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FINAL REPORT

AIRCRAFT CABIN NOISE PREDICTION AND OPTIMIZATION

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TABLE OF CONTENTS

1. INTRODUCTION

The transmission of noise into an aircraft cabin has a direct effect on the design of General Aviation, Turboprop, Advanced Turbo-prop (ATP) and other types of aircraft. Series of theoretical and experimental studies have been undertaken by NASA to advance the understanding of the mechanism of noise transmission and to develop improved acoustic treatments for noise attenuation. The present report is a summary of the research activity of the principal investigator extending over the period from 1976 to 1985. The main objective of the proposed study was to develop analytical models capable of predicting noise transmission into aircraft. These models were used to design acoustic sidewall treatments for interior noise control in light twin-engine turboprop aircraft. This final report highlights the key achievements, summarizes doctoral thesis research activities and presents a list of publications that have resulted from the sponsored work under this grant.

2. PERSONNEL

2.1 Principal Investigator

The principal investigator of the grant during the entire funding period was Professor Rimas Vaicaitis. For the period of 1976-1977 (13 months) and 1984 (7 1/2 months) Professor Vaicaitis was on a full time assignment at NASA, Langley Research Center, ANRD, SAB.

2.2 Research Associate

Dr. M. Slazak was appointed as research associate on a full time basis in the Department of Civil Engineering and Engineering Mechanics in 1979-1980 for a period of 10 months. During the summer of 1979 (2 months) Dr. Slazak was on a full time assignment at NASA Langley Research Center, ANRD, SAB.

2.3 Graduate Students

The following graduate students were sponsored by the present grant. (Full time for part time basis)

		Present Position
1.	M. Slazak, 1977-1979, Ph.D., 1979	Senior Research Engineer at Bell Telephone Co., New Jersey
2.	M.T. Chang, 1979-1980, Ph.D., 1980	Research Engineer at Brookhaven National Laboratory, New York
3.	H.-K. Hong, 1981-1982, Ph.D., 1982 (Partial support)	Associate Professor, National University of Taiwan, Taiwan
4.	D.A. Bofilios, 1982-1985, Ph.D., 1985	Assistant Professor at San Diego State University, San Diego, California
5.	R. Eisler, 1983, Prof. Degree, 1984	Research Engineer at McDonnel Douglas Astronautics Division Los Angeles, CA
6.	L.H. Hass, 1984, M.S., 1984	Engineer at North American Rockwell, Rocketdyne Division, Los Angeles, CA
7.	J. Tarter, 1985, M.S. 1986 (expected)	

2.4 Undergraduate Students

All undergraduate students were supported on a part time basis.

		Present Position
1.	E. Hanson, B.S. 1984	Graduate School, University of Texas Department of Aeronautical Engr.
2.	C. Brawand, B.S. 1984	Graduate School, Columbia University, Department of Civil Engineering & Engr. Mechanics
3.	C. Hagan, B.S. 1984	United States Air Force

3. PUBLICATIONS

The following is a list of publications which resulted from the research under the sponsorship of the present grant.

3.1 Archive Journal Papers

1. Mixson, J.S., Barton, C.K. and Vaicaitis, R., "Investigation of Interior Noise in a Twin-Engine Light Aircraft," Journal of Aircraft, Vol. 15, No. 4, April 1978.

2. Vaicaitis, R., "Noise Transmission into a Light Aircraft," Journal of Aircraft, Vol. 17, No. 2, Feb. 1980, pp. 81-86.

3. Vaicaitis, R. and Slazak, M., "Noise Transmission through Stiffened Panels," Journal of Sound and Vibration, 70(3), pp. 413-426, 1980.

4. Chang, M.T. and Vaicaitis, R., "Noise Transmission into Semicylindrical Enclosures Through Discretely Stiffened Curved Panels," Journal of Sound and Vibration, 85(3), 1982.

5. Vaicaitis, R., "Recent Research on Noise Transmission into Aircraft," The Shock and Vibration Digest, Vol. 14, No. 8, Aug. 1982 (Review Article).

6. Slazak, M. and Vaicaitis, R., "Response of Stiffened Sandwich Panels," to appear in AIAA Journal.

7. Mixson, J.S., Roussos, L.A., Barton, C.K., Vaicaitis, R., and Slazak, M., "Laboratory Study of Add-On Treatments for Interior Noise Control in Light Aircraft," Journal of Aircraft, AIAA, Vol. 20, No. 6, June, 1983.

8. Hong, H.K. and Vaicaitis, R., "Nonlinear Response of Double

Wall Sandwich Panels," to appear in Journal of Structural Mechanics.

9. Vaicaitis, R., Grosveld, F.W. and Mixson, J.S., "Noise Transmission Through Aircraft Panels," Journal of Aircraft, Vol. 22, No. 4, April, 1985.

10. Vaicaitis, R. and Mixson, J.S., "Theoretical Design of Acoustic Treatment for Noise Control in a Turboprop Aircraft," Journal of Aircraft, Vol. 22, No. 4, April, 1985.

11. Vaicaitis, R., "Noise Transmission Into Propeller Aircraft", to appear in Shock and Vibration Digest, 1985.

3.2 Conference Papers

1. Vaicaitis, R., Slazak, M., and Chang, M.T., "Noise Transmission-Turboprop Problem," AIAA 5th Aeroacoustics Conference, Paper No. 79-0645, March 12-14, 1979, Seattle, Wash.

2. Mixson, J.S., Barton, C.K. and Vaicaitis, R., "Interior Noise Analysis and Control for Light Aircraft," SAE 1977, Business Aircraft Meeting, March 29-April 1, 1977, Wichita, Kansas.

3. Vaicaitis, R., and McDonald, W., "Noise Transmission for Light G/A Aircraft," AIAA 16th Aerospace Sciences Meeting, January 16-18, 1978, Huntsville, Alabama, Paper No. 78-197.

4. Vaicaitis, R., Ishikawa, H. and Shinozuka, M., "Dynamic Response and Failure of Window Panes," Proceedings of ASCE Specialty Conference on Probabilistic Mechanics and Structural Reliability, January 10-12, 1979, Tucson, Arizona.

5. Vaicaitis, R., Slazak, M. and Chang, M.T., "Noise Transmission and Attenuation by Stiffened Panels," AIAA 6th Aeroacoustics Conference, June 1980, Paper No. 80-1034.

6. Vaicaitis, R., Slazak, M. and Chang, M.T., "Noise Transmission and Attenuation for Business Aircraft," 1981 SAE Business Aircraft Meeting and Exposition, No., 810561, Wichita, Kansas, April, 1981.

7. Slazak, M. and Vaicaitis, R., "Response of Stiffened Sandwich Panels," AIAA/ASME/ASCE/AHS, 22nd Structures Structural Dynamics and Materials Conference, Atlanta, Ga., April 1981, Paper No. 81-057.

8. Vaicaitis, R., and Chang, M.T., "Noise Transmission into Semicylindrical Enclosures, ASME/ASCE Mechanics Conference, Paper No. EM 7.5, June 1981, Boulder, Colorado.

9. Vaicaitis, R., "Noise Transmission by Viscoelastic Sandwich Panels," 15th Midwestern Mechanics Conference, March 23-25, 1977, Chicago, Illinois.

10. Vaicaitis, R., "Noise Transmission into an Enclosure," 1977 EMD Specialty Conference, ASCE, May 23-25, Raleigh, North Carolina.

11. Vaicaitis, R., "Transmission of Wind-Induced Noise," 3rd U.S. National Conference on Wind Engineering Research, Feb. 26-March 1, 1978, Gainesville, Florida.

12. Vaicaitis, R., Slazak, M. and Chang, M.T., "Noise Transmission and Attenuation for Business Aircraft," 1981 SAE Business Aircraft Meeting and Exposition, Wichita, Kansas, 1981.

13. Vaicaitis, R., "Cabin Noise Control for Twin Engine General Aviation Aircraft," 19th Annual Meeting, Society of Engineering Science, Rolla, Missouri, October, 1982.

14. Vaicaitis, R., "Testing for Theory and Validation, SAE and NASA Aircraft Interior Noise Meeting, Langley Research Center, Hampton, Virginia, June, 1982.

15. Vaicaitis, R., and Hong, H-K., "Noise Transmission Through Nonlinear Sandwich Panels," AIAA 8th Aeroacoustics Conference, 83-0696, Atlanta, Georgia, April, 1983.

16. Vaicaitis, R., and Hong, H-K., "Nonlinear Random Response of Double Wall Sandwich Panels," 24th AIAA/ASME/ASCE/AHS Structures, Structural Dynamics, and materials Conference, Paper No. 83-1037-CP, Lake Tahoe, Nev., May 1983.

17. Vaicaitis, R., Grosveld, F.W. and Mixson, J.S., "Noise Transmission Through Aircraft Panels," 25th AIAA/ASME/ASCE/AHS SDM Conference, AIAA-84-0911, Palm Springs, CA., May, 1984.

18. Vaicaitis, R., and Mixson, J.S., "Theoretical Design of Acoustic Treatment for Cabin Noise Control of a Light Aircraft," AIAA-84-2328, AIAA/NASA 9th Aeroacoustics Conference, Williamsburg, VA., October 1984.

19. Vaicaitis, R., "Theoretical Noise Transmission Prediction and Sidewall Acoustic Treatments," SAE/NASA 2nd Aircraft Interior Noise Meeting, Hampton, Va., October 1984.

20. Vaicaitis, R., and Bofilios, D.A., "Response of Double Wall Composite Shells," 26th AIAA/ASME/ASCE/AHS SDM Conference, Paper No. 85-0604-CP, Orlando, Fl., April, 1985.

21. Vaicaitis, R., and Mixson, J.S., "Review of Research on Structureborne Noise, 26th AIAA/ASME/ASCE/AHS SDM Conference, Paper No. 85-0786-CP, Orlando, FL., April, 1985.

22. Vaicaitis, R., and Bofilios, D.A., "Noise Transmission of Double Wall Composite Shells," ASME Conference on Vibration and Sound, Cincinnati, Ohio, Sept. 1985.

3.3 Reports

1. Vaicaitis, R., "Noise Transmission by Viscoelastic Sandwich Panels," NASA TND-8516, Langley Research Center, Hampton, Virginia, January, 1978.

2. McDonald, W.P., Vaicaitis, R., and Myers, M.K., "Noise Transmission through Plates into an Enclosure," NASA Technical Paper 1173, Langley Research Center, Hampton, Virginia, January, 1978.

3. Vaicaitis, R., Bofilios D.A., and Eisler, R., "Experimental Study of Noise Transmission into a General Aviation Aircraft," NASA CR-172357, June 1984.

3.4 Invited Talks (Presented by principal investigator)

1. University of Illinois, Dept. of Aeronautical and Astronautical Engineering. "Noise Transmission into Aircraft" 1977

2. Rutgers University, Dept. of Mechanical Engineering, "Response, Flutter and Noise Transmission of Panels" 1979

3. United Technologies, Inc., Hartford, Conn. "Noise Transmission into Aircraft" 1981

4. Fourth Science and Engineering Symposium, Chicago, Ill. "Noise Optimization for Light Aircraft" 1981

5. Gulfstream American, Inc., Bethany, Okla. "Noise Reduction for Propeller Driven Aircraft" 1981

6. General Motors Research Laboratories, Warren, Mich. "Noise Transmission into Enclosures 1982

7. Army Materials and Mechanics Research Center, Watertown, MA., "Noise Transmission for Army Applications" 1983

8. Duke University, Durham, North Carolina, "Noise Transmission and Control" 1984

9. Gulfstream American, Inc., Bethany, Oklahoma, "Noise Optimization for Commander 1000 Aircraft" 1984

10. United Technologies, Inc., Hartford, Conn. "Noise Transmission and Control" 1985

4.0 DOCTORAL THESES

The following includes title, abstract, highlights and conclusions of the doctoral theses sponsored by this grant.

1. M. Slazak, "Noise Transmission Through Stiffened Panels" 1979.

ABSTRACT

An analytical study is presented to predict low frequency noise transmission through stiffened panels into cavity backed enclosures. Noise transmission is determined by solving the acoustic wave equation for the interior noise field and stiffened panel equation for vibrations of the stiffened panel. The dynamic behavior of the panel is determined by the transfer matrix procedure. Also presented is a transfer matrix development for stiffened sandwich panels. Results include comparisons between theory and experiment, noise transmission through the sidewall of an aircraft.

HIGHLIGHTS

The problem geometry of the rectangular acoustic enclosure and a discretely stiffened panel are shown in Figs 1 and 2. The stiffened panel is taken to be flat and located at $z = 0$, $x = a_0$ and $y=b_0$. To evaluate the validity of noise transmission model for discretely stiffened panels, a comparison is made between theory and experiment. A quantitive comparison of interior noise, measured and calculated is shown in Fig. 3 for a simple rectangular enclosure and a stiffened aluminum panel. The input was generated by applying random white noise 100dB acoustic waves on the panel. The ζ_0 and ξ_0 are the modal damping coefficients corresponding to structural and acoustic fundamental modes. As can be observed from these results, the agreement between theory and experiment is good.

To illustrate the effect on noise transmission by structural models which account for interaction of stiffeners and panels, results were obtained for stiffened panels and equivalent panels in which the stiffeners are either assumed to be rigid (individual panel assumption) or smeared (orthotropic panel assumption). These results are shown in Figs. 4 and 5. The results presented indicate that significant errors might be introduced if the dynamic effects of discrete stiffening are not included.

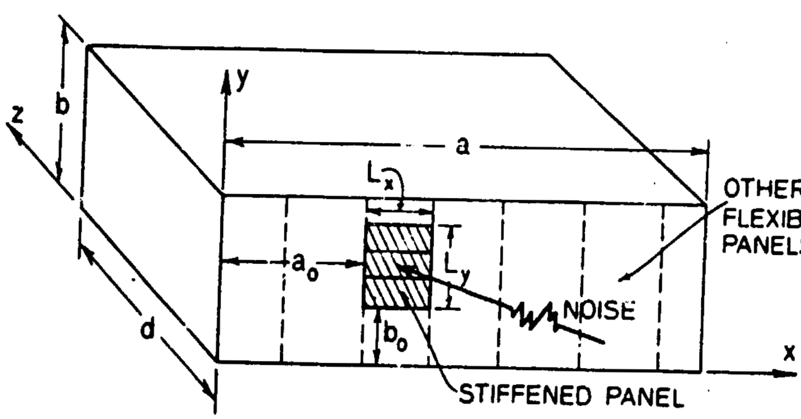


Fig. 1. Geometry of Rectangular Cavity Noise Model

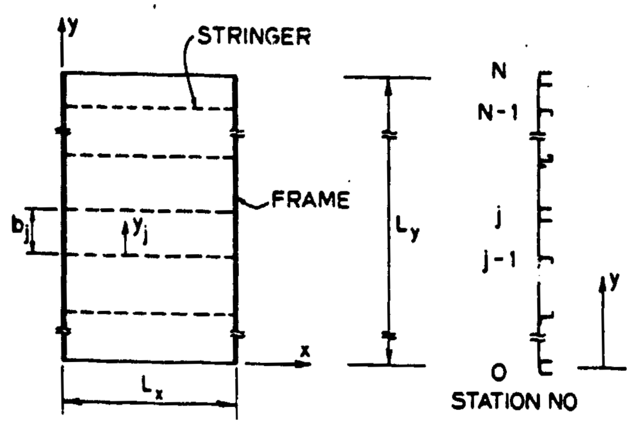
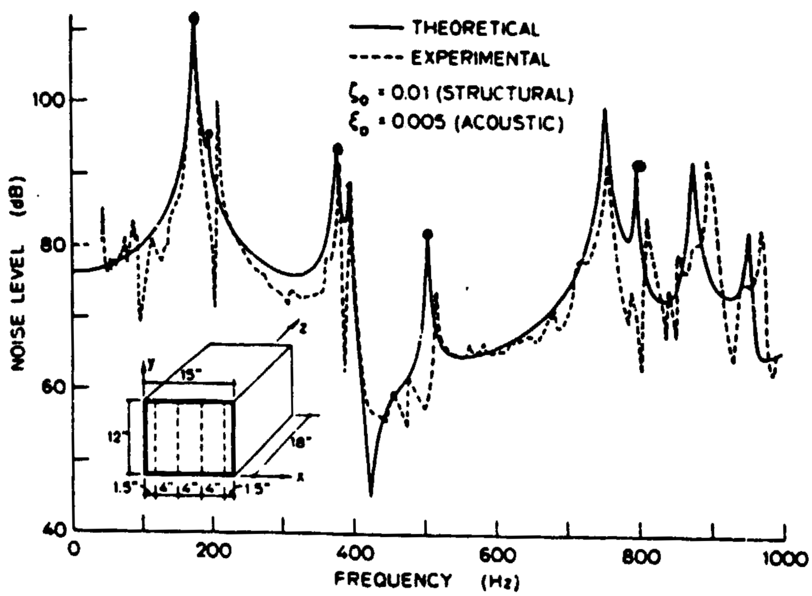


Fig. 2 Skin-Stringer Panel



(• denotes structural modes)

Fig. 3 Comparison of Measured and Calculated Interior Noise

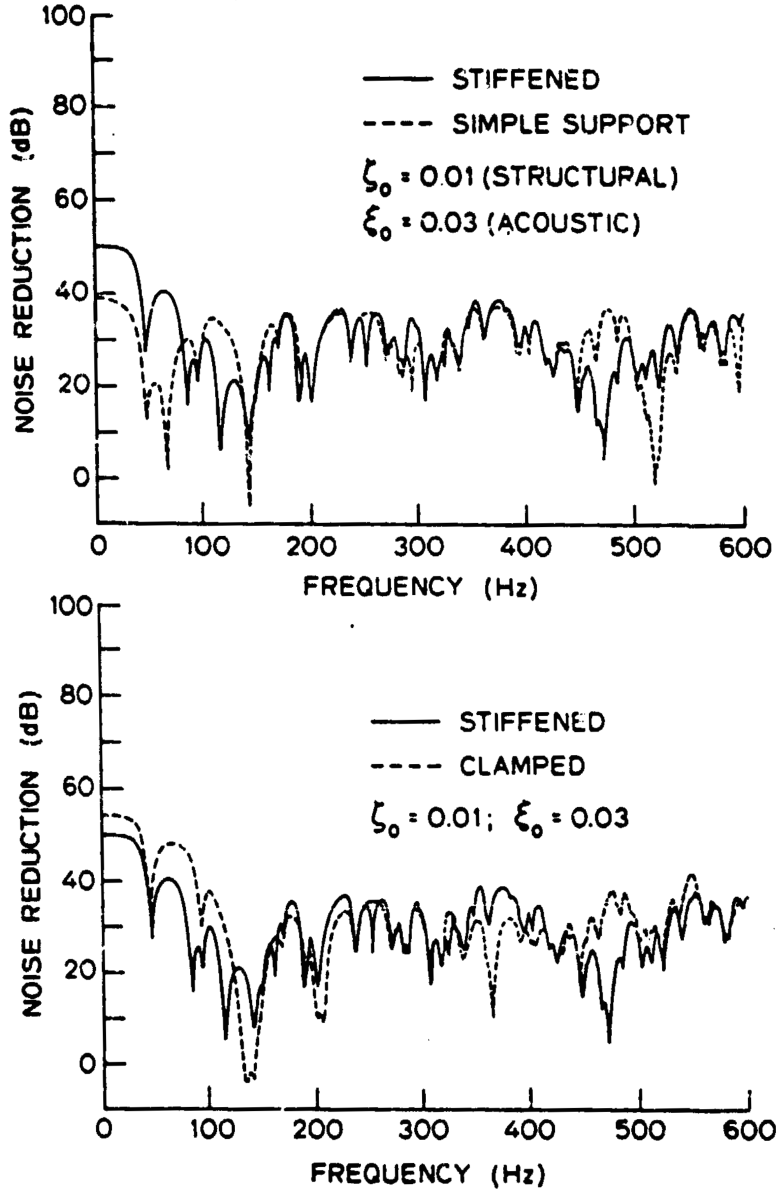


Fig. 4. Comparison of Noise Reduction between a Stiffened Panel and Individual Panel Model

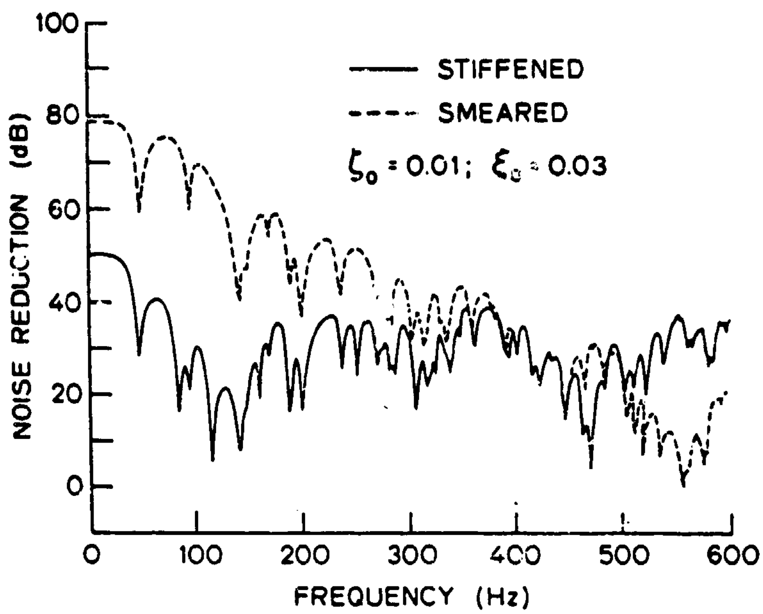


Fig. 5. Comparison of Noise Reduction between a Stiffened Panel Model and Smeared Orthotropic Model

CONCLUSIONS

Analytical studies on low frequency (up to 600 Hz) noise transmission through finite discretely stiffened panels are presented. Good agreement between theory and experiment have been obtained. Numerical procedures have been improved to determine natural frequencies and normal modes of skin-stringer panels for frequencies ranging from 0 to 600 Hz. The results indicate that neglecting the effect of discrete elastic stiffening could introduce significant errors when computing noise transmission through skin-stringer panels. To accurately evaluate the effect of low frequency attenuation in interior noise when adding damping, stiffening, or mass treatments to the vibrating sidewall, the dynamic interaction between panel and stiffeners should be included.

Transfer matrices for stiffened sandwich panels have been developed. It is demonstrated that for thin samdwich panels the sixth order equations can be reduced to a fourth order system with only minor effect on the numerical results considered. However, even thin sandwich panels could provide a significant amount of energy dissipation due to damping in the core. It is shown that sandwich panels, while providing the same stiffening benefits as equivalent elastic panels, can thus significantly reduce interior noise levels.

2. M.-T. Chang, "Noise Transmission Through Stiffened Curved Panels", 1981

ABSTRACT

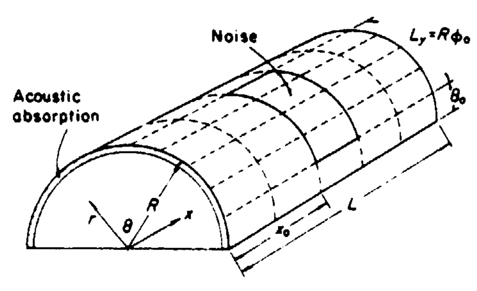
Presented here is the theory and solution of sound transmission into a semicylindrical enclosure through an elastic discretely stiffened curved panel. The transmitted sound is calculated by solving the coupled acoustic-structural equations using a modal analysis. The modes and the natural frequencies of the curved stiffened panels are estimated by the finite element-strip method while the acoustic modes are those of a semicylindrical enclosure. Numerical results include spectra of the interior sound pressure due

to white noise, turbulent boundary layer and propeller noise inputs.

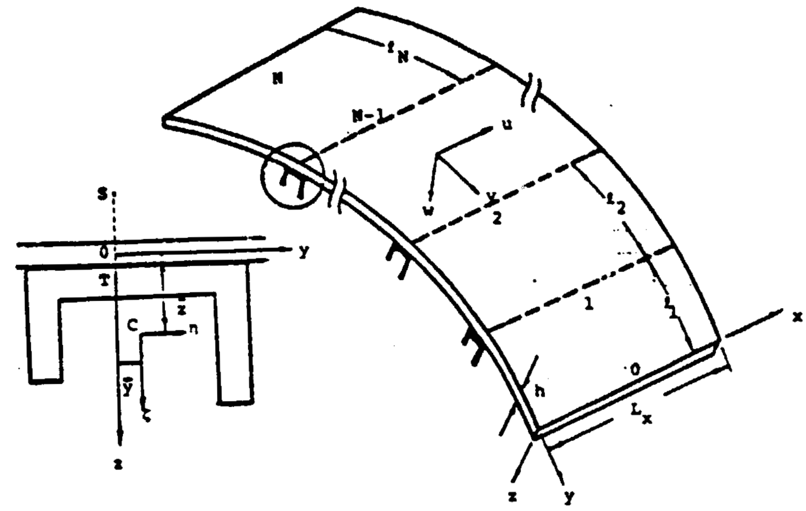
HIGHLIGHTS

The geometry of a semi-cylindrical acoustic enclosure and of a curved and discretely stiffened panel are shown in Fig. 6. The acoustic space occupies a volume $V = \pi RL$ and is surrounded by surface S of which the portion S_F is flexible, while the remaining surfaces are rigid. The curved panel shown in Fig. 6 is stiffened by two discrete stringers spaced at equal intervals and it is assumed to be simply supported around the edges. To show the effect of discrete stiffening on noise transmission, comparisons of interior sound pressure levels (SPL) are made between the curved stiffened panels, equivalent curved panels in which the stringers are assumed to be rigid, and "smeared" curved panel where the dynamic properties of stringers are "smeared" into an equivalent skin. In Fig. 7, results are shown for a discretely stiffened panel and single panels (three panels) which are taken to be simply supported on all edges. Since the stringers are assumed to be rigid, the coupling effects between adjacent panels are removed and each panel vibrates independently. Since all these individual panels are identical, structural resonances of all panels occur at the same frequency. However, a larger number of modes are excited for stiffened panels over the selected frequency range. Since the stringers are allowed to rotate and deflect, stronger coupling is observed between the acoustic modes and structural motions resulting in higher interior SPL when compared with the noise transmitted by single panels. These results indicate that significant errors might be introduced if the elastic stringers are replaced with rigid type supports.

The interior SPL for a discretely stiffened panel and an unstiffened curved single panel are given in Fig. 8. The geometric and material properties of the unstiffened panel are taken to be the same of the stiffened panel but with the stringers removed. The dynamic behavior of such a panel can represent a "smeared" model where the dynamic properties of stringers are averaged out into an equivalent skin. The results shown in Fig. 8 indicate that such an idealization could lead to significant differences in noise transmission between discretely stiffened panels and unstiffened panels. In view of the results presented, noise transmission through localized panel units should account for the actual modal shapes and the corresponding natural frequencies of discretely stiffened panels.



a) Semicylindrical Acoustic Enclosure



b) Geometry of Curved and Discretely Stiffened Panel

Fig. 6. Semicylindrical Acoustic Space and Panel Geometry

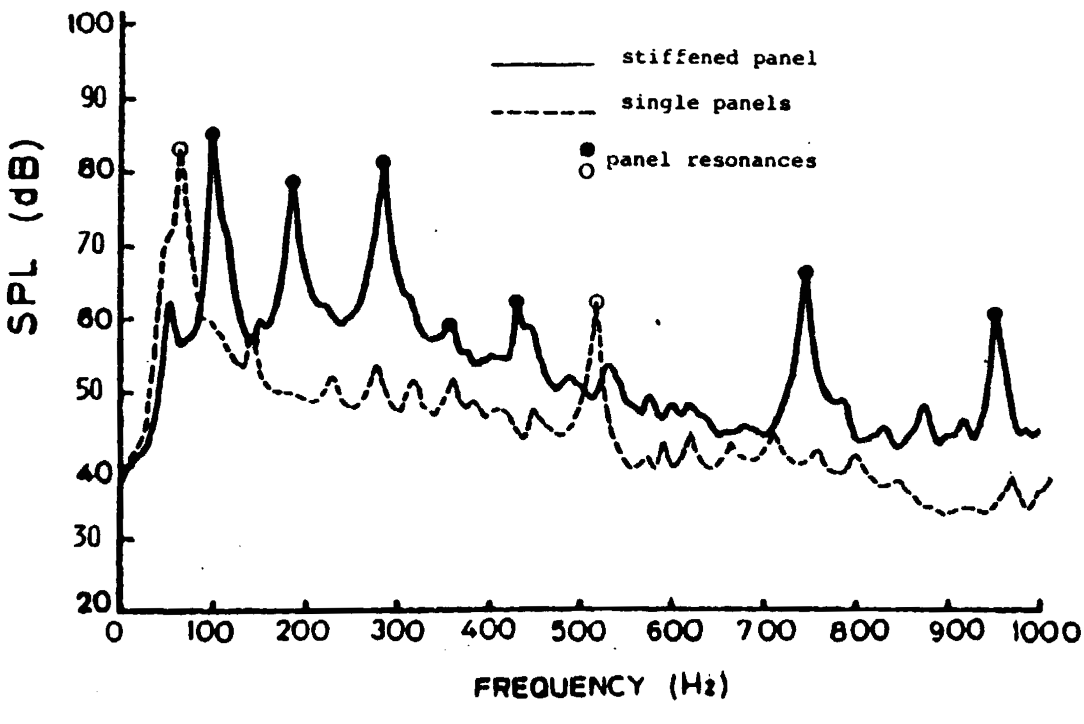


Fig. 7. Transmitted Noise Through a Stiffened Panel and a Single Panel (R = 78.74 in.)

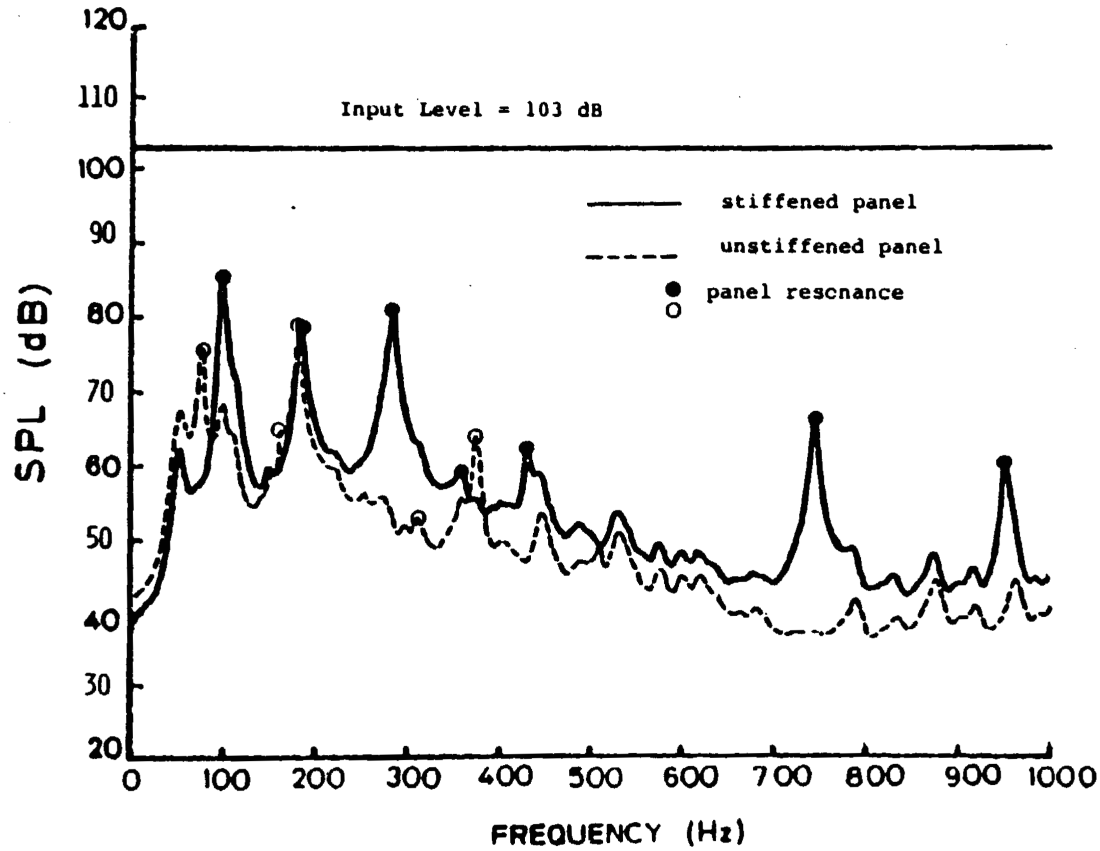


Fig. 8. Transmitted Noise for a Panel With and Without Discrete Stiffening (R = 78.74)

CONCLUSIONS

Analytical studies of noise transmission through single panels and curved discretely stiffened panels into a semicylindrical enclosure are presented. Procedures for estimating the modes and frequencies of curved skin-stringer panels are given utilizing the finite element-strip method. Good agreement between the transfer matrix approaches and the finite element strip method was reached when calculating the modes and frequencies. The coupling for a vibrating panel due to the aerodynamic surface flow and the back-up cavity pressure was investigated. It was found that their effects are negligible on sound transmission and thus can be eliminated from formulation. Pressurization of the cabin tends to increase the stiffness of the vibrating panels, which can alter the noise transmission characteristics in the low frequency region.

When computing noise transmission through curved skin-stringer panels, the results indicate that neglecting the effect of discrete stiffening can introduce significant errors. For lower frequencies, the noise transmission is also sensitive to the radius of the semicylindrical enclosure. When the input is propeller noise, the interior noise is dominated by distinct peaks due to the blade passage harmonics. The acoustic absorption at the interior walls can also have a positive effect on noise attenuation by suppressing the acoustic resonances.

3. H.-K. Hong, "Nonlinear Response and Noise Transmission of Double Wall Sandwich Panels," 1982

ABSTRACT

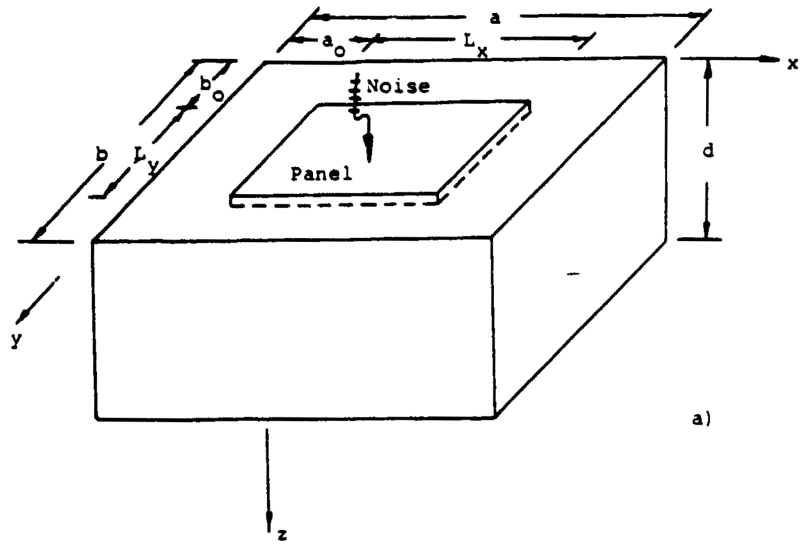
An analytical study is presented to predict the non-linear response of a double wall sandwich panel system subjected to random type loading. Viscoelastic and non-linear spring-dashpot models are chosen to characterize the behavior of the core. The noise transmission through this panel system into an acoustic enclosure of which the interiors are covered with porous absorption materials is deter-

mined. The absorbent boundary conditions of the enclosure are accounted for by a two-step transformation of the boundary effect into a wave equation which governs the acoustic pressure field inside the enclosure. The nonlinear panel response and interior acoustic pressure are obtained by utilizing modal analyses and Monte Carlo simulation techniques. Numerical results include the response spectral densities, root mean square responses, probability density function histograms, crossing rates, and noise reduction. It is found that by proper selection of the dynamic parameters and damping characteristics, the structural response and noise transmission can be significantly reduced.

HIGHLIGHTS

The results presented herein correspond to a rectangular acoustic enclosure and a double sandwich panel system shown in Fig. 9. It is assumed that noise enters only through the double wall panel. The top plate of the double wall construction is exposed to a uniformly distributed stationary and Gaussian random pressure. The large deflection theory allows for nonlinear deformation of the two aluminum face plates. The core material is taken to be relatively soft so that dilatational motions can be included. A nonlinear stiffness and nonlinear damping model is used for the core. The dimensions of the double wall panel and the enclosure are L_x = 20 in., L_y = 10 in., a = 72 in., b = 52 in., d = 50 in., h_s = 1 in. The panel is located at z = 0, a_0 = 26 in., and b_0 = 21 in. The structural response is computed at the center of the plate and the transmitted noise is calculated at x = 36 in., y = 26 in., and z = 10 in.

Shown in Fig. 10 are segments of the simulated time history of the input random pressure and the deflection response time histories of the top and bottom plates. The response of the bottom plate is linear (h_B = 0.064 in.) while the response of the tope plate is nonlinear (h_T = 0.032 in.). The noise reduction for different top plate thicknesses is shown in Fig. 11. For the calculation of noise transmission into the enclosure, it was assumed that a layer (1in.) of absorption material was applied uniformly on all the interior surfaces of the six walls. The results presented in Fig. 11 illustrate significant differences of noise transmission for linear and nonlinear panel motions. For h_T = 0.064 in., the top panel response is linear and distinct resonant peaks are observed at the natural frequencies of the double wall system. However, when the response reaches the nonlinear range (h_T = 0.032 in. and h_T = 0.016 in.), the resonant peaks are suppressed above the fundamental mode. The resonant frequencies are now functions of the deflection amplitude. It should be noted that by decreasing the thickness of the top plate, the surface density of the double wall construction is reduced. Thus, favorable gains in noise reduction can be achieved for a smaller amount of added weight for a design consisting of a thin top plate (nonlinear response) and a thicker bottom plate.



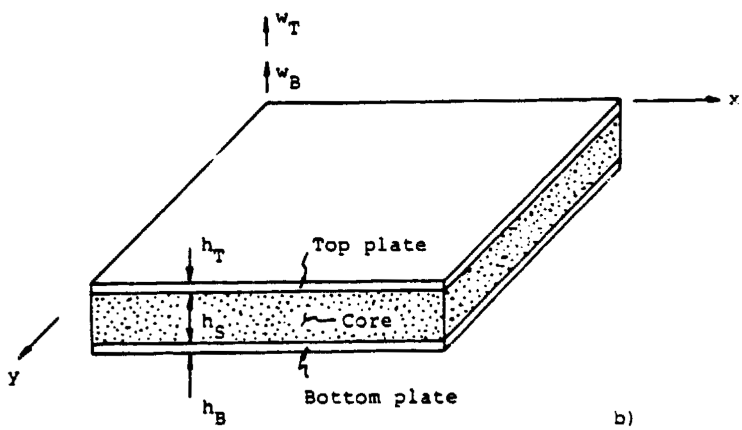


Fig. 9. Geometries of a) Acoustic Enclosure and b) Double Wall Sandwich Panel System

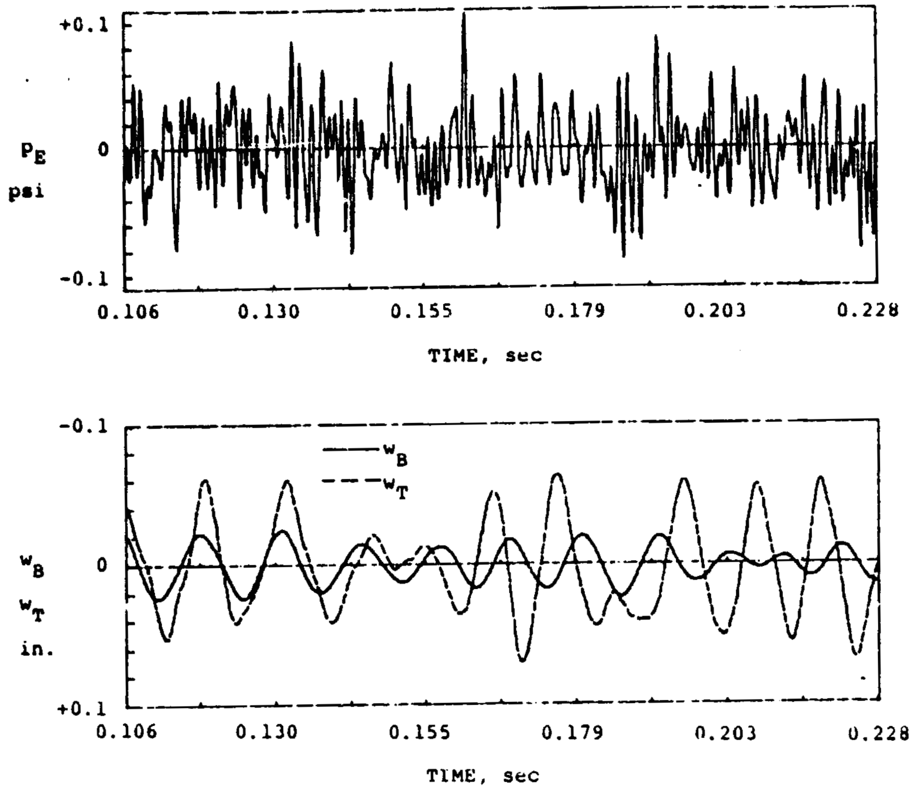


Fig. 10. Time Histories of Simulated Input and Plate Responses

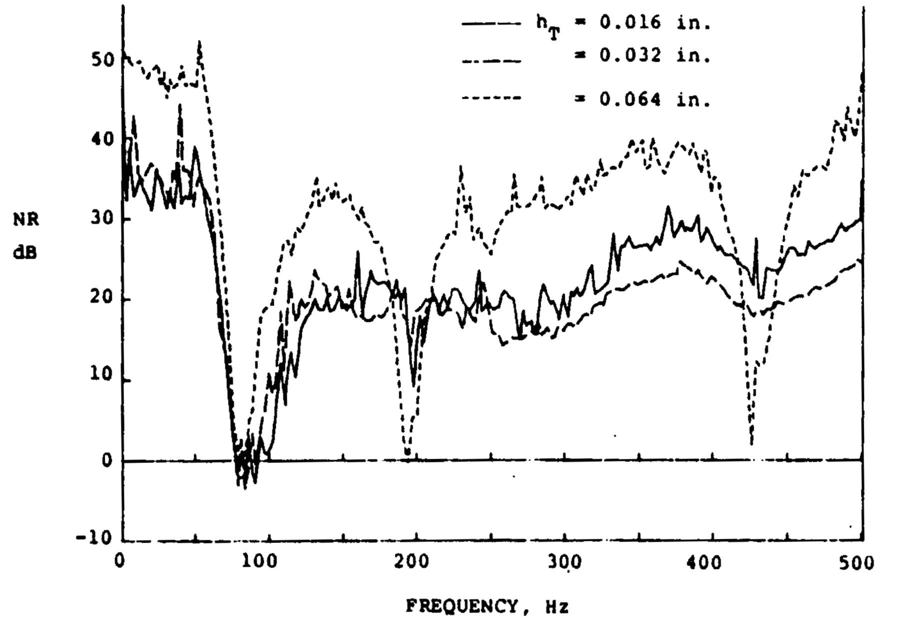


Fig. 11. Noise Reduction for Different Top Plate Thicknesses

CONCLUSIONS

An analytical model has been developed to predict the nonlinear response of a double wall sandwich panel system subjected to random type loading. The noise transmission through this coupled panel system into an enclosure of which the interiors are covered with porous absorption materials has been obtained. The results indicate that the response and noise transmission is strongly dependent on the geometric and material properties of the double wall sandwich construction. The deflection response of the bottom plate and, in turn, the noise transmission into the acoustic enclosure can be controlled by the proper selection of the core stiffnes and top plate thickness. For a soft core and nonlinear response of the top plate, no distinct resonance peaks were observed at frequencies above the fundamental resonance frequency of the coupled system. By increasing damping at the core, the response peak at the fundamental frequency can also be suppressed. This suggests that the viscoelastic damping factor of the core should be large in the vicinity of the fundamental natural frequency of the coupled system.

A better noise reduction is achieved for a core with a light mass than for a core with a heavy mass. Such an observation is contradictory to that of a single panel construction where more added weight is considered to be beneficial to noise attenuation. However, for a double wall construction, inertia coupling between the top and bottom plates is introduced through the core. A heavier core tends to induce stronger coupling and larger response of the bottom plate.

The effect of nonlinear stiffness and nonlinear damping of the soft core on noise reduction are in general favorable, but the contributions are not very large. The absorption materials can be used very effectively to suppress acoustic resonances. Since the added damping and mass of these materials to the flexible panels are very small, their effect on the structural response is negligible.

4. D.A. Bofilios, "Response and Noise Transmission of Double Wall Circular Plates and Laminated Composite Cylindrical Shells," 1985.

ABSTRACT

An analytical study is presented to predict the response and noise transmission of double wall circular plates and double wall laminated composite fiber reinforced cylindrical shells to random loads. The core of the double wall construction is taken to be soft so that dilatational motions can be modeled. The analysis of laminated shells is simplified by introducing assumptions similar to those in the Donnell-Mushtari theory for isotropic shells. The theoretical solutions of the governing acoustic-structural equations are obtained using modal decomposition and a Galerkin-like procedure. Numerical results include modal frequencies, deflection response spectral densities and interior sound pressure levels. From the parametric study it was found

that by proper selection of dynamic parameters, viscoelastic core characteristics and fiber reinforcement orientation, vibration response can be reduced and specific needs of noise attenuation can be achieved.

HIGHLIGHTS

The numerical results presented herein correspond to the double wall sandwich shell and circula. plate system shown in Fig. 12. The inputs to this system are either uniformly distributed random pressure or random point loads as presented in Fig. 13. The following set of parameters are selected for the study: The dimensions of the double wall shell are L = 25 ft., R = 58 in., h_s = 2 in. The shell response is computed at x = L/2 and θ = 45°. The thicknesses of the external and the internal shells are h_E = 0.032 in. and h_I = 0.1 in. The stiffness and material density of the core are k_3 = 4.17 lb_f/in^3 and $\rho_s = 3.4 \times 10^{-6}$ $lb_f - sec^2/in^4$. The outer shell consists of three laminae while the inner shell is composed of ten laminae. Fiberglass and graphite fibers are used to reinforce the Plexiglass material. The ratio of fibers volume to the Plexiglass volume is 0.2. The fiber orientation is prescribed by angle α (Fig. 12). The elastic moduli, Poisson's ratios and material densities are $E_f = 7.75 \times 10^6$ psi $\nu_f = 0.33$, $\rho_f = 0.0002$ $lb_f - sec^2/in^4$, $E_g = 10.5 \times 10^7$ psi $\nu_g = 0.33$, $\rho_g = 0.00015$ $lb_f - sec^2/in^4$, $E_p = 2.35 \times 10^5$ psi, $\nu_p = 0.35$, $\rho_p = 0.00011$ $\frac{lb_t - sec^2}{in^4}$ where f,g,p represent fiberglass, graphite and plexiglass, respectively. The fiber reinforcement (same pattern is used for internal and external shell) is arranged as follows: 1st layer fiberglass, 2nd layer graphite, 3rd layer fiberglass, and so on. For the aluminum shell, $E_a = 10.5 \times 10^6$ psi, $\nu_a = 0.30$, $\rho_a = 0.000254\ lb_f - sec^2/in^4$.

The viscous damping coefficients C_E and C_I are expressed in terms of modal damping ratios ζ_{mn}^E and ζ_{mn}^I corresponding to the external and internal shells respectively. Numerical results are obtained for constant values of modal damping. Damping in the soft core in introduced through the loss factor g_s for which values ranging from 0.02 to 0.1 are selected.

The input random pressures p^e, p^i, and points loads $F_j{}^e, F_j{}^i$ $(j = 1,2)$ are assumed to be characterized by truncated Gaussian white noise spectral densities

$$S_{p^e,\,p^i} = \begin{cases} 8.41 \times 10^{-5}\ (psi)^2/Hz & 0 < f < 1000\ Hz \\ 0 & \text{otherwise} \end{cases} \tag{1}$$

$$S^{e,i}_{F_1,F_2} = \begin{cases} 0.84\quad lb_f^2/Hz & 0 < f < 1000\ Hz \\ 0 & \text{otherwise} \end{cases} \tag{2}$$

The spectral densities given in Eq. 1 correspond to a 130 dB sound level. The random point loads were located at $x^e = x_1{}^e =$ 12.5 ft., $x_1{}^i = x_2{}^i = 12.5$ ft., $\theta_1{}^e = -90°$, $\theta_2{}^e = 90°$, $\theta_1{}^i = -90°$, $\theta_2{}^i = 90°$.

The dimensions of the double wall plates located at x = 0, L are taken to be $R^p = 58$ in., $h_s^p = h_s$. The thicknesses of the inner and outer plates are $h_T^p = h_B^p = 0.25$ in. The stiffness and material density of the core are the same as in the shell system, i.e., $k_s = 4.17\ lb_f/in^3$ and $\rho_s = 3.4 \times 10^{-6}\ lb_f - sec^2/in^4$. Both end plate systems are composed of aluminum, with elastic moduli $E_T = E_B = 10.5 \times 10^6\ lb_f/in^2$, Poisson's ratios $\nu_T = \nu_B = 0.3$ and material densities $\rho_T = \rho_B = 0.000259\ lb_f - sec^2/in^4$. The viscous damping coefficients C_T and C_B are expressed in terms of modal damping ratios ζ^T_{sq} and ζ^B_{sq} corresponding to the outer (top) and inner (bottom) plates respectively. The loss factor accounting for damping in the soft core of the double wall plate systems is taken to be $g_s = 0.02$. The input random pressures p^T, p^B, and the point loads P_j^T, P_j^B $(j = 1,2)$ was assumed to be characterized by truncated Gaussian white noise spectral densities.

$$S_{p^T, p^B} = \begin{cases} 8.41 \times 10^{-7}\ (psi)^2/Hz & 0 < f < 1000\ Hz \\ 0 & \text{otherwise} \end{cases} \tag{3}$$

$$S_{P_j^T,\ P_j^B} = \begin{cases} 0.84\ \ lb_f^{\,2}/Hz & 0 < f < 1000\ Hz \\ 0 & \text{otherwise} \end{cases} \tag{4}$$

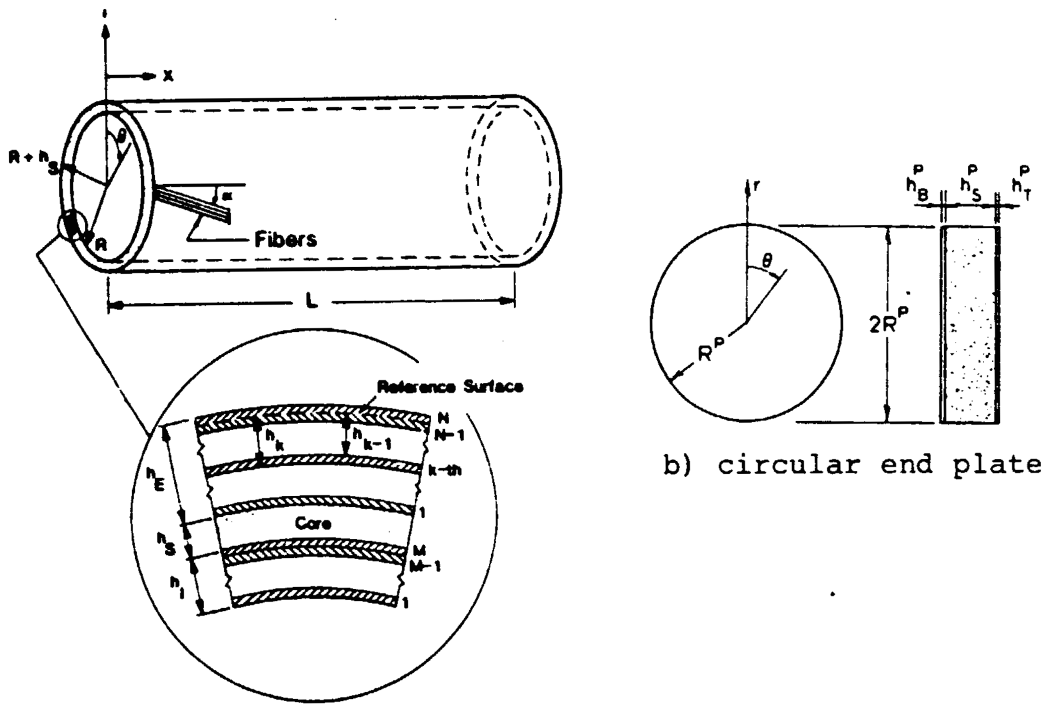
Numerical results are presented for noise transmitted and noise generated by vibrations of the cylindrical shell and circular plate systems. However, the vibrations and noise transmission of the shell and circular plates are assumed to be independent. Then, the total transmitted noise into the enclosure by the shell and the end plates can be obtained by superposition of the individual contributions. The sound pressure levels generated by an aluminum and fiber reinforced laminated shells due to point load ac on are given in Fig. 14. As can be observed from these results, the noise levels generated by a composite shell are higher at most frequencies than the noise levels of an aluminum shell. The mass of the composite shell is about one-half of the mass of the aluminum shell. However, the composite shell is much stiffer than aluminum shell. The effect on noise transmission due to fiber orientation is illustrated in Fig. 15. The fiber orientation of the three layers (Fig. 12) of the exterior shells is described in Fig. 15. The fiber orientation for the ten layers of the interior shell are: (A) 0°, 22.5°, 45°, 45°, 22.5°, 0°, 90°, 90°, 90°,90°, (B) 90°, 0°, 90°, 0°, 90°, 0°, 90°, 0°, 90°, 0°, (C) -45°, 45°, -45°, 45°, -45°, 45°, -45°, 45°, -45°, 45°. These results indicate that noise transmission through a composite shell is a function of reinforcing fiber orientation. The interior noise levels might be tailored to meet specific needs by selecting a suitable fiber orientation of a multilayered shell. However, the transmitted noise is a function of frequency and only specific frequency bands might be affected.

The interior sound pressure levels at $x = L/2$, $r = 23$ in., and $\theta = 45°$ due to noise transmitted by fiber reinforced composite shell and double wall end plate located at $x = L$ are shown in Figs. 16 and 17. In both of these cases, the end plate located at $x = 0$ is assumed to be rigid. As can be seen from these results, the noise transmitted by the end plate is primarily low frequency (below 200 Hz) while noise above 200 Hz is dominate by the shell motions.

CONCLUSIONS

An analytical model has been developed to predict vibration response and noise transmission of double wall circular plates and double wall laminated composite shells to random inputs. Results indicate that the shell response is strongly dependent on damping characteristics of the shell material and the core, location of the point load action, and reinforcing fiber orientation of the different laminae. In general, the response levels for a composite double wall shell are lower at most frequencies than those of an equivalent aluminum construction. The vibration response of the end caps (circular plates) are predominately low frequency with the largest peak occuring at the fundamental mode.

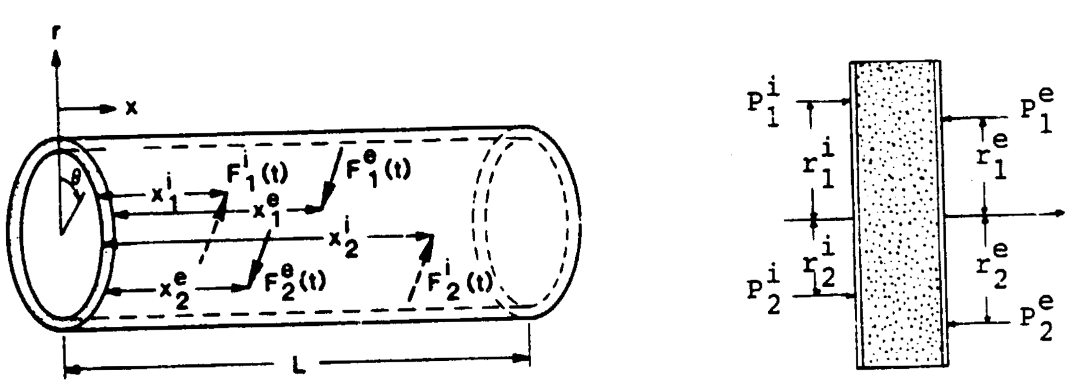
The interior noise is strongly dependent on damping characteristics of the shell and the core, location of the point load action, fiber orientation of the different laminae and wall absorption of the interior walls. A fiber reinforced composite double wall shell tends to generate more noise than an equivalent aluminum shell. This is due to the fact that the mass of the composite shell is about one half of the mass of the aluminum shell and increase of the modal frequencies of the stiffer composite shell could induce different coupling of structural-acoustic modes. The noise transmitted by the end caps is predominantly low frequency. Thus, neglecting noise transmitted by the end caps could underestimate interior sound pressure levels for the low frequency region. Furthermore, by a proper selection of structural damping, reinforcing fiber orientation, acoustic absorption and core stiffness, a significant amount of lower response and higher noise attenuation can be achieved by a design consisting of double wall laminated fiber reinforced composite shells and a soft viscoelastic core.



a) composite shell

b) circular end plates

Fig. 12 Geometry of double wall shell and end plate system



a) shell

b) circular plates

Fig. 13 Random point loads

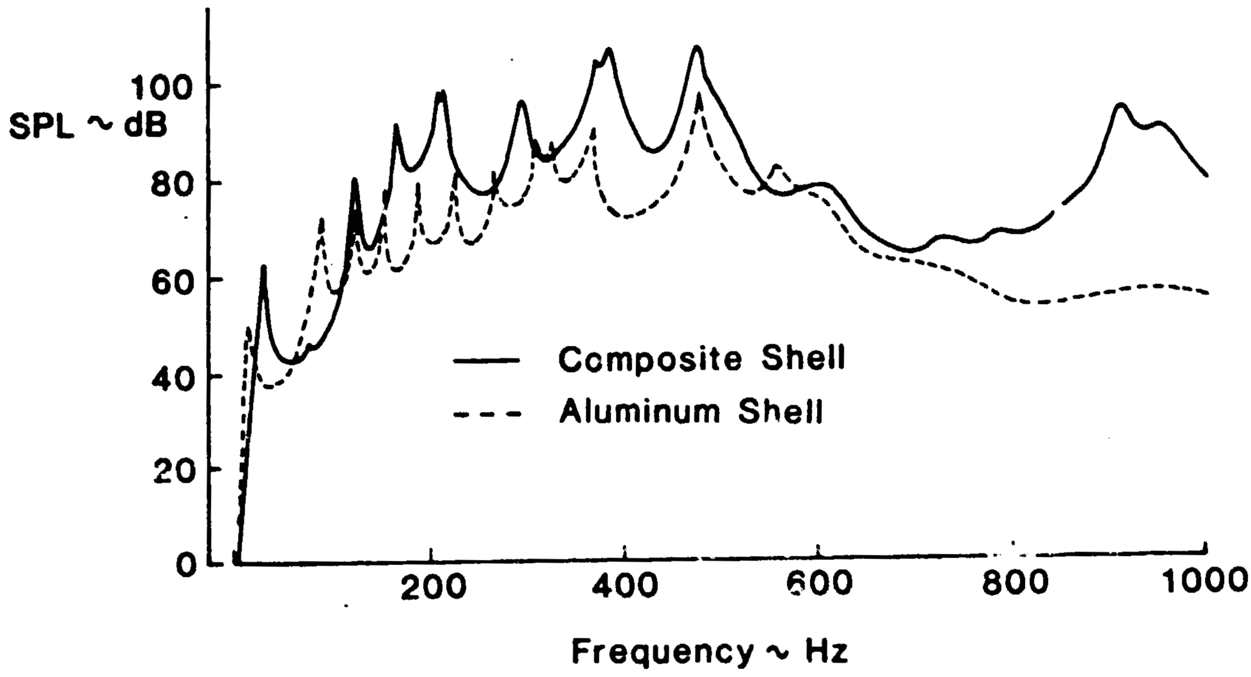


Fig. 14. Sound pressure levels for aluminum and composite shells (exterior point loads)

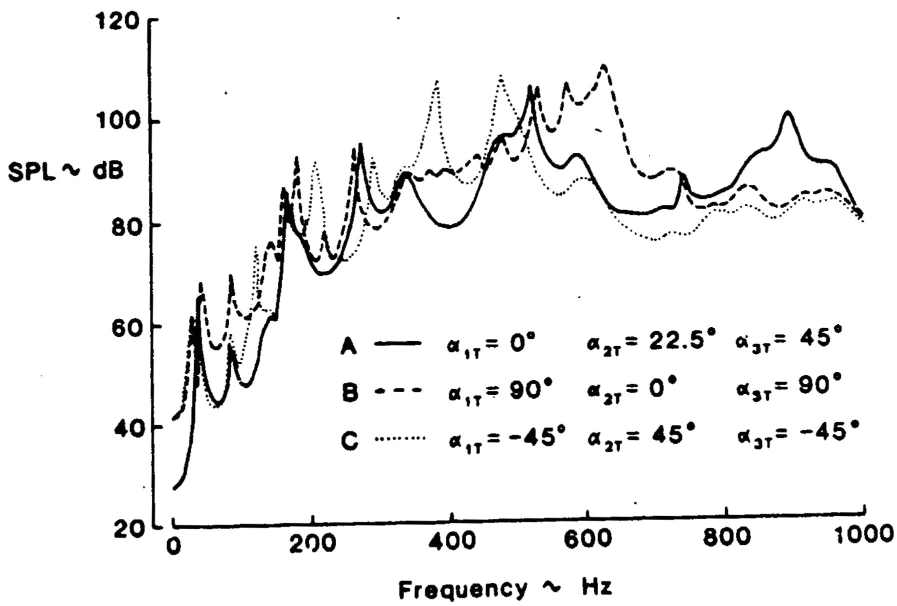


Fig. 15. Sound pressure levels in a composite shell for different fiber orientations (interior point loads).

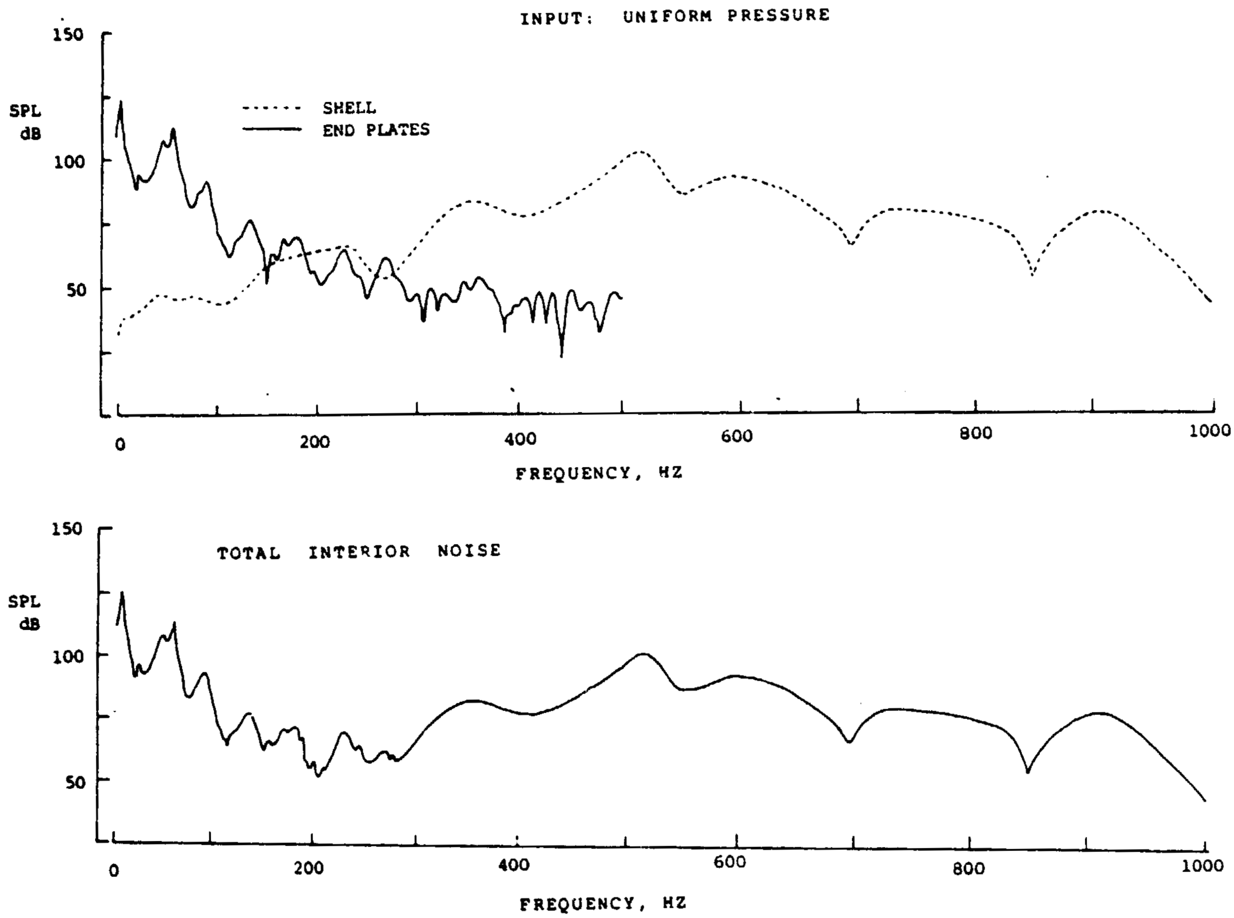


Fig. 16 Sound pressure levels due to noise transmitted by the shell and end plates (uniform pressure input)

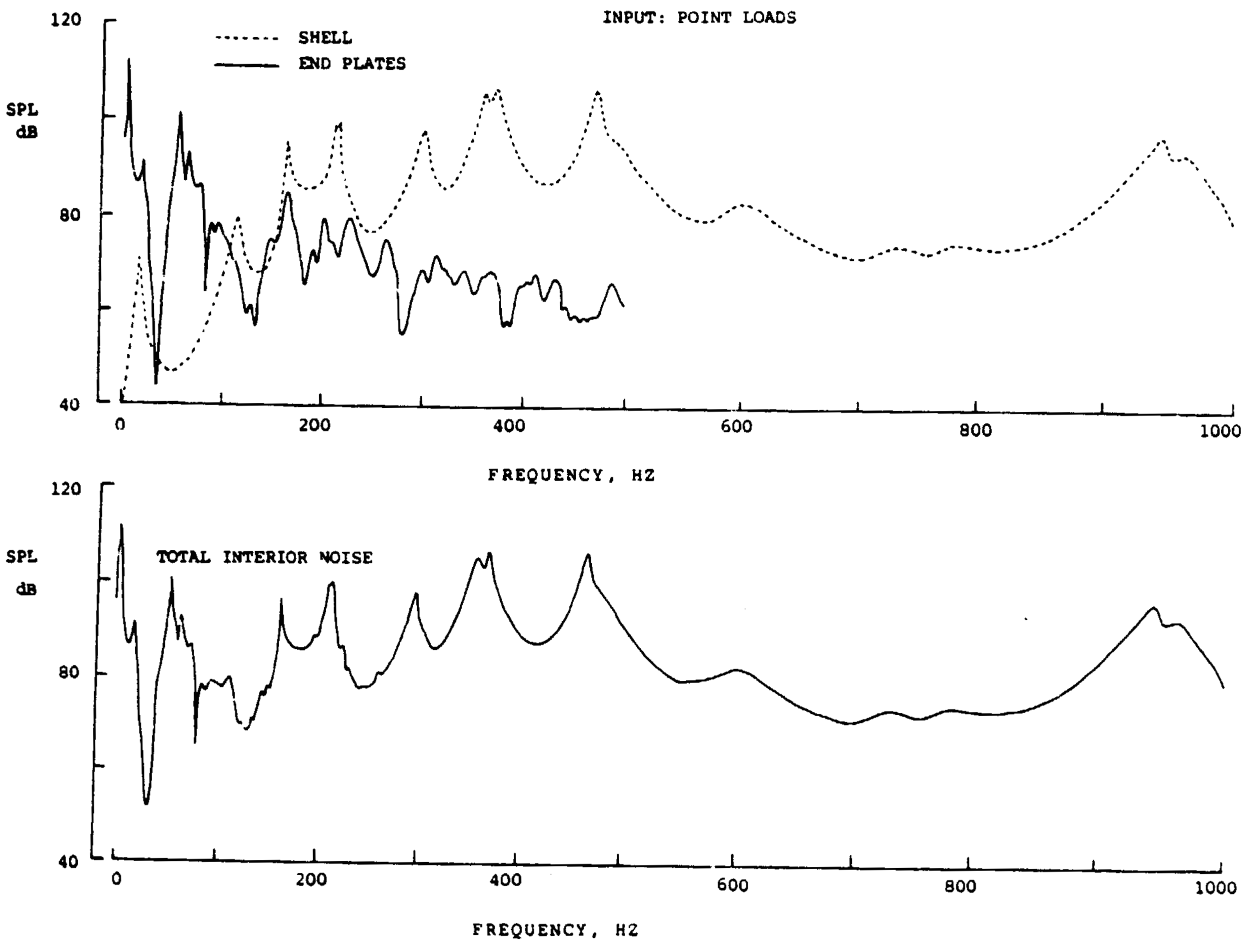


Fig. 17 Sound pressure levels due to noise transmitted by the shell and end plates (point load inputs)

5.0 RESEARCH HIGHLIGHTS

The following is a brief description of research highlights of the work sponsored by this grant from 1976 to 1985. A more detailed account of other accomplishments can be found in the numerous technical articles listed in Sec. 3.

5.1 Noise Transmission Into Small Rectangular Enclosures

5.1.1 Elastic Plates

The objective of this work was to develop an analytical model to predict the noise transmitted through a rectangular elastic plate into an otherwise hard-walled acoustic cavity. The major new analytic contribution here is the consideration of cases in which only a portion of one cavity wall is flexible. Nonoverlapping, independently vibrating panels can be treated by superposition of solutions for a single panel for applications to aircraft noise transmission. The plate is driven by an external noise pressure which is assumed to be a stationary random process, and the force exerted by the interior acoustic pressure on the plate is also taken into account. The plate displacement and the interior acoustic pressure are obtained in terms of the natural modes of the plate and cavity and are ultimately expressed in terms of spectral density functions.

A great deal of information on noise transmission can be gained through a study of simple structural and acoustic geometries. For application to aircraft interior noise, these models could serve as baseline considerations where the key structural and acoustic parameters are accounted. Aircraft sidewalls are generally composed of many individual panels stiffened by stringers and frames. In many cases these individual panels can be assumed to vibrate independently of each other, the total interior noise pressure being determined by superposition as if each panel were moving in an otherwise rigid wall.

Figure 18 illustrates the rectangular enclosure-panel geometry. The panel location is described by position coordinates x_0, y_0. The noise reduction values predicted by theory were compared with experimental values at a point x = 0.1594m, y = 0.193m, z = 0.127m in a cavity of dimensions a = 0.3099m, b = 0.386m, d = 0.4572m. The external-pressure excitation used was spatially uniform white noise at a level of 100 dB, and the aluminum panel of thickness 0.001524m occupied the entire face z = 0 of the cavity ($x_0 = 0$, $y_0 = 0$). Figure 19 shows calculated and measured 1/3-octave band noise reduction. As can be observed from these results, the agreement between theory and experiment is relatively good. Then, using the same analytical model parametric studies of noise transmission through elastic panels into rectangular enclosures were performed. From the results obtained, the following generalizations were made:

1. External pressure distribution - The specific form of the external loading can have a considerable effect on noise reduction.
2. Plate boundary conditions - A clamped-edge plate gives more noise reduction below its fundamental natural frequency than a simply supported one. However, the noise

reduction is about the same at higher frequencies, except that at plate natural frequencies the clamped-edge plate yields less noise reduction.

3. Fluid parameters - Increased acoustic damping can increase noise reduction at cavity natural frequencies. If the fluid density is doubled, noise reduction is decreased by 6 dB.

4. Structural parameters - Increased structural damping increases noise reduction at panel natural frequencies. Increasing the panel natural frequencies increases noise reduction. In addition, doubling plate material density or thickness increases noise reduction by 6 dB.

5. Geometrical parameters - Smaller panels in general yield more noise reduction. Replacing one panel by several smaller panels increases noise reduction below and near the fundamental natural frequency of the smaller panels, but has little effect on noise reduction at higher frequencies. Considerable variation of noise reduction in the direction normal to the plane of the plate is observed.

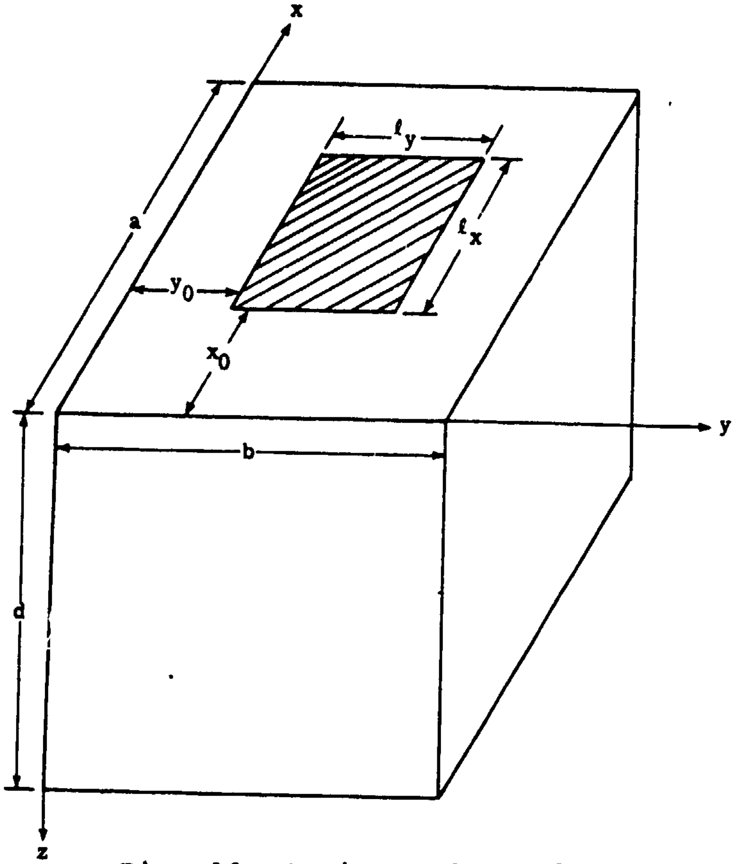


Fig. 18 Cavity and panel geometry.

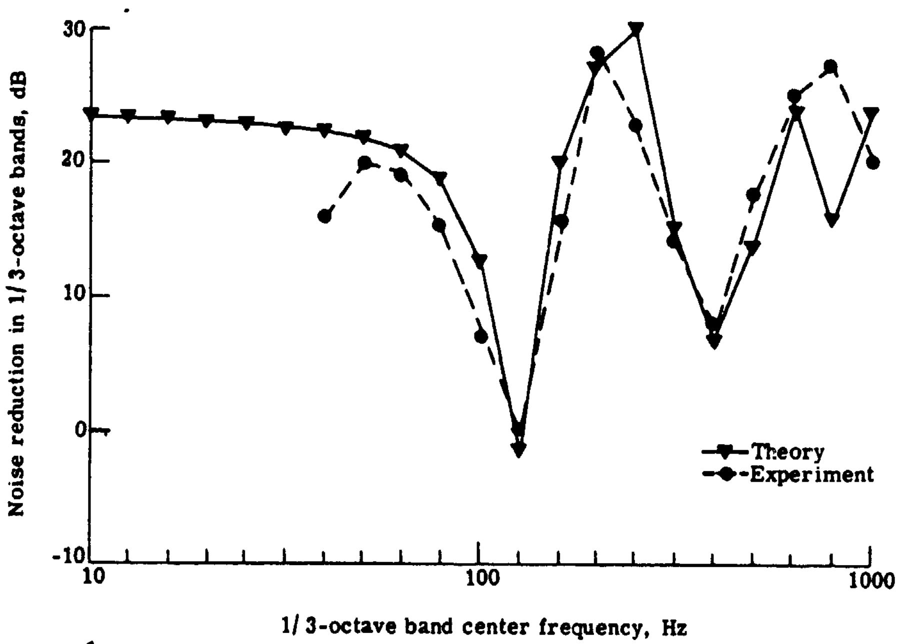


Fig. 19 Theoretical and experimental 1/3 octave band spectrum of noise reduction.

5.1.2 Viscoelastic Sandwich Panels

The information available in the literature and from ongoing research programs on interior aircraft noise indicates that noise in many aircraft and rotorcraft exceeds acceptable comfort limits. Since acoustic absorption materials used in aircraft constructions are not very effective in reducing interior noise at low frequencies, new means of providing noise attenuation at low frequencies need to be established. Interior aircraft noise in the structural resonance range is strongly controlled by the vibrational characteristics of the fuselage skin panels. Past studies have demonstrated that a viscoelastic material sandwiched between two elastic plates is a very efficient way of dissipating vibrational energy. Replacing some of the elastic skin panels with viscoelastic sandwich constructions should achieve additional amounts of noise reduction. Thus, an analytical study of this subject has been undertaken. The problem geomtery of acoustic enclosure and a viscoelastic sandwich panel is shown in Fig. 20. The dimensions of the flexible panel were chosen from typical aircraft skin panels. The solutions for noise transmission were developed for two limiting cases of core stiffness. In the first case, the viscoelastic material is taken to be very soft so that bending and shearing stresses can be neglected, and the core acts merely as a viscoelastic spring. For the sandwich construction with this soft core, the flexural vibration modes are governed by the stiffness of the two face plates, and the out-of-phase dilatational modes are controlled by the stiffness characteristics of the viscoelastic spring. In the second case, a stiff viscoelastic core material is used where the bending and shearing strains in the core are important. In both cases, the solution of the governing acousto-structural equations are obtained by using modal expansions and a Galerkin-like procedure. Since the boundary conditions for those equations are time dependent, the commonly used method of separation of variables cannot be applied to this system. The time dependence, however, is removed by splitting the solution into two parts: a solution corresponding to a nonhomogeneous differential equation with homogeneous boundary conditions and a solution on the boundary. Following this procedure, a Fourier series solution is developed which converges rapidly not only in the interior acoustic space but also on the boundary. These series have a computational advantage over the nonuniformly convergent series which usually converge very slowly.

The numerical results presented correspond to aluminum face plates and lightweight low modulus viscoelastic core material. The input random pressure acting on the top face plate is taken to be uniformly distributed truncated Gaussian white noise. The noise transmission through a sandwich construction with a soft core is shown in Fig. 21. These results are obtained from

$$NT = 10 \log S_p/S_p \tag{5}$$

where S_p is the spectral density of the acoustic pressure in the cavity at x = 0.102m, y = 0.152m, z = 0.254m and S_p is the spectral density of the input pressure. The dimensions of the acoustic enclosure and the sandwich panel are a = 0.250m, b = 0.508m, d = 0.762m, $h_1 = h_2 = 0.00051$m, $h_3 = 0.00635$m. The coefficient β indicates different values of damping loss factor in the core. When β = 0, damping is only present in the face plates and not in the core. As can be observed from these results, a strong peak occurs at the first dilatational modal frequency (401 Hz) when β = 0. The noise transmission characteristics of elastic panel (β > 0) with soft core are similar for frequencies up to about 250 Hz and above 600 Hz. However, in the intermediate frequency range, structural modes corresponding to dilatational vibrations have a significant effect on noise transmission. At the frequency of the first dilatational mode, a panel with a soft viscoelastic core can achieve about 20 dB more noise reduction. These results indicate that noise transmission into the enclosure is dominated by flexural modes for frequencies below 250 Hz, by dilatational and acoustic modes in the frequency region 250 to 500 Hz, and by acoustic modes above 500 Hz.

For the viscoelastic sandwich panel with a hard core, damping in the face plates was assumed to be negligible in comparison to damping in the core. A comparison of noise transmission by elastic and viscoelastic panels is shown in Fig. 22. The mass of the elastic panel was adjusted to be equivalent to the mass of the viscoelastic sandwich panel. Modal damping for the elastic panel was calculated from

$$\alpha_{mn} = \alpha_{11} (\omega_{11}/\omega_{mn}) \qquad (6)$$

where α_{mn} are modal coefficients and ω_{mn} are the natural frequencies of the elastic plate. The results shown in Fig. 22 correspond to $\alpha_{11} = 0.02$. As can be observed from these results, noise transmission for an elastic panel is strongly dominated by structural vibrations while the noise transmission by a viscoelastic sandwich panel is dominated by acoustic cavity modes. Significant differences of noise transmission can be achieved for viscoelastic cores with different values of shear modulus G_0.

An analytical study was conducted to determine noise transmission characteristics of viscoelastic sandwich panels. The results indicate that noise transmission by sandwich panels is strongly dependent on thickness, damping, and material properties of the viscoelastic core.

Sandwich panels with very soft viscoelastic cores transmit noise much like elastic panels except in the frequency range where dilatational (out-of-phase) modes are excited. About 20 dB more noise reduction can be achieved by visco-elastic sandwich panels in this frequency range (300 to 600 Hz).

The vibration response and noise transmission of sandwich panels with hard cores are lower when compared with equivalent elastic panels. As much as 50 dB more noise reduction can be achieved by viscoelastic panels at some frequencies in the low frequency range (below 200 Hz). At frequencies above 200 Hz, acoustic modes dominate interior noise for the acoustic enclosure chosen in this study. About 10 dB more overall noise reduction can be gained by increasing the core thickness 10 times (from 0.0025 to 0.025 m). With increasing core stiffness (from a shear modulus of 2 760 00 N/m^2 to a shear modulus of 68 900 000 N/m^2), noise reduction decreased by about 5 dB overall.

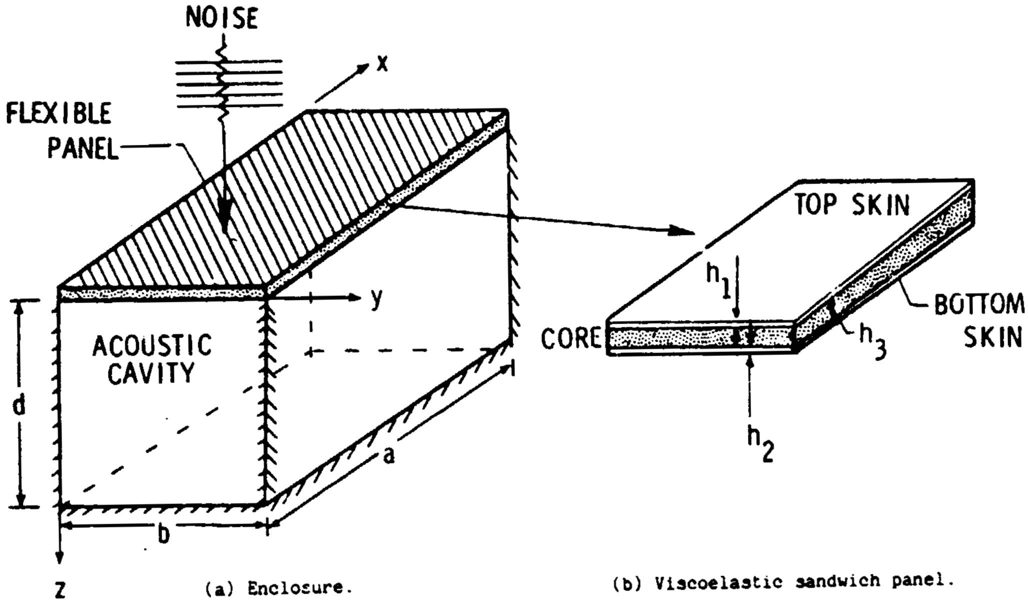


Fig. 20 Geometry of enclosure and viscoelastic sandwich panel.

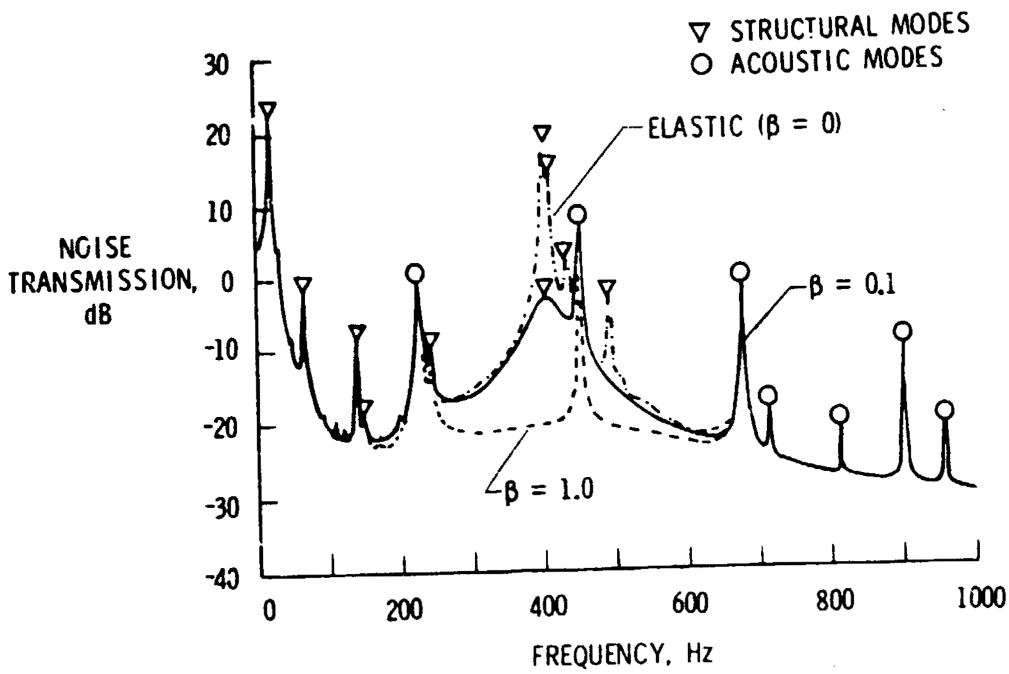


Fig. 21 Noise transmission by viscoelastic panel with soft core. h_3 = 0.00635 m; α_{001} = 0.01.

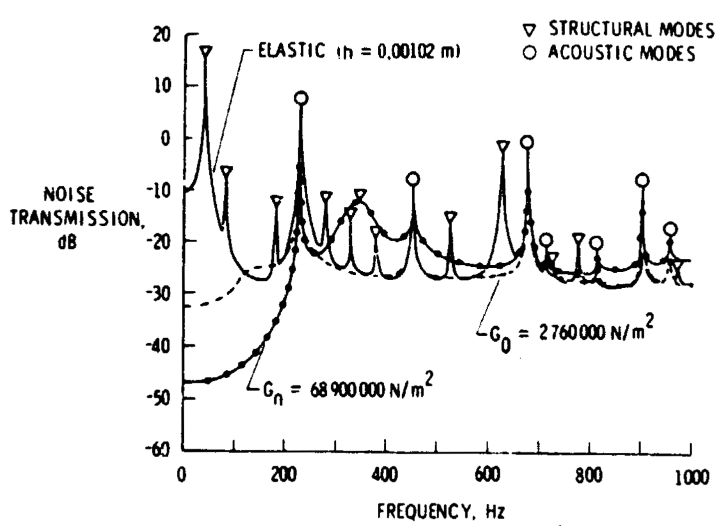


Fig. 22 Noise transmission by elastic and viscoelastic panels with hard cores. h_3 = 0.00635 m; β = 1.0.

5.2 Noise Transmission Through Stiffened Panels

Efforts to reduce cabin noise levels have created a need for better understanding of the interaction between the aircraft fuselage vibrations and the interior noise in the cabin. For example, effective utilization of add-on treatments, such as honeycomb panels, constrained layer damping, tuned damping and non-load carrying mass, requires detailed knowledge of the dynamic characteristics of multi-span structure of which many transportation vehicles are constructed. Most analytical studies on noise transmission into aircraft have been directed toward using "smeared" orthotropic plates or shells and individual panels. The effect of discrete stiffeners on the vibration and noise transmission was not considered. The objective of the present research was to develop analytical models capable of predicting the noise transmitted through flat and curved discretely stiffened panels. A brief description and highlights of this procedure was presented in Sec. 4. In what follows, key results of noise transmission through discretely stiffened panels are presented for laboratory models and light aircraft tested at ground conditions.

5.2.1 Transmission Loss Apparatus

The transmission loss apparatus located at NASA, Langley Research Center, ANRD is designed around two adjacent reverberant rooms of which the receiving room is acoustically and structurally isolated from the rest of the building (Fig. 23). The test specimen is mounted on a heavy, stiff frame which is installed as a partition between the two rooms. In the source room a diffuse noise field is produced by two reference sound power sources. Noise reduction is defined as the difference between the measured sound pressure levels of the microphones in the source and receiving rooms. The test "window" between the two rooms can accommodate test specimens that are similar in size to light aircraft fuselage side-walls. These specimens can range in size up to about 1.15m x 1.46m. Experiments were carried out by NASA to measure noise transmission through a stiffened panel built according to the specifications of an actual light turboprop aircraft. In addition, laboratory study was undertaken to define the noise transmission variables for a number of add-on treatments.

A theory based on a normal mode approach was developed to predict noise transmission through various panels installed in the noise transmission loss apparatus. The random pressure acting on the panel was modeled as a reverberant acoustic field for which the joint modal acceptances were calculated. The low frequency noise transmission (up to about 125 Hz) was calculated using a "smeared" orthotropic panel model. For the intermediate frequency range of 125 - 800 Hz, discretely stiffened panels were used for which the modes and modal frequencies were obtained by transfer matrix or finite element strip methods. Single panel model was used for those cases where a flexible panel was supported by heavy frames. The emphasis of the analytical study was on low frequency noise transmission (up to about 800 Hz).

The insertion losses due to add-on treatments which have no marked effect on the structural vibrations (porous acoustic materials, noise barriers, and trim panels) were calculated by the impedance transfer method. Numerical procedures were developed for the multi-layered treatments shown in Fig. 24. The sound pressure levels inside the enclosure were calculated from

$$SPL \Big|_{treated} = SPL \Big|_{untreated} - \Delta TL_0 - \Delta TL \qquad (7)$$

where ΔTL_0 are the noise losses due to treatments directly attached to the aircraft skin (honeycomb panels and damping tape) and ΔTL are the insertion losses due to all other treatments (porous blankets, septum barriers, trim).

As a first step in the validation of the theory, predictions and tests were performed for a stiffened panel shown in Fig. 25. The noise inputs were measured at several positions close to the panel surface by a stationary microphone. A spatially averaged value of source pressure was selected for noise reduction calculations and measurements. Transmitted noise was measured by a stationary microphone located at about the middle of the panel and 66 in. from the panel surface.

The analytical calculations were obtained for a constant structural modal damping ratio ζ_0 and acoustic modal damping $\alpha_{ijk} = \alpha_0(\omega^{\ell}/\omega_{ijk})$. ζ_0 and α_0 are the damping coefficients of the fundamental modes, ω_{ijk} the acoustic modal frequencies, ω^{ℓ} the lowest modal frequency in the receiving room, and i,j,k the modal indices. For the present sutdy, it was assumed that ζ_0 = 0.03 for the untreated sidewall panel and 0.05 for the case where honeycomb panels are attached to the interior surface (receiving room side) of the skin. These damping coefficients are taken to be representative of the overall damping which includes material, structural, and acoustic radiation damping. The acoustic damping in the receiving room α_0 is taken to be equal to 0.005. This damping model is assumed to be a simplified representation of equivalent acoustic damping in a room with hard walls.

To investigate the effect of skin stiffening, lightweight treatments in the form of honeycomb panels were attached to the interior side of the skin. For this purpose, aluminum honeycomb sandwich construction with a core thickness of 0.25 in. and a face plate thickness of 0.032 in. was selected. The surface density of this treatment is about 0.68 psf. Fig. 26 shows theoretical and experimental noise reductions of the stiffened panel treated with honeycomb. In general, the aggreement between theory and experiment is good in view of the complexities involved. About 4-5 dB of noise attenuation is achieved in the 80-400 Hz frequency range with the honeycomb add-on treatment. However, for frequencies above 400 Hz, lower values of noise reduction were measure for treated panels. The noise reduction in this frequency range seems to be influenced by the coincidence effect at the critical frequency. The critical frequency of a sandwich construction composed of 0.063-in. skin, a 0.25-in. honeycomb core, and a 0.032-in. face plate is about 1000 Hz. However, the presence of heavy stiffeners could reduce the critical frequency of this panel further.

To simulate a treated sidewall, porous acoustic materials and a 0.5 psf lead impregnated vinyl trim was added to the honeycomb-treated panel. These results are presented in Fig. 27. The theoretical predictions are given for two values of flow resistivity R. Data on flow resistivity of the acoustic material used for the present study do not seem to be available. However, it is assumed that these values are larger than R = 4.23×10^4 mks rayl/m commonly used for AA-type acoustic blankets. From the results shown in Fig. 27, it can be seen that similar noise reduction trends are predicted by theory and experiment. However, significant differences occur at some frequencies. These differences might be attributed to the limitations of the impedance transfer method used to calculate insertion loss for finite panels and uncertainties associated with structural and acoustic damping, and material properties of acoustic materials. Addition of a limp panel provides a significant amount of noise attenuation for frequencies above 125 Hz.

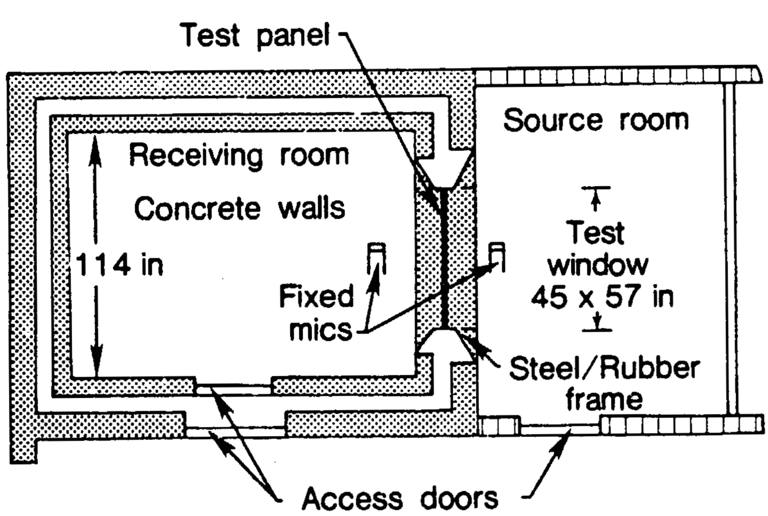


Fig. 23 Transmission loss apparatus

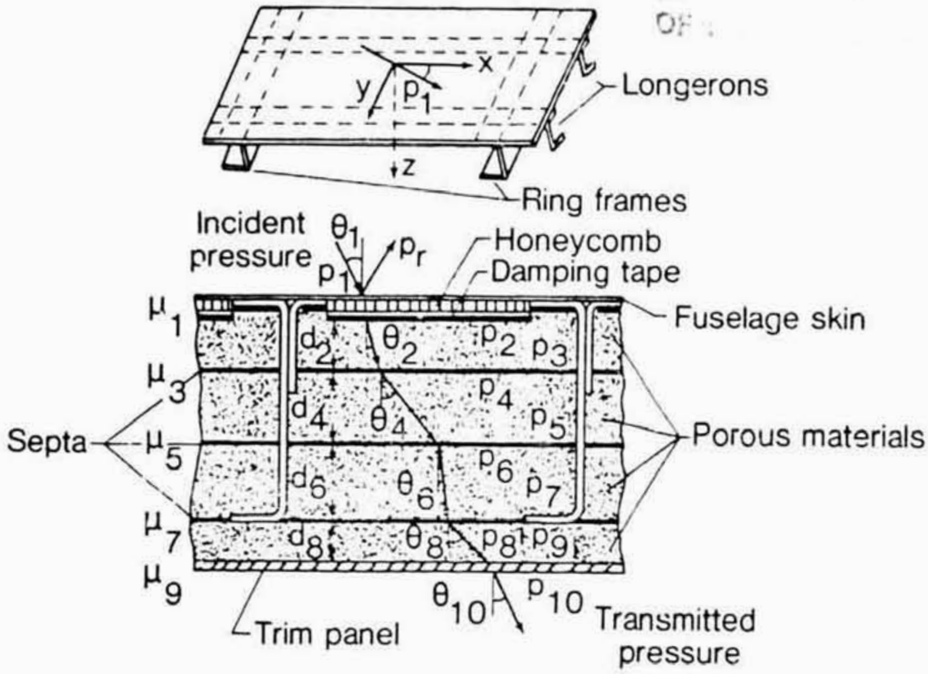


Fig. 24 Sidewall representation of multilayered add-on treatments

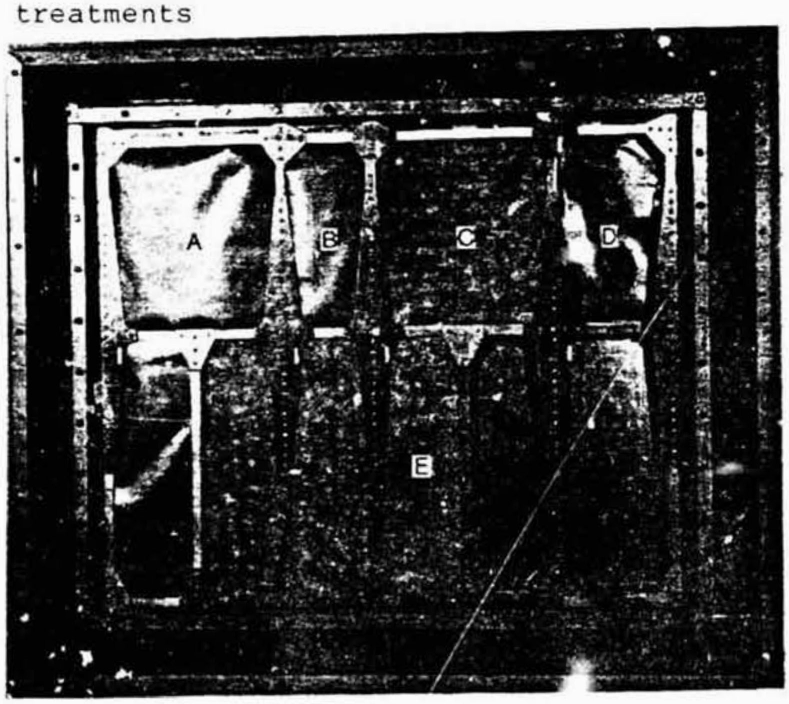


Fig. 25 Stiffened sidewall panel used for noise transmission study

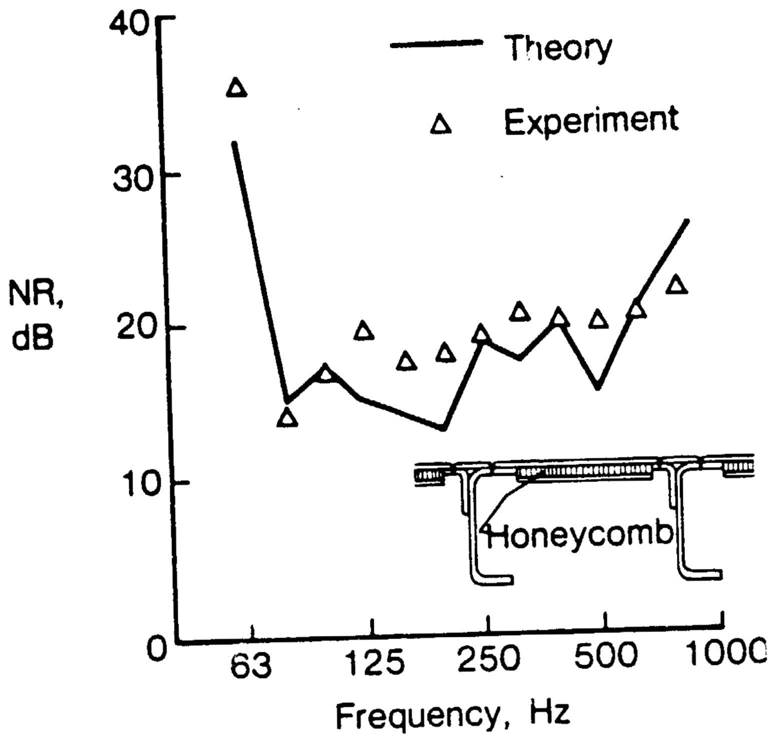


Fig. 26 Noise reduction of the panel treated with honeycomb

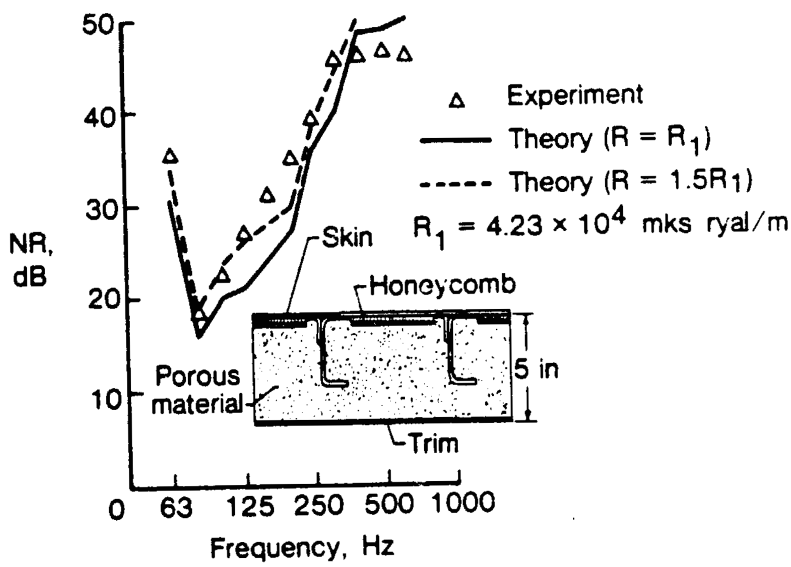


Fig. 27 Noise reduction of a sidewall panel treated with honeycomb, porous acoustic material and trim

5.2.2 Noise Transmission Into Aircraft (Ground Conditions)

The aircraft used in these studies is the Model 680 Aero Commander shown in Figs. 28 and 29. 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. The propeller tip clearance from the sidewall is about 5 inches and the propeller plane intersects the fuselage at approximately middle of the cabin. The cabin provides seats for pilot, copilot and four passengers.

The flexible portion of the aircraft sidewalls shown in Figs. 28 and 29 is composed of an external skin which is stiffened by stringers and frames, thermal and acoustic insulation, trim panels and several single- and double-wall window units. For the analytical study, the sidewall was subdivided into several smaller units as shown in Fig. 30. Such a segmentation offers significant advantages for noise transmission path identification and interior noise optimization. Due to the very stiff boundary conditions of some of the panel units, such an approximation seems to be justified for this type of fuselage construction. In the present analysis, simple panels, discretley stiffened panels, and double-wall windows were considered as structural models for noise transmission estimation through localized regions and the entire sidewall. The noise transmission through these regions into the cabin is obtained by solving the linearized acoustic wave equation for the interior noise field and the vibration equations for the sidewall panels. The solution to this system of equations is obtained by using modal expansions. The single panels are modeled by simple plate theory, discretely stiffened panels by the transfer matrix methods, and windows by a double-wall theory.

The numerical results presented in this report are noise transmission through sidewall panels and the entire sidewall of the aircraft shown in Figs. 28 and 29. The calculations are obtained for structural and acoustic modal damping ratios $\zeta_{mn} = \zeta_0 (\omega_{11}/\omega_{mn})$ and $\xi_{ijk} = \xi_0 (\omega^\ell/\omega_{ijk})$. The ζ_0 and ξ_0 are the damping coefficients of the fundamental mode, ω_{mn} and ω_{ijk} are the structural and acoustic modal frequencies, and ω^ℓ is the lowest acoustic modal frequency in the enclosure. For the present analytical study, it was assumed that ζ_0 equals 0.01 for single panels, 0.03 for honeycomb-treated panels, 0.04 for damping tape treated panels and 0.05 for the Plexiglas windows. The damping coefficient ξ_0 was assumed to be equal to 0.03 for a lightly treated cabin and 0.06 for a heavy treated cabin. Wall absorption was provided through a point impedance model.

Experiments were carried out for noise transmission estimation through localized regions and the entire sidewall. The localized inputs to the sidewall panels and windows were generated by an acoustic guide shown in Fig. 31. The basic features of the acoustic guide design include a high quality apeaker and a slowly diverging rectangular duct. To minimize noise leakage from the interior enclosure of the guide, two layers of noise barriers, each with a surface density of 1 lb/ft^2, were added to the exterior surfaces of the guide. Between the duct and the sidewall of the aircraft, soft insulation material (foam), ranging in thickness from about 2 in. to 4 in. was installed around the periphery of the guide. The present design with duct dimensions of either 20 in. x 20 in. or 30 in. x 30 in. can be used to generate acoustic inputs for small panels, windows, and larger areas of discretely stiffened panels. The noise-measuring system includes a microphone inside the guide at about 1 in. from the sidewall surface and a microphone inside the cabin at about 10 in. from the interior wall. For laboratory study of noise transmission through the entire sidewall, the acoustic inputs were generated by a two-speaker setup. A relatively uniform external noise pressure distribution over the surface of the sidewall was generated by the two speakers which were located at about 4 ft. from the sidewall.

Theory and Experiment

The noise transmission through each panel unit indicated in Fig. 30 was calculated and measured. Assuming independence of these noise transmission paths, the total interior sound pressure (theoretical) was determined by the superposition of the contributions of all panels locate on the sidewall. The noise reduction for a typical stiffened panel and the entire sidewall is given in Figs. 32 and 33. Similar results are presented in Fig. 34 for a double-wall window. These results correspond to untreated conditions and an interior location at ear level in the propeller plane and 10 in. from the sidewall. As can be observed from these results, the agreement between theory and experiment is relatively good in view of the complexities involved. The theoretical model tends to predict lower noise reduction at some structural modal resonance frequencies. These differences might be attributed to damping (structural and radiation) and idealistic mode shapes used in the theoretical calculations. Furthermore, the assumed spatially uniform inputs for the theoretical model cannot be accurately simulated with the present experimental setup.

Experimental Parametric Study of Add-on Treatments

The noise transmission data were obtained for a variety of aircraft sidewall treatments. These treatments include honeycomb panels, damping tapes, nonload carrying mass, acoustic barriers, multi-layered septum and trim panels. Furthermore, noise transmission through aircraft windows has been measured for several conditions of different acoustic interiors. The primary objective of this work was to evaluate a number of candidate treatments to be used for the optimization of interior noise in light aircraft. The description of add-on treatments is given in Table 1.

The noise reduction for untreated and treated sidewall is given in Fig. 35. These results indicate that 4-14 dB of additional noise reduction can be achieved in the frequncy range 50-1000 Hz with an add-on treatment composed of honeycomb, constrained layer damping tape and two layers of acoustic blankets. The added surface density is about 1 lb/ft^2. The results presented in Fig. 36 shown the effect of trim panel on noise reduction. For frequencies above 160 Hz, significant amount of noise reduction can be achieved with a heavy trim panel (about 1 lb/ft^2). The following conclusions were drawn from this parametric study:

1. An acoustic guide device can be used to generate noise inputs over localized regions of the sidewall.
2. Interior noise levels transmitted through localized panels or windows are function of measurement position in the cabin and conditions of treated interior.
3. Stiffening skin panels with honeycomb could provide 3-7 dB additional noise reduction. However, these gains are functions of panel geometry, installation conditions and frequency. The noise attenuation obtained for the entire sidewall treated with honeycomb panels is less than for some individual panels.
4. Constrained layer damping materials could provide 2-3 dB of noise reduction. However, these increases depend on frequency.
5. Porous acoustic blankets (2-3 layers) provide noise attenuation for frequencies only above 300 Hz. The insertion losses reach about 10-12dB at 1000 Hz.
6. A multilayered treatment composed of porous blankets and impervious vinyl septa does not provide additional noise reduction for frequencies up to about 500 Hz. In the frequency range below 500 Hz, several double wall resonances are observed.
7. Noise barriers composed of urethane elastomer and decoupler foam do not give noise attenuation for frequencies up to about 500 Hz.

8. A treatment composed of several layers of acoustic foams does not seem to provide noise attenuation. However, when a single layer is used as a trim panel, positive gains of noise reduction are achieved for frequencies above 600 Hz.

9. A trim with a surface density of lb/ft^2 panel which is isolated from the vibration of the main structures could provide insertion losses ranging from 3-20 dB for frequencies 160-1000 Hz. Negative values of noise attenuation were measured at the double wall resonance frequency of 125 Hz.

10. An acoustic treatment for noise control in this aircraft should be composed of honeycomb panels, constrained layer damping tape, several layers of porous acoustic materials, and limp trim panel which is isolated for the vibration of the main structure. Furthermore, additional stiffening to window supports, some frames and longerons would need to be implemented.

<u>Interior Noise Optimization: Theoretical Study</u>

A detailed analytical study of noise control and cabin noise optimation with add-on treatments was undertaken. 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. 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 was used. The interior noise in the cabin was optimized utilizing measured propeller noise inputs (ground conditions) and the following computation procedure. 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 treatments was increased until a selected target noise level at a critical point in the cabin was reached. The treatment or combination of treatments which reduces the transmitted noise to the selected optimization value for the least amount of added weight was assumed to be the best treatment. An 85 dBA overall average noise level was selected as the optimization target for the noise transmitted into the cabin. The optimized and baseline (untreated cabin) interior noise levels are shown in Fig. 37. The optimized treatment includes honeycomb panels, constrained layer damping tape, porous acoustic blankets and relatively soft trim which is isolated from the vibrations of the main structure. The added weight of the optimized treatment is about 1.5 lb/ft^2. The distribution of the surface densities (baseline and treated) for the optimized sidewall are given in Fig. 38. In this procedure, it is assumed that flanking paths can be eliminated. Based on this study, the following conclusions have been reached.

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 trnsmission 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 constrained layer damping tape applied to the aircraft skin seems to promise the required reduction 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 combination 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.

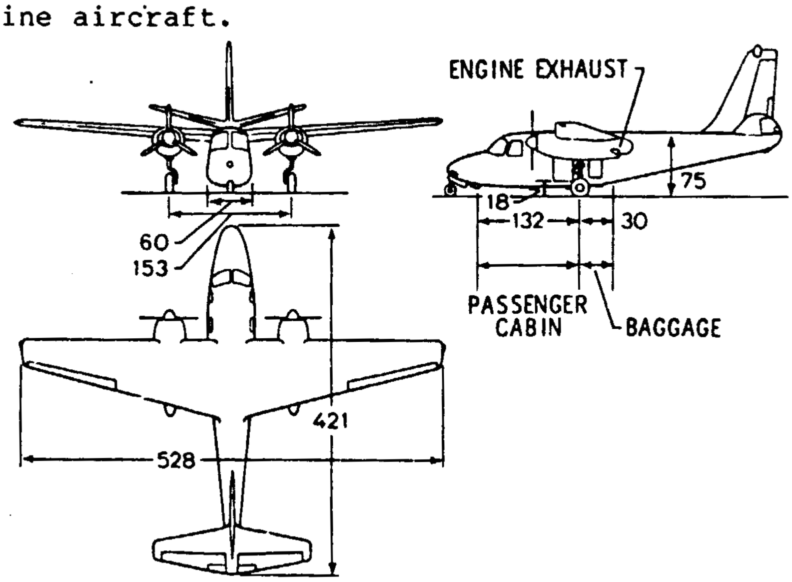


Fig. 28 Twin-engine aircraft used in the noise optimization study

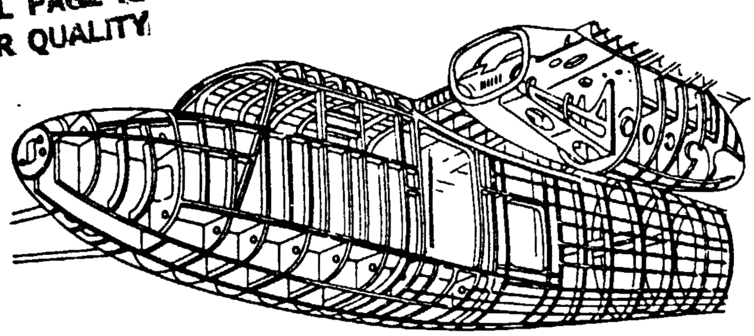


Fig. 29 Structural features of a twin-engine aircraft used for interior noise study

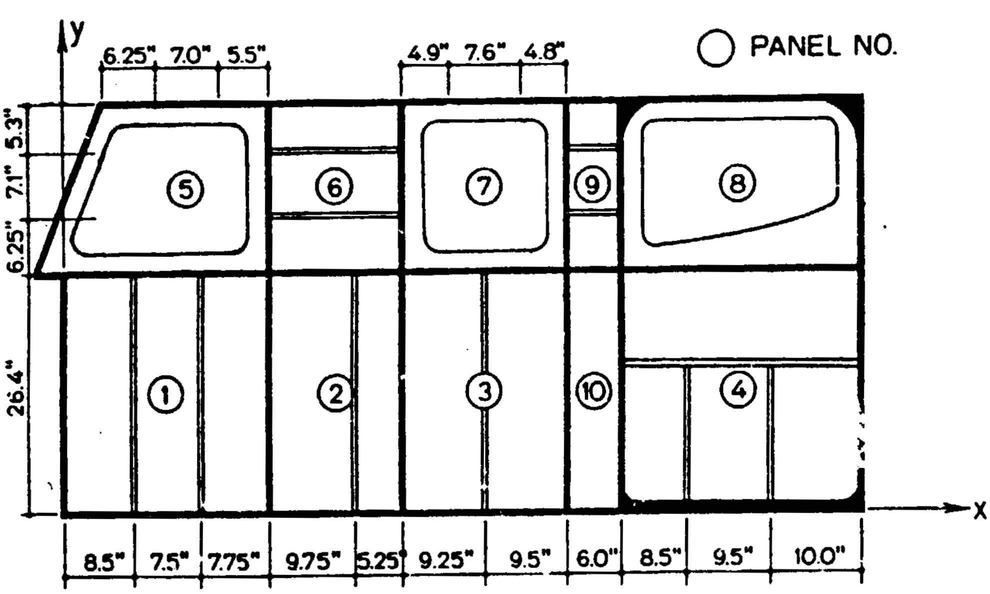


Fig. 30 Panel Identification for Port Side (P)

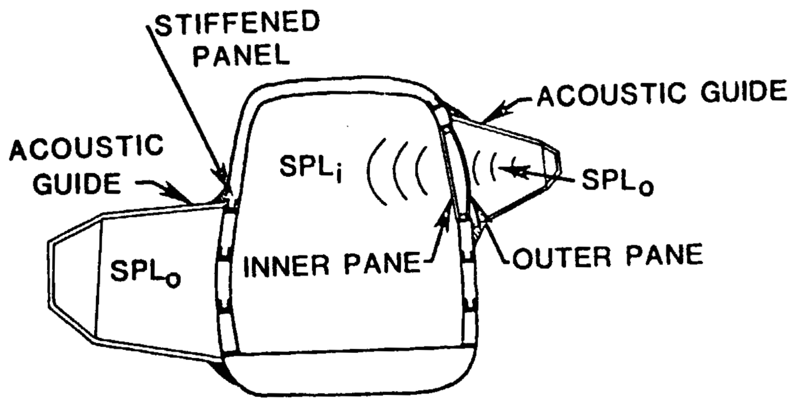


Fig. 31 Setup for laboratory noise transmission thorugh localized regions

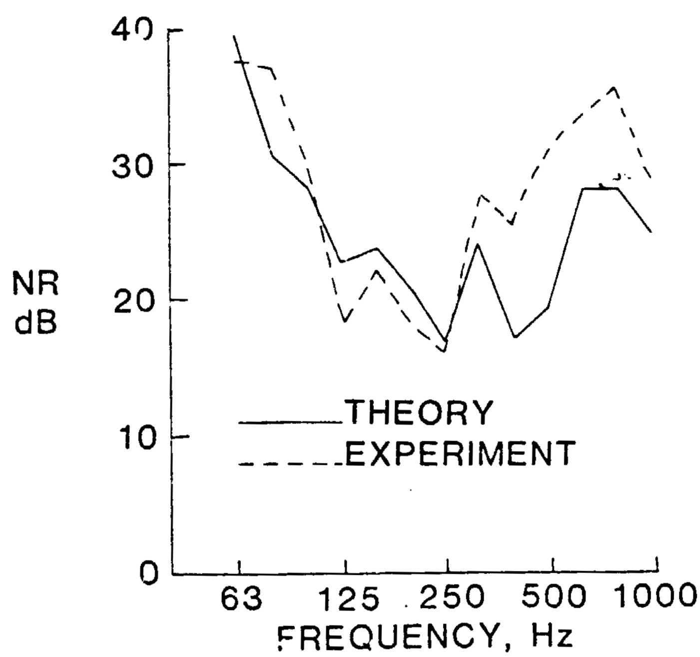


Fig. 32 Noise reduction of a stiffened aircarft panel: noise input from an acoustic guide

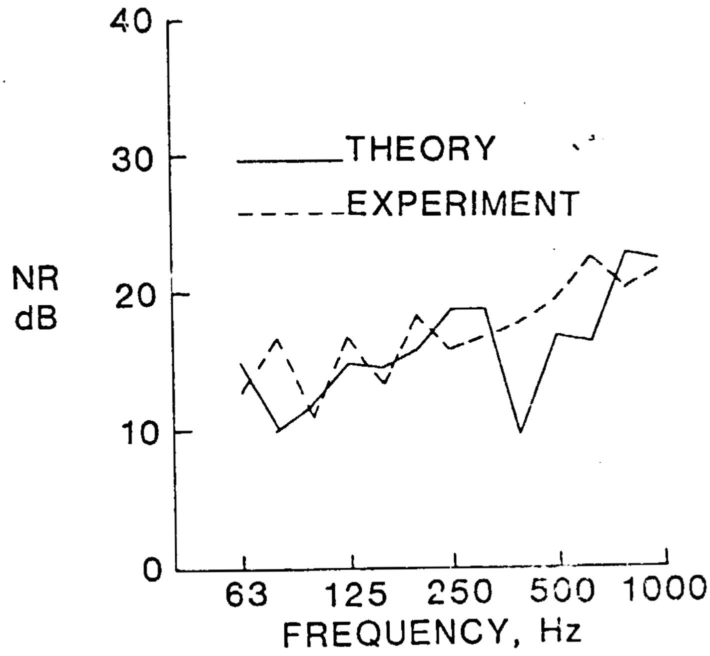


Fig. 33 Noise reduction of an aircraft sidewall: noise input from a diffuse field

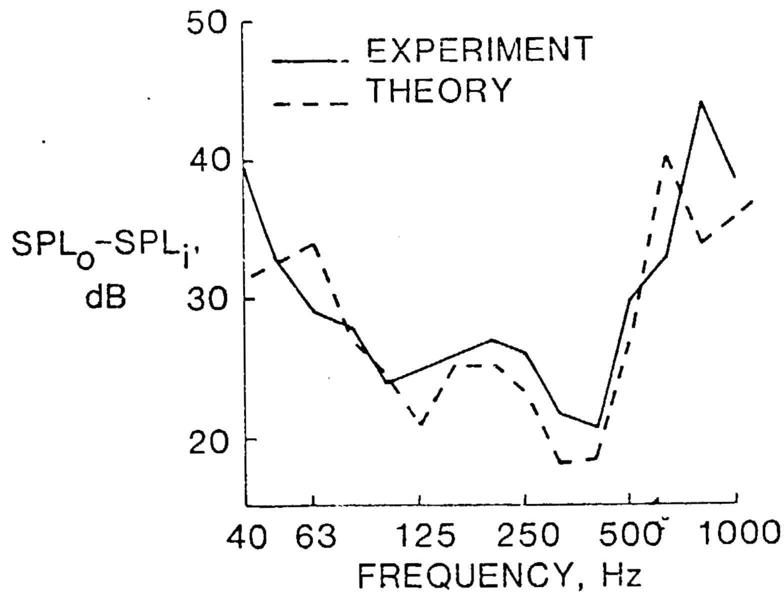


Fig. 34 Noise reduction of a double wall aircraft window: noise input from an acoustic guide

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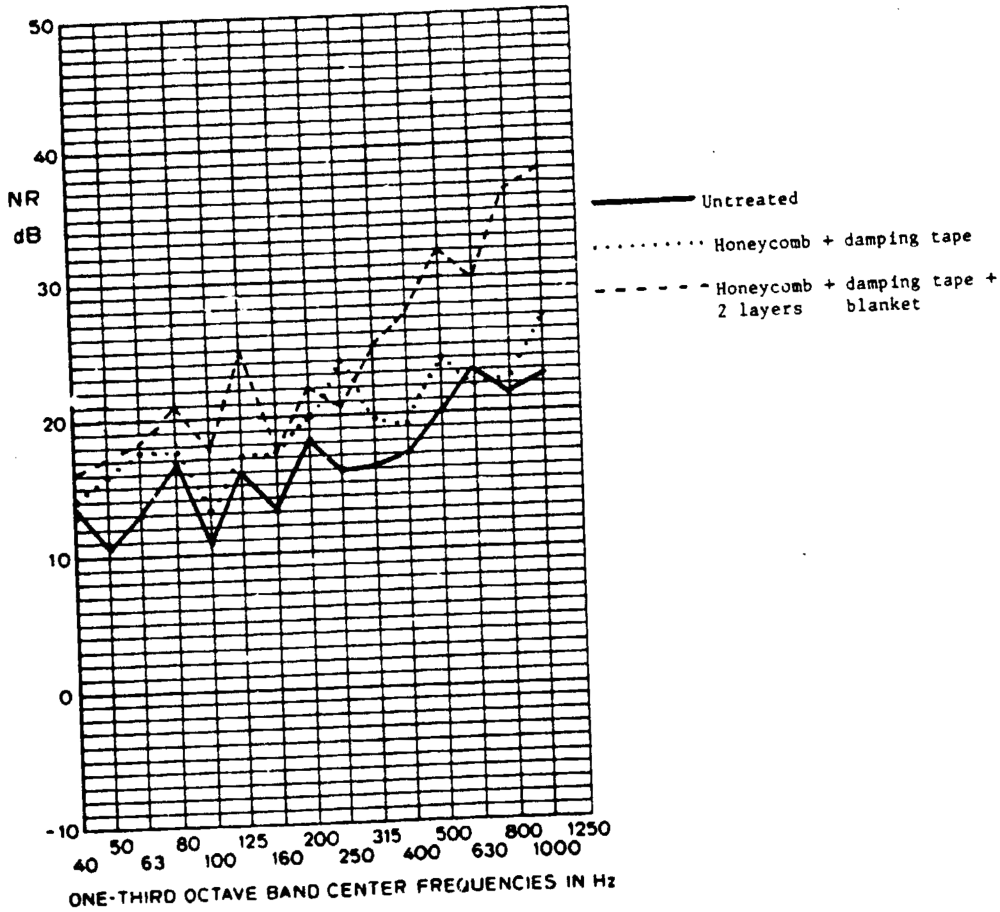

Fig. 35 The one-third octave noise reduction for treated and untreated sidewall

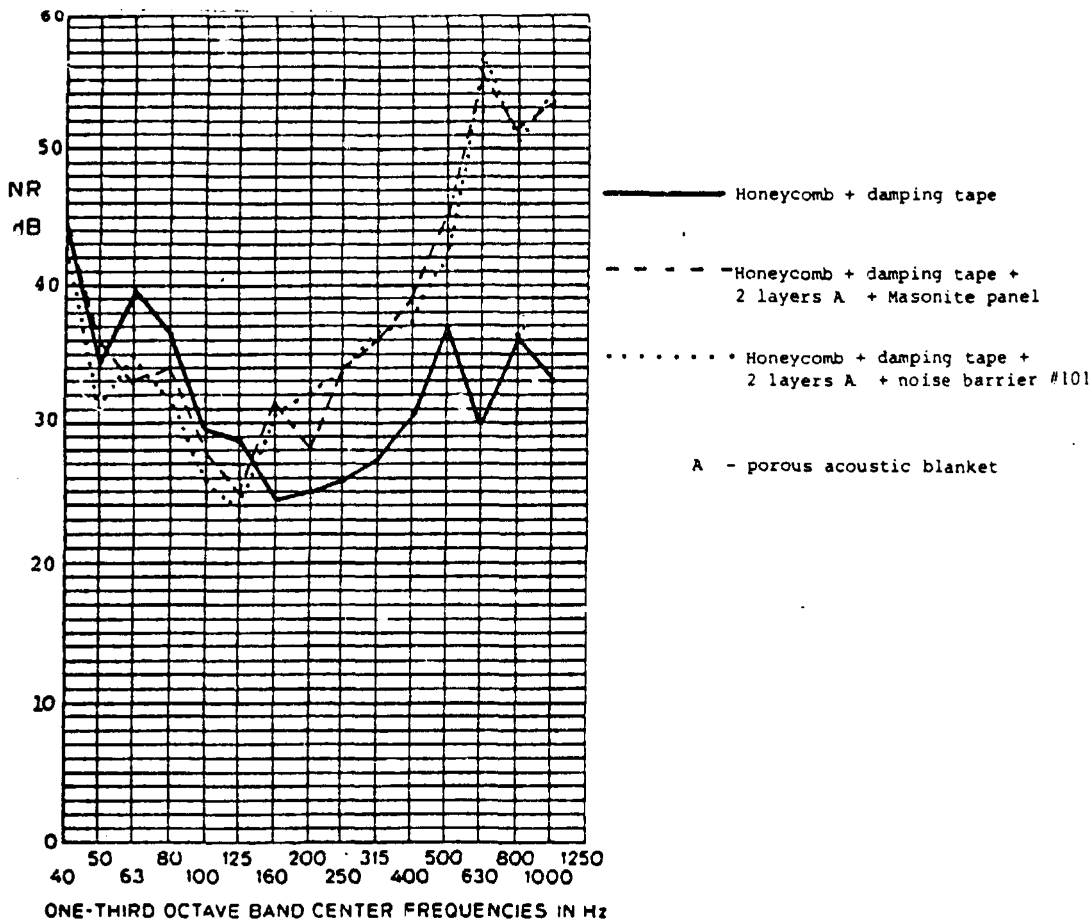

Fig. 36 Noise reduction with a heavy trim

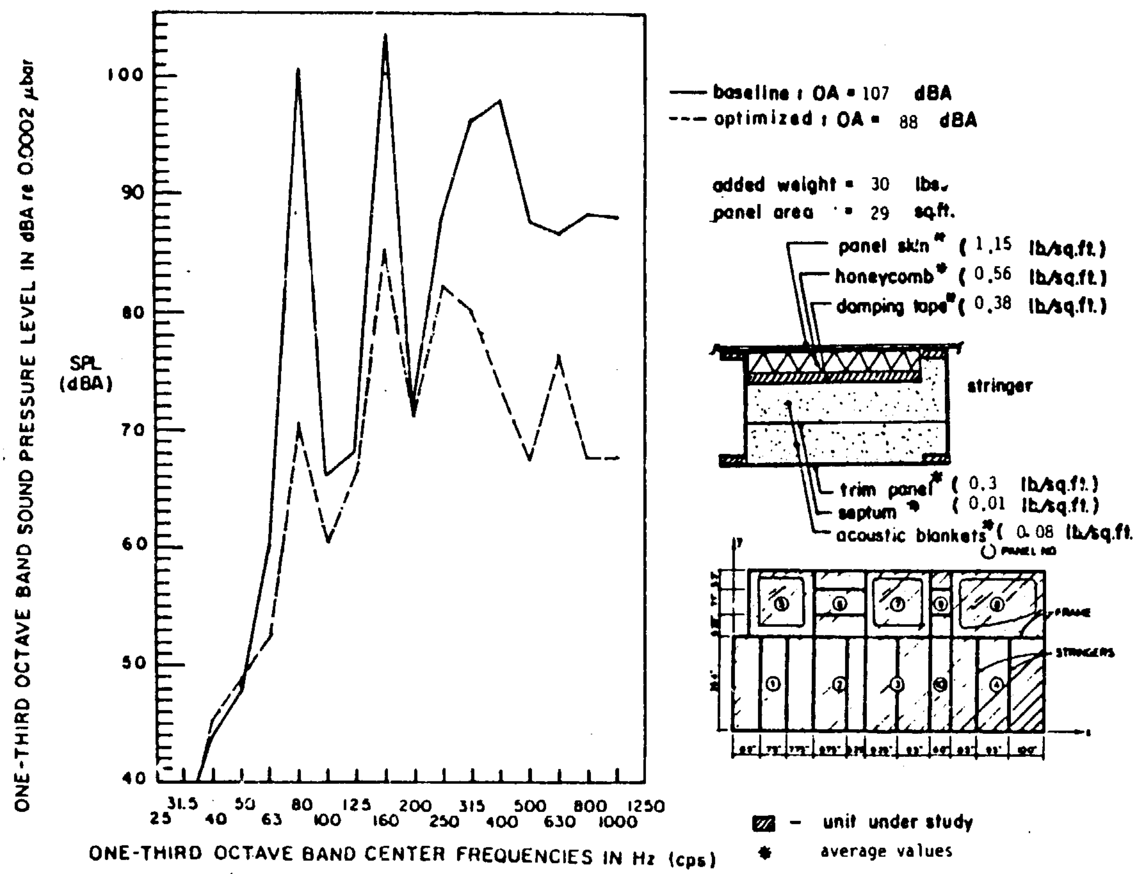

Fig. 37 Optimized interior noise (Sidewall)

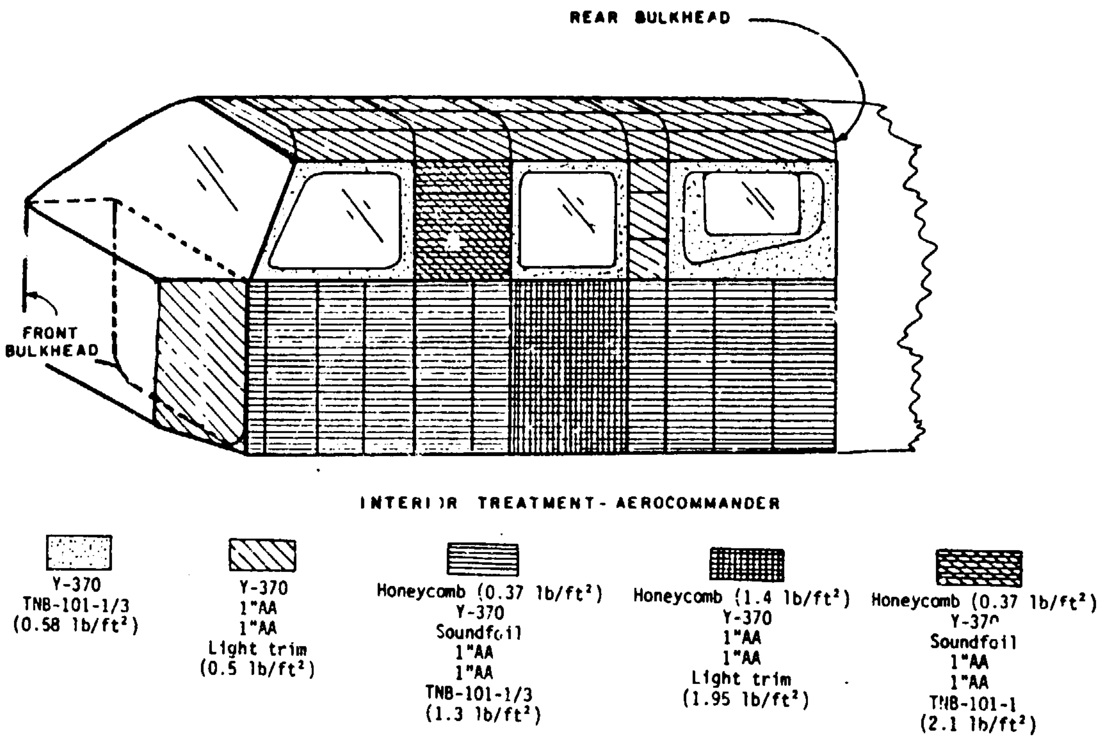

Fig. 38 The final configuration of add-on treatment for the AeroCommander aircraft

5.3 Noise Transmission Into Aircraft: Flight Conditions

The aircraft considered in the present study is a twin engine light aircraft, Fig. 39. Flight tests indicate that an acoustic treatment capable of reducing the average overall noise levels in the cabin by about 17 dB compared to the untreated cabin is desirable. Theoretical analyses of noise transmission have been developed to assist in the design of such a treatment. The basic concept of the theoretical model is that of modal analysis and the acoustic impedance transfer. A similar approach described in Sec. 5.2.2 has been used for interior noise optimization of a small twin-engine aircraft corresponding to inputs measured at the ground conditions. These methods have been extended for design of noise control treatments of the aircraft shown in Fig. 39 under flight conditions. Improved analytical methods of noise transmission prediction through flexible panels and double pane windows are incorporated in the present study. The analyses are compared with test data under laboratory and flight conditions.

The acoustic treatments considered in this paper include aluminum honeycomb panels, constrained layer damping tape, porous acoustic materials, noise barriers, limp trim panels, tuned dampers and changes in aircraft window design. This work is focused on evaluating the advantages and disadvantages of these treatments for noise control and then designing a candidate acoustic treatment which reduces noise in the cabin to a satisfactory level.

The sidewalls of the aircraft shown in Fig. 39 are composed of a 0.063 in. thick external skin which is stiffened by frames and longerons and several single- and dual-pane windows. The exact dynamic analysis of such a structure is too complicated and a simplified model needs to be constructed. For the structural model of the present study, Fig. 40, the aircraft sidewall is segmented into four stiffened skin-stringer panels, two single panels, and six windows. The noise transmitted through other surfaces of the sidewall is neglected. Such a segmentation offers significant advantages for noise transmission path identification and computational simplification. However, this procedure is best applicable for the intermediate to high frequency ranges. The lowest modal frequencies calculated for window unit No. 1 and stiffened panel unit No. 11 are 114 Hz and 134 Hz. Measurements tend to indicate the presence of several frame-wall modes in the frequency range of 48 Hz to 104 Hz. Thus, the segmented structural sidewall model does not account for the frame-wall modes below the frequency of about 100 Hz. To develop accurate solutions for low frequencies, numerical techniques such as the finite element method can be used. Since the A-weighted interior noise in this aircraft is dominated by the 2nd, 3rd and 4th (152 Hz, 228 Hz, 304 Hz) propeller blade passage harmonics, the idealization of the sidewall into several independent units seems to be justified.

The external surface pressure acting on the aircraft is propeller and turbulent boundary layer noise. For frequencies up to about 700 Hz, propeller noise due to blade passage harmonics is the dominant source of noise for this aircraft. In the present study, the input pressure field is treated as a random process with prescribed cross spectral density function. The sound pressure levels characterized by the spectal density are taken to be uniformly distributed over each panel surface, but varying in a step-wise fashion from one panel to another. These spectral densities are obtained from flight data. In addition, the empirical predictions of propeller noise are utilized to distribute the sound pressure levels over the aircraft fuselage. The trace velocity corresponding to the y- direction was taken as the propeller rotation tip speed while sonic trace velocities were assumed for the longitudinal direction x (normal to the propeller rotation tip plane). The turbulent boundary layer noise was taken to be fully correlated and uniformly distributed over each panel surface.

The insertion losses due to add-on treatments which have no marked effect on the structural vibrations (porous acoustic materials, noise barriers, trim panels) are calculated by the impedance transfer method. In this case, the treated structure is assumed to be an infinite, stiffened and pressurized panel. The sound impinges on the panel at an angle θ_1 relative to the normal and an azimuthal angle ϕ relative to the x-axis as shown in Fig. 24. The transmission coefficient τ and the insertion loss ΔTL can be obtained from

$$\tau(\omega,\theta_1,\phi) = \left|\frac{(p_1/p_2)_{untreated}}{(p_1/p_2\cdots(p_{n-1}/p_n)_{treated}}\right|^2 \qquad (8)$$

$$\Delta TL(\omega,\theta_1,\phi) = -10 \log \{\tau(\omega,\theta_1,\phi)\} \qquad (9)$$

where p_{n-1}/p_n are the pressure ratios across the different layers of the add-on treatments. The pressure ratio p_1/p_2 is across the skin-stiffener panel requiring the information of the bare fuselage impedance. The interior of the ·edium is assumed to extend to infinity with a. acoustic termination impedance ρc. Numerical results were obtained for treatments composed of up to eight different layers and the elastic skin-stiffener panel. These treatments include porous acoustic materials, noise barriers and limp trim panels.

Noise transmission measurements were obtained for the aircraft shown in Fig. 39 for normal cruise at 214 kts, maximum continuous power 96 percent RPM, and altitude of 16,000 ft. This aircraft has a maximum take-off weight of about 11,200 lbs, a standard cabin layout for a pilot and seven passengers, pressurized cabin environment and 35,000 ft operational ceiling. Operating at 1500 RPM during cruise, the propeller blade passage frequency is 75 Hz and the tip speed 692 ft/sec. The cabin height, width and length are, respectively, 4.76 ft, 4 ft, and 17.5 ft. Flight tests results are reported for two acoustic treatment configurations. Configuration 1 is referred as untreated which had no acoustic treatments and no carpet on the floor. However, four seats (pilot, co-pilot and two passengers) were present for the "untreated" and the "treated" cases. Configuration 2 is similar to the standard soundproofing treatment used in this aircraft. This configuration includes a constrained layer damping tape attached to the aluminum skin, four layers of porous acoustic blankets, two layers of lead vinyl septa, and a foam-rubber type sandwich noise barrier. The combination of the treatments and surface density varied with location in the cabin. The heaviest treatment of about 2.25 lb/ft^2 was applied in the vicinity of the propeller rotational plane. The hard-plastic trim panels were not installed for these tests. However, the trim condition of a "limp" panel was simulated by the presence of the noise barrier located at about 2.5 in to 3 in from the fuselage skin.

Cabin noise measurements were obtained at six location in the aircraft. Measured exterior sound pressures were used to model the inputs for the analytical noise transmission predictions. The one-third octave A-weighted cabin noise levels are shown in Figs. 41 and 42 for the untreated and treated conditions. These results correspond to a location in the propeller plane and about 8 inches from the skin. Similar trends of interior noise are predicted by theory and experiment. However, for the untreated cabin theory predicts significantly higher noise levels at the second propeller blade passage harmonic (150 Hz). The segmentation of aircraft sidewall into independent panel units, assumption of uniform pressure distribution over each panel surface and large scatter in measured data could have contributed to these differences. Furthermore, for untreated cabins the sound that is radiated out through the vibration of the entire fuselage might be significant. In the present analytical model, only the panels through which noise is being transmitted are taken to be flexible. The structural modal damping $\zeta_0 = 0.02$ and $\zeta_0 = 0.05$ were used for sidewall panels and dual pane windows, respectively, The equivalent acoustic modal damping of the untreated cabin was assumed to be equal to 0.03. To account for the acoustic absorption in a treated cabin, the acoustic modal damping coefficient was increased to 0.05. The main contribution to noise attenuation of the heavy treatment occurs at frequencies above 300 Hz. The results tend to indicate that the multi-layered acoustic treatments do not provide the required noise attenuation for frequencies below 300 Hz where the propeller inputs are the highest.

A theoretical parametric study has been carried out to evaluate a variety of candidate treatments for noise transmission control in this aircraft under flight conditions. The following add-on treatments were considered: honeycomb panels, constrained layer damping tapes, non-load carrying mass, porous acoustic blankets, noise barriers, trim panels and changes in window design. The results from the parametric study were used to design an acoustic treatment suitable for noise control in this aircraft.

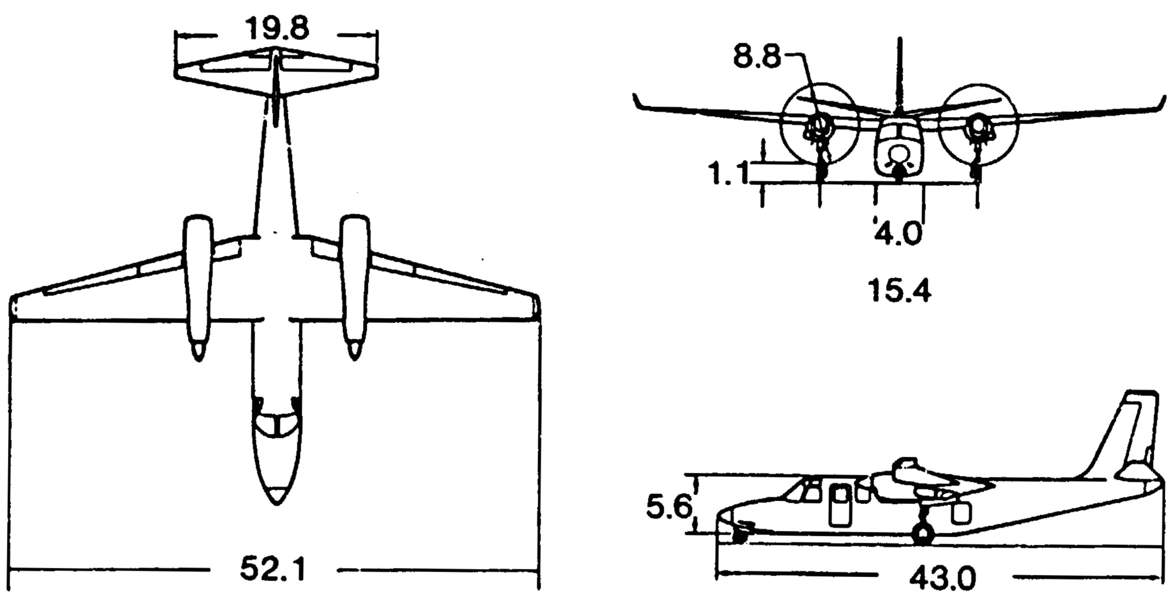

Fig. 39 Twin-engine aircraft used in noise transmission study (dimensions in feet)

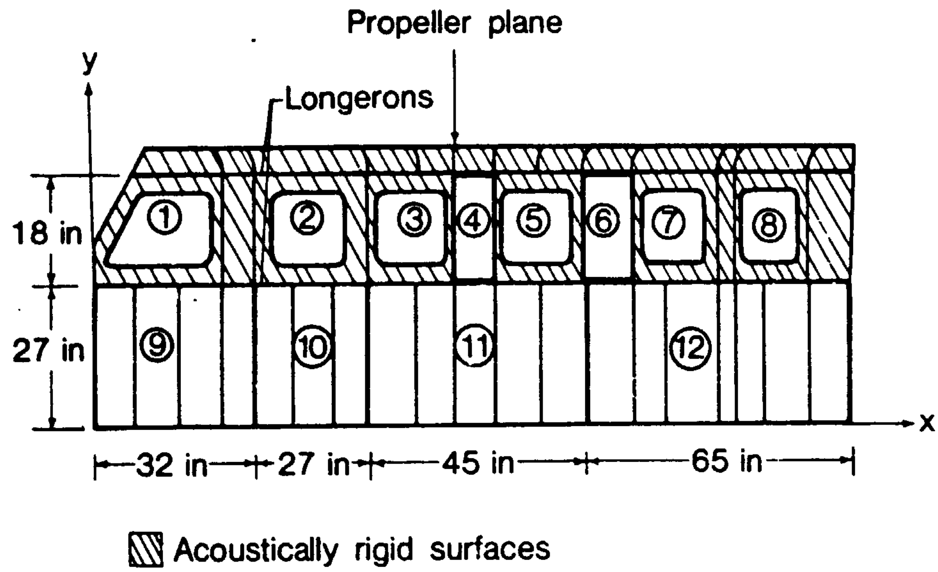

Fig. 40 Aircraft sidewall model used for noise transmission analysis

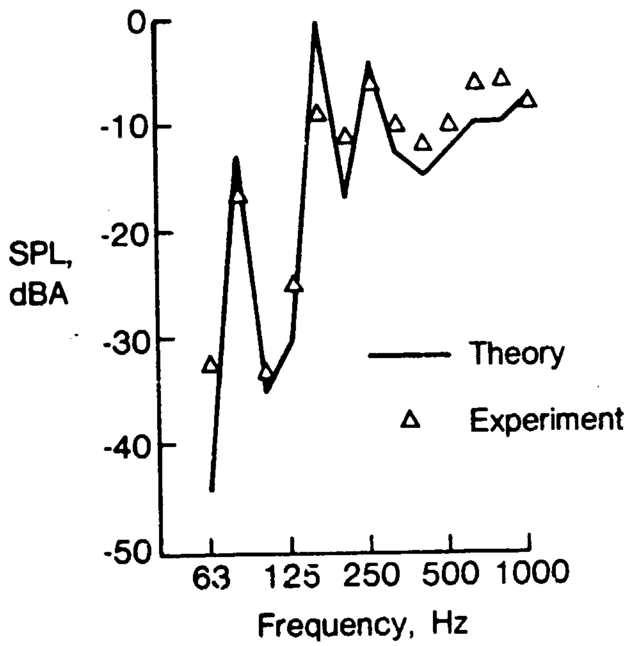

Fig. 41 Normalized interior noise levels in an untreated aircraft

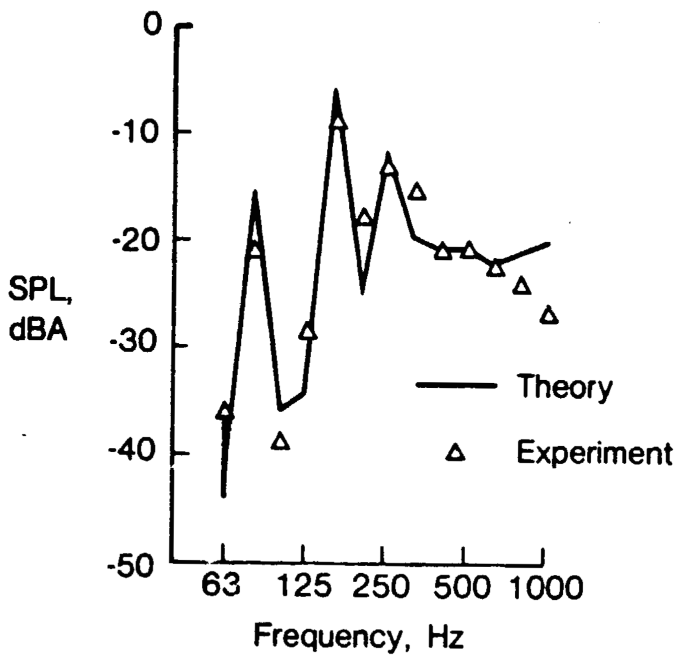

Fig. 42 Normalized interior noise levels in the aircraft treated with a heavy soundproofing package

5.4 New Proposed Acoustic Add-on Treatment

The results of the parametric study were used to design a treatment suitable to provide a more comfortable cabin noise level. The design objective was to reduce the overall interior noise level of the untreated cabin by 17 dB or better at standard cruise power and rpm. To achieve this goal, substantial reduction of noise in the frequency bands of 160 Hz, 250 Hz and 315 Hz (the propeller 2nd, 3rd and 4th harmonics) is needed. A treatment which was found to meet these design objectives utilizes the combination of honeycomb panels, constrained layer damping tape, porous acoustic materials, limp-isolated trim, tuned damping, and modifications in window design. These add-on treatments, except for windows, do not require structural changes of the fuselage.

Engineering judgment is exercised to limit the number of different treatments and weight configurations to design the optimized treatment. Furthermore, the function of this treatment for noise control is based on the condition that noise leakage through various flanking paths can be controlled. Installation of a barrier between the cockpit and the passenger cabin is likely to be needed to reduce noise flanking from the cockpit region. A treated aft bulkhead may also be needed to reduce noise entering from the rear of the aircraft, and improve the interior absorption in the cabin.

The multilayered acoustic treatment for noise control is designed to function as a unit rather than separate individual components. The honeycomb panels are attached to the aircraft skin and cover the regions bounded by frames and longerons. Aluminum honeycomb panels with a core thickness of 0.25 in. and the face plate thicknesses 0.032 in. (region of propeller plane) and 0.016 in. (otherwise) are selected. A constrained layer of damping tape is added to all skin surfaces including the ceiling which is not treated with honeycomb panels. A damping tape suitable for low temperatures should be used. Three layers (one inch thickness of each layer) of porous acoustic materials are added to all accessible surfaces of the cabin. The first two layers are tightly fitted in the regions between stiffeners while the third layer covers all the frames and longerons. Space permitting, a fourth layer should be added to further increase noise reduction and sound absorption capability. In order to design a practical acoustic treatment, a trim that will contain the porous acoustic material and presents an acceptable appearance needs to be installed. The main function of the trim for noise control is to provide additional noise attenuation as the sound enters through the sidewall. A trim which has a low value of stiffness (limp panel concept) and is isolated from the vibration of the main frame structure seems to be best suited for noise control in this aircraft. Difficulties might arise installing the limp-trim panels in the aircraft. Lightweight honeycomb (paper, nomex, etc.) or a layer of other acceptable materials can be attached to the limp septa to increase the stiffness so that installation requirements are satisfied. The basic features of the proposed treatment are shown in Fig. 43. The effectiveness of the new treatment for noise control has been evaluated experimentally using Transmission Loss Apparatus facility described in Sec. 5.2.1. These results are shown in Fig. 44 where a direct comparison of transmission loss for 695A (Acoustic treatment used for production aircraft) and the new treatment is presented. As can be observed from these results, the new treatment provided about 5-10 dB additional noise attenuation in the critical frequency range of 100-500 Hz. However, the new treatment is lighter than the 695A treatment.

Experiments indicate that vibration of frames and longerons are relatively large at the first two propeller blade passage harmonics. These vibrations are strongly coupled to the vibrations of panels and windows. A tuned damper could provide reduction of structural motions at a selected tuned frequency. Such a reduction of structural response could subsequently lead to noise reduction at that frequency. Thus, it is recommended that vibration dampers tuned at the first and the second propeller blade passage harmonics should be used to control the overall vibrations of the fuselage.

The tests and theoretical studies indicate that noise transmitted through windows at the 2nd. 3rd and 4th propeller blade passage harmonic could be a potential cause of high interior noise levels in a treated aircraft. Thus, heavy sidewall treatments might not solve the cabin noise problem if the windows are providing a weak link in noise transmission. The window supports of this aircraft are relatively flexible and the vibrations of the windows are strongly coupled to the vibrations of the supporting skin, longerons and frames. Thus, altering the window design might not improve the noise transmission characteristics if the boundary support conditions are not changed. As the first step to improve window design for noise transmission control, the stiffness of the boundary supports should be increased significantly. The theoretical parametric study suggests that one of the alternatives to increase noise reduction for windows is to increase the thickness of the exterior window pane. For the present design of the proposed treatment, the thickness of the exterior window pane should be increased to 0.5 in.

Figure 45 illustrates the treatment used in various cabin regions. The surface densities and thicknesses of these add-on treatments are given in Table 2. The total weight of this acoustic treatment is about 2% of the gross take-off weight of the aircraft. The proposed treatment is slightly lighter than the standard acoustic-thermal treatment used for this aircraft. However, the new treatment should provide about 4-10 dB more of noise reduction than the standard treatment. Laboratory tests tend to verify these predictions. A comparison of noise transmitted into the untreated cabin (measured) and into the cabin with the proposed treatment (calculated) is shown in Fig. 46. The average calculated overall noise levels of the untreated cabin will be reduced by about 17 dB with the new acoustic treatment.

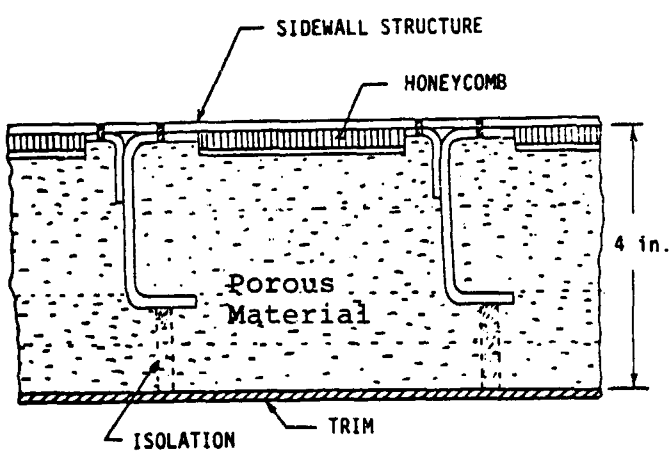

Fig. 43 Basic features of the proposed new acoustic treatment

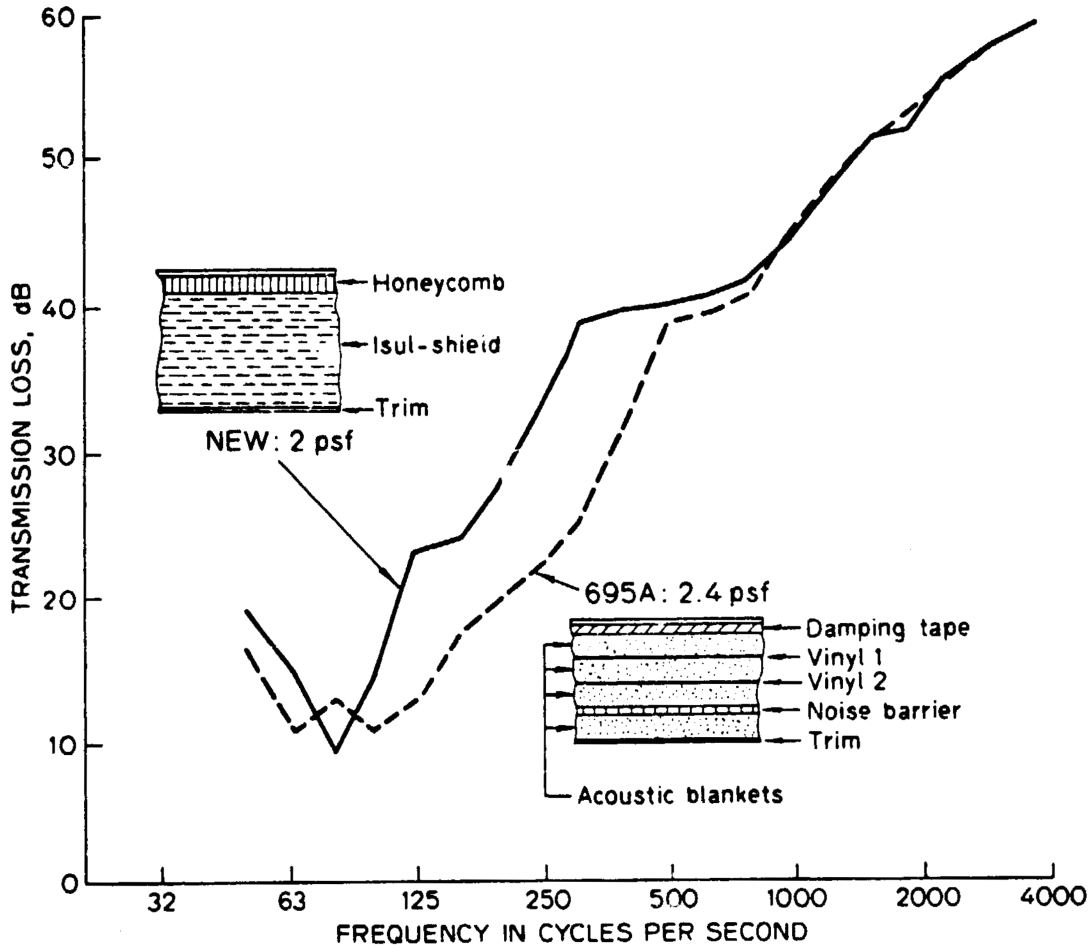

Fig. 44 Transmission loss of a panel for two acoustic treatment configurations

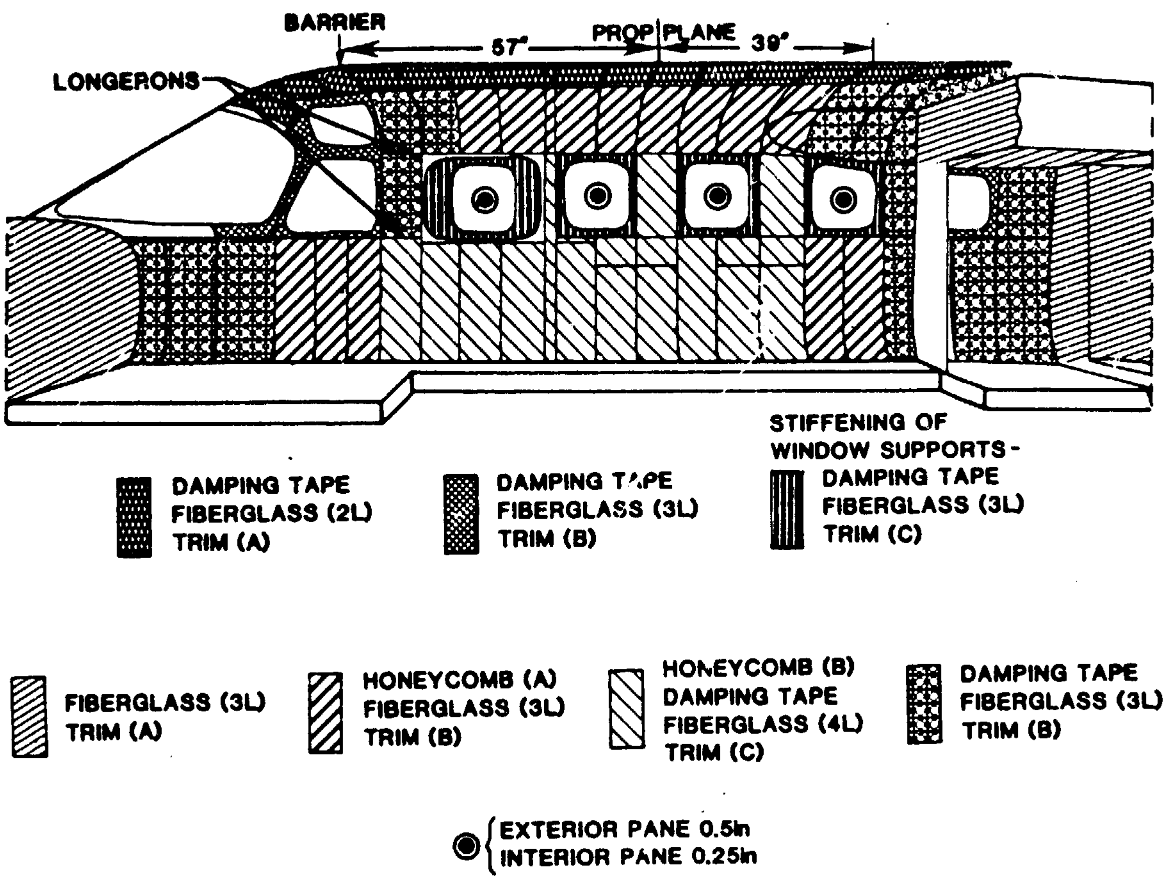

Fig. 45 Distribution of the proposed acoustic treatment for noise control in the aircraft

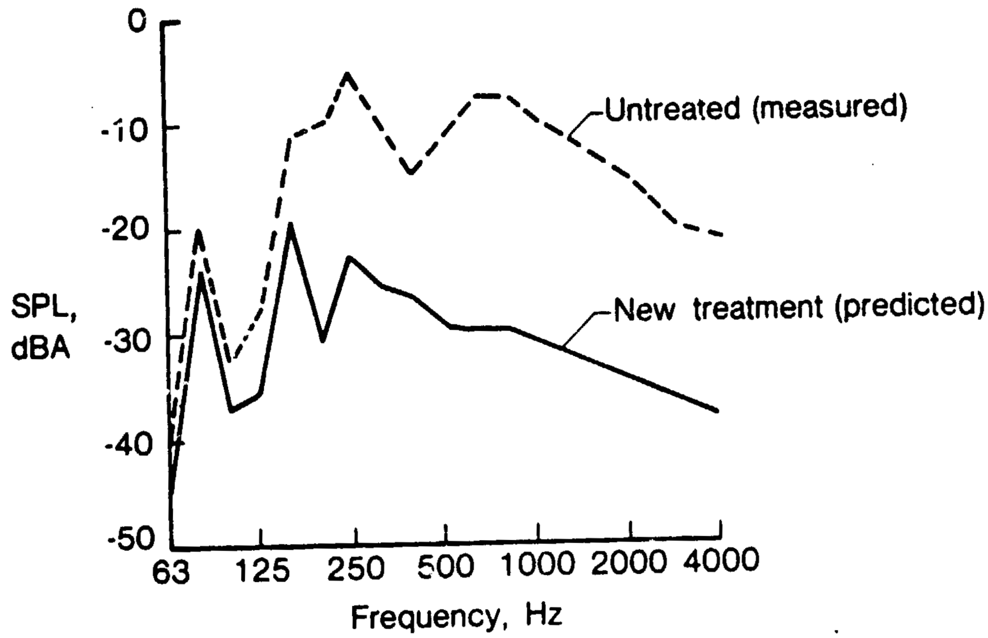

Fig. 46 Interior noise levels of untreated and treated cabins for flight conditions.

Table 1. Description of Add-on Treatments

Treatment	Materials and Specifications	Surface Density lb/ft^2
Honeycomb Panels	Aluminum: facing and core	0.60 - 0.66
Damping Tape	dense foam, adhesive and thin aluminum foil	0.296
Acoustic Blankets	Fiberglass/Thermal	0.04 (one 0.75 in thick layer)
Acoustic Foam	Open cell foam with thin aluminum foil facing	0.125 (one 0.5 in thick layer)
Noise Barriers	101: urethane elastomer bonded to decoupler foam 103: urethane elastomer bonded to acoustical foam 104: urethane elastomer bonded to decoupler and acoustical foams	0.5 - 1.0
Vinyl Septa	Lead impregnated vinyl fabric	0.26
Trim	Stiff masonite panel, noise barrier #101, or lead vinyl	1.06, 1.0, 0.26

TABLE 2 Surface densities and thicknesses of add-on treatments

TREATMENT	SURFACE DENSITY LB / FT^2	THICKNESS IN
Damping Tape	0.25	0.25
Fiberglass (2L)	0.16	1.80
Fiberglass (3L)	0.24	2.70
Fiberglass (4L)	0.32	3.60
Honeycomb (A)	0.40	0.27
Honeycomb (B)	0.68	0.28
Trim (A)	0.30	0.13
Trim (B)	0.40	0.13
Trim (C)	0.75	0.25
Increase in Exterior Window Pane Thickness	1.35	0.25

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15. Supplementary Notes

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16. Abstract

Theoretical and experimental studies were conducted to determine the noise transmission into acoustic enclosures ranging from simple rectangular box models to full scale light aircraft in flight. The structural models include simple, stiffened, curved stiffened, and orthotropic panels and double wall windows. The theoretical solutions were obtained by modal analysis. Transfer matrix and finite element procedures were utilized. Good agreement between theory and experiment has been achieved. An efficient acoustic add-on treatment was developed for interior noise control in a twin engine light aircraft.

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