



AIAA 2002-2416

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**8th AIAA/CEAS Aeroacoustics Conference
June 16–18, 2002/Breckenridge, CO**

Noise Transmission Characteristics of Damped Plexiglas Windows

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Most general aviation aircraft utilize single layer plexiglas material for the windshield and side windows. Adding noise control treatments to transparent panels is a challenging problem. In this paper, damped plexiglas windows are evaluated for replacement of conventional windows in general aviation aircraft to reduce the structure-borne and airborne noise transmitted into the interior. In contrast to conventional solid windows, the damped plexiglas window panels are fabricated using two or three layers of plexiglas with transparent viscoelastic damping material sandwiched between the layers. Results from acoustic tests conducted in the NASA Langley Structural Acoustic Loads and Transmission (SALT) facility are used to compare different designs of the damped plexiglas panels with solid windows of the same nominal thickness. Comparisons of the solid and damped plexiglas panels show reductions in the radiated sound power of up to 8 dB at low frequency resonances and as large as 4.5 dB over a 4000 Hz bandwidth. The weight of the viscoelastic treatment was approximately 1 % of the panel mass. Preliminary FEM/BEM modeling shows good agreement with experimental results for radiated sound power.

Introduction

General aviation aircraft windows are generally manufactured from a solid plexiglas material. In this paper, damped multi-layer plexiglas panels are evaluated as replacements for the solid windows to reduce the structure-borne and air-borne noise transmitted into the aircraft. Laminated glass has shown promise in enhancing noise transmission loss in automobile and architectural applications,¹ but little work has been demonstrated for plexiglas.

Damped plexiglas window panels were fabricated using two or three layers of plexiglas with transparent viscoelastic damping material sandwiched between the layers. Two sets of flat panels were fabricated. The first set was nominally 1/4" thick, representative of the front windshield, and consisted of a solid panel and four different layups for the damped plexiglas panels. The second set was nominally 1/8" thick, representative of side windows, and consisted of a solid panel and three different layups for the damped plexiglas panels. Within a given set of panels, the overall panel weight varied by less than ten percent. For this study, the panels were flat for ease of fabrication and testing.

Vibration and acoustic response measurements with the panels installed in the NASA Langley Structural Acoustic Loads and Transmission (SALT) Facility² provide the data for comparison of relative performance of the various panel layups. Ongoing work includes analytical Finite Element Modeling (FEM), Boundary Element Modeling (BEM) and layer optimization. Preliminary FEM/BEM results will be presented.

Table 1 Nominal 1/4" Inch Test Panel Configurations

Panel Label	Layer 1 (in)	Layer 2 (in)	Layer 3 (in)	Total (in)	Weight (lb)
0.216	0.216			0.216	10.78
0.114-0.114	0.114	0.114		0.232	11.15
0.030-0.175-0.030	0.030	0.175	0.030	0.239	11.65
0.060-0.114-0.060	0.060	0.114	0.060	0.238	11.85
0.175-0.060	0.175	0.060		0.239	11.53

Table 2 Nominal 1/8" Test Panel Configurations

Panel Label	Layer 1 (in)	Layer 2 (in)	Layer 3 (in)	Total (in)	Weight (lb)
0.114	0.114			0.114	5.67
0.060-0.060	0.060	0.060		0.122	N/A
0.030-0.060-0.030	0.030	0.060	0.030	0.124	6.15
0.030-0.085	0.030	0.085		0.117	5.48

Test Panel Description

A series of test panels were constructed using multiple layers of plexiglas and damping treatment. They can be classified into two categories: panels with nominal thickness of 1/4", and panels with nominal thickness of 1/8". In each thickness category several composite layups were made to compare relative performance to a solid panel of the same nominal thickness. The panels are all 48" x 24" and are mounted in a test frame which results in an exposed panel dimensions of 37.75" x 15.25". The nominal 1/4" test panel layups are shown in Table 1. A layer of 0.002" damping treatment is sandwiched between plexiglas layers. The weight shown in column 6 is for the total panel not the exposed area. The damping treatment is 0.002" thick, has a density of about 80 % that of plexiglas, and thus adds about 1 % to the overall panel weight. The nominal 1/8" test panel layups are shown in Table 2.

Test Facility

The Structural Acoustic Loads and Transmission (SALT)² facility located at the NASA Langley Research Center is shown in Figure 1. This facility consists of an anechoic chamber, a reverberation chamber, and a trans-

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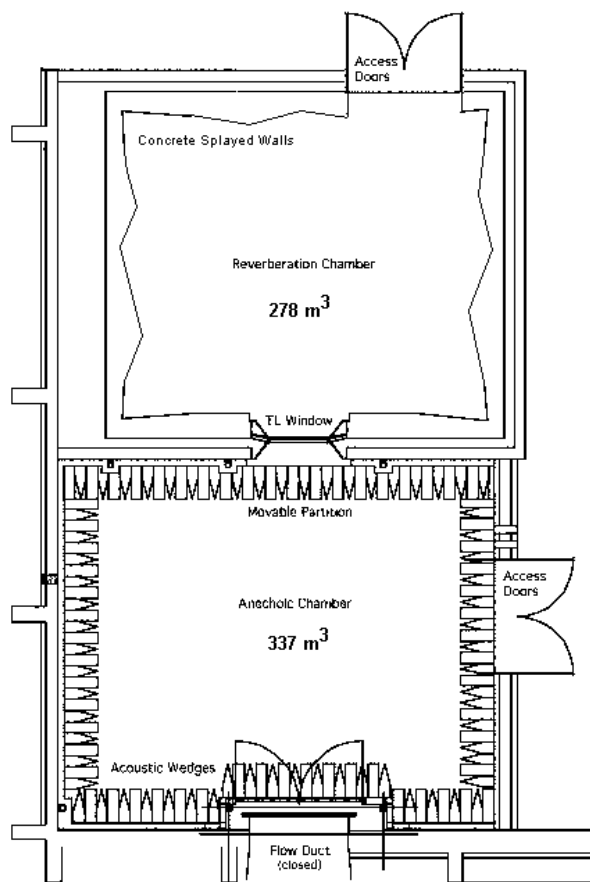


Fig. 1 Schematic of SALT Facility

mission loss (TL) window. The anechoic chamber has a volume of $11,900 \text{ ft}^3$ (337 m^3). Interior dimensions of the anechoic chamber, measured from wedge tip to wedge tip, are 15 ft (4.57 m) in height, 25 ft (7.65 m) in width, and 32 ft (9.63 m) in length. The reverberation chamber has approximate dimensions of 14.8 ft (4.5 m) in height, 21.3 ft (6.5 m) in width, and 31.2 ft (9.5 m) in length for a volume of $9,817 \text{ ft}^3$ (278 m^3). The reverberation chamber is structurally isolated from the rest of the building by suspension on large springs. The TL window accommodates test structures of up to $54''$ (1.41 m) \times $54''$ (1.41 m). A plexiglas test panel is shown installed in the SALT Facility in Figure 2. The plexiglas panels are clamped between two $0.75''$ thick aluminum frames to approximate a clamped boundary condition (see Figure 3). The frame assembly is then installed in a $3''$ thick fiberboard fixture in the transmission loss window. The visible dimensions of the plexiglas panels are $37.75 \times 15.25''$.

Vibro-Acoustic Tests

Point Force Excitation

For this set of tests, the SALT facility was setup as a transmission loss (TL) suite with a reverberation chamber on the source side, corresponding to the aircraft exterior,

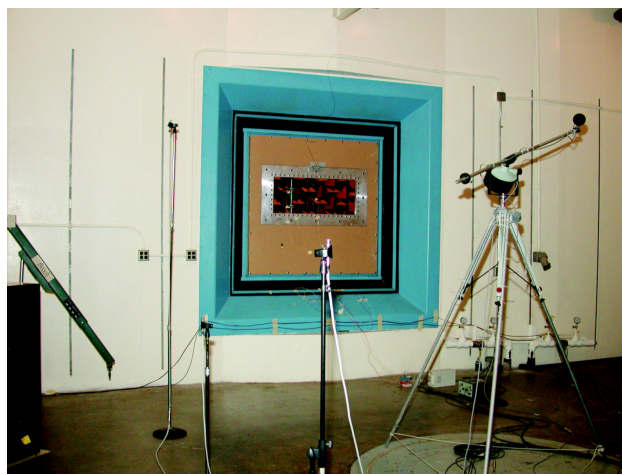


Fig. 2 plexiglas Panel Installed in SALT Transmission Loss Window. View From Source Room

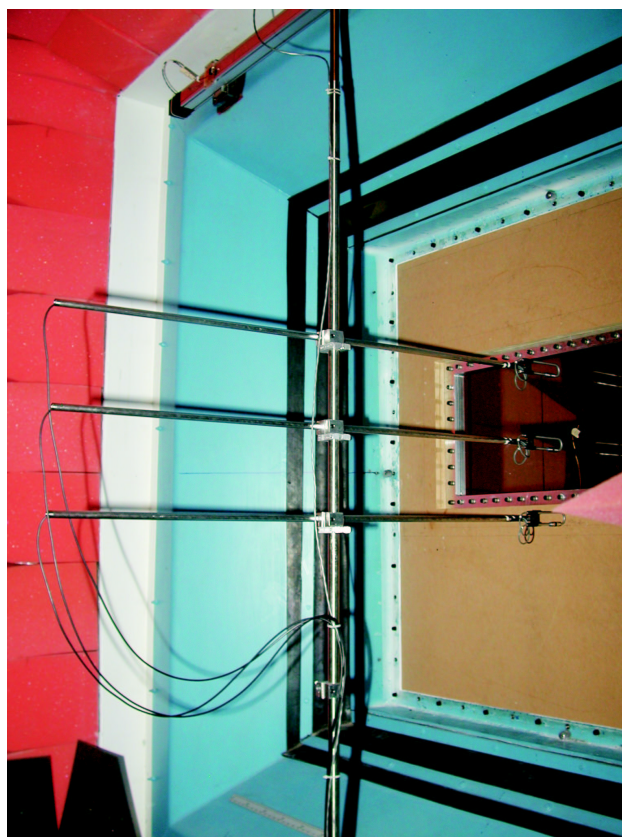


Fig. 3 Receiving Room of SALT Facility Showing Intensity Probes

and an anechoic chamber on the receiver side of the panel, corresponding to the aircraft interior. For the forced vibration response, surface velocities were measured using a scanning laser vibrometer while the panel was subjected to pseudo-random excitation by a shaker. Velocity scans were made at a spatial resolution of 1" and all data were recorded in the form of transfer functions relative to input force measured at the input shaker. From this data, complex velocity distributions at each frequency can be calculated. The velocity distributions can be used to estimate the total radiated sound power using a Rayleigh Integral formulation.³

Acoustic excitation

To measure the transmission loss of a window, the reverberation room was driven with 4 randomly placed speakers to produce a diffuse acoustic excitation of the window. The speakers were driven with white noise. Both radiated and incident sound power was measured. To compute the incident sound power, 6 quarter-inch condenser microphones were randomly distributed throughout the reverberation chamber shown in Figure 2. The incident sound power was computed from the average sound pressure level spectrum of the reverberation room as:

$$P_i(f) = \frac{A}{2\pi\rho c} \frac{1}{M} \sum_{k=1}^M p_k(f) \quad (1)$$

where p_k is the pressure spectrum of the k -th room microphone, M is the number of room microphones, A is the area of the panel, ρ is the density of air, and c is the speed of sound. To measure the sound power radiated from a window mounted in SALT, a traverse mechanism was installed on the anechoic side of the TL window.⁴ Three two-microphone intensity probes were mounted on a 2-D traverse as shown in Figure 3 to measure the intensity radiated from the window. The intensity can be measured as follows

$$I_p(f) = \frac{\text{Im}[G_{xy}(f)]}{4\pi\rho fL} \quad (2)$$

where I_p is the intensity corresponding to the p -th location of the grid, $G_{xy}(f)$ is the cross spectrum between the two probe microphones, and L is the distance between the microphones. The traverse mechanism was used to move the intensity probes and measurements were made at discrete points in a plane parallel to the test panel. The measured spatial intensity distribution was integrated to find the radiated power as shown below:

$$P_t(f) = \sum_{p=1}^N I_p(f) a_p \quad (3)$$

where a_p is the p -th elemental area, and N is the total number of measurement locations. The TL was computed as the ratio of the incident sound power to the radiated sound power as shown in equation 4.

$$TL(f) = 10\log_{10} \left(\frac{P_i(f)}{P_t(f)} \right) \quad (4)$$

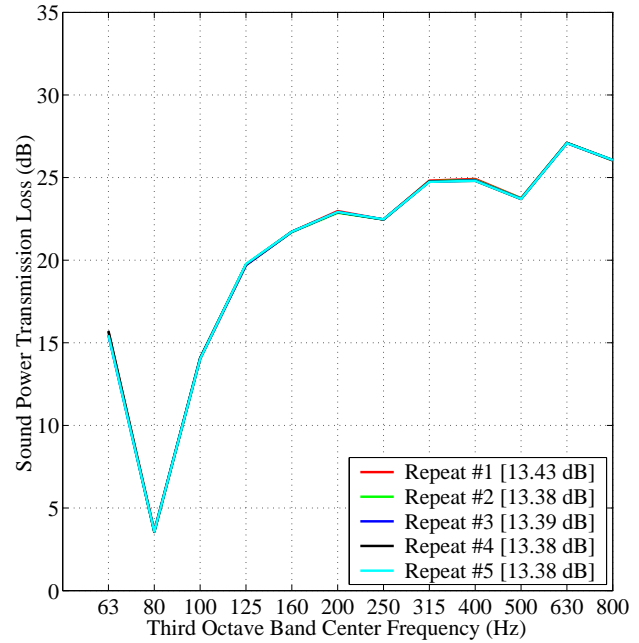


Fig. 4 Installation repeatability, measured transmission loss for back-to-back measurements

where TL is the transmission loss in dB.

Two different probe setups and spatial sampling grids were used to measure the intensity radiated from the windows into the anechoic room over a frequency range from 50 Hz to 10000 Hz. For frequencies below 800 Hz, a 2 inch (5.08 cm) by 2 inch spatial sampling grid and a 1.96 inch (4.98 cm) intensity probe spacer were used. For frequencies above 800 Hz, a 1 inch (2.54 cm) by 1 inch spatial sampling grid and the 0.334 inch (0.848 cm) intensity probe spacer were used.

Measurement Repeatability

To evaluate noise control treatments applied to a window, the variation in the measured transmission loss must be significantly smaller than the change caused by a treatment. The variability of the TL measurement of a typical window was studied to ensure the quality of the measurements. Both the back-to-back measurement variation and repeat installation variation were investigated. The back-to-back measurement variation was determined by repeating a TL measurement five times during a 4 hour period. The TL measurement of the five back-to-back tests is illustrated from 63 to 800 Hz in Figure 4. The standard deviation of the TL measurements is illustrated in Figure 5. The frequency averaged standard deviation of the transmission loss for back-to-back measurements during a single day is 0.03 dB. This is much smaller than expected and will not limit the evaluation of the performance of noise control treatments applied to a window.

The variation due to repeat installations was determined by measuring the TL of a typical plexiglas window four times over a period of six weeks. The fixture that held the plexiglas window in the TL window was completely disassembled and reassembled before each test. The ambi-

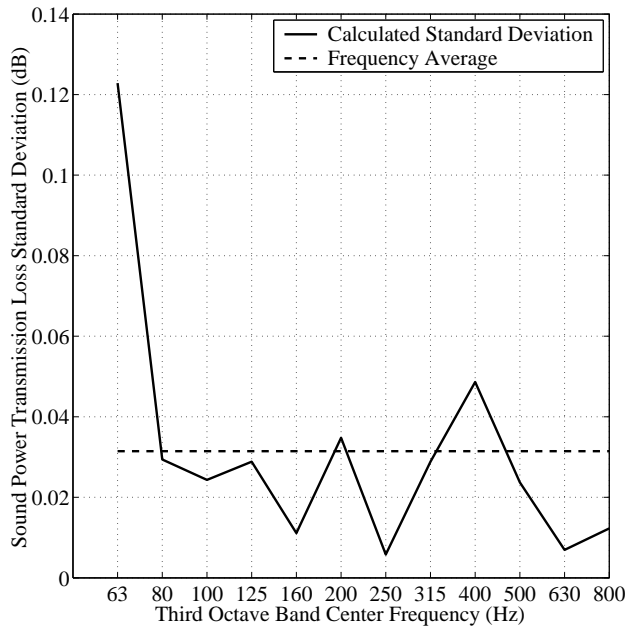


Fig. 5 Installation repeatability, standard deviation of back-to-back measurements

ent temperature and pressure varied significantly between tests. The intensity probes and the reverberation room microphones were calibrated before each test with the same pistonphone. The measured TL for the 4 tests from 63 to 800 Hz is shown in Figure 6. The standard deviation of the measured TL for the repeat installations is shown in Figure 7. The frequency averaged standard deviation of the measured TL for the repeat installations is 0.5 dB. The variation due to repeat installations over a six week span is significantly higher than the variation due to repeat measurements (Figures 5 and 7). However, the frequency averaged standard deviation of 0.5 dB is dominated by the variation in the 63 Hz one-third octave band (Figure 7). The standard deviation decreases as frequency increases (Figure 7) and at higher frequencies is typically between 0.25 and 0.3 dB. At frequencies above 125 Hz the standard deviation of the TL measurement due to repeat installations is acceptable for evaluation of the performance of noise control treatments applied to a window with a 95% confidence band of ± 0.6 dB.

Results and Discussion

In this section several experimental test results will be presented. First the panels were subjected to broadband force excitation from a shaker and complex velocity measurements were made using a scanning laser vibrometer. The data was post processed to estimate the total radiated sound power using a discrete Rayleigh Integral approach.³ Next the panel was subjected to random diffuse acoustic input from the reverberant chamber and the acoustic intensity distribution was measured using the aforementioned scanning intensity system shown in Figure 3. The intensity data was used to visualize the spatial distribution of the radiation and quantify the one-third octave band transmission

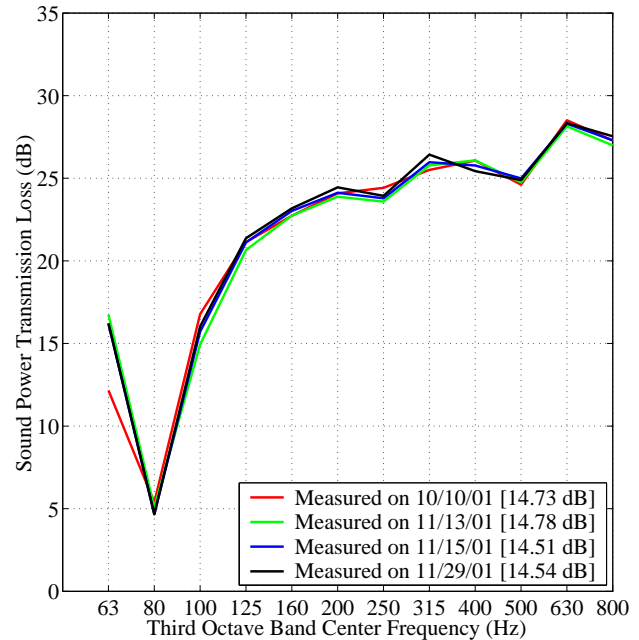


Fig. 6 Installation repeatability, measured TL for repeat installations

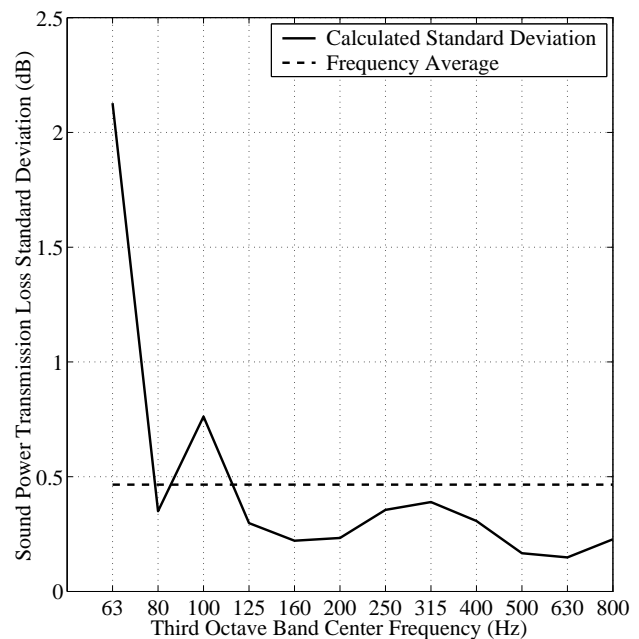


Fig. 7 Installation repeatability, standard deviation of TL for repeat installations

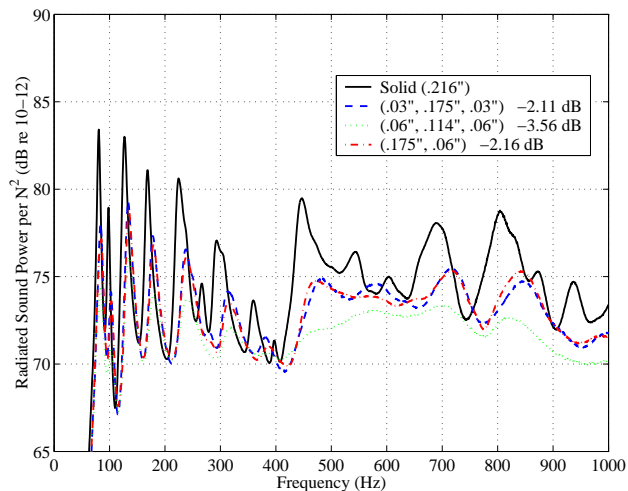


Fig. 8 Predicted radiated sound power using measured surface velocity data for nominally 1/4" panels.

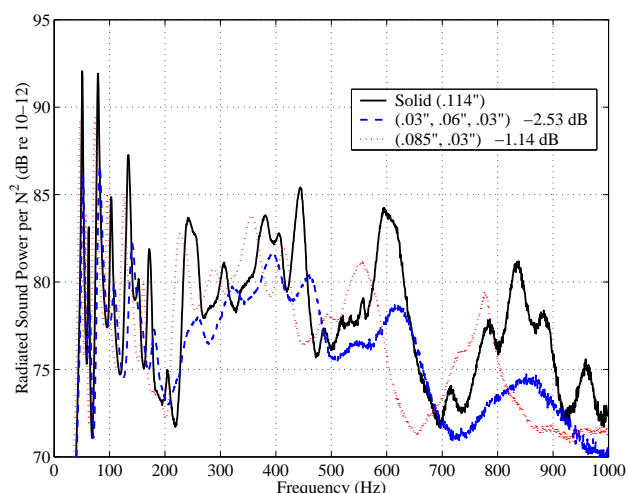


Fig. 9 Predicted radiated sound power using measured surface velocity data for nominally 1/8" panels.

loss of each panel.

Radiated Sound Power Prediction

The surface velocity data for broadband forced vibration input (shaker input) were used as inputs to a free field radiation prediction program using a Rayleigh Integral approach.³ Figures 8 and 9 show the predicted sound powers using measured surface velocity data for some of the 1/4" and 1/8" nominal windows, respectively. For the 1/4" case, the radiated sound power was dominated by low frequency resonances. The first five resonances occur at 81, 98, 131, 169, and 220 Hz. As seen in Figure 8, the best performance was from the 0.060-0.114-0.060 panel which reduced the radiated sound power by approximately 8 dB at low frequency resonances and 3.6 dB integrated over a bandwidth of 0 - 1000 Hz. For the 1/8" case, the performance was not quite as good. The best performance occurred with the 0.030-0.060-0.030 panel which reduced the radiated sound power approximately 5 dB at resonances and 2.5 dB integrated over the bandwidth of 0 - 1000 Hz.

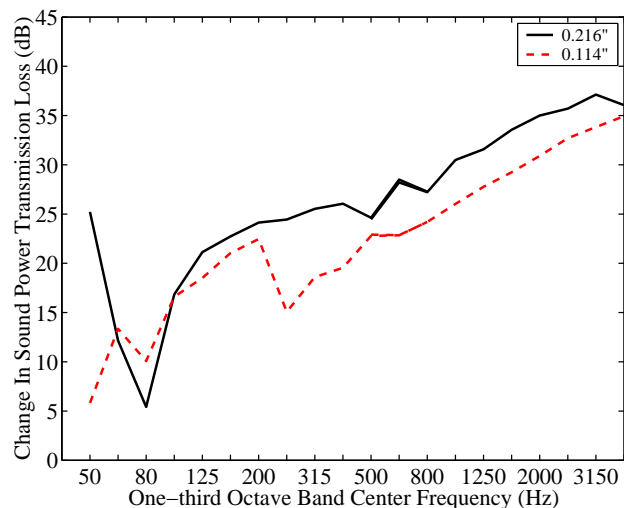


Fig. 10 Transmission Loss of 1/4" and 1/8" Solid Windows

Transmission Loss Measurements

The transmission loss (TL) results of the nominal 1/4" and 1/8" solid windows are shown in Figure 10. The 1/4" window shows a minimal 5 dB TL at the 80 Hz one-third octave band due to the lightly damped first mode of the panel acting as a strong radiator. The TL increases to about 35 dB at 4 kHz. The 1/8" solid panels exhibits less transmission loss over most of the bandwidth compared to the 1/4" panel. The TL of the other nominal 1/4" panels relative to the TL of the solid panel is shown in Figure 11. The TL is enhanced by as much as 7 dB in the 80 Hz one-third octave band using the damping treatment. For most of the frequency range the (0.060-0.114-0.060) panel provided the highest increase in TL.

The best performing 1/4" panel is compared to the solid panel up to a bandwidth of 8 kHz in Figure 12. Two features to note are that the exceptional performance of the damped panel is demonstrated up to 8 kHz, and the coincidence frequency dip can be seen at ~6 kHz. The integrated performance of the 1/4" windows compared to the baseline solid case is shown in Table 3. The TL integrated over the bandwidth of 50 - 4000 Hz was increased by as much as 4.5 dB (0.060-0.114-0.060 panel). These values must be interpreted carefully because most of the transmitted sound power occurs at low frequencies. Thus increases in TL in the 50 - 125 Hz one-third octave bands dominate the total TL. The designer should consider the nature of the forcing function, panel thickness and subsequent first mode resonant frequency when estimating the level of increased performance from applied damping. Thus Figure 11 might be a more useful indicator of performance. The results presented are significant in that the reductions were achieved with a modest increase in weight of 8 % due to difference in overall panel thicknesses. This increase in weight due to panel thickness could account for only about 0.6 dB increase in TL (assuming mass law). The total weight of the damping treatment (0.004") was on the order of 1 % of the panel weight.

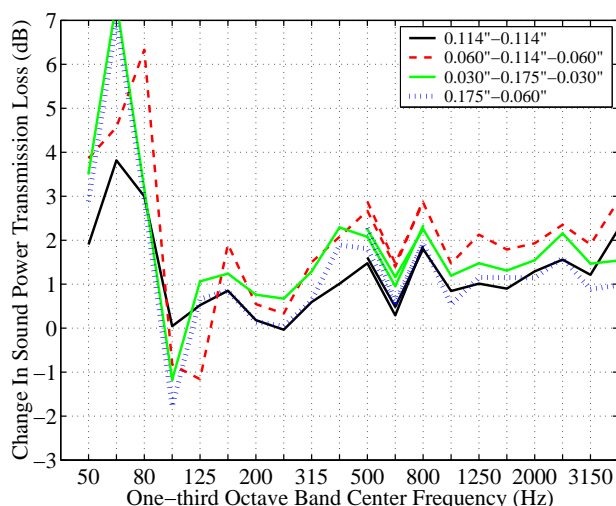


Fig. 11 Transmission Loss of 1/4" Damped Windows Relative to the Solid Window

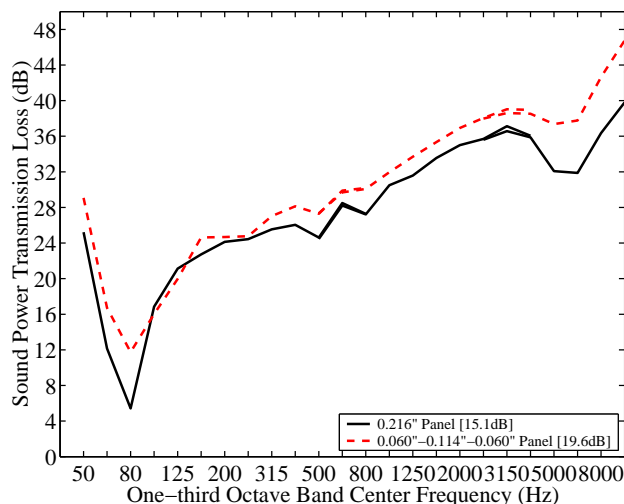


Fig. 12 Transmission Loss of 1/4" Windows, High Bandwidth

Table 3 Integrated TL Performance (50 - 4000 Hz), 1/4"Nominal Windows

Panel Label	TL (dB)	Increased TL (dB)
0.216	16.9	-
0.114-0.114	19.5	2.6
0.030-0.175-0.030	19.9	3.0
0.060-0.114-0.060	21.4	4.5
0.175-0.060	19.6	2.7

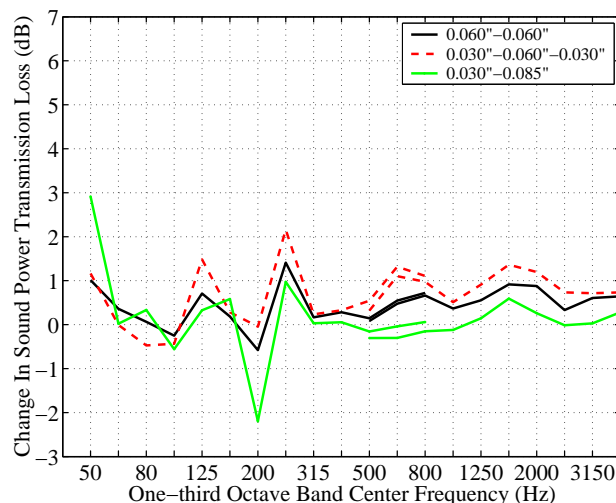


Fig. 13 Transmission Loss of 1/8" Damped Windows Relative to the Solid Window

Table 4 Integrated TL Performance (50 - 4000 Hz), 1/8"Nominal Windows

Panel Label	TL (dB)	Increased TL (dB)
0.114	15.7	-
0.060-0.060	16.4	0.7
0.030-0.060-0.030	16.3	0.6
0.030-0.085	17.0	1.3

As shown in Figure 13, the TL results for the 1/8" windows relative to the baseline solid window show similar trends, but reductions are less. The 0.030-0.060-0.030 panel shows the best performance at 125 Hz and above. The integrated performance of the 1/8" windows compared to the baseline solid case is shown in Table 4. As mentioned previously, the integrated TL performance is dominated by effect of the lowest few one-third octave bands. The 0.030-0.085 panel demonstrates the largest increase in TL over the entire bandwidth because of the 3 dB increase at 50 Hz. It is noted that the values presented for 50 and 63 Hz one-third octave bands are less accurate due to the repeatability standard deviation shown previously in Figure 7.

The outer layers of plexiglas in the panel layout could be considered constraining layers of a conventional constrained layer damping (CLD) treatment. There appears to be a relationship of TL to constraining layer thickness and damping location within the panel as the 0.060-0.114-0.060 panel consistently performed better than the 0.030-0.175-0.030 panel. Also the two layer panels only included half of the damping material of the three layer panels and this could have contributed to the relatively poor performance of the two layer compared to the three layer windows. The reduced performance of the 1/8" windows compared to the 1/4" windows is not well understood. An FEM/BEM model of the system is being developed to predict performance of both 1/4" and 1/8" windows, and determine the optimal panel configuration. This modeling should help answer some of these questions. Preliminary FEM/BEM modeling results will be presented later in this paper.

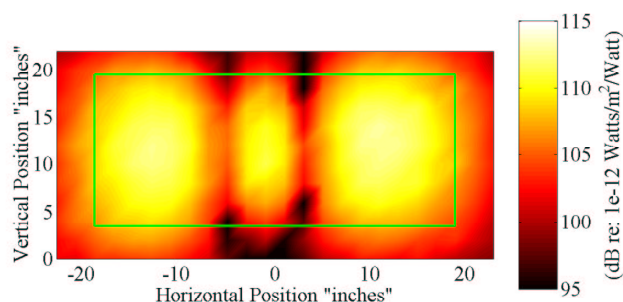


Fig. 14 Active Intensity Level at 131 Hz for 0.216" Solid Panel

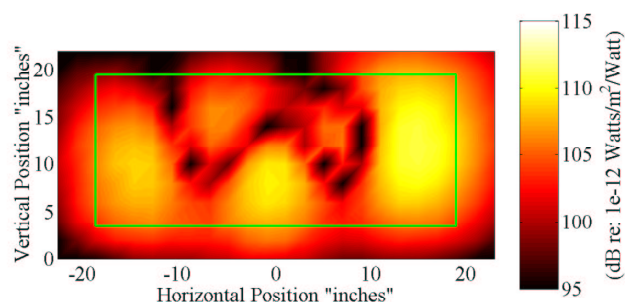


Fig. 16 Active Intensity Level at 220 Hz for 0.216" Solid Panel

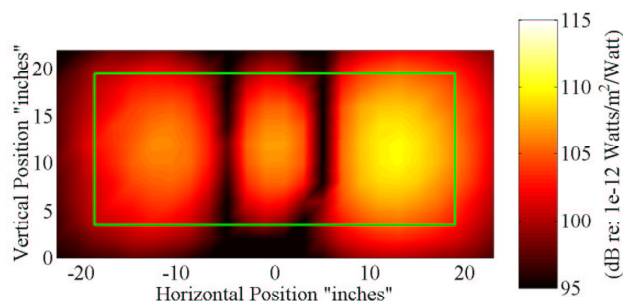


Fig. 15 Active Intensity Level at 131 Hz for 0.060 - 0.114 - 0.060 panel

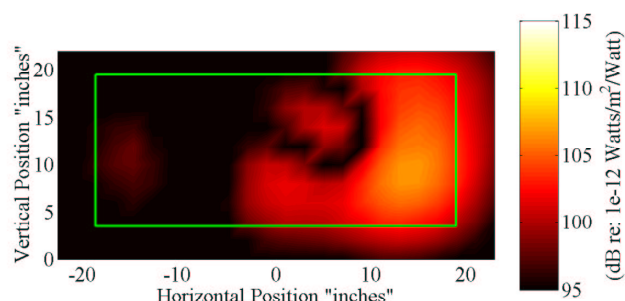


Fig. 17 Active Intensity Level at 220 Hz for 0.060 - 0.114 - 0.060 panel

Spatial Intensity Distribution

Complete spatial intensity scans were made for both point force and diffuse excitation. In this section the spatial distribution of radiated acoustic intensity will be examined for the point force excitation. The shaker was located at horizontal position of 12.5 inches and a vertical position of 8.5 inches for coordinates shown on Figures 14 through 17. The *active* intensity is associated with real energy leaving the structure that will propagate to the far-field. The *reactive* intensity is the imaginary portion which corresponds to energy that sloshes back and forth in the near-field, but does not propagate to the far-field. It can be seen in Figure 8 that the third resonance occurs at 131 Hz. Figures 14 and 15 show the spatial active intensity distribution at 131 Hz for the solid and best damped 1/4" panels. Careful examination of the figures shows reduction of peak energy on the order of 6 dB and the energy distribution is slightly skewed toward the shaker location indicative of a damped structural resonance.

It can be seen in Figure 8 that the fifth resonance occurs at 220 Hz. The spatial active intensity distribution at 220 Hz is shown in Figures 16 and 17 for the solid and best damped 1/4" panels, respectively. The damping treatment reduces the peak active intensity by 10 dB at this frequency. The overall active intensity is significantly reduced and heavily skewed toward the shaker location. The only significant radiation occurs local to the point force.

Preliminary Numerical Modeling and Results

There is an ongoing effort to analytically study and optimize the behavior of damped windows. Preliminary finite element models were developed in MSC.PATRAN and an-

alyzed using MSC.NASTRAN. Initial models consist of the plexiglas and viscoelastic layers being modeled with eight node solid elements with constant material properties. A 152 x 60 element mesh per layer was used. Clamped boundary conditions were prescribed at "the outer surface nodes. Velocity predictions for a point force excitation consistent with the test setup were generated for the panel. The predicted panel surface velocities were interpolated to the 39 x 16 mesh of the COMET boundary element model. The boundary element analysis consisted of the panel with symmetric boundary conditions radiating into a free-field. Predictions of the radiated sound power for the point force excitation are shown in Figure 18. This compares well with the results shown in Figure 8 for the measured velocity data propagated with the Rayleigh method up to 400 Hz. For the 0.060-0.114-0.060 panel, the overall reductions in radiated sound for the measured and predicted cases are within 0.5 dB. Further investigations will examine the use of frequency dependent material properties to improve the predictions in the 400 to 1000 Hz range.

Conclusions

Multi-layer damped plexiglas windows provide significantly enhanced transmission loss compared to conventional solid windows of similar thickness and weight. Reductions of radiated sound power of as much as 3.5 dB were demonstrated for point force excitation and 4.5 dB over a frequency range of 0 - 4000 Hz for diffuse acoustic excitation compared to the baseline window. These reductions are achieved with minimal additional weight.

Future work will concentrate on FEM/BEM modeling of the windows based on promising preliminary results. Once

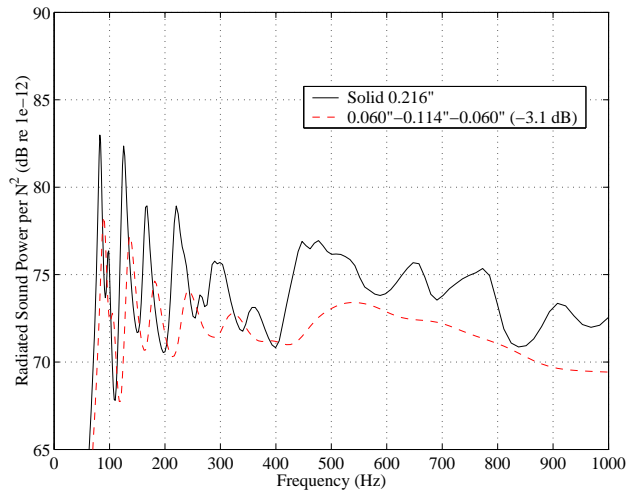


Fig. 18 Radiated Sound Power from FEM/BEM Prediction

accurate modeling techniques are developed, an optimization procedure will be employed to determine the optimal panel configuration. A new panel will be constructed to verify the optimization results.

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