   50   80   125   200   315   500   800   1250
25   40   63   100   160   250   400   630   1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

——— baseline : OA = 95 dBA
----- honeycomb : OA = 99 dBA

added weight = 1.90 lbs.
panel area = 5.13 sq.ft.

panel skin (0.75 lb/sq.ft.)
honeycomb (0.37 lb/sq.ft.)

y
PANEL NO

⑤   ⑥   ⑦   ⑨   ⑧   FRAME

①   ②   ③   ⑩   ④   STRINGERS

x

83" 75" 77.5" 67.5" 57.5" 67.5" 83" 60" 83" 83" 100"

☒ — unit under study

Fig. 24 Interior noise levels with honeycomb treatment (Panel No. 4)

Fig. 25 Interior noise levels with honeycomb treatment (Panel No. 6)

Fig. 26   Interior noise levels with honeycomb treatment (Panel No. 10)

Fig. 27 Interior noise levels with honeycomb treatment (Sidewall)

Fig. 28 Interior noise levels with heavy honeycomb treatment (Panels 1 and 2)

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Fig. 29   Interior noise levels with heavy honeycomb treatment (Panel No. 3)

Fig. 30 Interior noise levels with heavy honeycomb treatment (Panel No. 4)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

100

90

SPL
(dBA)

80

UPPER LIMIT

70

60

50

40

31.5 · 50 80 125 200 315 500 800 1250
25 40 63 100 160 250 400 630 1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

—— baseline : OA = 106 dBA
---- honeycomb : OA = 90 dBA

added weight = 2.72 lbs.
panel area · = 1.94 sq.ft.

panel skin (0.86 lb/sq.ft.)
honeycomb (1.4 lb/sq.ft.)

O PANEL NO.

FRAME

STRINGERS
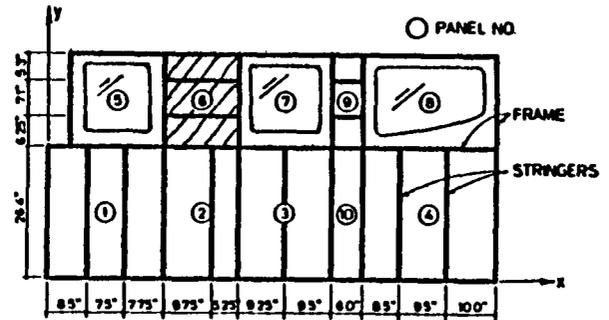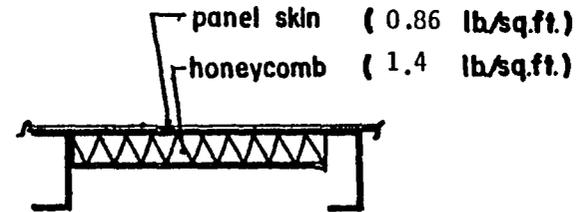
— unit under study

Fig. 31 Interior noise levels with heavy honeycomb treatment (Panel No. 6)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 µbar

SPL (dBA)

UPPER LIMIT

100

90

80

70

60

50

40

31.5   50   80   125   200   315   500   800   1250
25   40   63   100   160   250   400   630   1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)
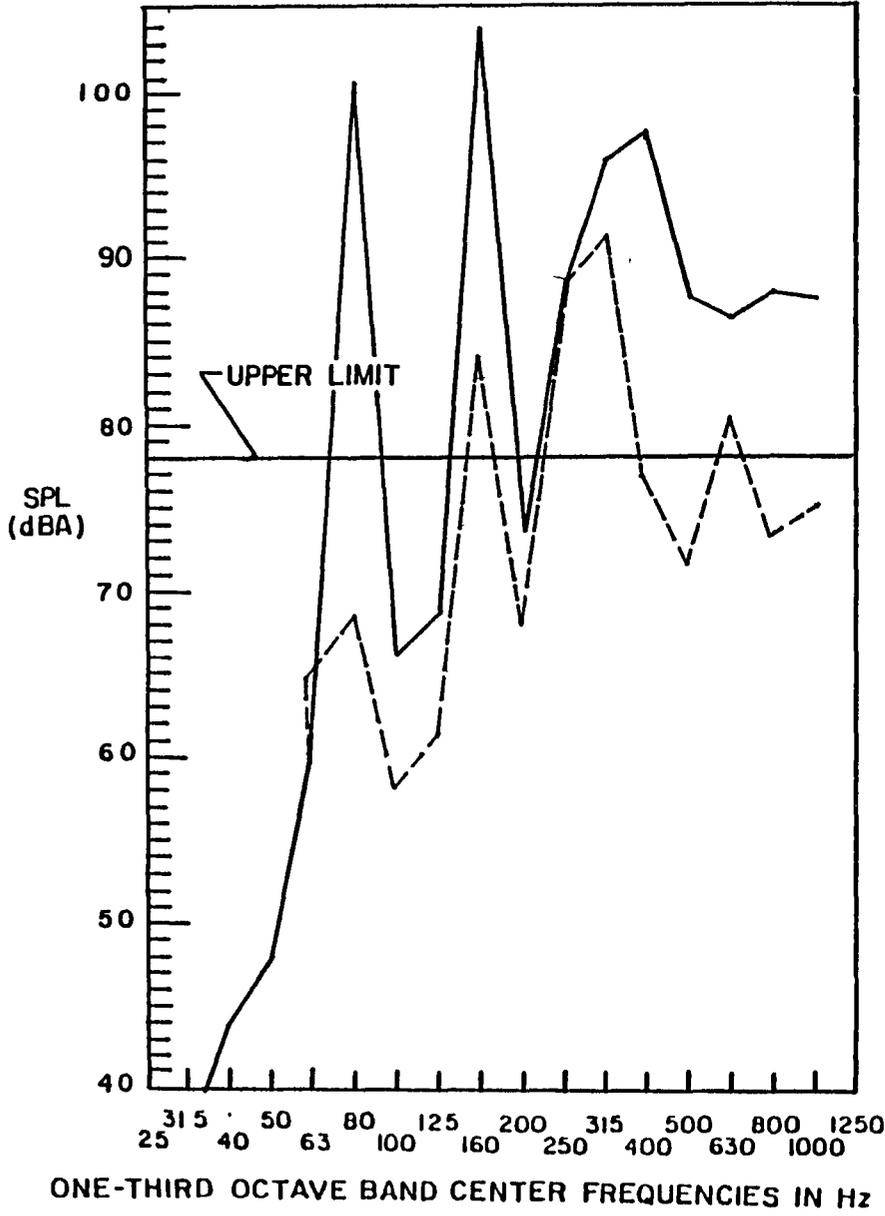
—— baseline : OA = 107 dBA
--- honeycomb : OA = 94 dBA

added weight = 25 lbs.
panel area = 29 sq.ft.

panel skin (1.15 lb/sq.ft.)
honeycomb (1.4 lb/sq.ft.)

O PANEL NO.

FRAME

STRINGERS
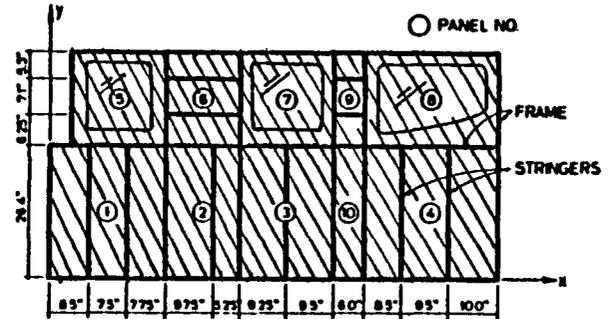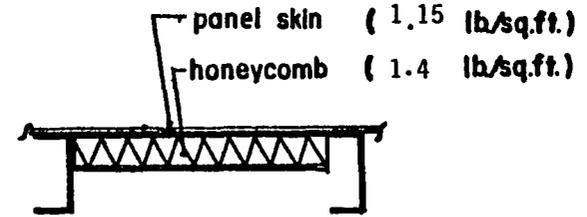
▨ — unit under study

Fig. 32   Interior noise levels with heavy honeycomb treatment (Sidewall)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL
(dBA)

UPPER LIMIT
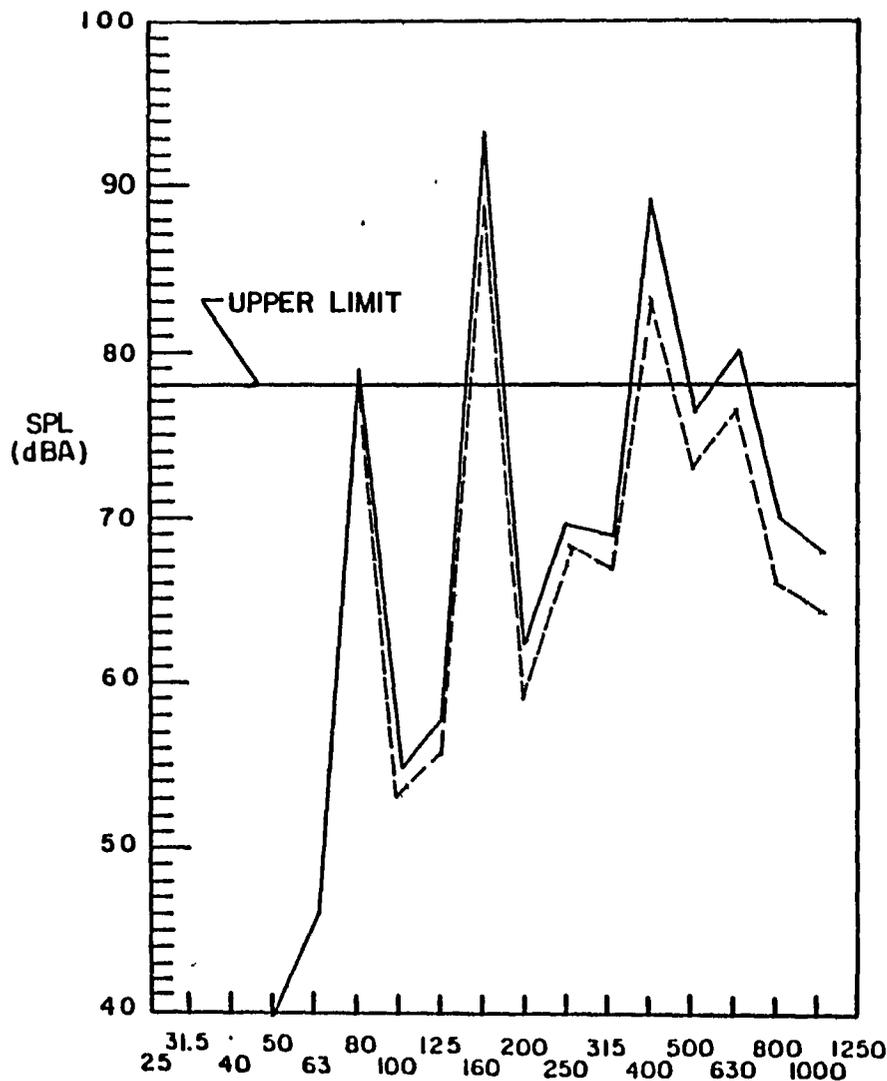
—— baseline : OA = 94 dBA
- - - damping tape : OA = 89 dBA

added weight = 1.67 lbs.
panel area ' = 4.4 sq.ft.

panel skin (0.73 lb./sq.ft.)
damping tape (0.38 lb./sq.ft.)

O PANEL NO.

FRAME

STRINGERS
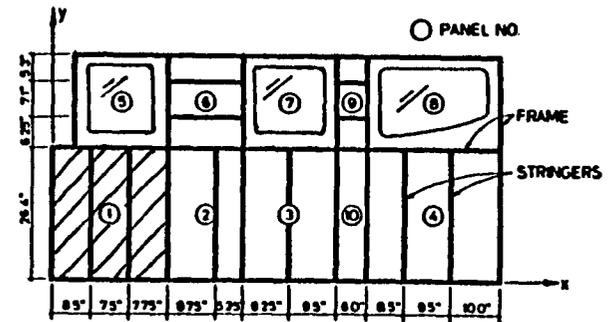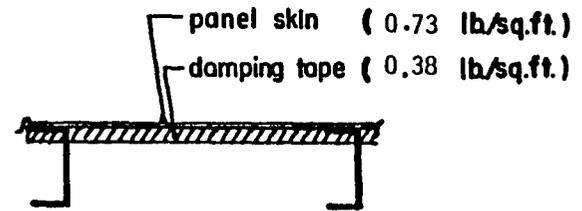
▨ — unit under study

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

Fig. 33   Interior noise with damping tape treatment (Panel No. 1)

Fig. 34 Interior noise with damping tape treatment (Panel No. 2)

Fig. 35   Interior noise with damping tape treatment (Panel No. 3)

Fig. 36 Interior noise with damping tape treatment (Panel No. 4)

Fig. 37 Interior noise with damping tape treatment (Panel No. 6)

The figure shows a plot of ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar, with SPL (dBA) on the vertical axis ranging from 40 to 100, and ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps) on the horizontal axis (25, 31.5, 40, 50, 63, 80, 100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250).

Legend:
— baseline : OA = 106 dBA
--- damping tape : OA = 96 dBA

added weight = 0.74 lbs.
panel area   = 1.94 sq.ft.

panel skin (0.86 lb/sq.ft.)
damping tape (0.38 lb/sq.ft.)

O PANEL NO.
FRAME
STRINGERS

▨ — unit under study

Fig. 38   Interior noise with damping tape treatment (Panel 9)

-88-

Fig. 39   Interior noise with damping tape treatment (Sidewall)

Fig. 40   Interior noise with damping tape and mass treatment (Panel No. 1)

Fig. 41   Interior noise with damping tape and mass treatment (Panel No. 2)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

-92-

100

90

80

SPL
(dBA)

70

60

50

40

UPPER LIMIT

31.5   50   80   125   200   315   500   800   1250
25   40   63   100   160   250   400   630   1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)
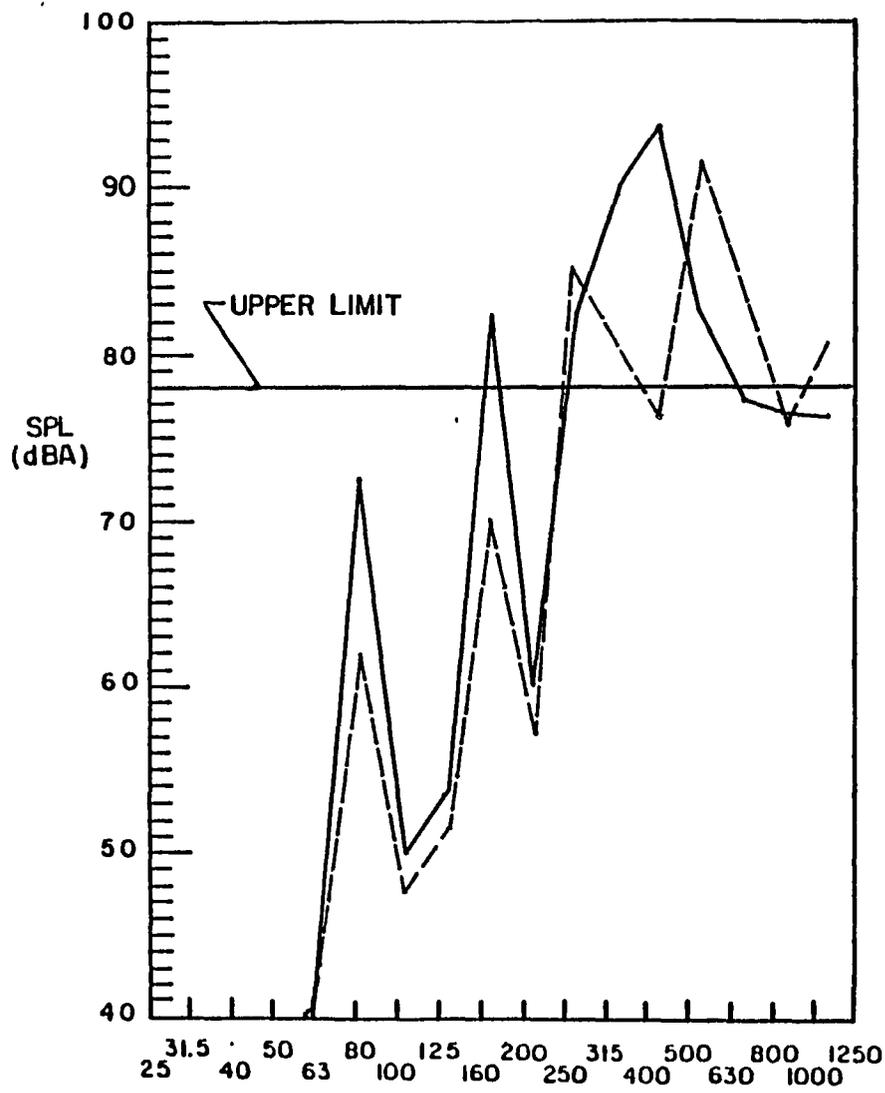
—— baseline : OA = 95 dBA
---- damping tape : OA = 92 dBA
+ added mass

added weight = 3.80 lbs.
panel area = 5.13 sq.ft.

panel skin (0.75 lb/sq.ft.)
damping tape (0.38 lb/sq.ft.)
added mass (0.36 lb/sq.ft.)

y   O PANEL NO

FRAME
STRINGERS
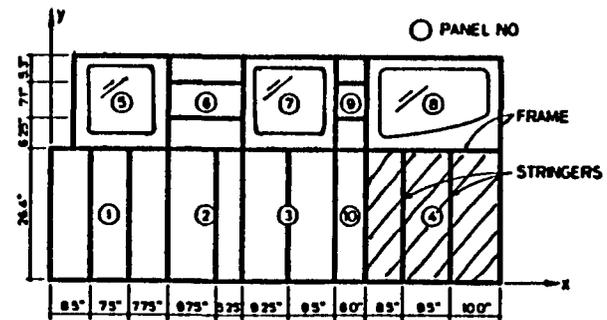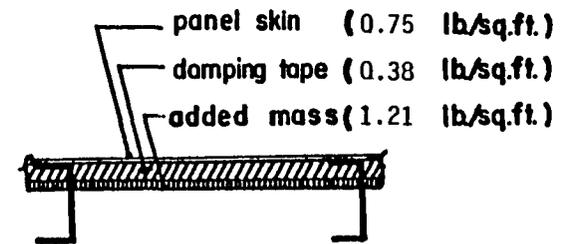
— unit under study

Fig. 42   Interior noise with damping tape and mass treatment (Panel No. 4)

Fig. 43 Interior noise with damping tape and heavy mass treatment (Panel No. 4)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 µbar

-94-

SPL
(dBA)

UPPER LIMIT
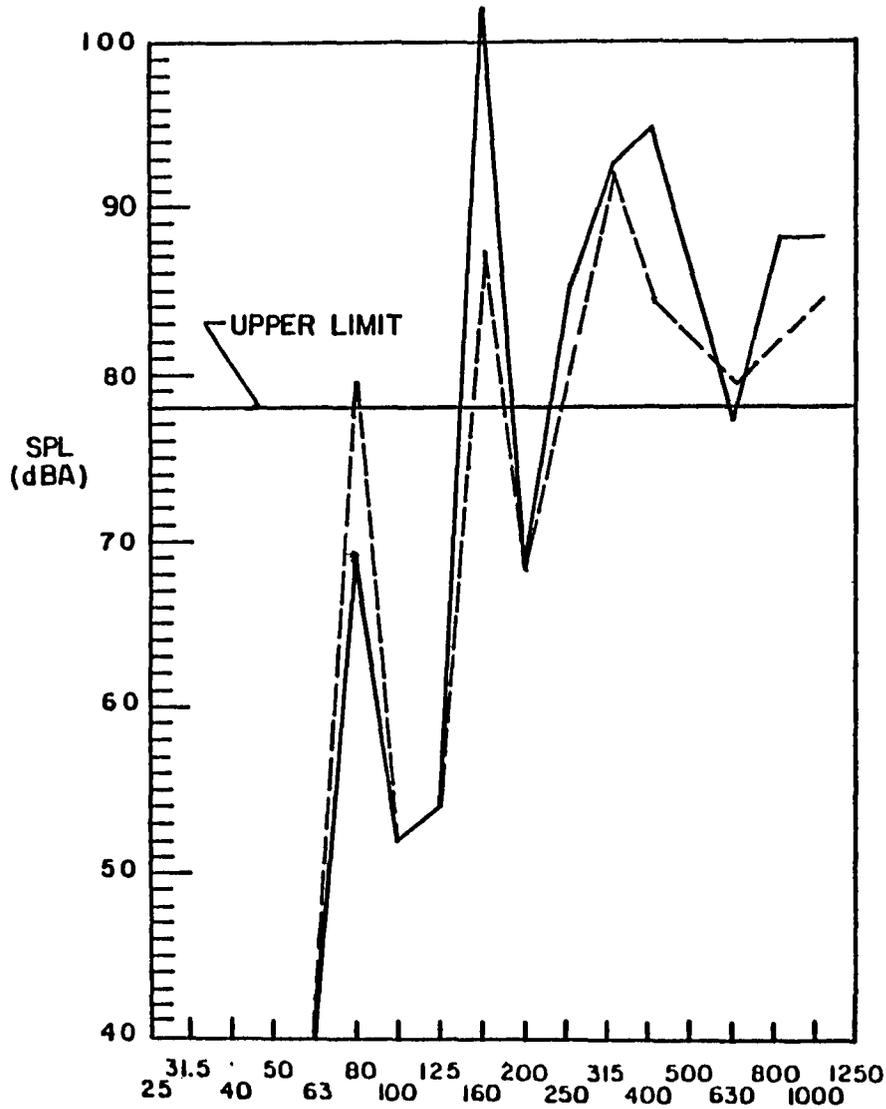
baseline : OA = 106 dBA
damping tape : OA = 95 dBA
+ added mass

added weight = 1.44 lbs.
panel area = 1.94 sq.ft.

panel skin (0.86 lb/sq.ft.)
damping tape (0.38 lb/sq.ft.)
added mass(0.36 lb/sq.ft.)

O PANEL NO

FRAME

STRINGERS
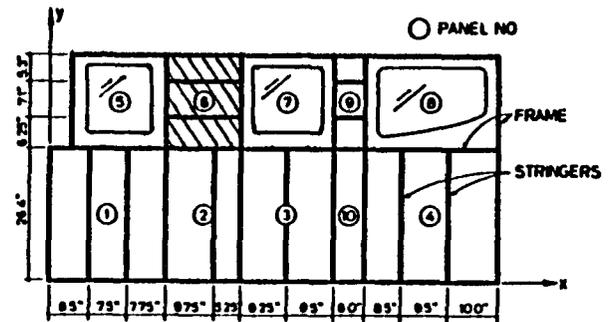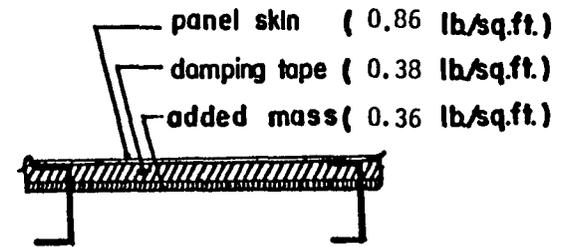
▨ — unit under study

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

Fig. 44   Interior noise with damping tape and heavy mass treatment (Panel unit No. 6)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL (dBA)

UPPER LIMIT
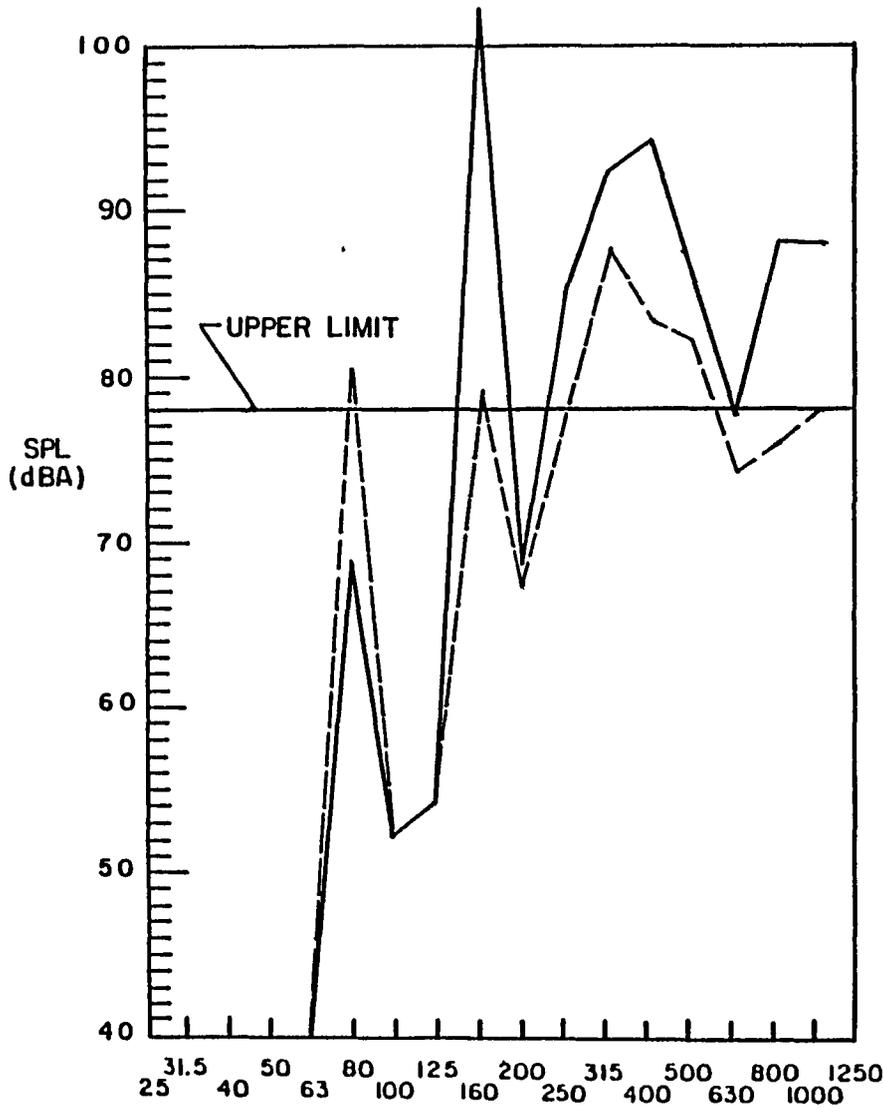
—— baseline : OA = 106 dBA
——— damping tape : OA = 91 dBA
+ added mass

added weight = 6.36 lbs.
panel area = 1.94 sq.ft.

panel skin (0.86 lb./sq.ft.)
damping tape (0.38 lb./sq.ft.)
added mass(2.9 lb./sq.ft.)

○ PANEL NO

FRAME

STRINGERS
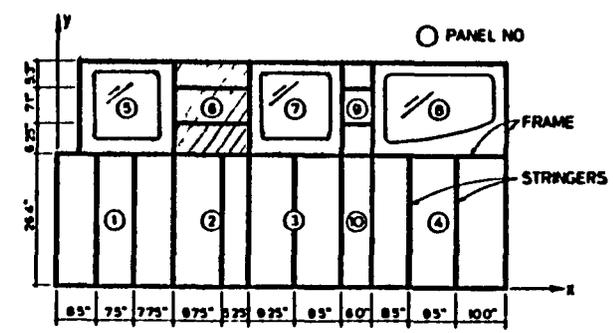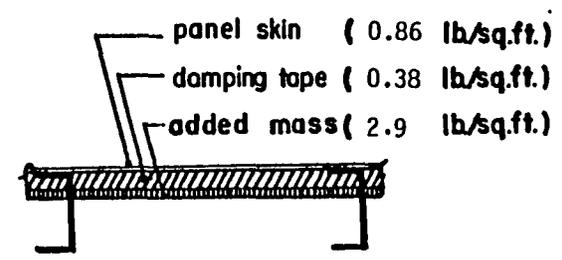
▨ — unit under study

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

Fig. 45 Interior noise with damping tape and heavy mass treatment (Panel No. 6, Add-on = 3.28 lb/ft^2)
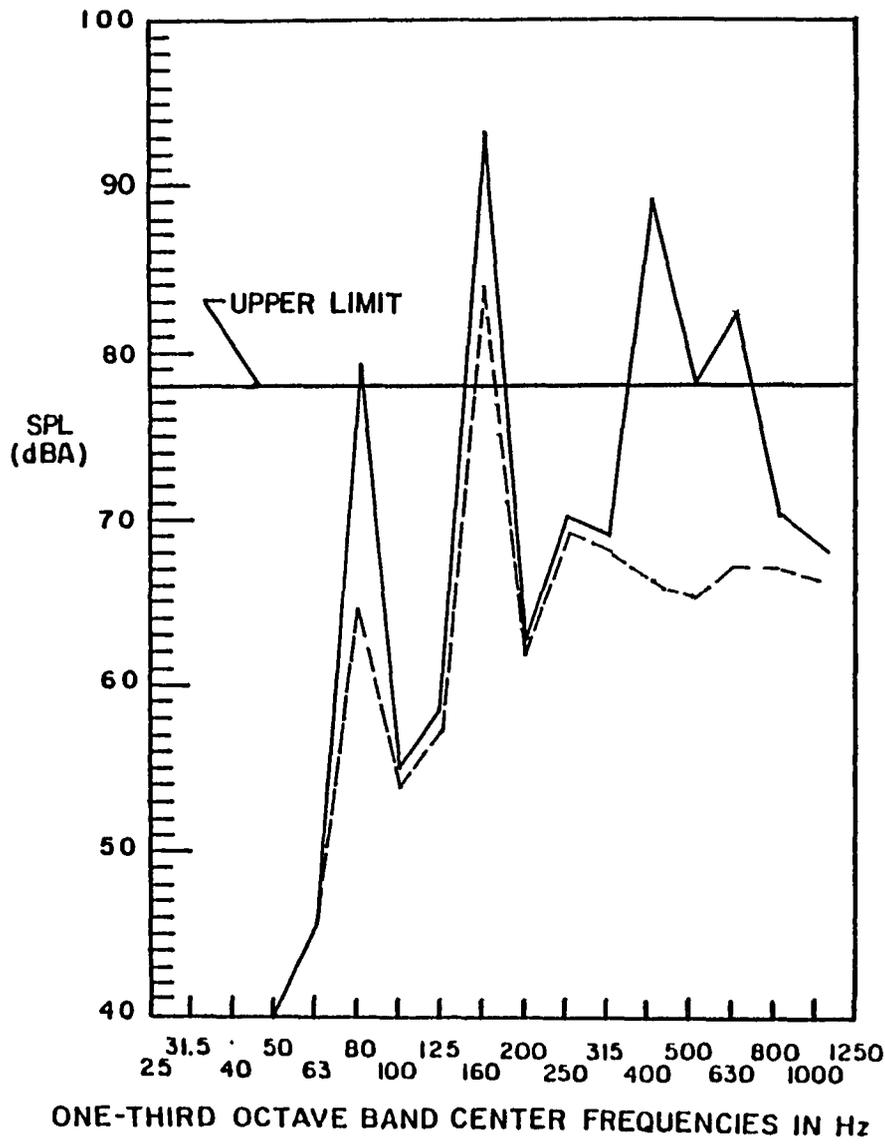
Fig. 46 Interior noise with honeycomb-damping tape treatment (Panel No. 1)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL (dBA)

UPPER LIMIT

100
90
80
70
60
50
40
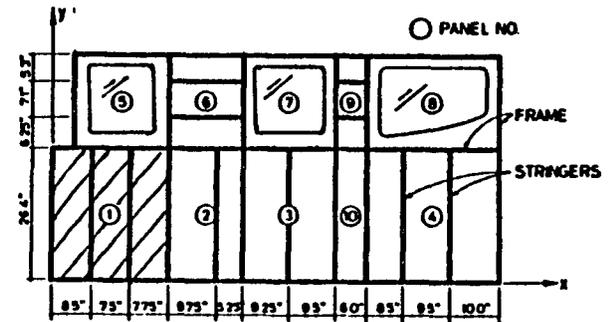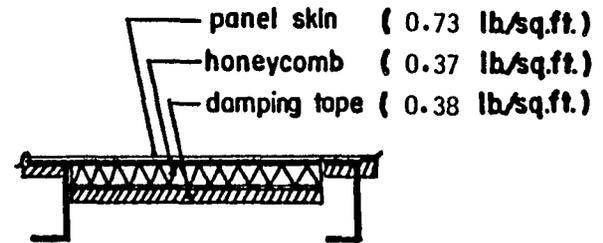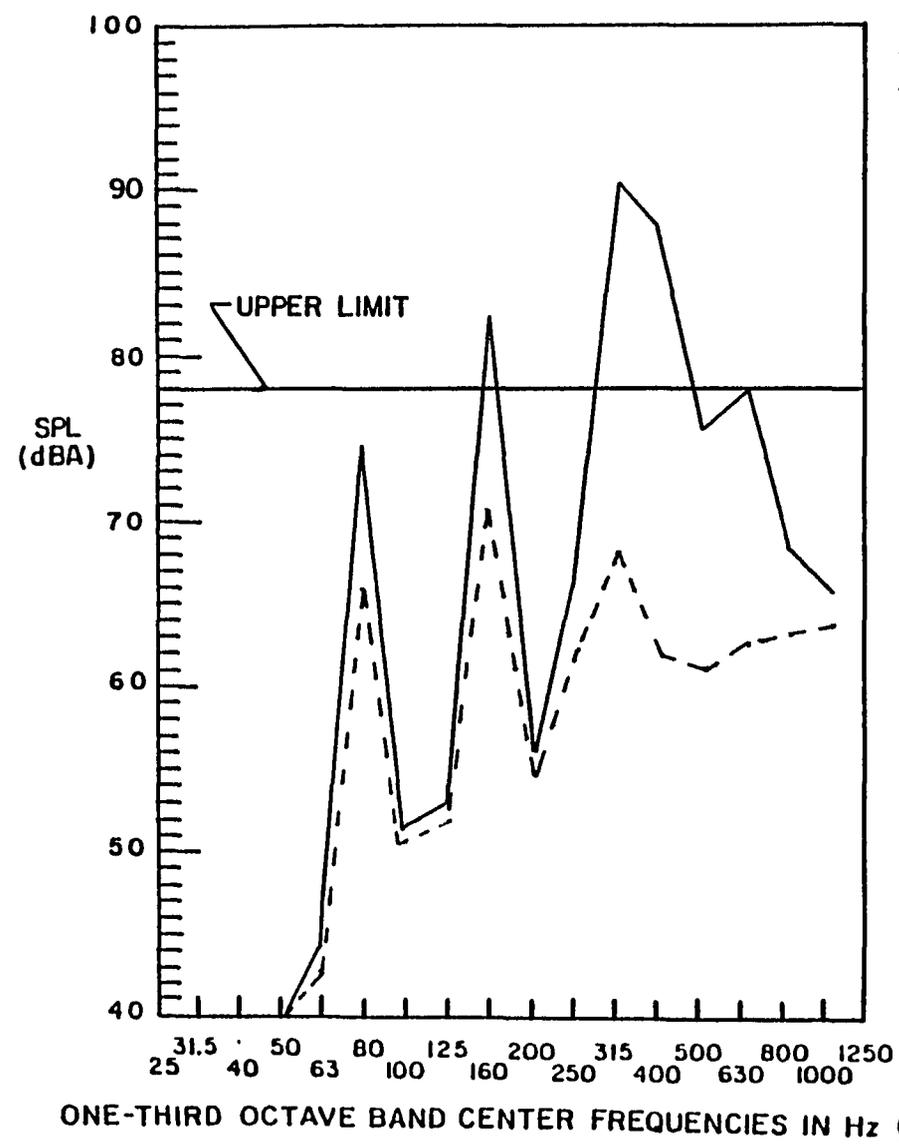
—— baseline : OA = 92 dBA
--- honeycomb : OA = 72 dBA
+ damping tape

added weight = 2.06 lbs.
panel area = 2.75 sq.ft.

panel skin (1.6 lb./sq.ft.)
honeycomb (0.37 lb./sq.ft.)
damping tape (0.38 lb./sq.ft.)

O PANEL NO.

FRAME
STRINGERS

⑤ ⑥ ⑦ ⑨ ⑧
① ② ③ ⑩ ④

6.5" 7.5" 7.75" 9.75" 9.75" 9.25" 9.5" 10.0" 8.5" 9.5" 10.0"

⬜ — unit under study

25 31.5 40 50 63 80 100 125 160 200 250 315 400 500 630 800 1000 1250

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

Fig. 47 Interior noise with honeycomb-damping tape treatment (Panel No. 2)

-97-

Fig. 48   Interior noise with honeycomb-damping tape treatment (Panel No. 3)

Fig. 49 Interior noise with honeycomb-damping tape treatment (Panel No. 4)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL (dBA)

UPPER LIMIT

100
90
80
70
60
50
40

31.5   50   125   200   315   500   800   1250
25   40   63   80   100   160   250   400   630 1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

—— baseline : OA = 106 dBA
---- honeycomb : OA = 92 dBA
+ damping tape

added weight = 1.46 lbs.
panel area = 1.94 sq.ft.

panel skin   (0.86 lb/sq.ft.)
honeycomb   (0.37 lb/sq.ft.)
damping tape (0.38 lb/sq.ft.)

O PANEL NO

FRAME

STRINGERS

⬚ — unit under study

Fig. 50   Interior noise with honeycomb-damping tape treatment (Pane. No. 6)

Fig. 51   Interior noise with honeycomb-damping tape treatment (Panel No. 10)

Fig. 52   Interior noise with honeycomb-damping tape treatment (Sidewall)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

-103-

100

90

80

SPL
(dBA)

70

60

50

40

UPPER LIMIT
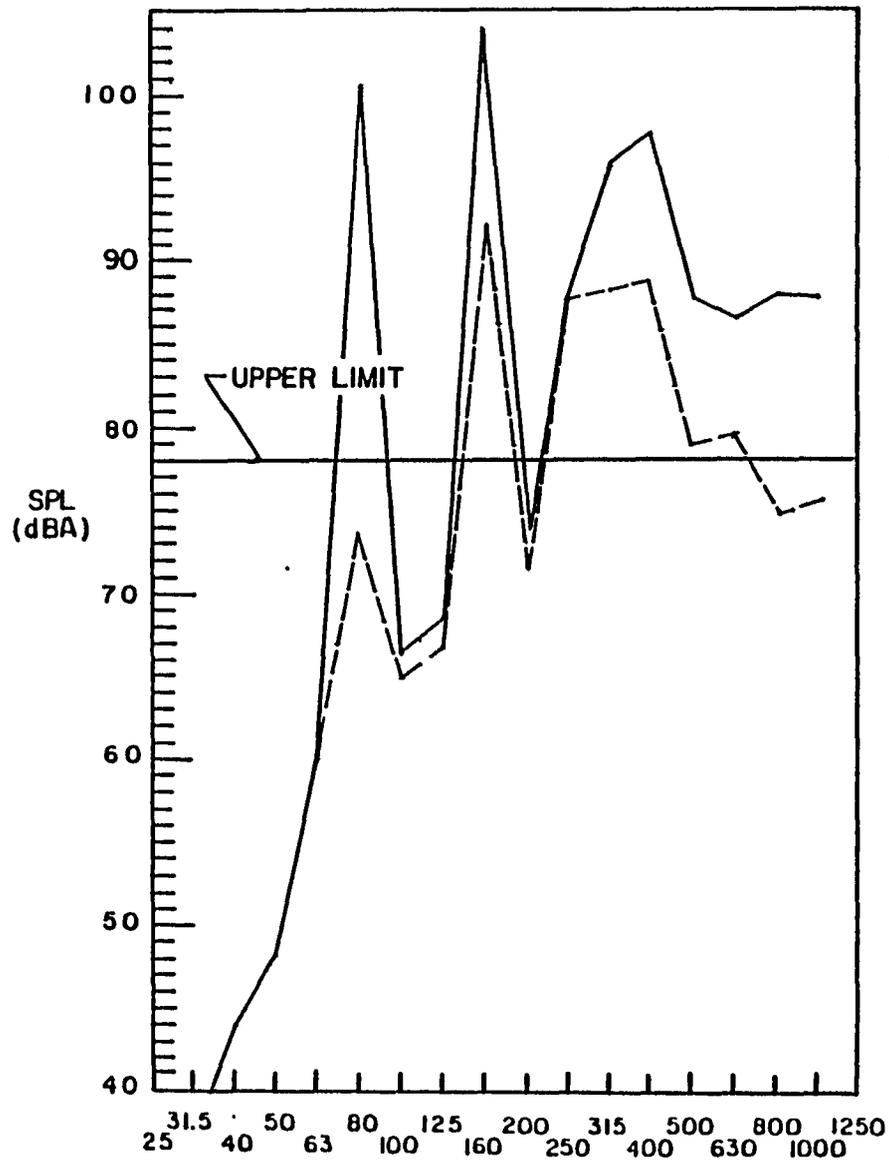
— baseline : OA = 102 dBA

--- honeycomb : OA = 78 dBA
+ damping tape

added weight = 5.68 lbs.
panel area = 3.44 sq.ft.

panel skin (0.89 lb/sq.ft.)
honeycomb (1.4 lb/sq.ft.)
damping tape (0.25 lb/sq.ft.)

O PANEL NO

FRAME

STRINGERS

⬚ — unit under study

315   50   80   125   200   315   500   800   1250
25   40   63   100   160   250   400   630   1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)
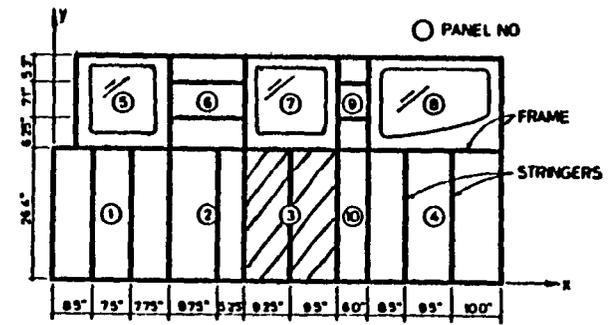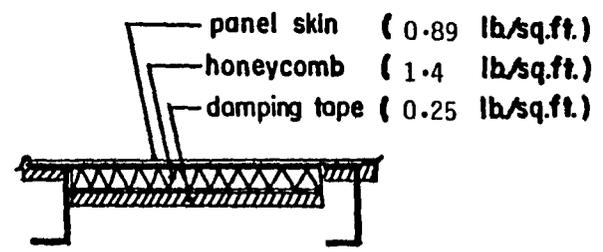
Fig. 53   Interior noise levels with heavy honeycomb-damping tape treatment (Panel No. 3)

Fig. 54 Additional transmission losses for acoustic blankets and semi-rigid materials (μ_5 = 0.01 lb/ft^2)

Fig. 55   Additional transmission losses
for acoustic blankets and semi-rigid
materials   (μ_5 = 0.358 lb/ft^2)

ΔTL
(dB)

50

40

—————— acoustic blankets

– – – – – semi-rigid materials

30

μ_1 = 1.15 lb/ft²

μ_3 = 0.01 lb/ft²

$d_2 = d_4$ = 0.073 ft

20

10

0

-10

31.5   50   80   125   200   315   500   800   1250
25   40   63   100   160   250   400   630   1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

μ_1

d_2

μ_3

d_4

μ_5
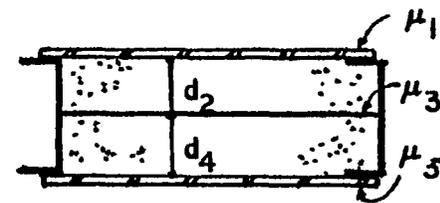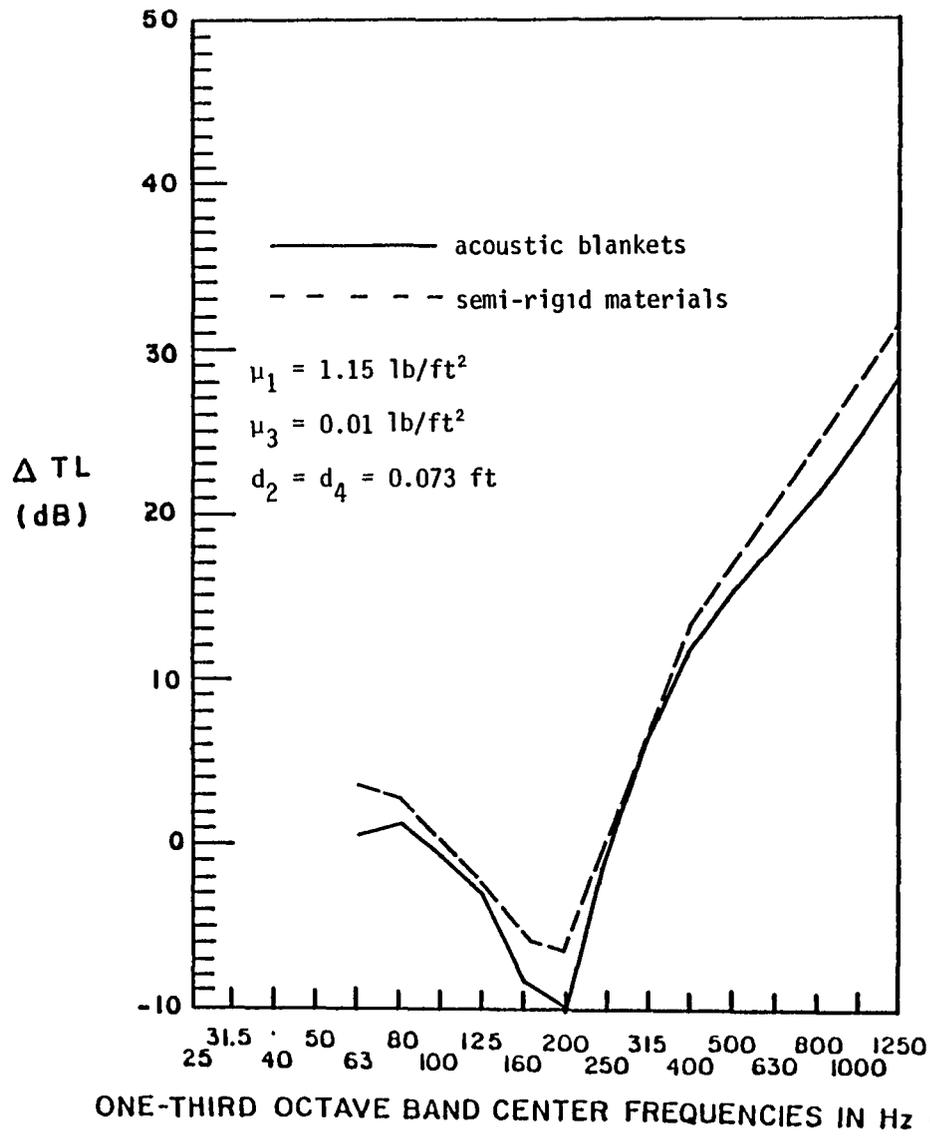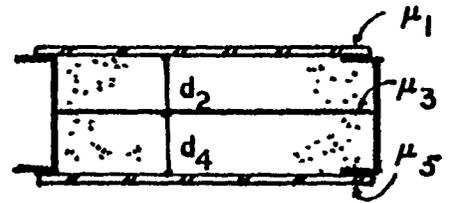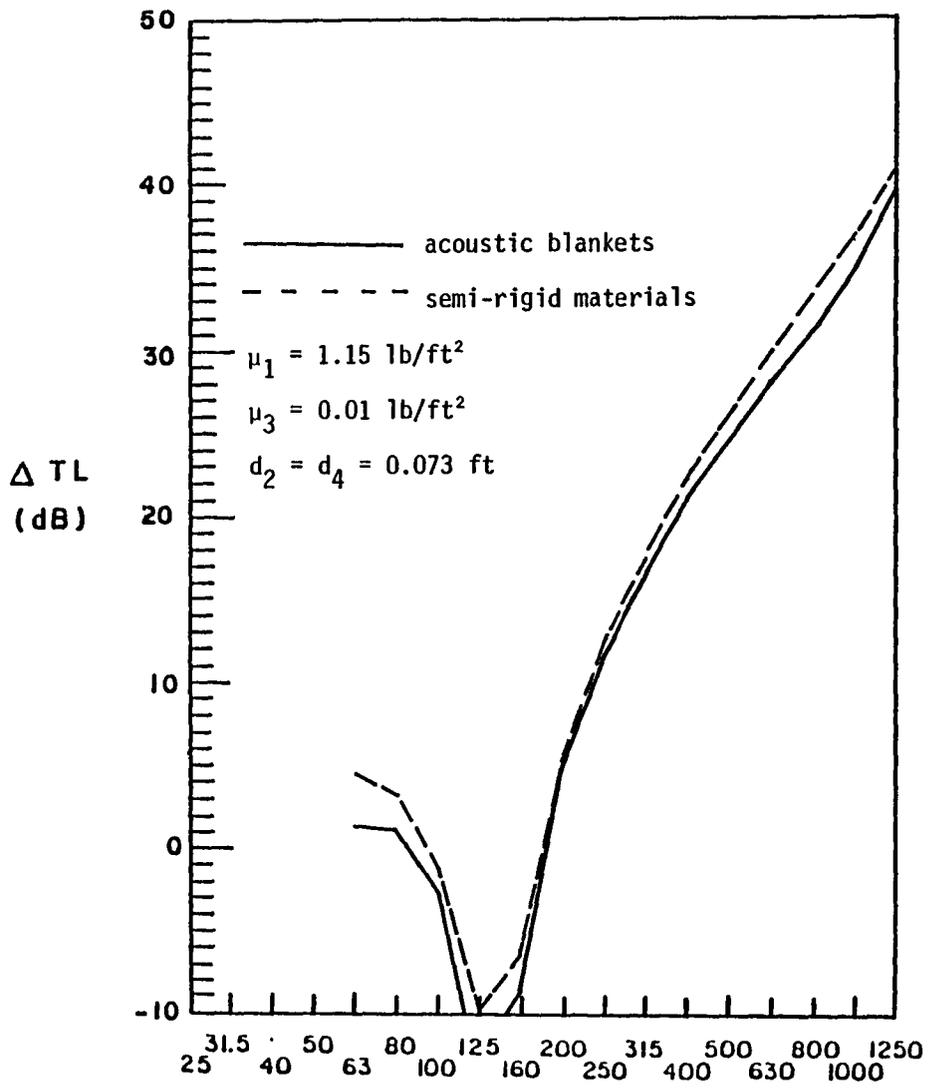
SIDEWALL SECTION

Fig. 56   Additional transmission losses
for acoustic blankets and semi-rigid
materials   (μ_5 = 1.0 lb/ft²)
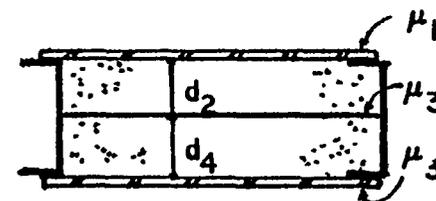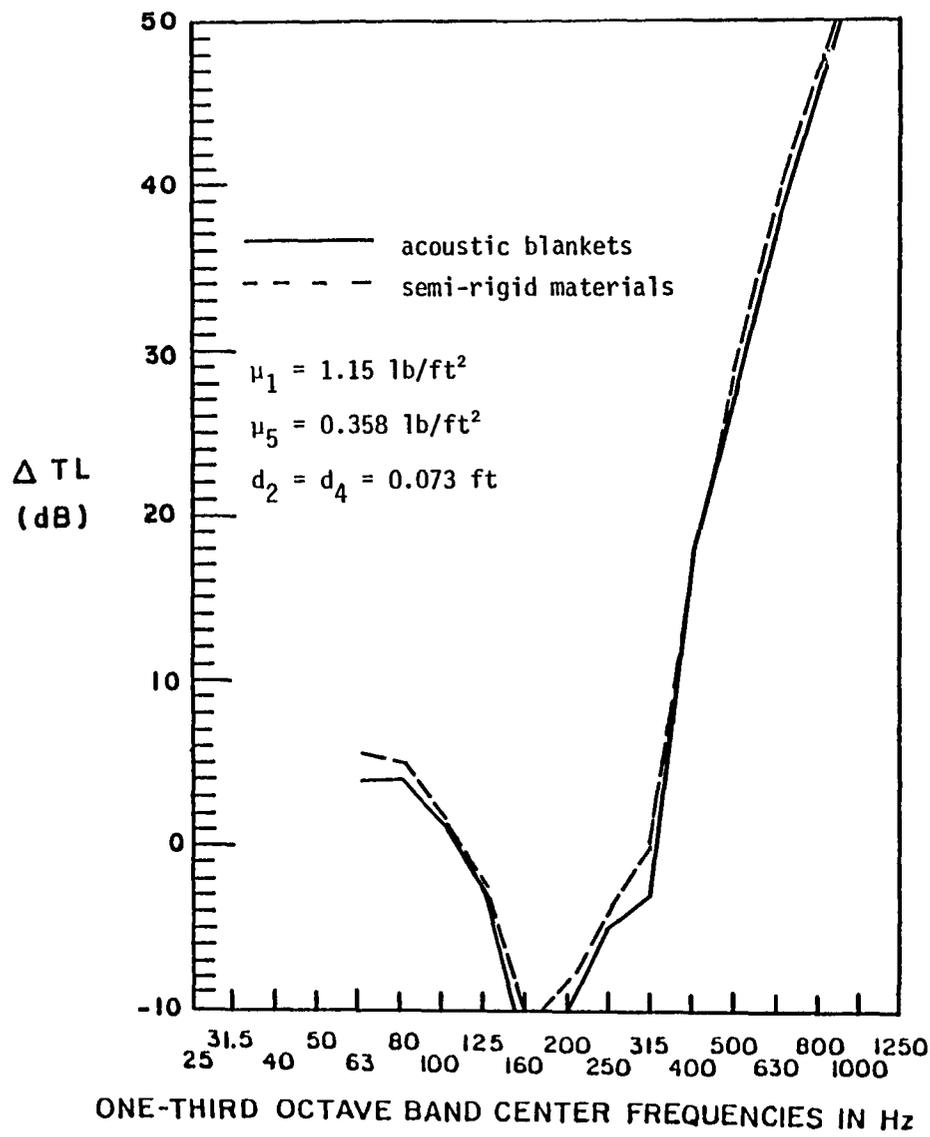
Fig. 57 Additional transmission losses for acoustic blankets and semi-rigid materials (μ_3 = 1.0 lb/ft^2)

SIDEWALL SECTION

acoustic blankets
semi-rigid materials

μ_1 = 1.15 lb/ft^2

μ_5 = 0.358 lb/ft^2

d_2 = d_4 = 0.073 ft

Δ TL (dB)

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)
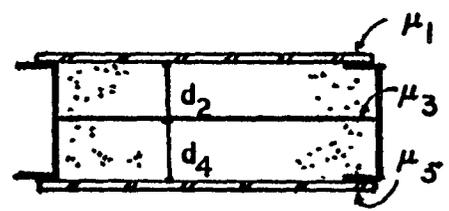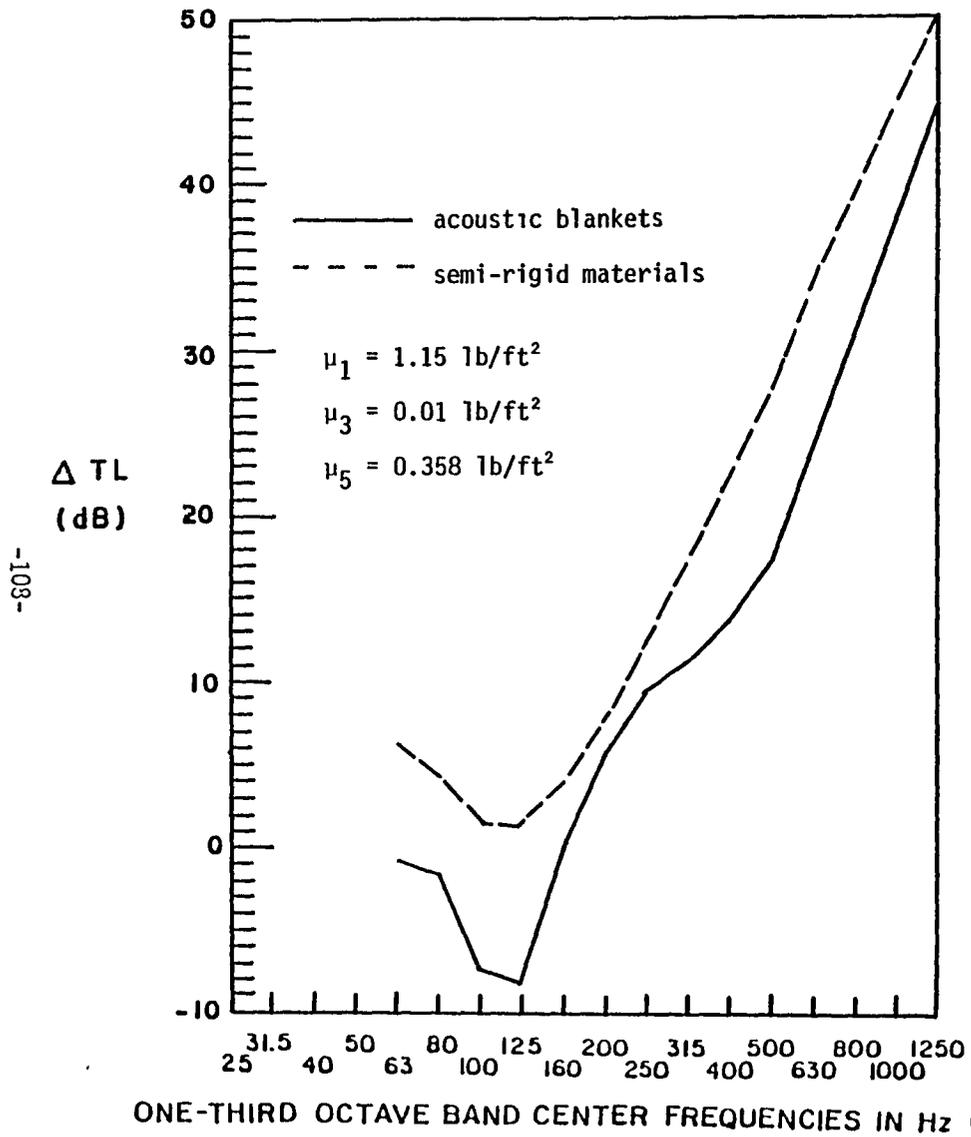
Fig. 58 Additional transmission losses for acoustic blankets and semi-rigid materials (d_2 = d_4 = 0.167 ft)
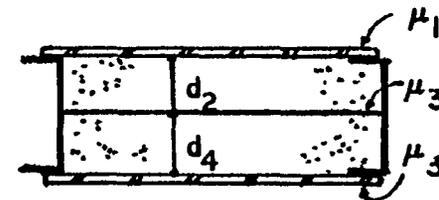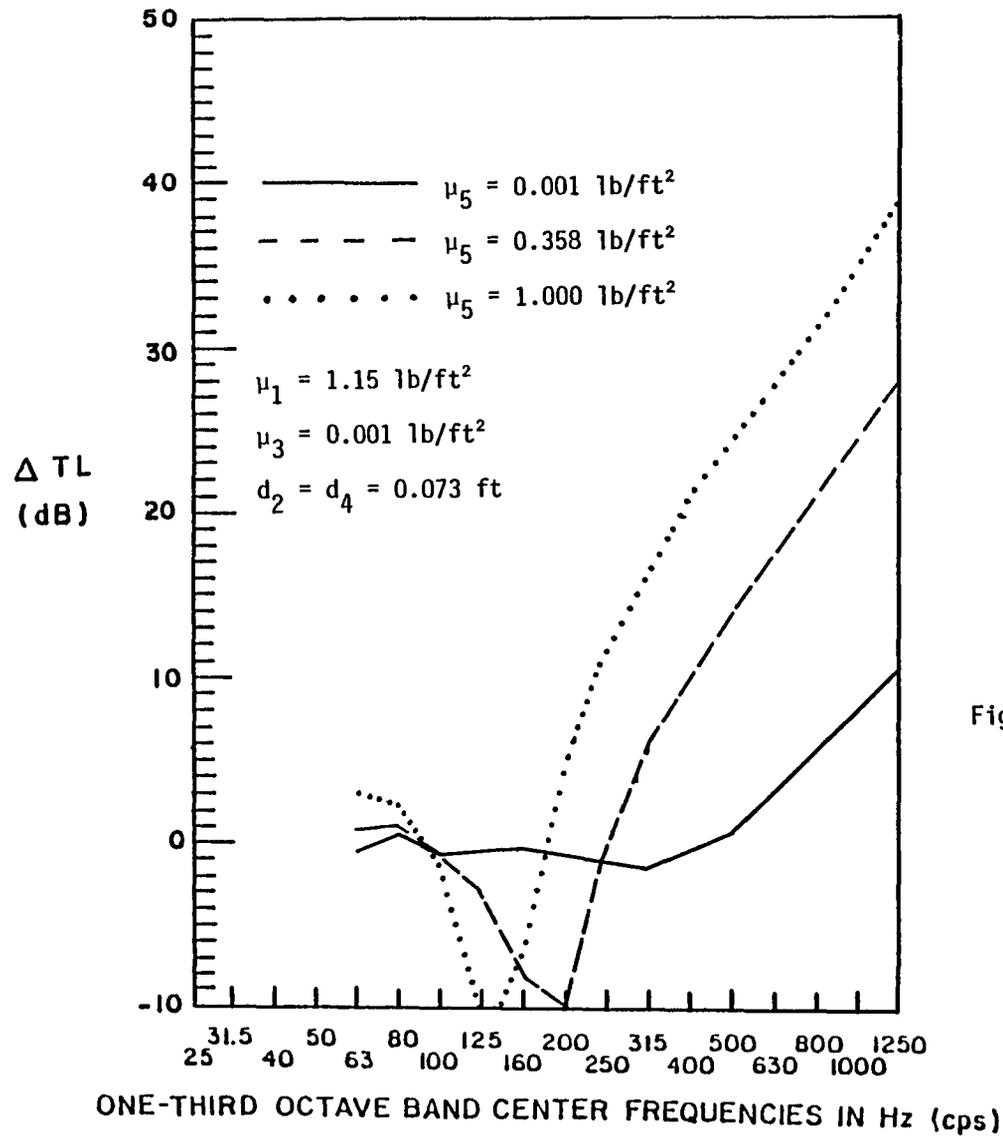
Fig. 59   Additional transmission losses for different trim panel surface densities
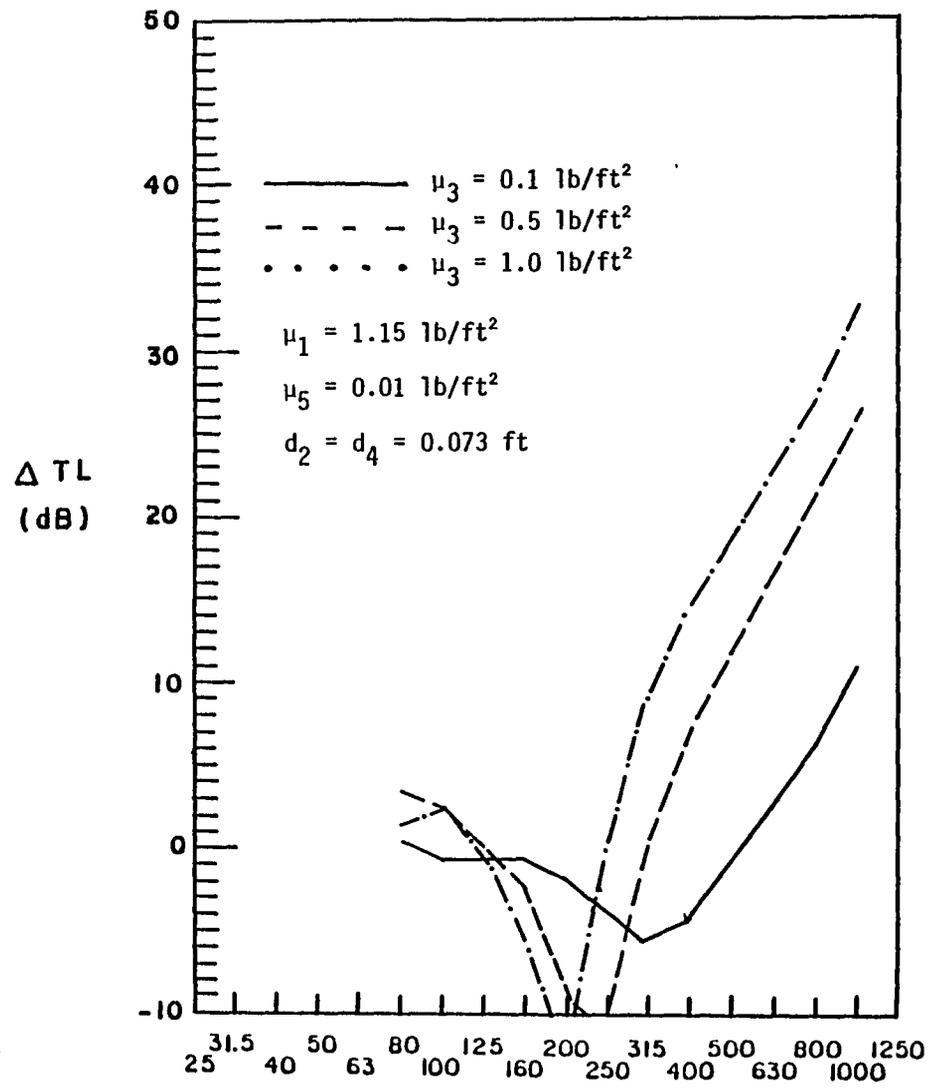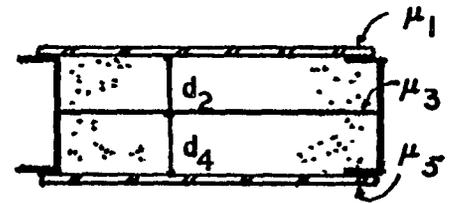
Fig. 60   Additional transmission losses
for add-on treatment
$(\mu_1 = 1.15, \mu_5 = 0.01)$
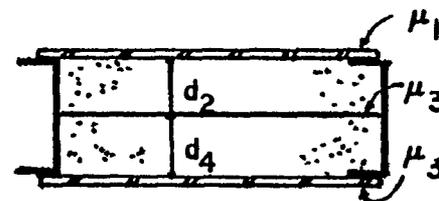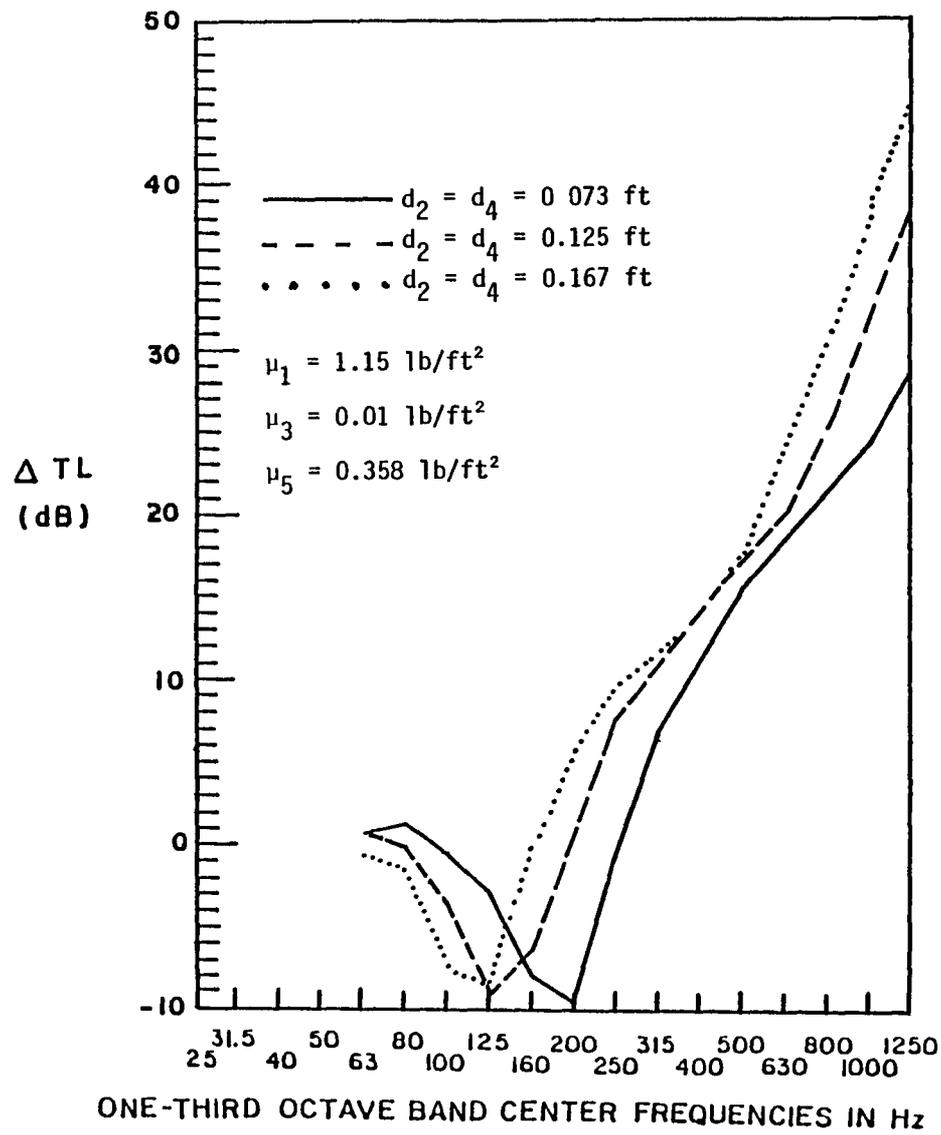
Fig. 61 · Additional transmission losses for different distances between the elastic panel and the trim panel
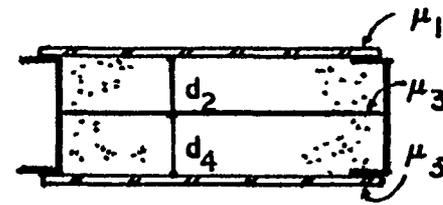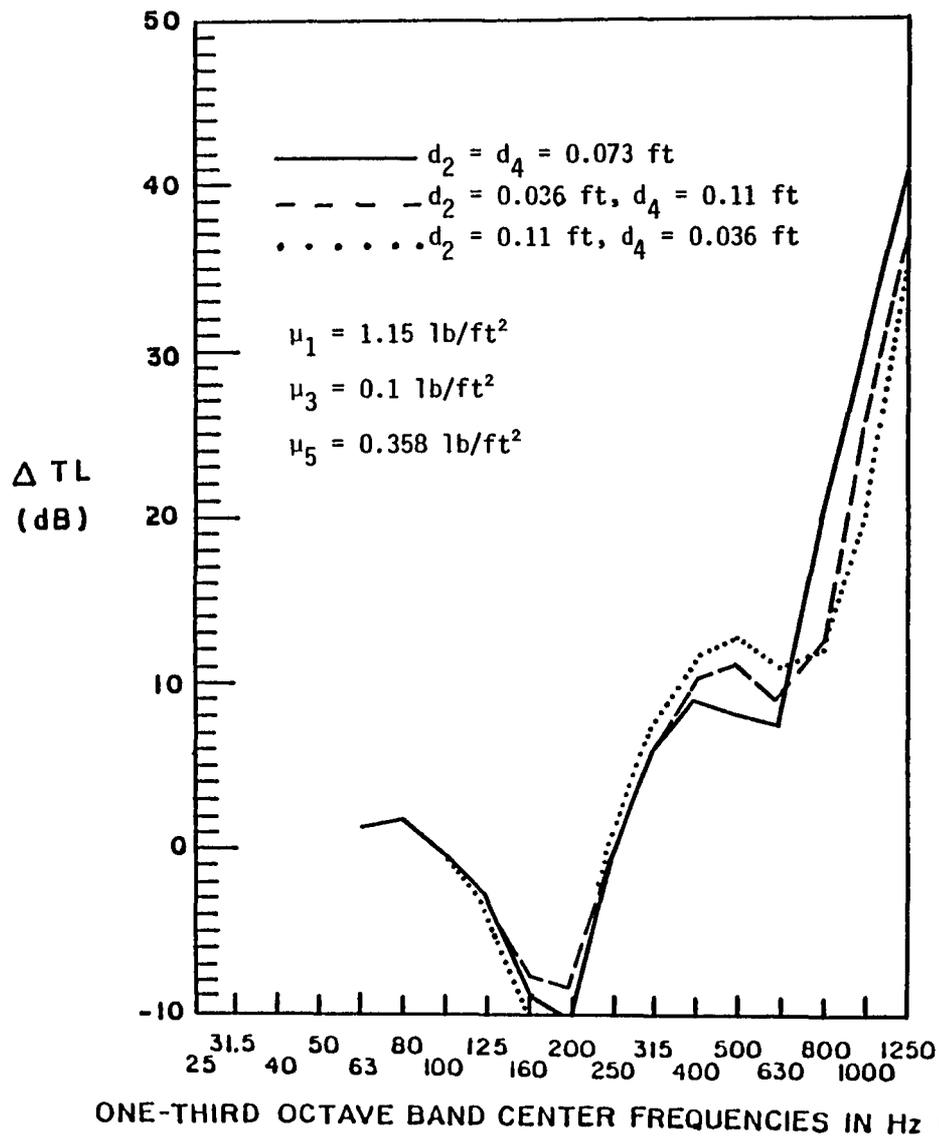
SIDEWALL SECTION

Fig. 62 Additional transmission losses for a light septum and different distances between the elastic panel and the septum
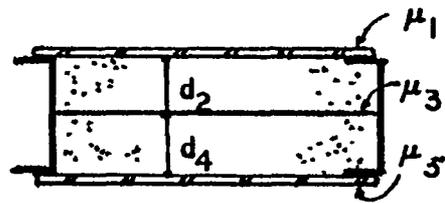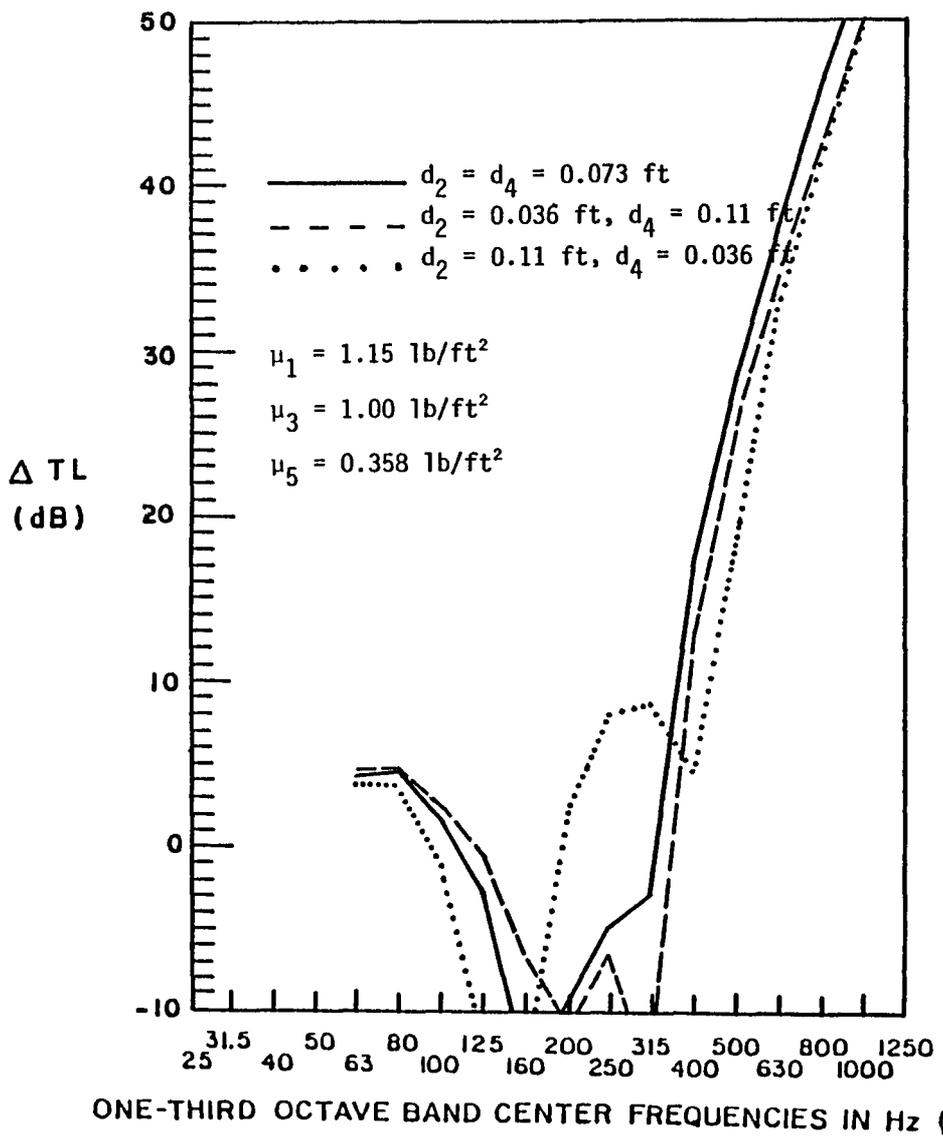
Fig. 63 Additional transmission losses
for different distances between the
elastic panel and the septum

SIDEWALL SECTION
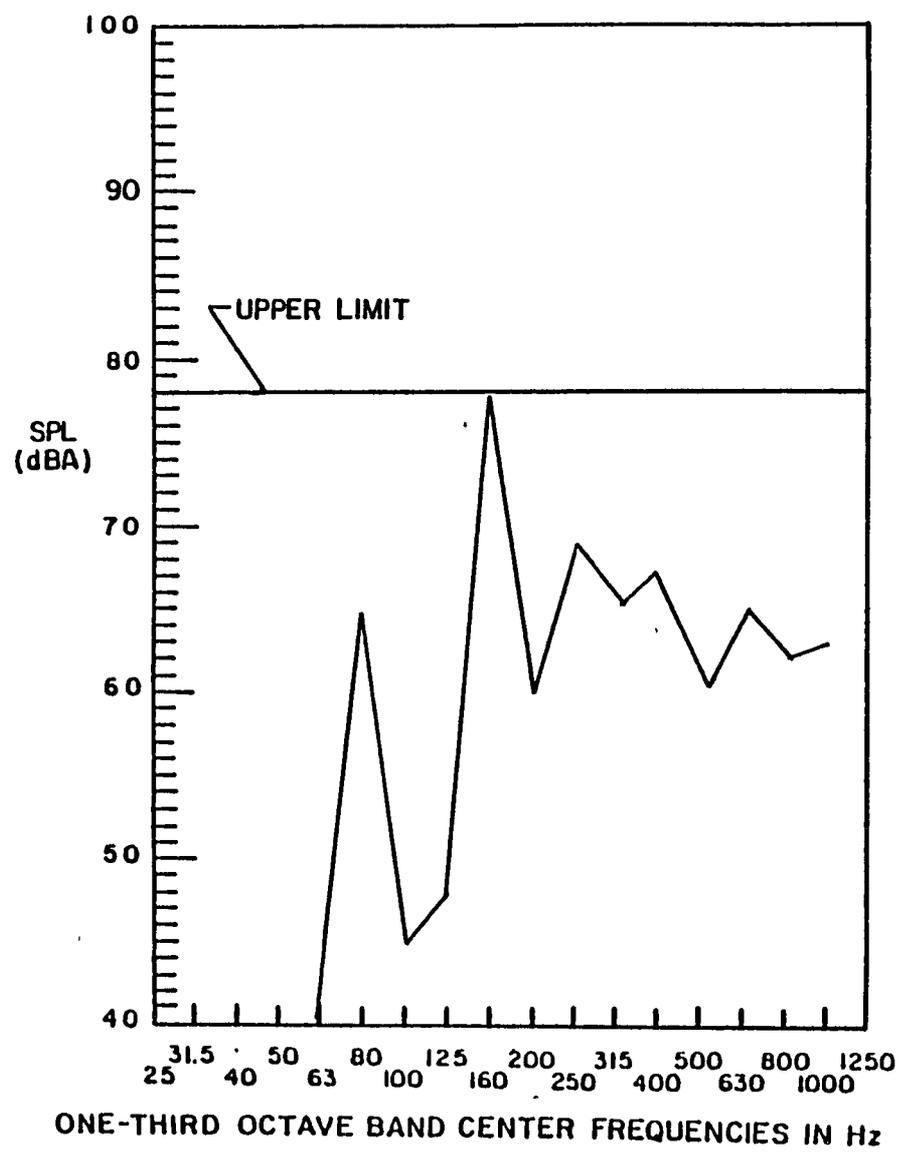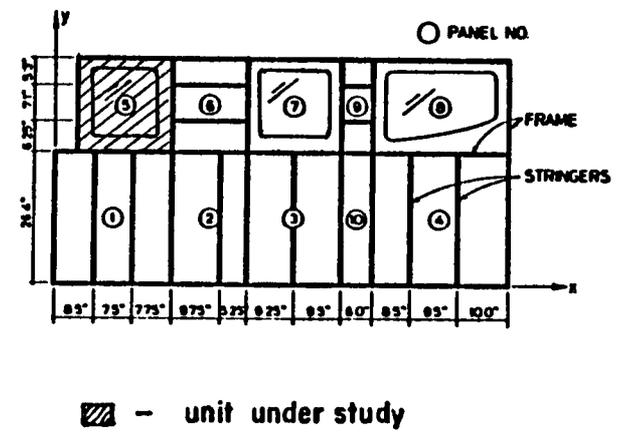
Fig. 64   Interior noise transmitted by a window (Panel No. 5)
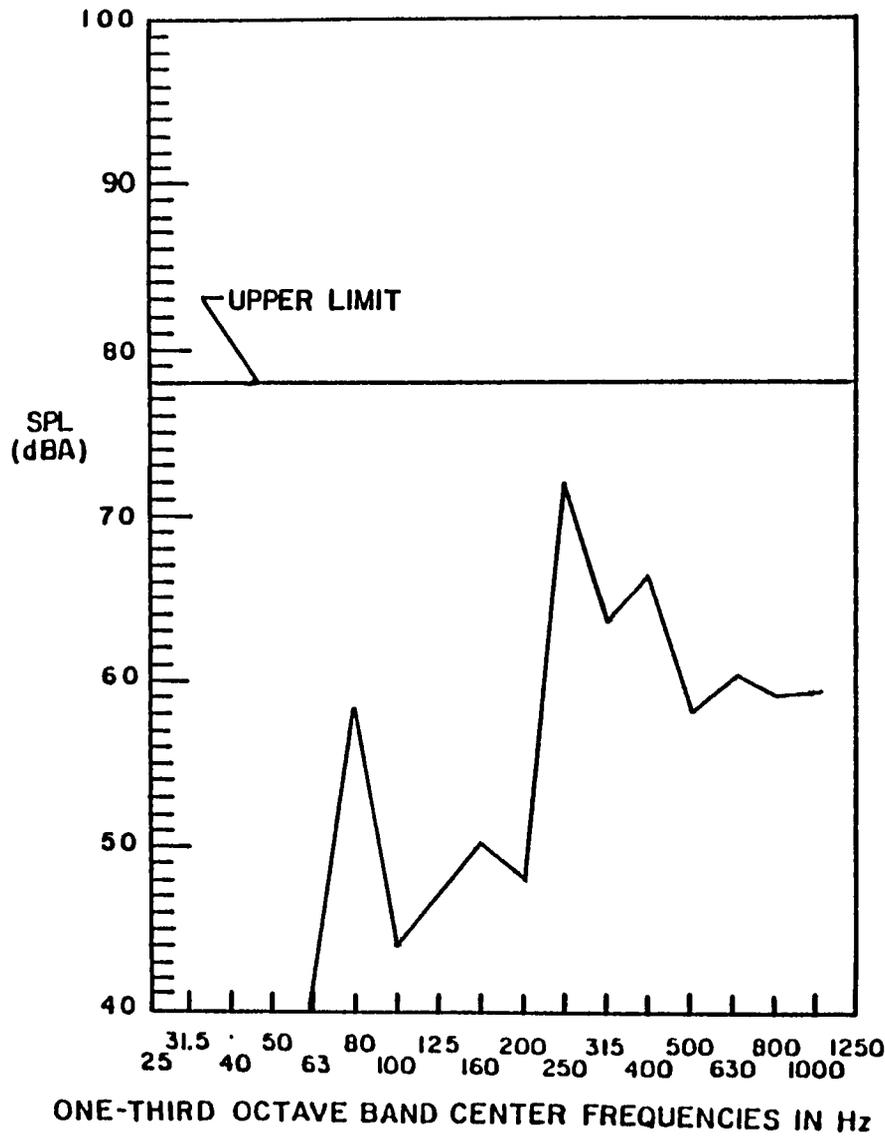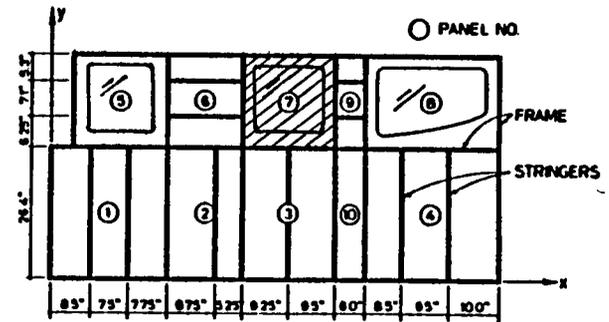
Fig. 65   Interior noise transmitted by a window (Panel No. 7)

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ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar
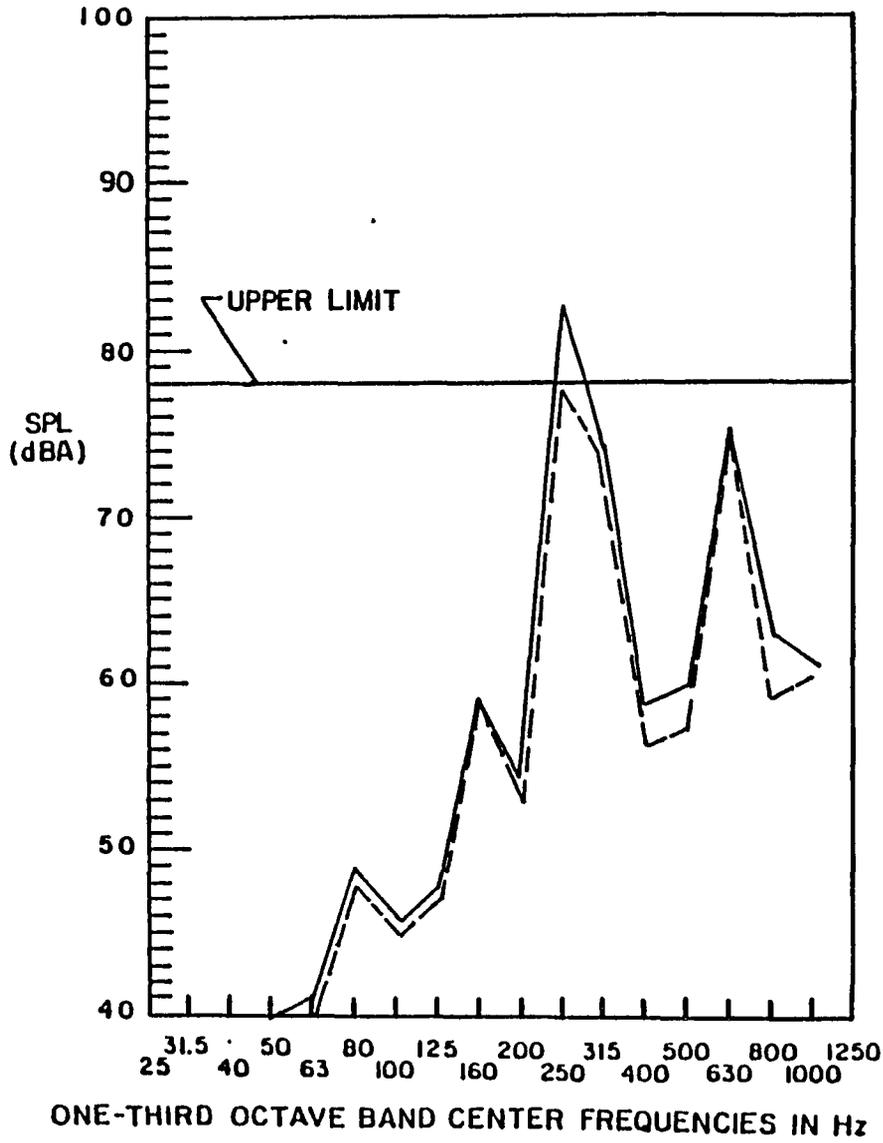
—— baseline : OA = 85 dBA
--- damping tape : OA = 80 dBA

added weight = 1.0 lbs.
panel area · = 3.6 sq.ft.

UPPER LIMIT

SPL (dBA)

damping tape
(0.38 lb/sq.ft.)

O PANEL NO.

FRAME

STRINGERS

y

x

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

25 31.5 40 50 63 80 100 125 160 200 250 315 400 500 630 800 1000 1250

100 90 80 70 60 50 40
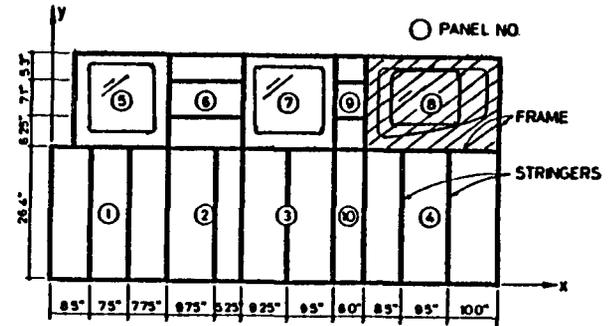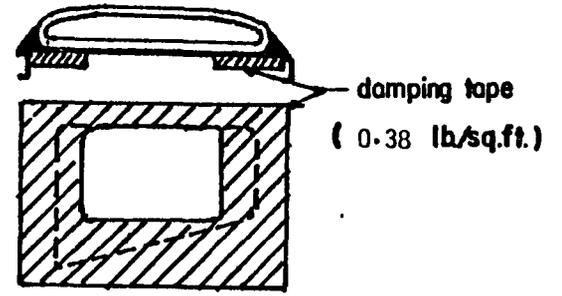
▨ — unit under study

Fig. 66 Interior noise transmitted by a window (Panel No. 8)

Fig. 67 Optimized interior noise (Panel No. 1)
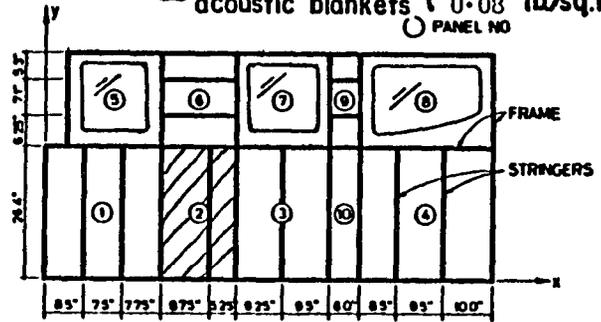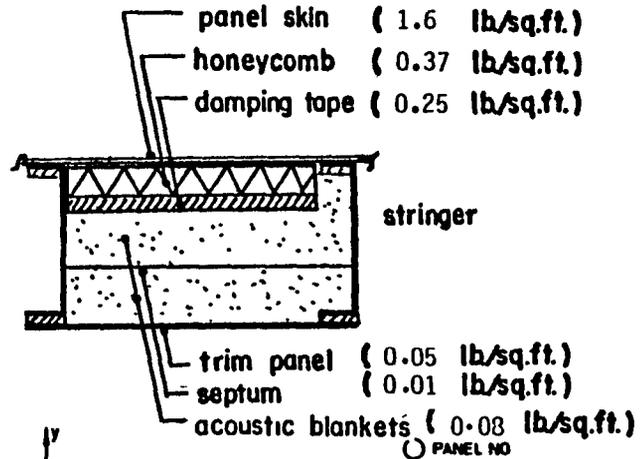
Fig. 68 Optimized interior noise (Panel No. 2)
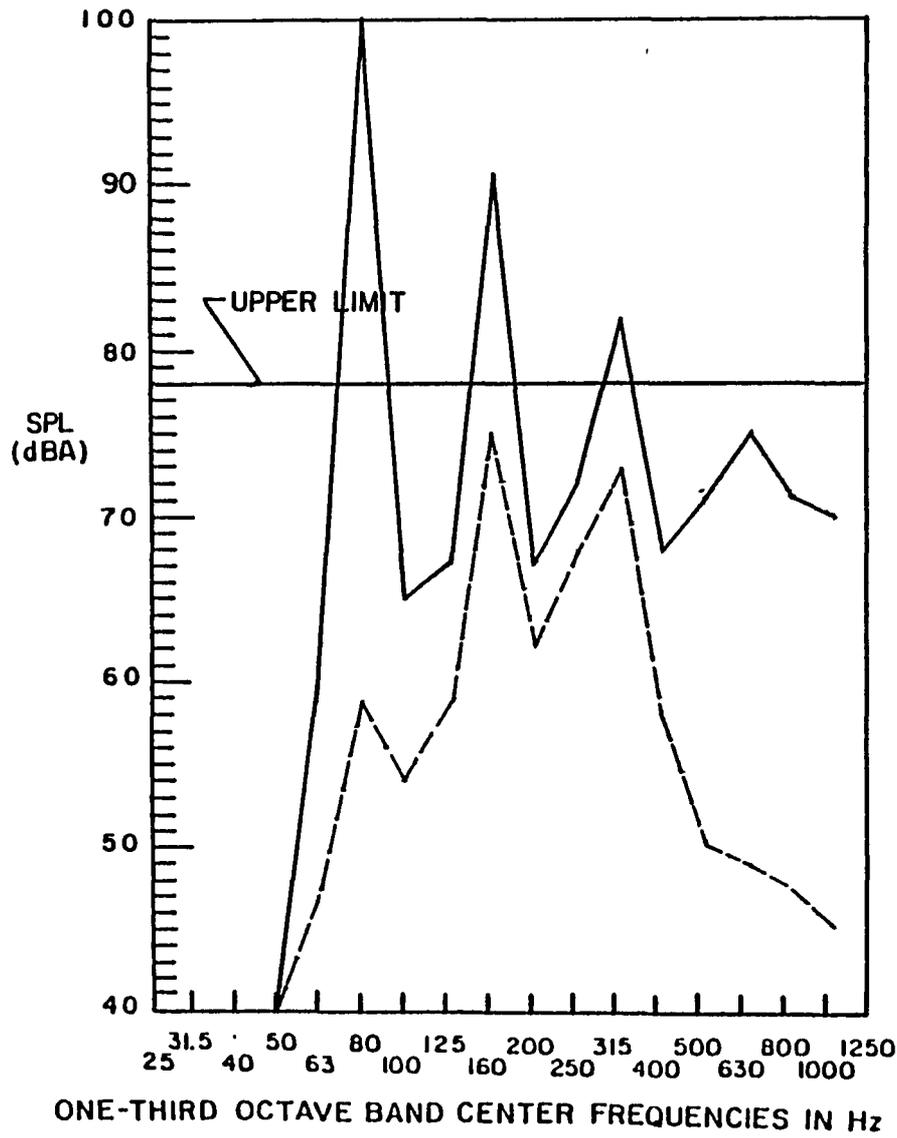
-118-

Fig. 69 Optimized interior noise (Panel No. 3)

Fig. 70 Optimized interior noise (Panel No. 4)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

-121-

100

90

80

SPL
(dBA)

70

60

50

40

UPPER LIMIT

31.5   50   80   125   160   250   400   630   1250
25   40   63   100   160   250   400   630   1000
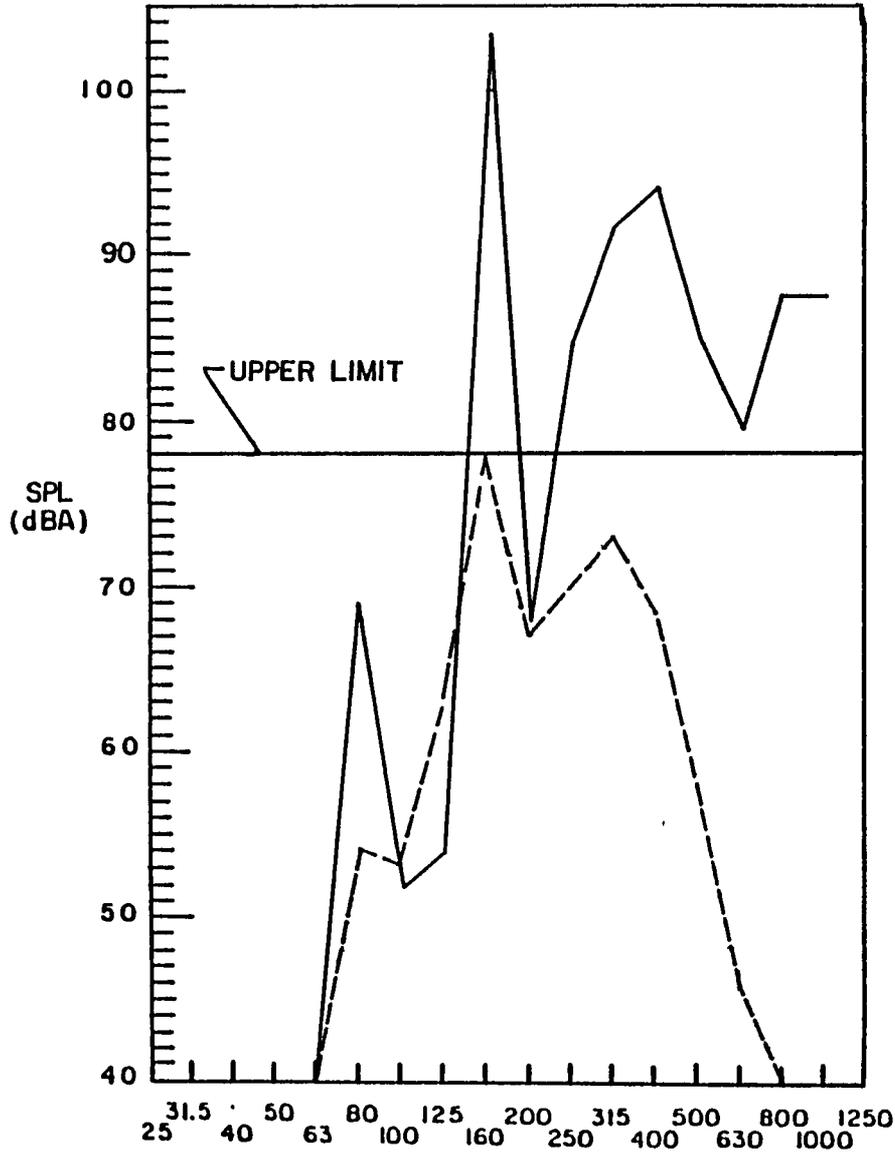
ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

——— baseline : OA = 106 dBA
——— optimized : OA = 78 dBA

added weight = 3.91 lbs.
panel area = 1.94 sq.ft.

panel skin   (0.86 lb/sq.ft.)
honeycomb   (0.37 lb/sq.ft.)
damping tape (0.38 lb/sq.ft.)

stringer

trim panel   (1.0   lb/sq.ft.)
septum   (0.01 lb/sq.ft.)
acoustic blankets (0.08 lb/sq.ft.)
◯ PANEL NO.

FRAME

STRINGERS
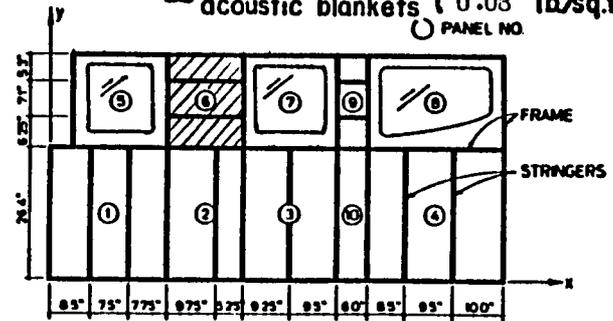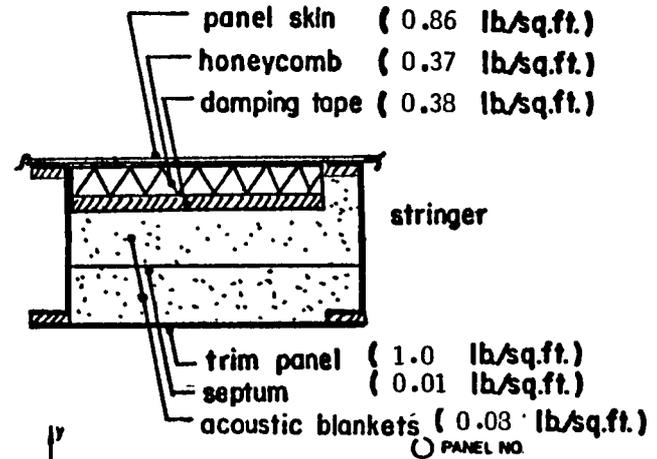
— unit under study

Fig. 71 Optimized interior noise (Panel No. 6)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL
(dBA)

UPPER LIMIT

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

25 31.5 40 50 63 80 100 125 160 200 250 315 400 500 630 800 1000 1250
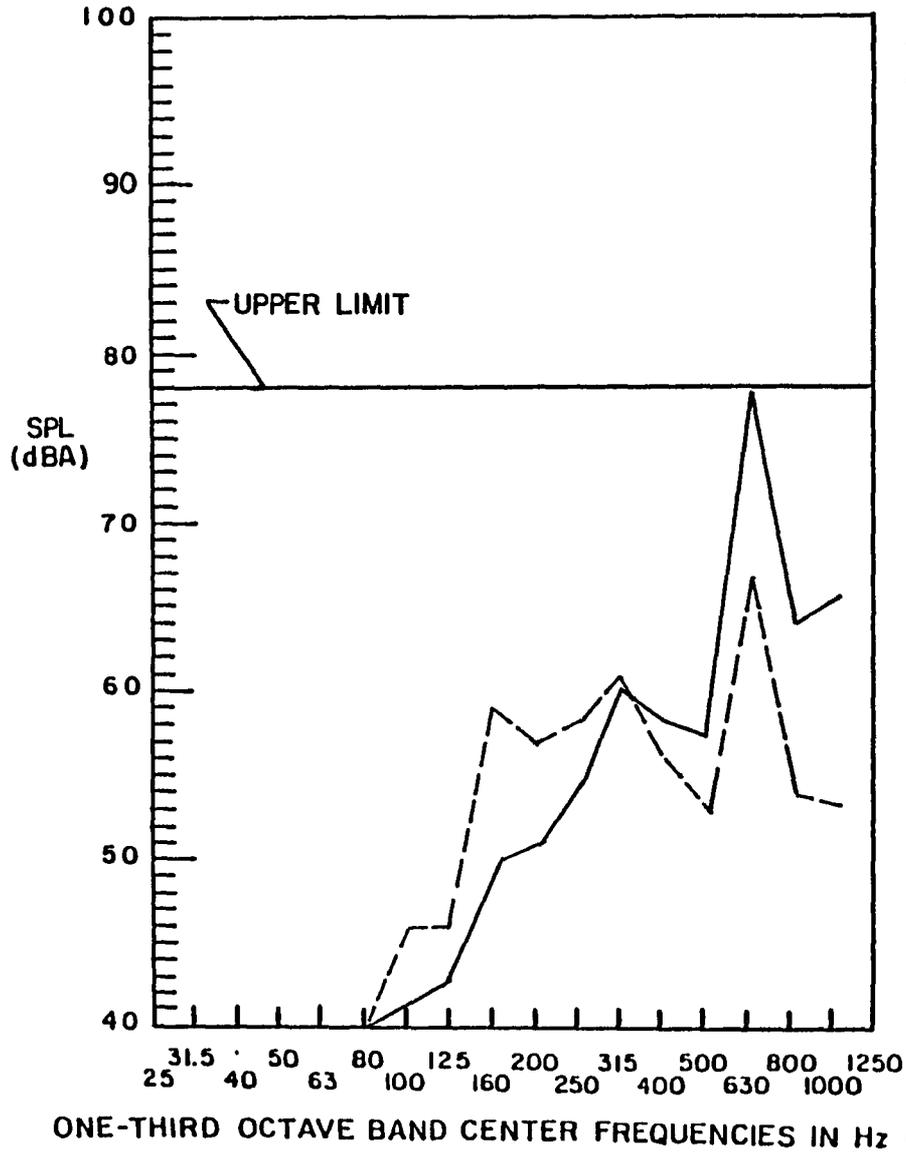
—— baseline : OA = 78 dBA
--- optimized : OA = 69 dBA

added weight = 0.42 lbs.
panel area = 0.77 sq.ft.

panel skin (1.06 lb/sq.ft.)

damping tape (0.25 lb/sq.ft.)

stringer

trim panel (0.1 lb/sq.ft.)
septum (0.01 lb/sq.ft.)
acoustic blankets (0.08 lb/sq.ft.)
○ PANEL NO.

FRAME

STRINGERS
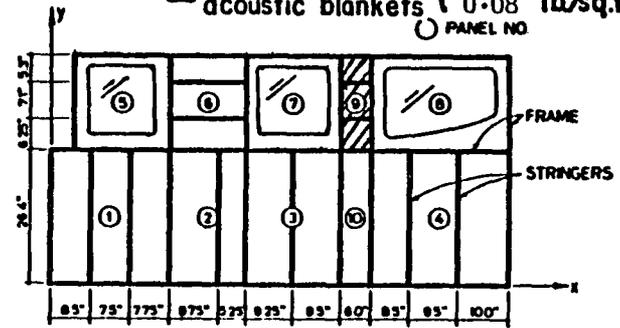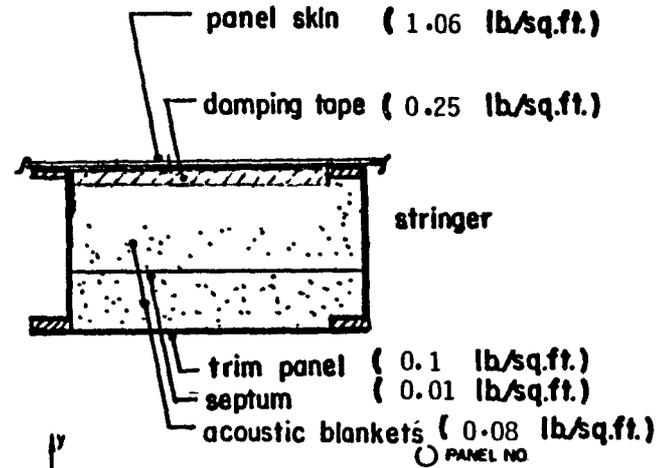
▨ — unit under study

Fig. 72 Optimized interior noise (Panel No. 9)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

SPL (dBA)

UPPER LIMIT

——— baseline : OA = 86 dBA
- - - optimized : OA = 73 dBA

added weight = 1.14 lbs.
panel area = 1.1 sq.ft.
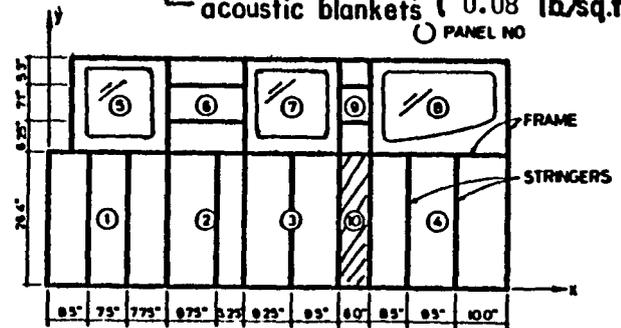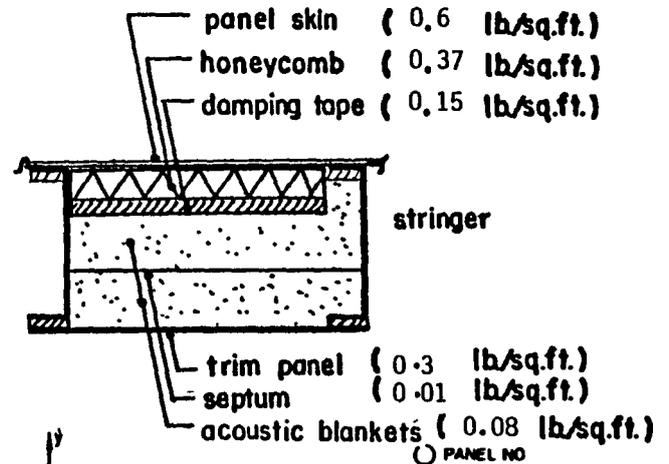
panel skin (0.6 lb/sq.ft.)
honeycomb (0.37 lb/sq.ft.)
damping tape (0.15 lb/sq.ft.)

stringer

trim panel (0.3 lb/sq.ft.)
septum (0.01 lb/sq.ft.)
acoustic blankets (0.08 lb/sq.ft.)
◯ PANEL NO

FRAME
STRINGERS

⬛ — unit under study

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)

Fig. 73 Optimized interior noise (Panel No. 10)

ONE-THIRD OCTAVE BAND SOUND PRESSURE LEVEL IN dBA re 0.0002 μbar

-124-

SPL
(dBA)

100

90

80

70

60

50

40

25 31.5 50   80   125   200   315   500   800   1250
   40   63   100 160 250 400 630 1000

ONE-THIRD OCTAVE BAND CENTER FREQUENCIES IN Hz (cps)
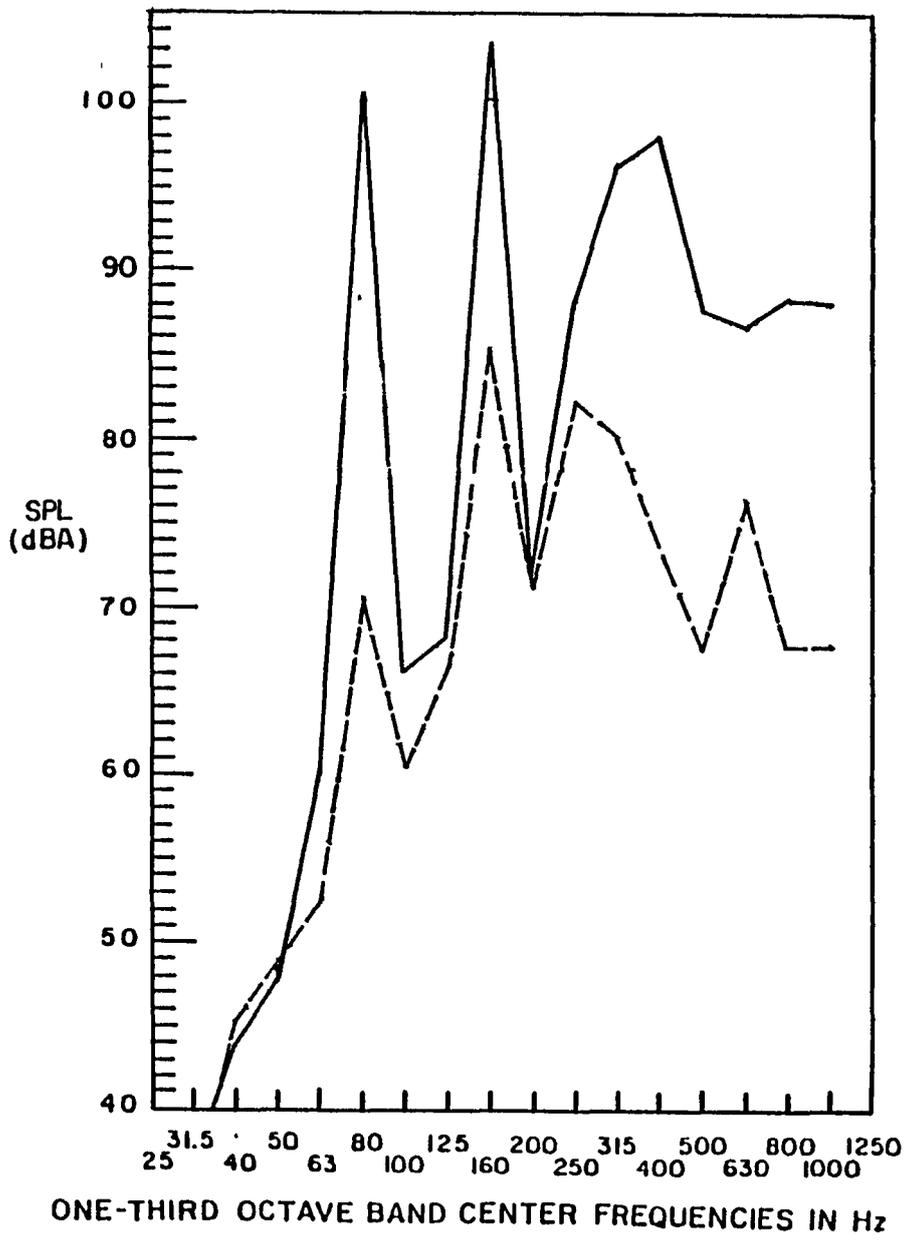
—— baseline : OA = 107 dBA
--- optimized : OA = 88 dBA

added weight = 30 lbs.
panel area = 29 sq.ft.

panel skin* (1.15 lb/sq.ft.)
honeycomb* (0.56 lb/sq.ft.)
damping tape*(0.38 lb/sq.ft.)

stringer

trim panel* (0.3 lb/sq.ft.)
septum * (0.01 lb/sq.ft.)
acoustic blankets*(0.08 lb/sq.ft.)
O PANEL NO.

FRAME

STRINGERS
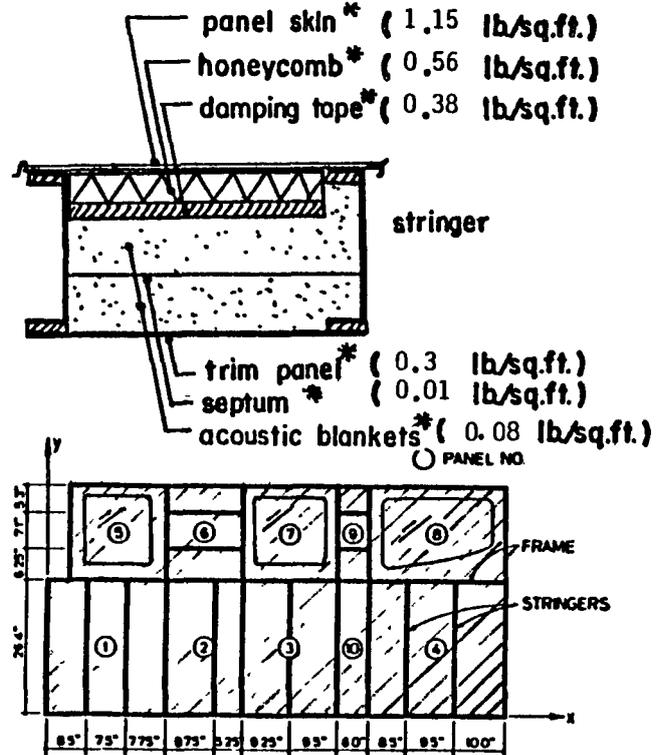
— unit under study
* average values

Fig. 74 Optimized interior noise (Sidewall)
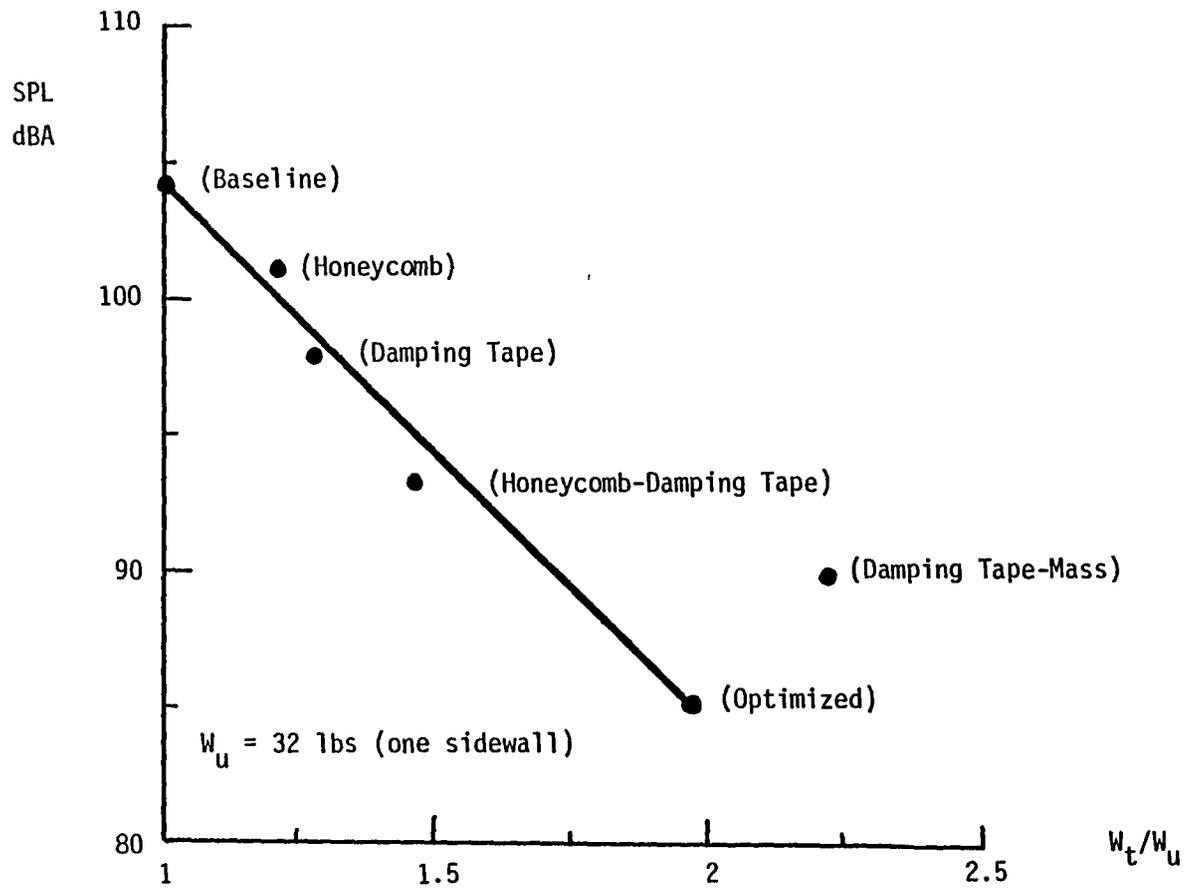
Fig. 75   Interior sound pressure levels vs the ratio of treated/untreated weight
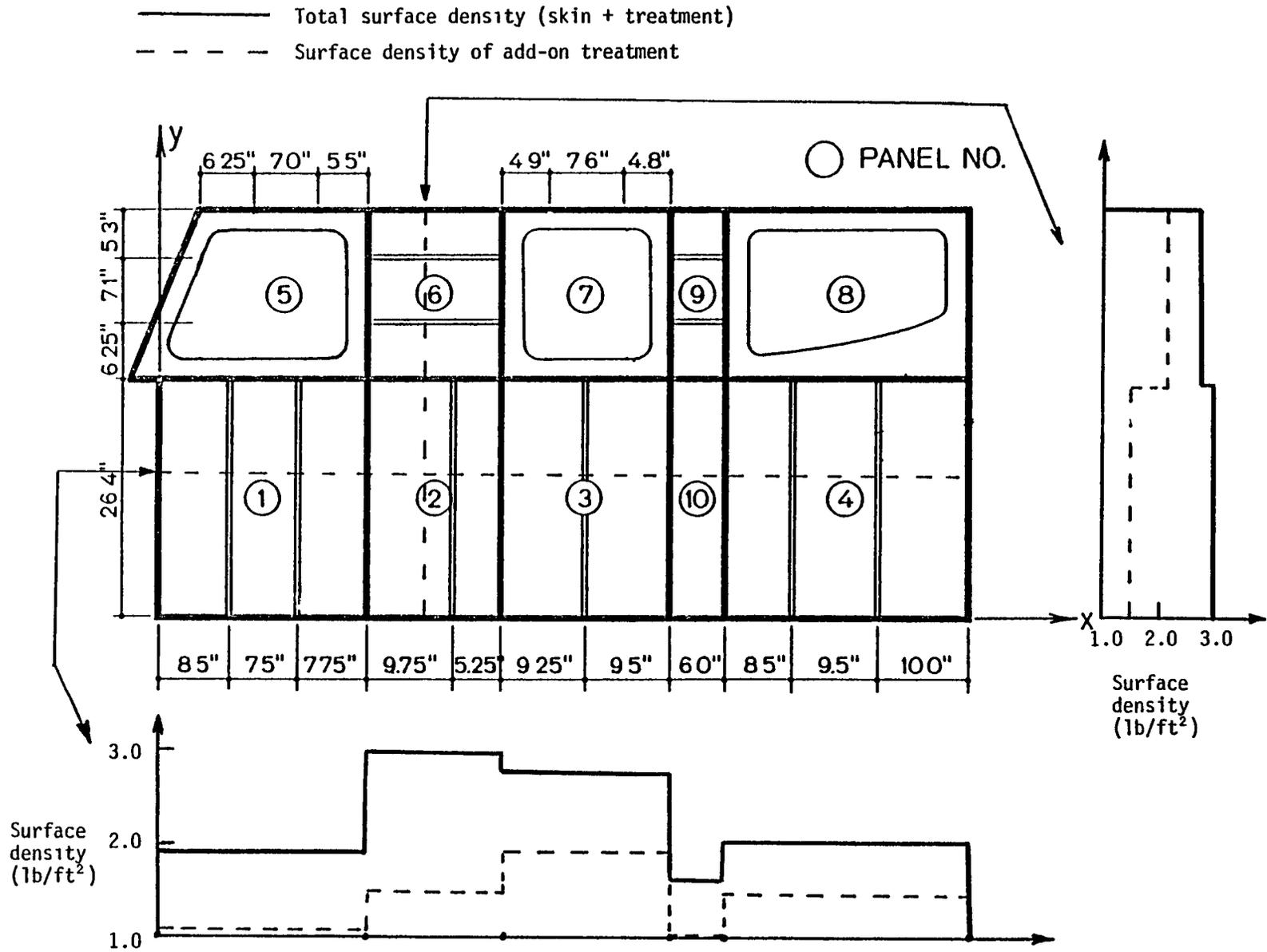
Fig. 76 Distribution of surface density

REAR BULKHEAD

FRONT BULKHEAD
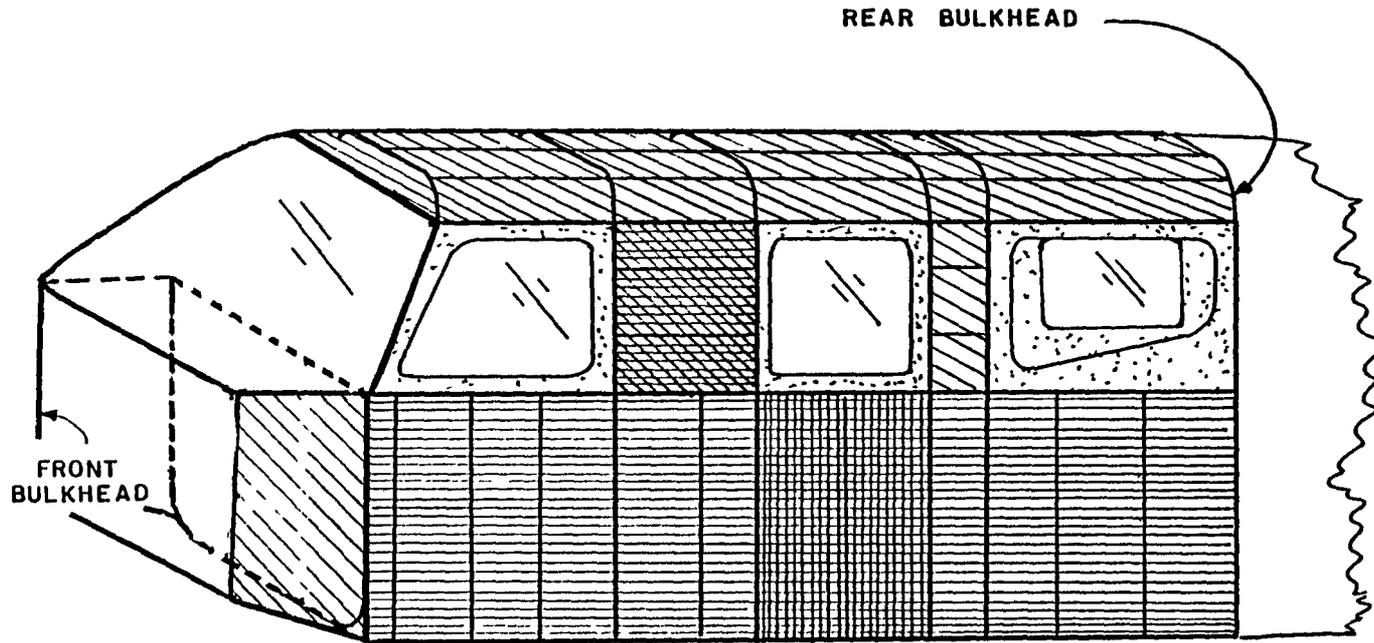
INTERIOR TREATMENT- AEROCOMMANDER

Y-370
TNB-101-1/3
(0.58 lb/ft^2)

Y-370
1"AA
1"AA
Light trim
(0.5 lb/ft^2)
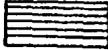
Honeycomb (0.37 lb/ft^2)
Y-370
Soundfoil
1"AA
1"AA
TNB-101-1/3
(1.3 lb/ft^2)
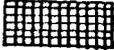
Honeycomb (1.4 lb/ft^2)
Y-370
1"AA
1"AA
Light trim
(1.95 lb/ft^2)

Honeycomb (0.37 lb/ft^2)
Y-37ᵒ
Soundfoil
1"AA
1"AA
TNB-101-1
(2.1 lb/ft^2)

Fig. 77 The final configuration of add-on treatment for the AeroCommander aircraft

APPENDIX A

Elements of Transfer Matrices

The field and point matrices for the elastic stiffened panel from Ref. 16 are as follows:

(1) Point transfer matrix [G] is given for the stiffener shown in Fig. 16 by

$$[G] = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ d' & c' & 1 & 0 \\ -e' & -d' & 0 & 0 \end{bmatrix} \tag{A-1}$$

$$c' = E_s C_{ws} \left(\frac{n\pi}{L_x}\right)^4 + G_s C \left(\frac{n\pi}{L_x}\right)^2 - \rho_s I_s \omega^2 \tag{A-2a}$$

$$d' = E_s I_{\eta\xi} s_z \left(\frac{n\pi}{L_x}\right)^4 - \rho_s A c_y \omega^2 \tag{A-2b}$$

$$e' = E_s I_\eta \left(\frac{n\pi}{L_x}\right)^4 - \rho_s A \omega^2 \tag{A-2c}$$

where E_s, G_s and ρ_s are the modulus of elasticity, shear modulus, and mass density of the stringer, respectively. The warping constant with respect to the shear center C_{ws} is given by

$$C_{ws} = C + s_z^2 I_\xi \tag{A-3}$$

where C = St. Venant's constant of uniform torsion about the centroid, s_z is the perpendicular distance between the panel and an axis parallel to the η centroidal axis of the stiffener and passing through the shear center, I_ξ

is the centroidal moment of inertia of the stiffener about the ξ centroidal axis of the stiffener and normal to the panel. The transformed inertia term I_s is given by

$$I_s = I_c + A[c_y^2 + (c_z - s_z)^2] \tag{A-4}$$

where I_c is the polar moment of inertia of the stiffener about the centroid, A is the area of the stiffener, c_y is the perpendicular distance between the shear center and ξ centroidal axis and c_z is the perpendicular distance between the shear center and η centroidal axis of the stiffener. $I_{\eta\xi}$ is the product of inertia of the stiffener.

(2) Field matrix $[F] = [B][R]([B]^{-1})$ where

$$[B] = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ -D(\frac{n\pi}{L_x})^2 \nu & 0 & D & 0 \\ 0 & -D(2-\nu)(\frac{n\pi}{L_x})^2 & 0 & D \end{bmatrix} \tag{A-5}$$

$$([B])^{-1} = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ (\frac{n\pi}{L_x})^2 \nu & 0 & 1/D & 0 \\ 0 & (2-\nu)(\frac{n\pi}{L_x})^2 & 0 & 1/D \end{bmatrix} \tag{A-6}$$

where D and ν are the bending stiffness and Poisson's ratio for the panel,

respectively.

$$[R] = \begin{bmatrix} C_0 & S_{-1} & C_{-2} & S_{-3} \\ S_1 & C_0 & S_{-1} & C_{-2} \\ C_2 & S_1 & C_0' & S_{-2}' \\ S_3 & C_2 & S_1 & C_0' \end{bmatrix} \qquad (A-7)$$

where

$$C_{-2} = (\cosh\sigma_1 y_j - \cos\sigma_2 y_j)/s^2$$

$$C_0 = (\sigma_2^2\cosh\sigma_1 y_j + \sigma_1^2\cos\sigma_2 y_j)/s^2$$

$$C_0' = (\sigma_1^2\cosh\sigma_1 y_j + \sigma_2^2\cos\sigma_2 y_j)/s^2$$

$$C_2 = \sigma_1^2\sigma_2^2(\cosh\sigma_1 y_j - \cos\sigma_2 y_j)/s^2$$

$$S_{-3} = \{(1/\sigma_1)\sinh\sigma_1 y_j - (1/\sigma_2)\sin\sigma_2 y_j\}/s^2 \qquad (A-8)$$

$$S_{-1}' = (\sigma_1\sinh\sigma_1 y_j + \sigma_2\sin\sigma_2 y_j)/s^2$$

$$S_1 = \sigma_1\sigma_2(\sigma_2\sinh\sigma_1 y_j + \sigma_2\sin\sigma_2 y_j)/s^2$$

$$S_1' = (\sigma_1^3\sinh\sigma_1 y_j - \sigma_2^3\sin\sigma_2 y_j)/s^2$$

$$S_3 = \sigma_1^2\sigma_2^2(\sigma_1\sinh\sigma_1 y_j + \sigma_2\sin\sigma_2 y_j)/s^2$$

$$s^2 = \sigma_1^2 + \sigma_2^2 \qquad (A-9)$$

$$\sigma_1^2 = \{\sqrt{\rho_p h_p / D} + (n\pi/L_x)^2\}$$

$$\sigma_2^2 = \{\sqrt{\rho_p h_p / D} - (n\pi/L_x)^2\}$$

$$(A\text{-}10)$$

List of Symbols

A	= stringer cross-sectional area
a	= cabin dimension
a_o, b_o	= distances from x and y axes, respectively, to panel location
b	= cabin dimension
b_2, b_4	= propagation constants in porous materials 2 and 4, respectively, $b = \alpha + i\beta$
C	= Saint-Venant constant of uniform torsion
C_w	= warping constant of stringer cross-section
C_{ws}	= $C_w + I_\zeta s_z^2$
c_y, c_z	= distances defined in Fig. 16
c	= speed of sound
$c_1, \ldots, c_6$	= speed of sound in regions 1, ..., 6, respectively
D	= elastic panel stiffness
D_t	= stiffness of honeycomb panel
D_x, D_y	= bending rigidities of skin-stringer panels
d	= cabin dimension
d_2	= distance between elastic panel and septum barrier
d_4	= distance between septum barrier and trim panel
E	= elastic modulus of the panel
E_s	= elastic modulus of the stringer
E_1, E_2	= elastic moduli of the facings of the honeycomb panel
$[F]$	= field transfer matrix
$[G]$	= point transfer matrix
G_s	= shear modulus of the stringer
H	= cross-rigidity of skin-stringer panel

H_{mn}	= frequency response function of the panel
h	= thickness of honeycomb core
h_p	= elastic panel thickness
I_c	= polar moment of inertia about stringer centroid
$I_\eta, I_\xi, I_{\eta\xi}$	= stringer cross-section polar moments of inertia and product of inertia, respectively, about the ξ and η axes
I_s	= transformed polar moment of inertia about point of attachment
$\underset{\sim}{i}$	= $\sqrt{-1}$
i,j,k,ℓ,m,n,r,s	= indices
K_2, K_4	= compressibility ratios in regions 2 and 4, respectively
L_{ijmn}	= acoustic-structural modal coupling terms
L_x, L_y	= longitudinal and transverse dimensions of a panel, respectively
M_{mn}	= generalized mass
M_n	= bending moment amplitude
m_s	= panel mass per unit area
NR	= noise reduction
p^r_{mn}	= generalized random forces
p	= acoustic pressure
p_0	= reference acoustic pressure
$\bar{p}$	= Fourier transform of acoustic pressure
$\bar{p}^r$	= Fourier transform of input random pressure
Q_n	= shear amplitude of panel motions
$\bar{q}_{mn}$	= generalized coordinates of the panel
$R(\omega)$	= resistance of acoustic material
R_1	= flow resistivity of the porous material
S_F	= surface of a flexible wall
S_i	= spectral density of input noise for the i-th panel

S_{mnrs} = cross-spectral densities of generalized random forces

S_p = spectral density of acoustic pressure

S_w = deflection response spectral density of the panel

S^e = cross-spectral density of pressure

S_p^e = spectral density of external pressure

S_p^i = spectral density of sound pressure transmitted by the i-th panel

SPL = sound pressure levels

SPL^T = total sound pressure levels transmitted by all panels

s_z = distance defined in Fig. 16

[T] = transfer matrix

t = time

t_{ij} = elements of transfer matrix

t_1', t_2 = thicknesses of honeycomb panel facings

V = volume of airplane cabin

V_x, V_y = convection velocities of propeller noise corresponding to direction along propeller rotation and perpendicular to it, respectively

$\{W_n\}$ = state vector

w = panel deflection

$\bar{w}$ = Fourier transform of panel deflection

$X(\omega)$ = reactance of porous material

X_{ij} = acoustic longitudinal-transverse modes

x,y,z = spatial coordinates

$\bar{x}$ = spatial separation along the x axis, $\bar{x} = x_1 - x_2$

Y_{mn} = structural modes

Y_2, Y_4 = porosity for porous materials in regions 2 and 4, respectively

y_q = local coordinate

$\bar{y}$	= spatial separation along the y axis, $\bar{y} = y_1 - y_2$
Z	= impedance
Z_A	= acoustic impedance
Z_p	= impedance of untreated panel
Z_2, Z_4	= impedance of porous acoustic blankets in regions 2 and 4, respectively
Z_3, Z_5	= septum impedances
Z_6	= termination impedance
α_2, α_4	= real parts of propagation constants b_2 and b_4, respectively
β	= acoustic damping coefficient
β_2, β_4	= imaginary parts of propagation constants b_2 and b_4, respectively
$\Delta\omega$	= frequency bandwidth
ΔTL	= additional noise transmission losses
δ_n	= deflection amplitude
ζ	= structural damping coefficient
ζ_{mn}	= structural modal damping coefficients
θ_n	= slope amplitude of panel motions
$\theta_1, \ldots, \theta_6$	= incidence angles for different media
η	= loss factor of the sidewall
μ_1, μ_3, μ_5	= surface densities of elastic panel, septum barrier, and trim panel, respectively
ν	= Poisson's ratio
ξ_o	= acoustic damping coefficient
ξ_{ij}	= acoustic modal damping coefficients
ρ	= air density
$\rho_1, \ldots, \rho_6$	= air densities for regions 1, ..., 6, respectively
ρ_p	= material density of the panel

ρ_s = material density of the stringer

τ = transmission coefficient of add-on sidewall treatment

ϕ = asimuthal angle

ω = angular frequency

ω_{ij} = acoustic modal frequencies

ω_{mn} = structural modal frequencies

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15. Supplementary Notes

Langley Technical Monitor: Dr. John S. Mixson

16. Abstract

    An analytical model based on modal analysis has been developed to predict the noise transmission into a twin-engine light aircraft. The model has been applied to optimize the interior noise to an A-weighted level of 85 dBA. To achieve the required noise attenuation, add-on treatments in the form of honeycomb panels, damping tapes, acoustic blankets, septum barriers and limp trim panels were added to the existing structure. The added weight of the noise control treatment is about 1.1% of the total gross take-off weight of the aircraft.

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