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NOISE TRANSMISSION THROUGH AN ACOUSTICALLY TREATED AND HONEYCOMB STIFFENED AIRCRAFT SIDEWALL

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Abstract

The noise transmission characteristics of test panels and acoustic treatments representative of an aircraft sidewall are experimentally investigated in the NASA Langley Research Center transmission loss apparatus. The test panels were built to represent a segment sidewall in the propeller plane of a twin-engine, turboprop light aircraft. It is shown that an advanced treatment, which uses honeycomb for structural stiffening of skin panels, has better noise transmission loss characteristics than a conventional treatment. An alternative treatment, using the concept of limp mass and vibration isolation, provides more transmission loss than the advanced treatment for the same total surface mass. Effects on transmission loss of a variety of acoustic treatment materials (acoustic blankets, septa, damping tape, and trim panels) are presented. Damping tape does not provide additional benefit when the other treatment provides a high level of damping. Window units representative of aircraft installations are shown to have low transmission loss relative to a completely treated sidewall.

Introduction

Propeller noise transmitted through the fuselage sidewall of a turboprop aircraft is a major contributor to the noise in the passenger cabin. Conventional sidewall acoustic treatment has been evaluated for a high wing twin-engine turboprop aircraft for which acoustic measurements are available from flight tests as well as from laboratory simulation.[1,2] Flight measurements have indicated that interior noise levels during standard cruise flight conditions are high enough to require improved sidewall treatment.

An improved acoustic treatment, developed through theoretical analysis,[3-5] has been designed to lower cabin overall sound pressure levels by 7 dB (A) or more compared with the conventional treatment. This advanced design utilizes a combination of honeycomb panels, constrained layer damping tape, absorptive acoustic blankets, and an isolated limp trim panel. The purpose of this paper is to evaluate and analyze the noise transmission loss characteristics of the advanced treatment as compared to a conventional treatment and to consider some alternative noise control measures. The noise attenuation characteristics of the individual elements of the sidewall treatments are systematically investigated in the NASA Langley transmission loss apparatus. Measurements are made also of the transmission loss contribution of the window units relative to the total sidewall structure.

Noise Transmission Loss Apparatus

To experimentally establish the noise transmission loss characteristics of the test structure and the add-on treatments, the aircraft sidewall panel is mounted as a partition between two adjacent reverberant rooms which are designated source and receiving room. A schematic of the transmission loss apparatus is depicted in Fig. 1. In the source room which measures 3.35 m by 3.66 m by 3.94 m, a diffuse field is produced by two reference sound power sources that generate random noise over a wide frequency range. Sound from the source room is transmitted into the receiving room only by way of the test panel, which has a sound exposed area of 1.15 m by 1.46 m. The test structure is accommodated by a steel and rubber mounting frame, which is designed for minimum acoustical and structural flanking. A space and time average of the sound pressure levels in each of the rooms is accomplished by means of a windscreen covered microphone mounted at the end of a 0.91 m long rotating boom which has a rotational speed of 16^{-1} revolutions per second. The microphones complete two full rotations during the 32 seconds linear time averaging analysis which is performed by a digital one-third octave band frequency analyzer. To obtain the noise reduction characteristics of the test structure in terms of transmission loss the "Plate Reference Method" is employed which is described in detail in Refs. 2 and 6. The measurements presented in this paper cover a frequency range extending from the 63 Hz one-third octave band up to and including the 4000 Hz one-third octave band. The accuracy of the measurements is within 1.5 dB in the very low frequency bands (<200 Hz) and within 0.5 dB for the higher one-third octave bands.

Test Panel Structure

The test panel structure used in the laboratory measurements was designed after a part of the fuselage sidewall of a twin-engine turboprop aircraft. This high wing aircraft has a maximum takeoff weight of 5080 kg, a standard cabin layout for a pilot and seven passengers and is powered by two turbo shaft engines which are flat rated to a maximum of 611 kW. The synchrophased, three bladed propellers incorporate supercritical airfoil sections and have a fuselage clearance of approximately 0.14 times the prop diameter. For an RPM of 1500 the blade passage frequency is calculated to be 75 Hz and the tip speed 211 m/s.

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It can be concluded that for the tests and treated sidewalls discussed here the damping tape primarily adds mass.

Effect of Acoustic Blankets

Porous acoustic blankets are used to absorb acoustic energy. The sound waves passing through the blanket cause motion of the fibers and the air around the fibers. Acoustic energy is thus converted into heat. At low frequencies, for wavelengths greater than ten times the thickness of the blanket, the acoustic blanket will move as a whole, following the movement of the panel to which it is attached, and the sound absorption mechanism described above cannot take place.[12] For calculating TL at these low frequencies it is therefore assumed that the sound attenuation by the acoustic blanket is zero. Reference 13 presents an empirical power law approximation for the propagation constant α which gives the sound attenuation in a semirigid material per unit thickness:

$$\alpha = k\left[1.64\left(\frac{\rho f}{R}\right)^{-0.595}\right] \tag{8}$$

where $k = 2\pi f/c$ is the wavelength constant, ρ is the gas density, f is the frequency and R is the flow resistivity which is 4.1×10^4 mks rayls/m for the fiberglass blankets (bulk density of the fiberglass is 9.5 kg/m^3). To include the viscous and inertial effects of the gas contained in a soft porous material an effective gas density, as given in Ref. 13, has been used in the calculation. The approximation in equation (8) is valid for values of $\alpha t > 9$ dB, where t is the total thickness of the blanket.[12] A prediction has been made for 3 in. (0.076 m) of acoustic blankets 1 and is shown in Fig. 15 as a solid line. A smooth curve is faired between the zero attenuation at 315 Hz and the 10 dB transmission loss at 630 Hz. The TL shown by the data for frequencies below 315 Hz is thought to be due to damping of the second structural resonance by the acoustic blankets, which are pressed tightly against the skin of the panel. The acoustic blankets have been compressed into the 2 in. (50.8 mm) depth between the stiffeners. Reference 9 indicates the availability of acoustic blankets 2 with better sound absorbing properties at the lower frequencies. This material has been tested for its TL characteristics and two thicknesses are compared with acoustic blankets 1 in Fig. 15. From this figure it can be seen that at the BPF harmonics acoustic blankets 2 perform better in terms of TL than acoustic blankets 1. As they are also more rigid, they are thought to provide more damping to the subpanels and the structure. In conclusion it can be said that in addition to thermal insulation, the acoustic blankets provide sound absorption, sound transmission loss and structural damping which shows that they are a very important component for interior noise control.

Effect of Vinyl Septa and Noise Barrier

The role of the vinyl septa and the noise barrier is to provide additional transmission loss in a limp, resonance-free panel. They also provide damping for the structural resonances of the sidewall. Reference 2 describes the beneficial effect of these noise control materials in a conventional configuration on the TL of the sidewall structure. Double and triple wall resonances might have an adverse effect on the TL in the lower frequency region. Table 1 provides information concerning weights and surface masses.

Effect of the Trim Panel

Adding a trim panel to the basic treatment package serves the purpose of interior decoration, protection for treatment and aircraft skin, thermal insulation, and acoustic attenuation. The effect of adding a stiff trim panel to the conventional treatment was discussed in Ref. 2 and found to be of little benefit in terms of TL. The advanced treatment was designed to have optimum acoustic attenuation characteristics with a light vinyl trim panel installed. Figure 16 shows the effect of an alternative trim panel on the TL of the treated sidewall when compared with the light vinyl trim panel. The different trim panels are described in terms of thickness, mass, and surface mass in Table 1. At the BPF and the first five harmonics only the heavy vinyl trim panel 4 shows an improvement in acoustic attenuation. However, the increase in TL/mass ratio is small at these frequencies and even shows a decrease for frequencies higher than 500 Hz. It can be concluded that a limp trim panel, isolated from the structural members of the sidewall by an acoustic blanket, is the most effective.

Alternative Treatments

In this paper it has been shown experimentally that the advanced treatment has better TL characteristics than the conventional treatment (Fig. 9) especially in the frequency region below 500 Hz which contains the BPF harmonics. At the BPF the first structural resonance of the test panel occurs and as the honeycomb treatment does not provide as much damping as the conventional treatment, the TL for the advanced treatment is very low. Investigating the effects of the individual elements of the treatment packages it appeared that the honeycomb could be replaced by a limp mass and still provide at least the same TL. At the same time, the limp mass would give better damping to the sidewall structure, helping to get a higher TL at the first structural resonance. From a practical point of view this would simplify the installation of the treatment tremendously, especially when the aircraft skin is slightly curved. It was also found that the damping tape is not very functional, as the other treatment will provide similar damping characteristics. It has been shown that the alternative acoustic blanket 2 gives higher TL values in the low frequency region and, as it is more rigid, provides better damping properties for the structure. Lightweight trim panels do not seem to have a great effect on the total TL of the sidewall structure. For practical reasons, a trim panel that can be molded to the contours required in the cabin would be most desirable. In Ref. 2 it was concluded that highest TL is achieved when a limp mass is either directly attached to the skin or as far away from the skin as possible in a double wall mechanical strength improves as temperature decreases. Trim panel 5 consists of compressed fiberglass with a perforated, thin, vinyl cover often used for sound proofing in rooms and buildings.

Laboratory Results

Advanced and Conventional Treatment Comparisons

The transmission loss of the sidewall test structure with the conventional treatment[2] and with the advanced treatment is shown in Fig. 9. Also indicated in this figure are the one-third octave bands in which the blade passage frequency (BPF = 75 Hz) and the first five harmonics occur. Highest excitation levels are experienced for these frequencies with the first harmonic (160 Hz one-third octave band) being most important for A-weighted interior noise level.[1] Figure 9 shows that the transmission loss (TL) of the advanced treatment is as much as 14 dB higher (315 Hz) than the TL of the conventional treatment, with an average gain of 8 dB at the BPF harmonics. A surface mass reduction of 2.25 kg/m^2 has been obtained, making the advanced treatment superior to the conventional treatment in terms of the ratio of transmission loss to surface mass. Two possible disadvantages have to be noted. At the BPF the transmission loss of the sidewall with the advanced treatment is approximately 5 dB less than the transmission loss of the sidewall with the conventional treatment. Reasons for this will be examined later. Also, the advanced treatment is about 2 inches thicker than the conventional treatment due to the use of thicker acoustic blankets. In the following sections the effect of each of the elements on the transmission loss of the treated sidewall will be discussed.

Effect of Honeycomb Treatment

High stiffness to mass ratio materials such as honeycomb are used to raise the fundamental frequency of a panel such that it will no longer coincide with frequencies of highest excitation.[3,5,7,8] In addition, treatments such as acoustic blankets are more effective at these higher frequencies.[2] The resonance frequencies of each of the 10 subpanels of the sidewall structure (Fig. 3) were established in Ref. 2 and are presented in Table 2. The resonance frequencies of the whole structure, including the skin, the structural members and the supporting frame along with the critical frequency of the aluminum skin are also given in Table 2. Adhering the honeycomb to the skin makes the area within the boundaries very stiff relative to the boundaries themselves (only aluminum skin). It therefore seems justified to assume simply supported edge conditions. The resonance frequency then is given by

$$f_r = \frac{\pi}{2}\left(\frac{1}{a^2} + \frac{1}{b^2}\right)\sqrt{\frac{B}{m}} \qquad (1)$$

where B is the bending stiffness, m is the surface mass, and a and b are the (sub) panel dimensions. The critical frequency, which is the lowest frequency at which the acoustic wavelength matches the bending wavelength in the panel, is given by

$$f_c = \frac{c^2}{2\pi}\sqrt{\frac{m}{B}} \qquad (2)$$

where c is the speed of sound in air. The bending stiffness for a homogeneous panel is defined by

$$B = \frac{Et^3}{12(1 - \nu^2)} \qquad (3)$$

where E is the elasticity modulus, t is the thickness of the panel and ν is Poisson's ratio. The bending stiffness of the honeycomb panel is, assuming the core has no flexural rigidity

$$B = \frac{E}{1 - \nu^2}\left[\frac{t_1^3}{12} + \frac{t_2^3}{12} + \frac{t_1 t_2}{t_1 + t_2}\left(\frac{t_1}{2} + \frac{t_2}{2} + d\right)^2\right] \qquad (4)$$

where t_1 is the thickness of the skin, t_2 is the thickness of the facing plate, and d is the core thickness. These formulas have been shown to provide reasonable agreement with measured frequencies of honeycomb stiffened panels similar to the panels of the present study.[10] The resonance frequencies and critical frequency are calculated for the honeycomb subpanels and tabulated, along with a few experimental values in Table 2.

The transmission loss of the bare test panel and the panel with the honeycomb applied are compared in Fig. 10. In the low frequency region (<315 Hz), where the BPF and the strongest harmonics occur, an average increase in transmission loss of 4 dB is observed due to the honeycomb application. This increase in TL is not necessarily due to an increase in the stiffness of the panels since mass has also been added. To investigate the mass effect, the TL of each panel is compared to its mass law TL and the difference is plotted in Fig. 11. Mass law TL was calculated using the skin mass for the bare panel, and the mass of skin and honeycomb for the honeycomb stiffened panel. Figure 11 shows that in the frequency region at and below the 315 Hz one-third octave band, the two sidewall configurations have the same deviation from mass law except for the 63 Hz one-third octave band, which shows a 5 dB larger TL for the honeycomb stiffened panel. This suggests that the same increase in TL might be achieved by applying a limp mass to the skin of the sidewall. The ΔTL for the test panel with rubber panels attached to the skin is also plotted in Fig. 11. The test data is taken from Ref. 2. The ΔTL for the rubber treated panel follows mass law much more closely at frequencies below 315 Hz than does the ΔTL for the other two panels. This can be explained by the damping properties of the rubber which raises the ΔTL at the second structural resonance of the sidewall plus supporting frame (200 Hz) up to its mass law level. The first structural resonance (80 Hz) is lowered in frequency (opposite to the honeycomb application) and some damping is provided. At the 63 Hz one-third octave band the rubber treated sidewall provides the least TL of the three configurations but this frequency is below the BPF of the propeller. These results indicate that at least the same or more TL can be obtained with limp mass applications as with a honeycomb treated sidewall in a laboratory test with a diffuse source sound field. The limp mass-like behavior of the honeycomb at low frequencies (<315 Hz) can be explained by the hypothesis that it is applied to the sidewall skin, thus adding stiffness to the subpanels but not to the total sidewall panel. As no stiffness is added, the first structural resonance will occur at about the same frequency and the frequency region above the first structural resonance (80 Hz) will be mass controlled. In the 63 Hz one-third octave band, which is below the fundamental resonance frequency in the stiffness controlled region, the honeycomb does add stiffness to the sidewall panel and thus raising its TL.

Previous tests of honeycomb stiffening used a horn noise source and an aircraft fuselage,[10] and showed that the honeycomb stiffening provided more noise attenuation than an equal weight of limp mass, at low frequencies (<200 Hz). The effect of honeycomb may be associated with the nature of the source field, the dynamics of the sidewall structure, the method of gluing the honeycomb, or the attachment of the honeycomb only to the skin but not to the stiffening frames. Determination of the governing effects appears to be important, if the full potential benefits of honeycomb are to be realized.

Referring again to Fig. 10, it is shown that between the 315 Hz and the 1000 Hz one-third octave bands the TL of the honeycomb treated sidewall is less than the TL of the bare sidewall structure. As shown in Table 2, the resonances of the subpanels of the honeycomb stiffened sidewall fall in this frequency range. The critical frequency of the honeycomb stiffened panel occurs in the 1000 Hz one-third octave band and coincidence resonances take place at this and higher frequencies as a function of the angle of sound incidence.

To investigate the effect of the honeycomb in the frequency region above 315 Hz a 1.15 m by 1.46 m unstiffened aluminum panel with the same thickness as the sidewall skin was treated with the same type honeycomb and tested in the TL apparatus. The measured results are shown in Fig. 12. The first two structural resonances now appear to occur in the 200 Hz and 400 Hz one-third octave bands. This would imply that the honeycomb has stiffened the bare aluminum panel more effectively than the combination of frames and honeycomb treated subpanels found in the sidewall panel for which the first two structural resonances were found at 80 Hz and 200 Hz (Table 2, Fig. 10). The transmission loss coefficient of an infinite panel, taking into account coincidence effects, is given according to Ref. 11

$$\tau(\theta) = \left\{\left[1 + \eta\left(\frac{\omega m}{2\rho c}\cos\theta\right)\left(\frac{\omega^2 B}{c^4 m}\sin^4\theta\right)\right]^2 + \left[\left(\frac{\omega m}{2\rho c}\cos\theta\right)\left(1 - \frac{\omega^2 B}{c^4 m}\sin^4\theta\right)\right]^2\right\}^{-1} \quad (5)$$

where the transmission loss coefficient τ is a function of the angle of sound incidence θ, η is the loss factor of the panel, ω is the circular frequency and ρ is the air density. Integrating over all angles of sound incidence up to a limiting angle θ_{lim} yields the average transmission loss coefficient

$$\bar{\tau} = \frac{\int_0^{\theta_{lim}} \tau(\theta)\cos\theta\sin\theta\, d\theta}{\int_0^{\theta_{lim}} \cos\theta\sin\theta\, d\theta} \quad (6)$$

The average transmission loss coefficient is related to the transmission loss by

$$TL = 10\log\left(\frac{1}{\bar{\tau}}\right) \quad (7)$$

Using the surface mass and bending stiffness equation (4) of the aluminum/honeycomb combination and an estimated loss factor of $\eta = 0.015$, the TL is predicted for the 1.15 m by 1.46 m panel and plotted in Fig. 12 along with its mass law. Very reasonable agreement between measurement and prediction is obtained.

The TL of the unstiffened honeycomb treated aluminum panel is compared with the honeycomb treated aircraft sidewall panel in Fig. 13, where the important resonances are indicated. The shift in structural resonances (structure plus supporting frame) can easily be seen to result in different TL values at those frequencies. Between 315 Hz and 1000 Hz the TL of the honeycomb treated aircraft sidewall is lower due to the resonances of the subpanels. Above the critical frequency of 1000 Hz both TL curves are very close and thus can be favorably compared with the predictions of the basic theory.

Effect of Damping Tape

The purpose of the damping tape is to suppress the damping controlled resonant structural vibrations of the sidewall structure and thus to prevent reradiation of noise on the receiver side. It has been shown in Ref. 2 that damping tape 1, when applied directly to the subpanels of the structure, will effectively damp vibrational resonances except for the first structural resonance of the entire structure. The transmission loss curve of the sidewall structure follows the mass law of the total surface mass of the skin and the damping tape. Figure 14 shows that when damping tape 2 is applied to the honeycomb treated sidewall, damping is provided for the second structural resonance (200 Hz), the subpanel resonances between 315 Hz and 1000 Hz and coincidence resonances above 1000 Hz. The effect of the damping tape at the BPF and the first five harmonics, however, is minimal. Applying damping tape 2 to the structural stiffener members of the sidewall did not alter the total damping nor the transmission loss. Measurements also indicated that applying damping tape 2 on a sidewall with other acoustic treatment does not have a beneficial effect when damping is already provided by the other treatment components.

It can be concluded that for the tests and treated sidewalls discussed here the damping tape primarily adds mass.

Effect of Acoustic Blankets

Porous acoustic blankets are used to absorb acoustic energy. The sound waves passing through the blanket cause motion of the fibers and the air around the fibers. Acoustic energy is thus converted into heat. At low frequencies, for wavelengths greater than ten times the thickness of the blanket, the acoustic blanket will move as a whole, following the movement of the panel to which it is attached, and the sound absorption mechanism described above cannot take place.[12] For calculating TL at these low frequencies it is therefore assumed that the sound attenuation by the acoustic blanket is zero. Reference 13 presents an empirical power law approximation for the propagation constant α which gives the sound attenuation in a semirigid material per unit thickness:

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where $k = 2\pi f/c$ is the wavelength constant, ρ is the gas density, f is the frequency and R is the flow resistivity which is 4.1×10^4 mks rayls/m for the fiberglass blankets (bulk density of the fiberglass is 9.5 kg/m^3). To include the viscous and inertial effects of the gas contained in a soft porous material an effective gas density, as given in Ref. 13, has been used in the calculation. The approximation in equation (8) is valid for values of $\alpha t > 9$ dB, where t is the total thickness of the blanket.[12] A prediction has been made for 3 in. (0.076 m) of acoustic blankets 1 and is shown in Fig. 15 as a solid line. A smooth curve is faired between the zero attenuation at 315 Hz and the 10 dB transmission loss at 630 Hz. The TL shown by the data for frequencies below 315 Hz is thought to be due to damping of the second structural resonance by the acoustic blankets, which are pressed tightly against the skin of the panel. The acoustic blankets have been compressed into the 2 in. (50.8 mm) depth between the stiffeners. Reference 9 indicates the availability of acoustic blankets 2 with better sound absorbing properties at the lower frequencies. This material has been tested for its TL characteristics and two thicknesses are compared with acoustic blankets 1 in Fig. 15. From this figure it can be seen that at the BPF harmonics acoustic blankets 2 perform better in terms of TL than acoustic blankets 1. As they are also more rigid, they are thought to provide more damping to the subpanels and the structure. In conclusion it can be said that in addition to thermal insulation, the acoustic blankets provide sound absorption, sound transmission loss and structural damping which shows that they are a very important component for interior noise control.

Effect of Vinyl Septa and Noise Barrier

The role of the vinyl septa and the noise barrier is to provide additional transmission loss in a limp, resonance-free panel. They also provide damping for the structural resonances of the sidewall. Reference 2 describes the beneficial effect of these noise control materials in a conventional configuration on the TL of the sidewall structure. Double and triple wall resonances might have an adverse effect on the TL in the lower frequency region. Table 1 provides information concerning weights and surface masses.

Effect of the Trim Panel

Adding a trim panel to the basic treatment package serves the purpose of interior decoration, protection for treatment and aircraft skin, thermal insulation, and acoustic attenuation. The effect of adding a stiff trim panel to the conventional treatment was discussed in Ref. 2 and found to be of little benefit in terms of TL. The advanced treatment was designed to have optimum acoustic attenuation characteristics with a light vinyl trim panel installed. Figure 16 shows the effect of an alternative trim panel on the TL of the treated sidewall when compared with the light vinyl trim panel. The different trim panels are described in terms of thickness, mass, and surface mass in Table 1. At the BPF and the first five harmonics only the heavy vinyl trim panel 4 shows an improvement in acoustic attenuation. However, the increase in TL/mass ratio is small at these frequencies and even shows a decrease for frequencies higher than 500 Hz. It can be concluded that a limp trim panel, isolated from the structural members of the sidewall by an acoustic blanket, is the most effective.

Alternative Treatments

In this paper it has been shown experimentally that the advanced treatment has better TL characteristics than the conventional treatment (Fig. 9) especially in the frequency region below 500 Hz which contains the BPF harmonics. At the BPF the first structural resonance of the test panel occurs and as the honeycomb treatment does not provide as much damping as the conventional treatment, the TL for the advanced treatment is very low. Investigating the effects of the individual elements of the treatment packages it appeared that the honeycomb could be replaced by a limp mass and still provide at least the same TL. At the same time, the limp mass would give better damping to the sidewall structure, helping to get a higher TL at the first structural resonance. From a practical point of view this would simplify the installation of the treatment tremendously, especially when the aircraft skin is slightly curved. It was also found that the damping tape is not very functional, as the other treatment will provide similar damping characteristics. It has been shown that the alternative acoustic blanket 2 gives higher TL values in the low frequency region and, as it is more rigid, provides better damping properties for the structure. Lightweight trim panels do not seem to have a great effect on the total TL of the sidewall structure. For practical reasons, a trim panel that can be molded to the contours required in the cabin would be most desirable. In Ref. 2 it was concluded that highest TL is achieved when a limp mass is either directly attached to the skin or as far away from the skin as possible in a double wall configuration. Taking all these considerations into account, two alternative treatment packages were designed, tested in the TL apparatus, and compared with the TL results of the advanced treatment in Fig. 17. The total surface mass of the alternative treatments and advanced treatment are about the same and the elements are given in Fig. 17. This figure shows that alternative treatment 1 (9.7 kg/m^2) gives an improvement in TL of 1.5 dB to 8 dB over the advanced treatment at the BPF and its first five harmonics. The addition of mass rather than stiffness (like the honeycomb) shifts the first structural resonance down to the 63 Hz one-third octave band which is below the BPF. The two acoustic blankets 2 provide damping, absorption, transmission loss, and isolation of the trim panels. The first trim panel is limp to provide mass and the second trim panel has some rigidity so it can be molded to the requirements and necessities in the cabin. A comparison of this alternative treatment and the advanced treatment with the TL for the bare sidewall panel is depicted in Fig. 18.

Windows

Installation of the double pane windows in the bare sidewall structure increases the TL as the TL of the windows is higher than the TL of the aluminum skin. This is illustrated in Fig. 19. The window units dampen the structural resonances of the bare sidewall, so that the increase in TL is greatest in the 100 Hz and 200 Hz one-third octave bands. Although the windows increase the bare sidewall TL, installation of the window units in a treated sidewall has the opposite effect. This is pictured in Fig. 20 for two different sidewall treatments. To determine the effect of windows on the sidewall TL, the windows were left uncovered in one test and were covered with two layers of heavy noise barrier material in another. Although differences for covered and uncovered windows are relatively small (<4 dB) improvements in window design might be desirable. Window TL can be improved in a number of ways including the use of thicker and/or curved panes, different distances between panes, smaller windows, vibration isolation, Helmholtz resonators, and depressurization of the air in between the panes.[2,4,14-17]

Conclusions

The noise transmission loss characteristics of an aircraft test panel having conventional, advanced and two alternative sidewall treatments were experimentally investigated. For the aircraft the highest excitation levels are generated at the propeller blade passage frequency (75 Hz) and its first five harmonics defining a frequency range of interest from the 80 Hz up to and including the 500 Hz one-third octave band. From the results of the noise transmission loss test discussed in this paper the following conclusions can be derived:

(1) Honeycomb stiffening of the skin panels raised the subpanel resonance frequencies, but the increase of TL associated with honeycomb installation was close to the increase predicted by mass law for installation of an equivalent amount of mass.

(2) Damping tape provides little beneficial effects when combined with other treatments that provide damping.

(3) Highest transmission loss is achieved by attaching a limp mass directly to the skin or locating it as far away from the panel as possible in a double wall configuration.

(4) An alternative treatment, consisting of acoustic blankets, soft trim panel and stiff trim panel performed better than the other treatments in terms of noise transmission loss.

(5) Windows provide extra transmission loss for a bare sidewall but represent a sound leak when installed in a completely treated sidewall.

The behavior of the honeycomb observed in these tests is thought to be associated with the application of the honeycomb directly to the skin without extending it to the framework. Improved low frequency transmission loss might be obtained by rigidly coupling the structural frames of the sidewall to the honeycomb on the skin. This is recommended for further investigation. The alternative treatment may be more practical than the advanced treatment as no honeycomb panels have to be permanently adhered to the skin, which is especially difficult when the skin is curved. The acoustic blankets and soft trim panel provide vibration isolation and the stiff trim panel can be molded to the specifications of the aircraft cabin.

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Table 1. Thickness, mass, and surface mass of the elements used in noise transmission loss tests.

Component	Thickness (mm)	Mass (kg)	Surface mass (kg/m^2)
Bare sidewall structure	N/A	18.43	N/A
Skin	1.61	N/A	4.66
Honeycomb	28.2	4.32	3.31
Damping tape 1	6.35	2.01	1.54
Damping tape 2	.40	1.51	1.16
Acoustic blanket 1	25.4	.40	.24
Acoustic blanket 2	76.2	3.00	1.92
Vinyl septum 1	1.02	2.89	1.79
Vinyl septum 2	.61	2.21	1.37
Vinyl septum 3	2.78	8.97	4.88
Rubber mass	6.35	12.68	8.13
Noise barrier	8.26	9.21	4.96
Trim panel 1	1.42	4.54	2.49
Trim panel 2	2.79	8.97	4.88
Trim panel 3	6.35	1.76	.98
Trim panel 4	3.81	2.33	1.28
Trim panel 5	6.35	3.43	1.87
Conventional treatment	87.9	20.1	11.9
Advanced treatment	129.1	16.3	9.6
Alternative treatment 1	133.4	16.4	9.7
Alternative treatment 2	130.8	16.9	10.0

Table 2. Resonance frequencies of the bare and honeycomb treated sidewall structure.

Structure component	Bare sidewall	Honeycomb treated sidewall	
	Experimental	Theoretical (Eqs. (1) + (4))	Experimental
Sidewall structure	92 and 185		
Subpanel A	72	307	330
Subpanel B	155	1036	
Subpanel C	71	307	
Subpanel D	133	635	734
Subpanel E	140	985	
Subpanel F	140	767	
Subpanel G	140	968	
Subpanel H	141	1002	
Subpanel K	142	756	
Subpanel L	125	561	605
Critical frequency	7865	1115 (Eqs. (2) + (4))	1000

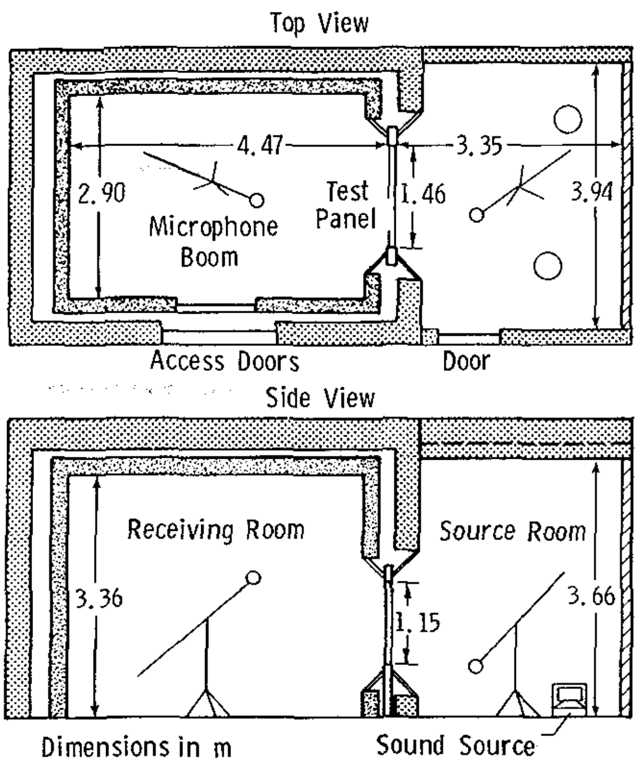

Fig. 1 Schematic of noise transmission loss apparatus.

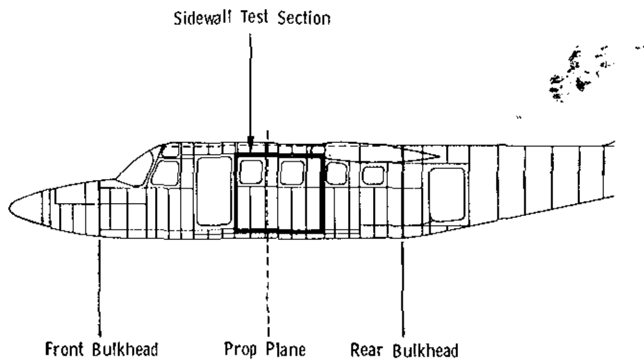

Fig. 2 Location of the sidewall test section on the fuselage of the light twin-engine turboprop.

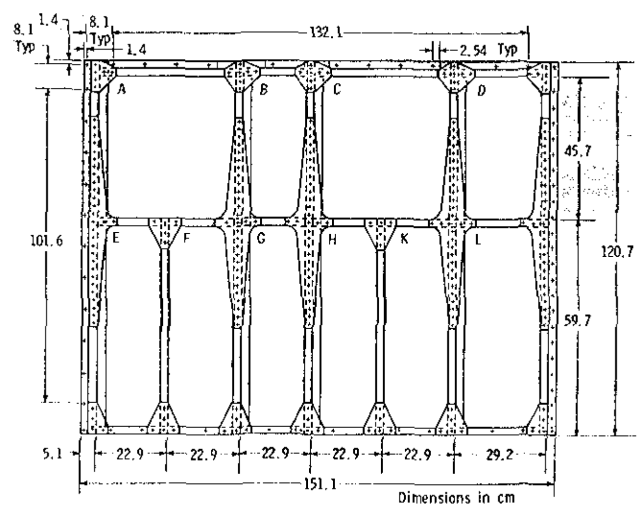

Fig. 3 Engineering drawing of the test structure.

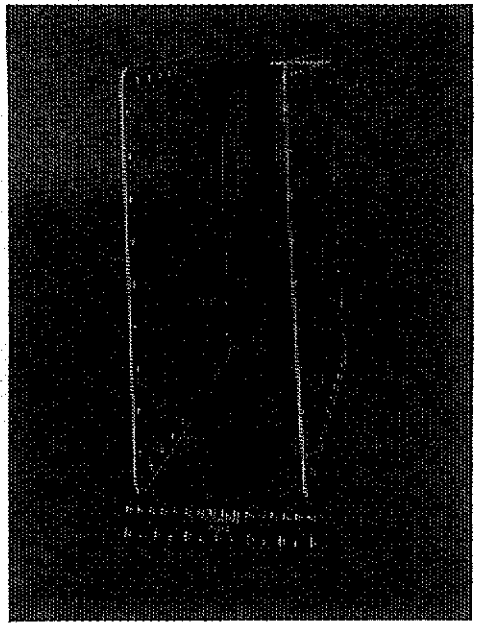

Fig. 4 Components of the laboratory double pane window unit.

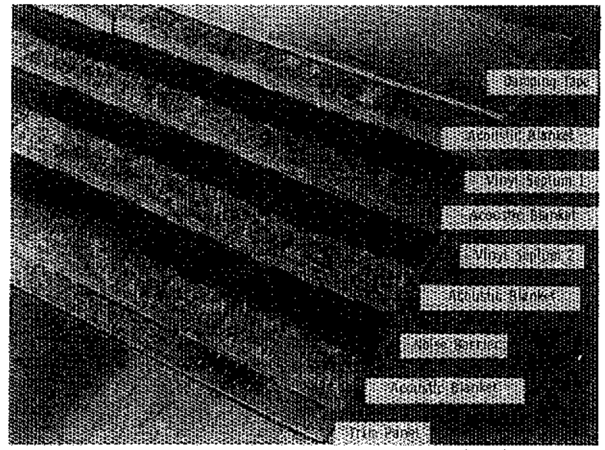

Fig. 6 Elements of the conventional treatment package.

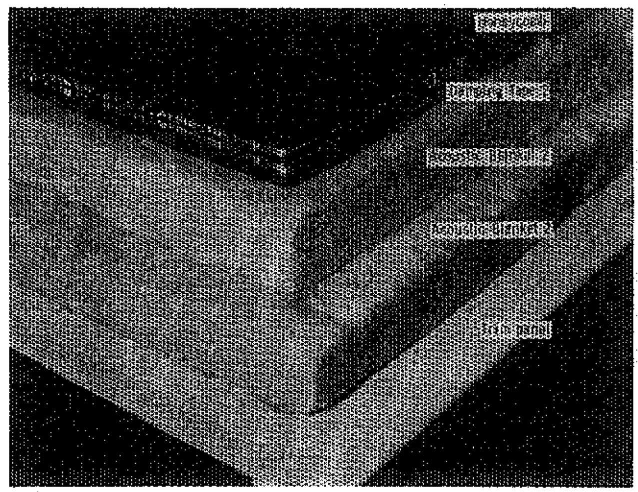

Fig. 7 Elements of the advanced treatment package.

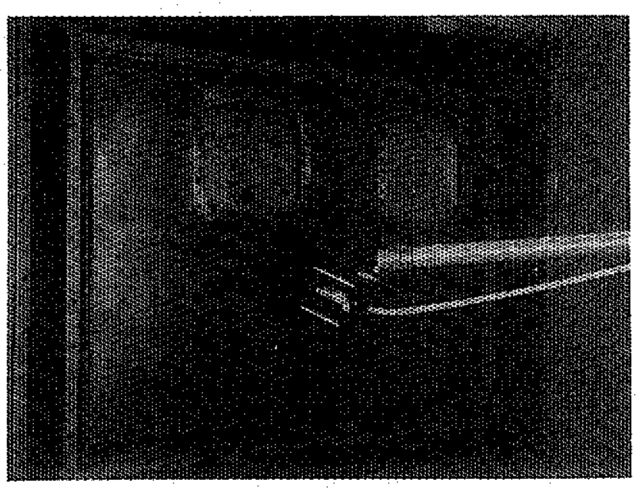

Fig. 5 Sidewall test panel as viewed from the source room.

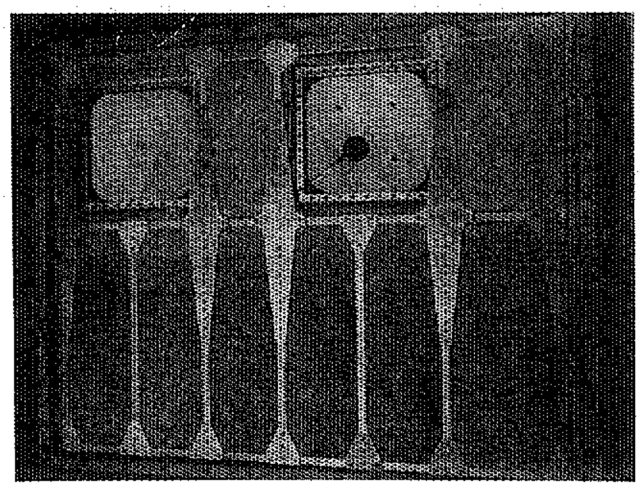

Fig. 8 Sidewall test panel showing partial fiberglass treatment (view from the receiving room).

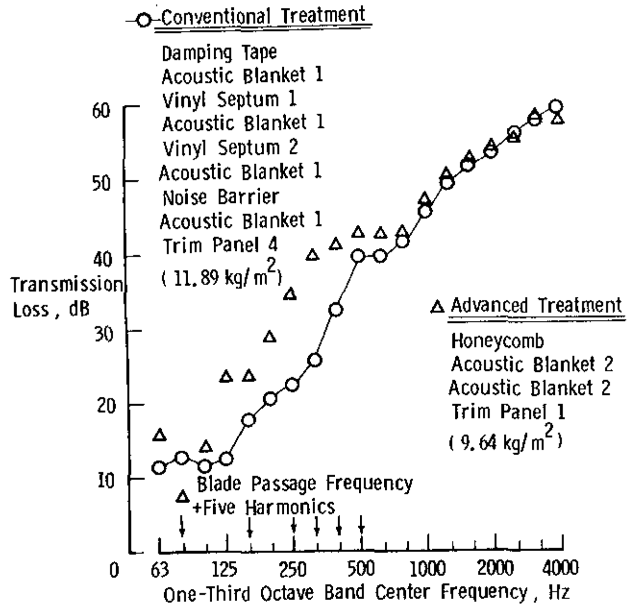

Fig. 9 Transmission loss of the conventional and advanced treatment applied to the sidewall test panel.

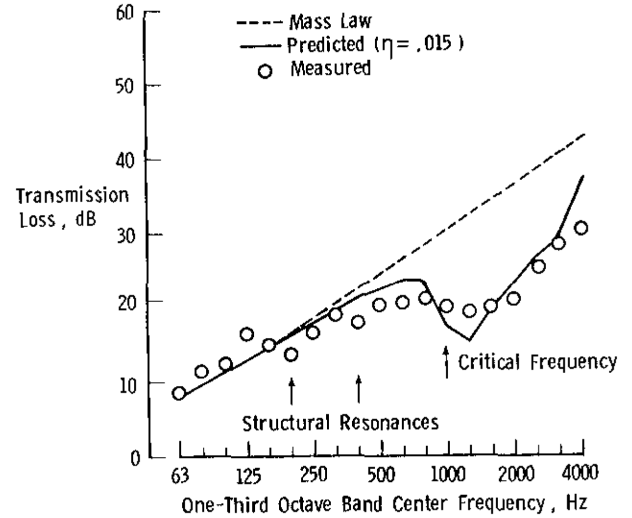

Fig. 12 Measured transmission loss of honeycomb treated aluminum panel compared with theoretical prediction and mass law.

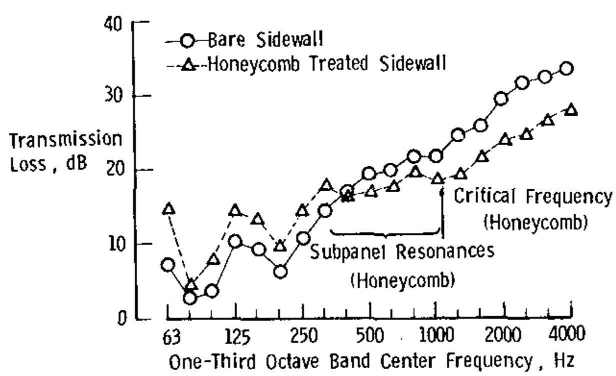

Fig. 10 Transmission loss of the bare sidewall test panel and the honeycomb treated sidewall.

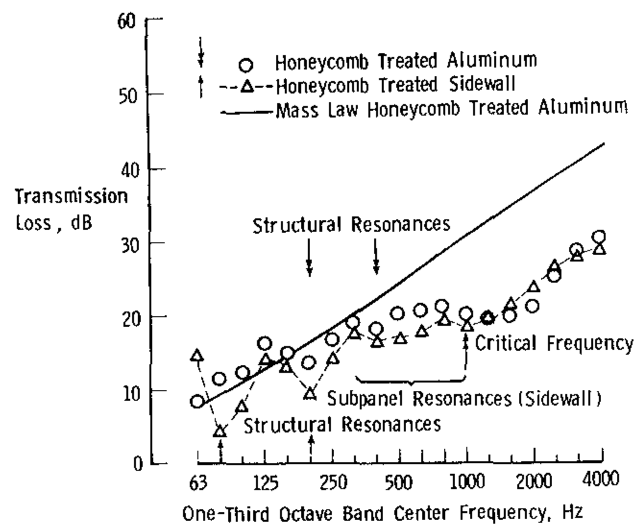

Fig. 13 Transmission loss of honeycomb treated aluminum and honeycomb treated sidewall panel.

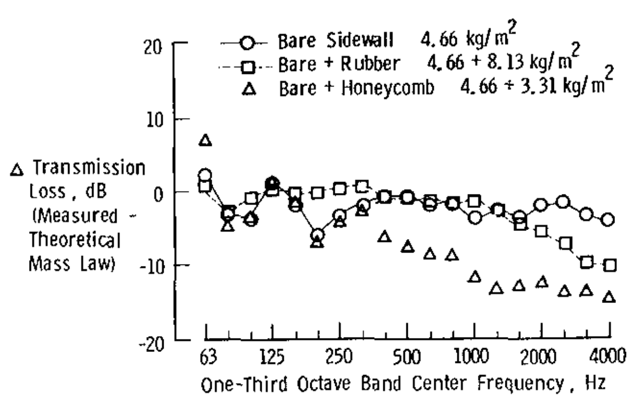

Fig. 11 Difference between measured transmission loss and mass law for the bare, rubber treated and honeycomb treated sidewalls.

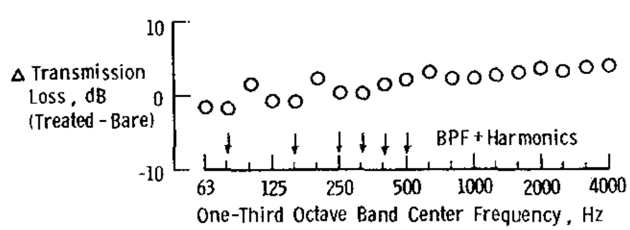

Fig. 14 Difference between transmission loss of the honeycomb sidewall treated with damping tape 2 and the bare honeycomb sidewall.

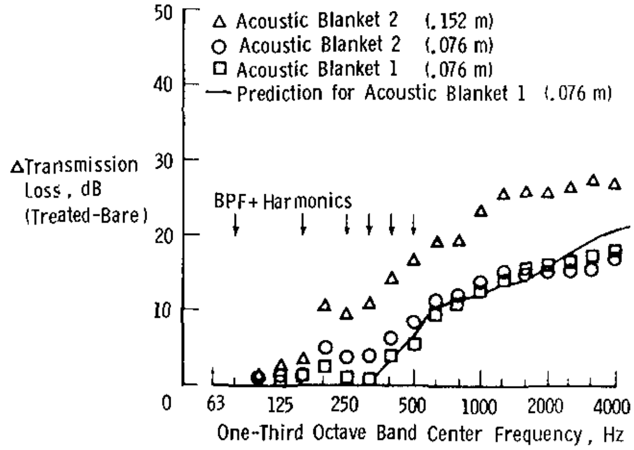

Fig. 15 Measured and predicted transmission loss of acoustic blanket 1 and transmission loss for two thicknesses of acoustic blanket 2.

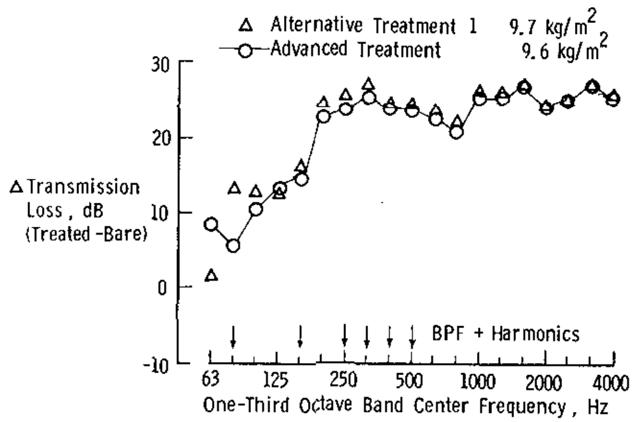

Fig. 18 Difference between transmission loss of alternative treatment 1 and advanced treatment packages.

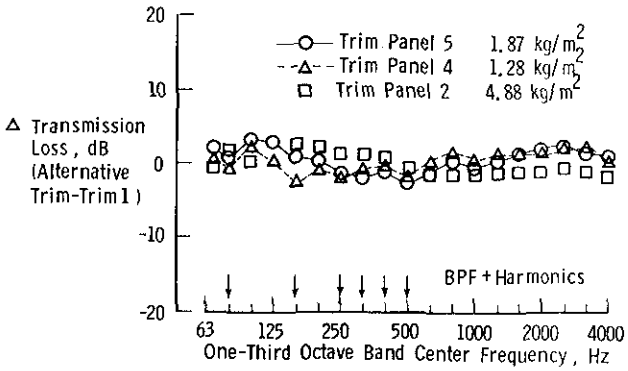

Fig. 16 Difference between transmission loss of three alternative trim panels and light vinyl trim panel 1.

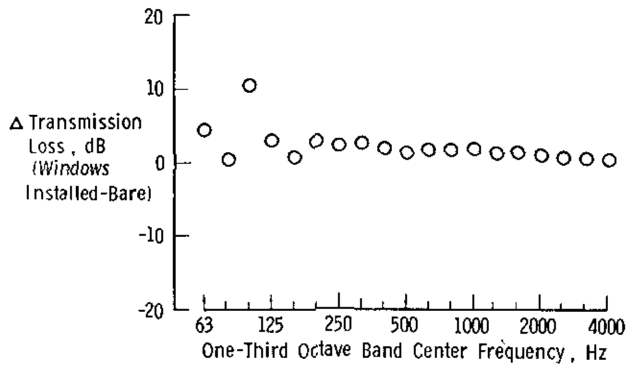

Fig. 19 Difference between transmission loss of the bare sidewall panel with and without the window units installed.

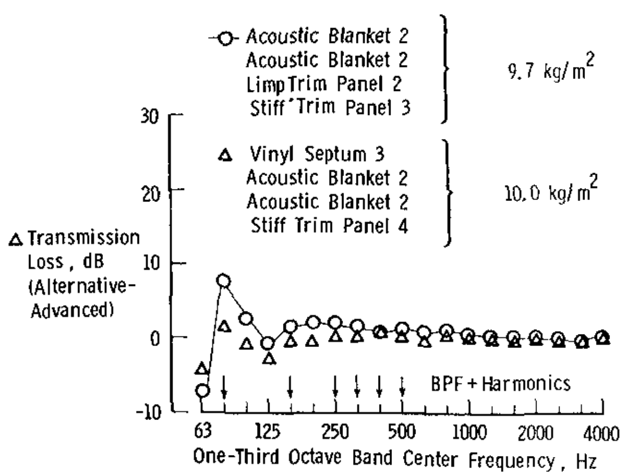

Fig. 17 Difference between transmission loss of two alternative treatments and the advanced treatment package (9.6 kg/m²).

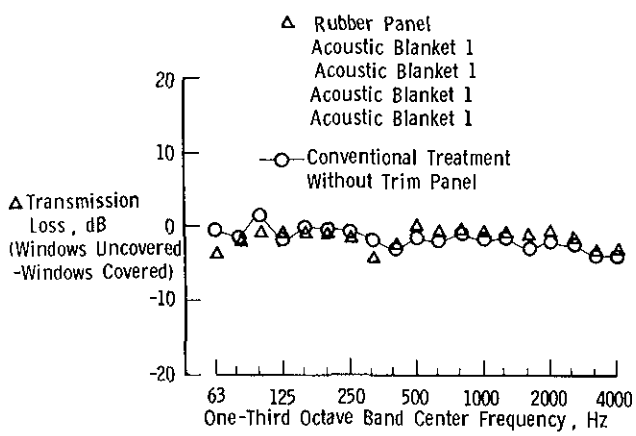

Fig. 20 Difference between transmission loss with and without the windows covered for two acoustic treatments on the sidewall panel.