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Damped Windows for Aircraft Interior Noise Control

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ABSTRACT

Windows are a significant path for structure-borne and air-borne noise transmission into aircraft. To improve the acoustical performance, damped windows were fabricated using two or three layers of plexiglas with transparent viscoelastic damping material sandwiched between the layers. In this paper, numerical and experimental results are used to evaluate the acoustic benefits of damped windows. Tests were performed in the Structural Acoustic Loads and Transmission Facility at NASA Langley Research Center to measure the transmission loss for diffuse acoustic excitation and radiated sound power for point force excitation. Comparisons between uniform and damped plexiglas windows showed increased transmission loss of 6 dB at the first natural frequency, 6 dB at coincidence, and 4.5 dB over a 50 to 4k Hz range. Radiated sound power was reduced up to 7 dB at the lower natural frequencies and 3.7 dB over a 1000 Hz bandwidth. Numerical models are presented for the prediction of radiated sound power for point force excitation and transmission loss for diffuse acoustic excitation. Radiated sound power and transmission loss predictions are in good agreement with experimental data. A parametric study is presented that evaluates the optimum configuration of the damped plexiglas windows for reducing the radiated sound power.

1. INTRODUCTION

Interior noise studies on general aviation aircraft[1,2] have shown that windows are a significant noise path. This noise arises predominantly from external pressure excitations that are distributed over the fuselage, such as turbulent boundary layer pressure fluctuations, external acoustic sources, or dynamic flow structures produced by propellers. For a single engine propeller aircraft, considerable variability in the cabin noise levels exist with the sound pressure levels near the windshield usually being the highest at most frequencies.[1] From a recent flight test by Unruh and Till,[2] it was concluded that treating the windshield and forward cabin windows with increased mass loading or damping treatments offered good potential for cabin noise reduction. This flight test provided the inspiration for the present work on damped windows for aircraft applications.[3,4]

Laminated glass has been shown to provide benefits for noise reduction in automotive and architectural applications.[5-10] Recent work at NASA Langley Research Center[3,4] has examined the acoustic benefits of laminated windows consisting of two or three layers of plexiglas with transparent viscoelastic damping material sandwiched between the plexiglas layers as shown in Figure 1. These damped plexiglas windows were evaluated as replacements for conventional solid aircraft windows to reduce the structure-borne and air-borne noise transmitted into the interior of general aviation aircraft. For windows that were nominally 6.35 mm thick, reductions in the radiated sound power as large as 3.7 dB over a 1000 Hz bandwidth were shown experimentally when comparing damped and uniform plexiglas windows.[3] An increase in transmission loss (TL) of up to 4.5 dB over a 50 to 4000Hz frequency range was also shown.[3] While this work focused on single-pane laminated windows, it has also been shown for architectural applications[10] that laminated glass provides increased transmission loss for window assemblies consisting of two glass panels separated by an air gap. This has the potential for broader application of the laminated windows to multi-pane window configurations applicable to passenger windows for advanced turboprop and jet aircraft.

The work presented in this paper extends the plexiglas window study[3-4] to include transmission loss predictions based on statistical energy analysis (SEA) methods and TL equations in the literature.[5] Transmission loss predictions are shown for finite element/boundary element predictions below 1 kHz, and SEA and TL equations in the range from 100 to 10k Hz. Finite element/boundary element predictions of the radiated sound power are also presented. The predictions are compared with acoustic response measurements for the windows installed in the NASA Langley Structural Acoustic Loads and Transmission (SALT) Facility.[11] Good agreement is obtained between measurements and predictions. Using the validated numerical models, a parametric study examined the optimum layup of the damped plexiglas windows. The test facility, hardware, vibro-acoustic tests, and numerical models are described.

2. TEST FACILITY

The Structural Acoustic Loads and Transmission (SALT) Facility[11] located at the NASA Langley Research Center is shown in Figure 2. The SALT facility consists of an anechoic chamber, a reverberation chamber, and a shared transmission loss (TL) window. The anechoic chamber has a volume of 337 cubic meters with interior dimensions, measured from wedge tip to wedge tip, of 4.57 meters in height, 7.65 meters in width, and 9.63 meters in length. The reverberation chamber has dimensions of 4.5 meters in height, 6.5 meters in width, and 9.5 meters in length for a volume of 278 cubic meters. A shared TL window accommodates test structures of up to 1.41- by 1.41-meters.

3. HARDWARE DESCRIPTION

A plexiglas window is shown installed in the SALT Facility in Figure 3. The SALT facility was setup as a transmission loss suite. A 1.22- by 0.61-meter plexiglas window was clamped between two 19-mm thick aluminum frames to approximate a fixed boundary condition. After mounting in the frame, the exposed portion of the plexiglas window measured 0.96- by 0.39-meters. The frame assembly was installed in a 76-mm thick fixture composed of medium density fiberboard, which was mounted in the transmission loss (TL) window.

The window test configurations are listed in Table 1 and include a uniform window and four different layups of damped plexiglas windows. The windows are labeled based on the thickness (in thousandths of an inch) of the layers in the laminate. For example, the 175,2,60 window has a 4.44-mm (0.175-inch) thick plexiglas layer, a 0.051-mm (0.002-inch) thick visoelastic layer, and a 1.52-mm (0.060-inch) thick plexiglas layer. All damped window test configurations used a 0.051-mm (.002-inch) thick 3M™ viscoelastic damping polymer (Product No. 112P02). Due to the thickness of the available plexiglas sheets, overall window thickness and corresponding window weight varied by approximately ten percent. The weight estimates are based on the exposed portion of the window (0.96- by 0.39-meters) and manufacturer provided density values. For this study, the windows were all manufactured flat for ease of fabrication and testing.

4. VIBRO-ACOUSTIC TESTS

Measurements of the radiated sound power and transmission loss of damped and uniform plexiglas windows were made by Gibbs et.al.[3] A brief summary of the test setup and results are provided below.

A. Radiated Sound Power

A plexiglas window was mounted in the TL window of the SALT facility and mechanically excited with psuedo random excitation by a shaker. The applied force was measured using a force transducer mounted to the window at the drive point. The spatial intensity was measured[12] on the anechoic side of the window using two-microphone intensity probes and the intensity distribution was integrated over the measurement surface to obtain the radiated sound power. The results for several of the tested 6.35-mm thick windows are shown in Figure 4. The damped windows are shown to reduce the radiated sound power by 7 dB at some of the lower resonance frequencies and up to 3.7 dB over the 1000Hz bandwidth.

B. Transmission Loss

The SALT facility was setup as a transmission loss suite. The reverberation room was driven with four speakers, each directed into a corner, to produce a diffuse acoustic excitation of the window. The transmission loss was found from measurements of the incident and transmitted sound power as described by Klos and Brown.[12]

The transmission loss of the uniform (216) window and the damped (60,2,114,2,60) window are shown in Figure 5. The addition of the constrained layer damping into the window provides a significant increase in the transmission loss at the resonance dip of 80 Hz and in the mass controlled region. A 4.5 dB increase in transmission loss over the 50 to 4000Hz frequency range is observed. For a homogeneous panel,[13] the most significant improvement due to the addition of damping is expected at the lower resonant frequencies and above the critical frequency. The critical frequency is the lower limiting frequency for coincidence, where the projected wavelength of an incident sound wave equals the bending wavelength in the panel. At coincidence, the sound waves and bending waves reinforce one another resulting in increased panel vibration. This results in a reduction in the transmission loss that is often referred to as the "coincidence dip." For the nominally 6.35-mm thick plexiglas window the critical frequency is approximately 6 kHz. As shown in figure 5, the damped window shows an increase in transmission loss of 6 dB at and above the critical frequency.

C. Vibroacoustic Test Summary

The addition of a constrained viscoelastic damping layer provided a significant improvement in acoustical performance for the 6.35-mm thick windows. The 60,2,114,2,60 window resulted in the best acoustical performance of the windows tested. However, the question of the optimum layup was not resolved by the experiment. It was desirable to develop and validate methods to predict the observed changes in radiated sound power and transmission loss due to the incorporation of constrained layer damping in the plexiglas windows. This led to the numerical studies described in the remaining sections.

5. NUMERICAL MODELS

A. Finite Element Model

MSC.Nastran 2001 was used for the finite element analyses of the window configurations. The viscoelastic layer was modeled with eight node solid elements with isotropic material properties. Concerns over high aspect ratio (length or width to thickness ratio) elements for the thin viscoelastic layer lead to a mesh refinement study[4] that determined a 76- by 30-element mesh was sufficient to model the dynamic response to 1 kHz. Buehrle et. al.[4] evaluated the use of both plate (CQUAD4) and solid (CHEXA) elements for the plexiglas base and constraining layers. It was determined that using a plate element for the plexiglas resulted in decreased stiffness above 400 Hz. Therefore, solid elements were used for both the viscoelastic and plexiglas layers in subsequent analyses of the damped windows.

The material properties for the plexiglas and viscoelastic damping material are listed in Table 2. Isotropic material properties were assumed for the plexiglas. Estimates of the damping properties of the plexiglas were based on comparisons with experimental results for the uniform window. The loss factor for both materials was entered as structural element damping (GE) on the material card of the finite element model. The viscoelastic damping material has frequency and temperature dependent properties. An ambient temperature of 20 degrees Celsius was assumed.

A modal frequency response solution that included 200 modes, up to a frequency of over 4000 Hz, was used to predict the response to unit force excitation over the 60 to 1000 Hz range. As noted previously, the viscoelastic material had frequency dependent material properties. The modal frequency response solution does not allow for frequency dependent material properties, so predictions are based on constant viscoelastic material properties corresponding to the values at 100 Hz and 20˚C. Incorporation of frequency dependent material properties requires the direct frequency response solution but this significantly increases computation time and was deemed unnecessary. As will be shown in the results section, the modal solution provided predictions consistent with the measured results in the frequency range considered. For each of the window layups, surface velocities were predicted for a given excitation and used as boundary condition input to the boundary element model.

B. Boundary Element Model

The boundary element model was developed in COMET/Acoustics. A grid of 39 elements along the length by 16 elements along the width was used to obtain adequate spatial resolution of the structural mode shapes up to 1000 Hertz. A symmetric boundary was used to simulate the acoustic radiation of the baffled window into the anechoic chamber. The direct exterior boundary element analysis was used to predict the radiated sound power from imposed surface velocity data. Surface velocity data was based on finite element analyses with either point force excitation or a diffuse acoustic pressure excitation applied to the finite element model. In order to evaluate the numerical data, the finite element velocity predictions were first interpolated to the reduced mesh size of the boundary element model, incorporated into the boundary element model, and then analyzed.

C. Diffuse Acoustic Excitation Model

The finite element models of the uniform (216) window and the damped (60,2,114,2,60) window were used to predict the change in the sound power transmission loss due to the addition of constrained layer damping. To use the finite element models to predict the transmission loss, a simulation of the diffuse acoustic excitation mechanism present in the experiment is needed.

A diffuse acoustic excitation of the finite element models was developed based on plane wave propagation. A large number, N, of plane waves having random angles of incidence, random magnitudes, and random temporal phase angles were summed together to simulate a diffuse field excitation. A plane wave incident on the surface of a window at angles θ_n and ψ_n is shown in Figure 6. The angles θ_n and ψ_n are uniformly distributed random numbers on the intervals $[0,\pi]$ and $[0,2\pi]$ respectively and represent the angles of propagation in spherical coordinates. The n^{th} plane wave has a magnitude of $P_n\cos(\theta_n)$, where P_n is a uniformly distributed random number on the interval [0,1]. Thus the steady state pressure can be described in space and time by the equation

$$P_n(x,y,z,t) = P_n\cos(\theta_n)e^{-ik_x x}e^{-ik_y y}e^{-ik_z z}e^{i(\omega t+\phi_n)} \quad (1)$$

where ω is the angular frequency, ϕ_n is a random temporal phase angle uniformly distributed on the interval of $[0,2\pi]$, and k_x, k_y and k_z are the wavenumber in the x, y and z directions, respectively, found from

$$k_x = k\sin(\theta_n)\cos(\psi_n) \quad (2a)$$

$$k_y = k\sin(\theta_n)\sin(\psi_n) \quad (2b)$$

$$k_z = k\cos(\theta_n) \quad (2c)$$

where $k=\omega/c$, is the wavenumber in air at a particular analysis frequency and c is the speed of sound in air. The random temporal phase angle is introduced to prevent the N plane waves from having the same phase angle at the x, y, z coordinates of (0,0,0). A weighting function of $\cos(\theta_n)$ is included in the pressure magnitude to correct for the probability distribution of incident plane waves likely present in the experimental excitation.[14] The random variables θ_n, ψ_n, P_n and ϕ_n are unique for each of the N plane waves. Assuming steady state simple harmonic motion and a nearly rigid boundary condition at the surface of the window, the spatial pressure distribution exciting the window can be approximated by

$$\hat{P}_n(x,y,z,\omega) = 2P_n\cos(\theta_n)e^{i(\phi_n - k_x x - k_y y - k_z z)} \quad (3)$$

where x, y and z are evaluated on the surface and k_x, k_y and k_z are evaluated at a particular angular frequency ω. The pressure acting on the surface of each element, e, in the finite element model, due to the N plane waves, was computed using the x, y and z coordinates of the element center x_e, y_e and z_e. The total pressure at the center of each element, P_e, due to N incident plane waves is

$$\hat{P}_e(\omega) = \sum_{n=1}^{N} 2P_n\cos(\theta_n)e^{i(\phi_n + k_x x_e + k_y y_e + k_z z_e)} \quad (4)$$

where N is the number of plane waves used to approximate the diffuse field. This pressure distribution acting on the surface elements of the finite element model was used as an excitation. The velocity response of the finite element model of the windows was predicted due to the pressure excitation. The predicted velocities were imported into the boundary element model of the window and the transmitted sound power, Π_t, was predicted.

To compute transmission loss, the ratio of the incident to transmitted sound power is needed. The incident sound power is computed from the intensity vector of each of the *N* plane waves. The intensity vector, $\vec{I}_n$, of the *n*th plane wave is

$$\vec{I}_n = -\frac{[P_n \cos(\theta_n)]^2}{\rho c}[\sin(\theta_n)\cos(\psi_n)\vec{i} + \sin(\theta_n)\sin(\psi_n)\vec{j} + \cos(\theta_n)\vec{k}] \tag{5}$$

The sound power incident on a window of area A due to the *n*th plane wave is

$$\Pi_{i,n} = A\cos(\theta_n)\frac{[P_n \cos(\theta_n)]^2}{\rho c} \tag{6}$$

The total incident sound power for all *N* plane waves is

$$\Pi_i = \sum_{n=1}^{N} \Pi_{i,n} \tag{7}$$

With the transmitted sound power computed from the finite element and boundary element models, the predicted transmission loss of the window is computed from the ratio of the incident and transmitted sound power.

$$TL = 10\log_{10}\left(\frac{\Pi_i}{\Pi_t}\right) \tag{8}$$

D. Statistical Energy Analysis

The statistical energy analysis code AutoSEA2 was used to estimate the transmission loss of the uniform and damped plexiglas windows in the frequency range from 100 to 10k Hz. The model consisted of a plate contained between two acoustic cavities as recommended in reference 15. For the uniform window, the uniform plate model was used with plexiglas material properties and a constant damping loss factor of 7 percent. The damped windows were evaluated using the AutoSEA2 general laminate model, where the frequency dependent material properties were prescribed for the viscoelastic damping layers. This resulted in a frequency dependent damping loss factor, which for the case of the 60,2,114,2,60 window varied from 9 percent at 100 Hz to 15 percent at 10k Hz. For a given window, the transmission loss was calculated from[15]

$$TL = 10\log_{10}\left(\frac{A\omega}{8\pi^2 n_1 \eta_2 c^2}\left(\frac{E_1}{E_2} - \frac{n_1}{n_2}\right)\right) \tag{9}$$

where the subscripts 1 and 2 define the source and receiving cavities, respectively, E is the energy in a cavity, n is the modal density of a cavity, A is the panel area, c is the speed of sound, η is the loss factor of a cavity, and ω is the excitation frequency.

E. Transmission Loss Equations

The transmission loss was also calculated using the equations outlined by Callister et. al.[5] This consists of using the equation for forced vibration transmission derived by Sewell[16] for frequencies less than one-half the critical frequency. Sewell's equation can be written:

$$TL = -10\log_{10}\left(\frac{\left[\ln(k\sqrt{A}) + 0.16 - U(\Lambda) + \frac{1}{4\pi k^2 A}\right]}{\left[\left(\frac{m\pi f}{\rho c}\right)\left(1 - \frac{f^2}{f_c^2}\right)\right]^2}\right) \tag{10}$$

where k is the acoustic wavenumber in air, A is the area of the plate, f is the frequency, f_c is the critical frequency, m is the mass per unit area, ρ is the density of air, c is the speed of sound in air, and $U(\Lambda)$ is a shape factor correction.[16]

For frequencies above the critical frequency, the transmission loss can be found from the following equation [5] derived by Cremer.

$$TL = 20\log_{10}\left(\frac{\pi f m}{\rho c}\right) + 10\log_{10}\left(\frac{2\eta f}{f_c}\right) - 5 \tag{11}$$

where η is the loss factor for the panel. The damping loss factors estimated from the AutoSEA2 general laminate model were used for the damped window configurations. Linear interpolation was utilized between the transmission loss found from equation (10) at one-half the critical frequency and equation (11) at the critical frequency.

6. RESULTS AND DISCUSSION

A. Radiated Sound Power Prediction

The measured and predicted sound power for a uniform plexiglas window and a damped plexiglas window are shown in Figures 7 and 8, respectively. Discrepancies at the first resonant frequency are associated with a coupling of the window mode and a mode of the supporting fiberboard test fixture. The test fixture is not included in the finite element model and therefore its dynamic effects are not captured in the predicted results. Above 500 Hz, response predictions are more heavily damped than the experiment but consistent trends in magnitude are observed. This difference in damping is attributed to the constant properties assumed for the viscoelastic material in the finite element model (see section 5A). The predicted sound power for several damped plexiglas window layups are shown in Figure 9. Reductions in the radiated sound power of as much as 7 dB at some of the lower frequencies and 3.3 dB over the 1kHz bandwidth are demonstrated. These predicted reductions are consistent with experimental results shown previously in Figure 4. Thus, the chosen numerical modeling methods are able to predict the effects of the constrained layer damping.

B. Parametric Study

A parametric study was performed to examine the optimum placement of viscoelastic damping material within a nominally 6.35-mm thick plexiglas window. Utilizing the test verified finite element models, windows with two and three plexiglas layers with viscoelastic damping material sandwiched between each layer were analyzed. The total amount of viscoelastic material was held constant for all of the window layups. The two layer windows used a single 0.102-mm thick viscoelastic layer while the three layer windows had two 0.051-mm thick viscoelastic layers. The results of the parametric study are shown in Figure 10. For a symmetric three-layer plexiglas window, the radiated sound power decreased as the center plexiglas layer thickness decreased and the optimum design converged to a window with two equal thickness plexiglas layers. Results for the two layer designs also showed the greatest reductions were obtained for two plexiglas layers of equal thickness with the viscoelastic material at the mid-plane. This is consistent with published[17] panel damping design recommendations when the base and constraining layers are made of the same material. It should also be noted that the same viscoelasitic material was used throughout. In contrast to this result, Grootenhus[18] evaluated multi-layer laminates and found that an unsymmetrical design using two different damping materials with significantly different shear modulus had the potential to broaden the frequency range for optimum damping. The validated numerical tools can be used to evaluate a multitude of parameter variations of which only one was presented here.

C. Transmission Loss Prediction

The transmission loss of the uniform (216) and damped (60,2,114,2,60) plexiglas windows were studied. The measured change in the transmission loss caused by the addition of the constrained layer damping was computed from the measured transmission loss data shown in Figure 5. The measured change in transmission loss is shown in Figure 11, red line. The change in transmission loss was most significant at the first resonance of the panel where an increase of 6 dB was observed. In the mass controlled region, an increase of 1 to 2 dB was observed in the transmission loss. Using the combined finite element, boundary element and diffuse acoustic field excitation models outlined in sections 5A-C, predictions of the transmission loss of the uniform (216) and the damped (60,2,114,2,60) plexiglas windows were made in the frequency range from 50 to 1000 Hz. The predicted change in transmission loss was computed (Figure 11, green curve) as the difference between these two predictions. There is good agreement between the measured and predicted change in transmission loss caused by the addition of the damping layer. In general, the predicted change in transmission loss is within 0.5 dB of the measurement (Figure 11). The finite element model of the panel incorporating the modal frequency response solution was used. The damping loss factor and shear stiffness of the viscoelastic damping layer were assumed to be constant with respect to frequency. However, the properties of the viscoelastic damping layer used in the experiment varied with respect to frequency. The errors observed when comparing the measured and predicted change in transmission loss are likely due to difficulty estimating the viscoelastic properties of the damping layer and variation of the viscoelastic properties with frequency.

For the frequency range from 100 to 10k Hz, transmission loss predictions from SEA and the TL equations outlined in sections 5D and 5E are shown along with the measured data in figures 12 and 13 for the uniform (216) and damped (60,2,114,2,60) windows, respectively. The TL equations are in better agreement with the measurements in the 1/3-octave bands from 125 to 400 Hz (see Figures 12-13). In the 500 to 3150 Hz 1/3-octave bands, both predictions are generally within 3 dB of the measurements. In the coincidence dip, more variation exists between the predicted and measured data. However, the change in TL due to the addition of the viscoelastic damping is well modeled as shown in Figure 14. For the SEA predictions, there is a large decrease in TL at 5000 Hz. This is associated with a shift in the critical frequency due to a change in window thickness for the uniform and damped window. In general the predicted TL increases are within 2 dB of the measured changes (Figure 14). This demonstrates that these prediction tools can be used to assess the acoustic effects of the addition of constrained layer damping in laminated window designs.

7. SUMMARY AND CONCLUSIONS

The potential of damped windows for aircraft noise control is demonstrated through experiments and numerical predictions. The incorporation of constrained layer damping into the windows increased transmission loss by 6 dB at the first natural frequency, 6 dB at coincidence, and 4.5 dB over a 50 to 4k Hz range. Radiated sound power was reduced up to 7 dB at the lower natural frequencies and 3.7 dB over a 1000 Hz bandwidth.

Numerical modeling of plexiglas windows showed good agreement with experimental data. The 60,2,114,2,60 window resulted in the best acoustical performance of the windows tested. The numerical models were used to determine the optimal damped plexiglas widow configuration. Based on a parametric study of two and three layer plexiglas windows with equal amounts of viscoelastic material, the optimum design converged to a window with two equal thickness plexiglas layers. These results show that it is most advantageous to place all of the damping material at the mid-plane where the shear strains are the largest. The modeling approach has been validated and can be used to evaluate other design configurations.

The change in transmission loss caused by the constrained layer damping was predicted using a simulated diffuse acoustic excitation of the finite element model for frequencies from 50 to 1000 Hz. In the frequency range from 100 to 10k Hz, SEA and TL equations were used to predict the TL for uniform and damped window designs. There was good agreement between the measured and predicted change in transmission loss caused by incorporation of the viscoelastic damping layer into the window. These prediction methods can be used to study sound transmission properties of windows and other structures that incorporate constrained layer damping.

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Table 1. Nominally 6.35-mm Test Windows

Window Configuration	Overall Thickness mm (inch)	Weight of Exposed Panel N (lbs)
216	5.49 (0.216)	23.9 (5.38)
114,2,114	5.84 (0.230)	25.4 (5.72)
175,2,60	6.02 (0.237)	26.2 (5.90)
30,2,175,2,30	6.07 (0.239)	26.4 (5.94)
60,2,114,2,60	6.04 (0.238)	26.3 (5.91)

Table 2. Material Properties

Material	Poisson Ratio	Density kg/m^3	Shear Modulus MPa	Loss Factor
Plexiglas	0.35	1200	1700	.07
Viscoealstic (100 Hz)	0.49	1000	2.5	1
Vicsoelastic (300 Hz)	0.49	1000	5	.85
Viscoelastic (500 Hz)	0.49	1000	7	.7
Viscoelastic (1000 Hz)	0.49	1000	9	.6

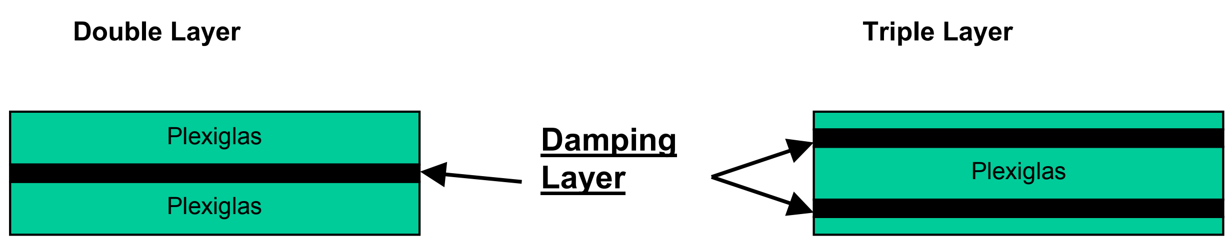

Figure 1. Damped plexigas window configurations.

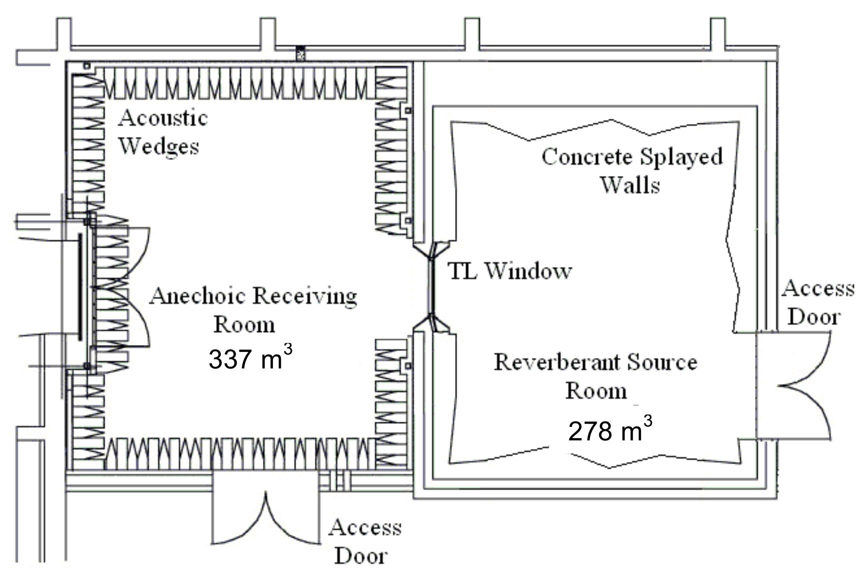

Figure 2. Schematic of SALT Facility.

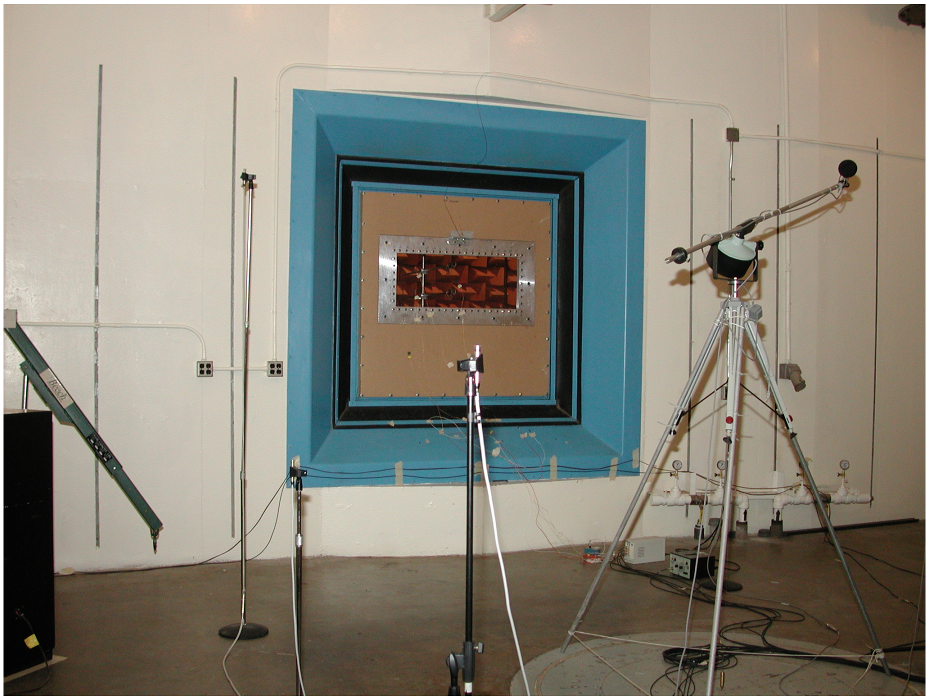

Figure 3. Plexiglas window installed in SALT TL window, view from reverberation chamber.

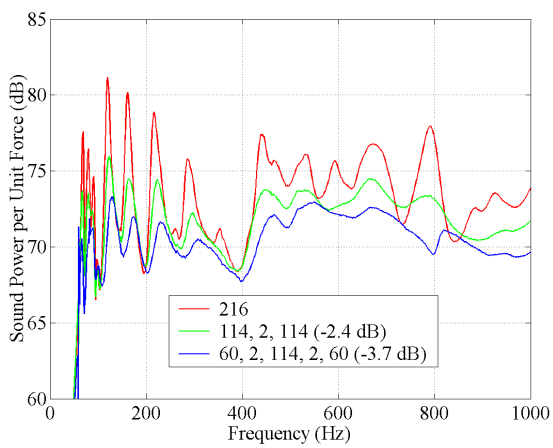

Figure 4. Measured sound power for nominally 6.35-mm thick windows installed in SALT.

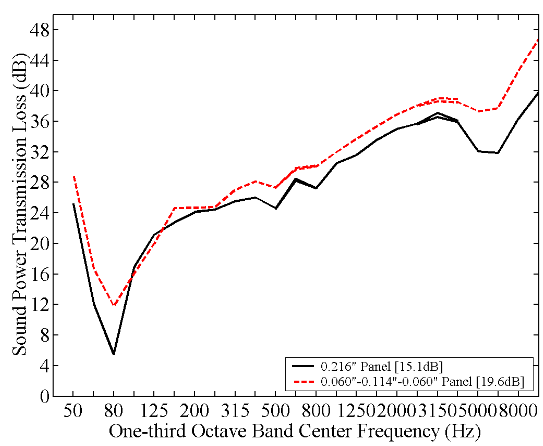

Figure 5. Measured transmission loss of the undamped "—" and damped "----" windows.

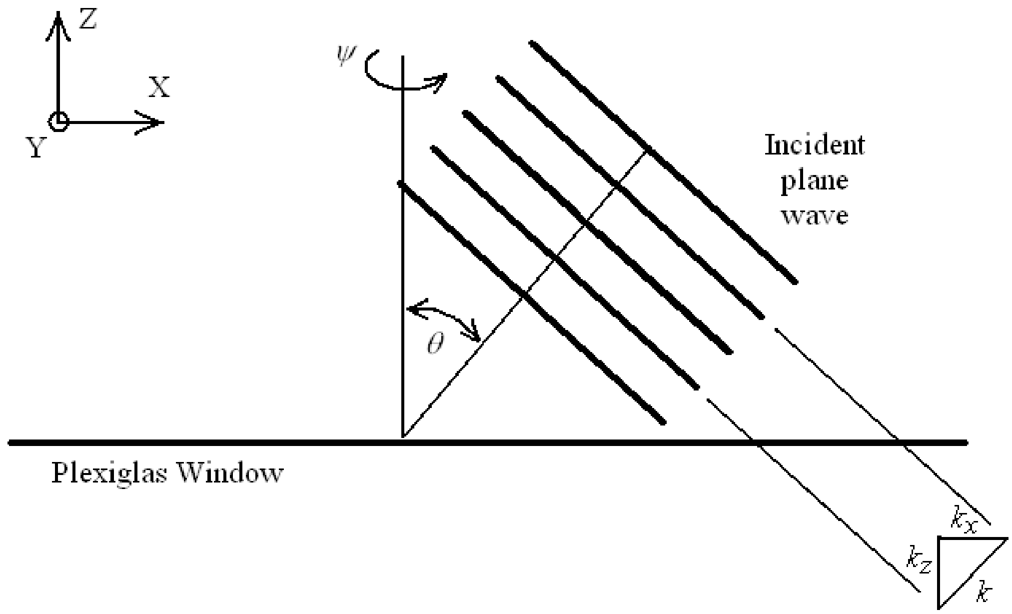

Figure 6: Plane wave incident on a Plexiglas window. The plane wave is shown propagating in the x-z plane, the y axis is into the page. The angle ψ represents a rotation of the heading of the plane wave about the z axis.

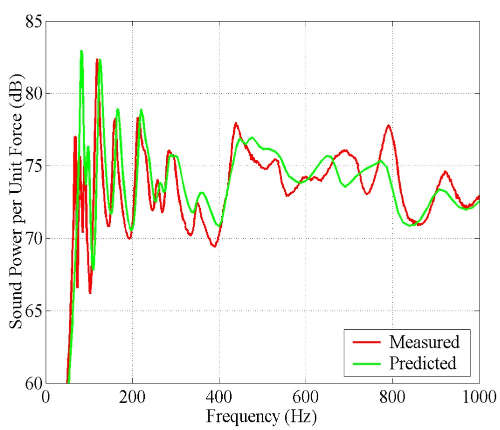

Figure 7. Measured and predicted radiated sound power for 216 panel.

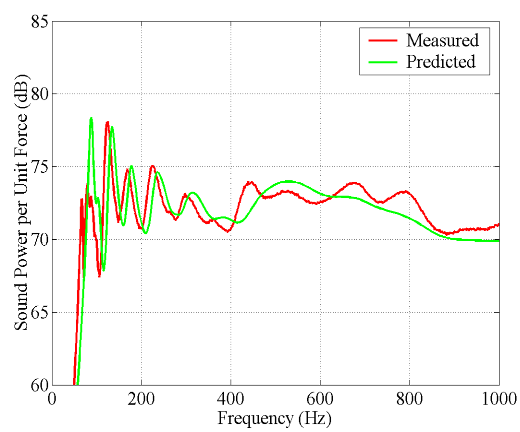

Figure 8. Measured and predicted radiated sound power for 114,2,114 panel.

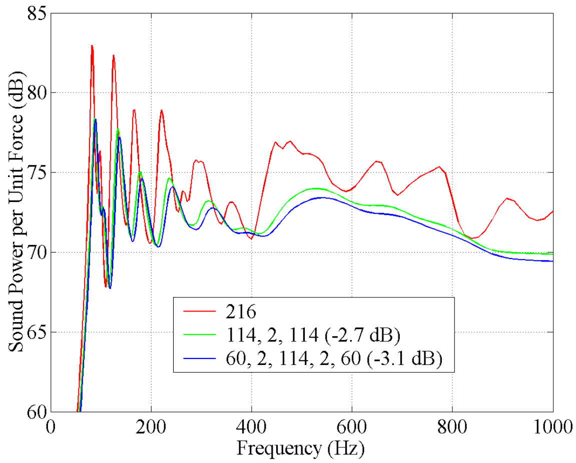

Figure 9. Predicted radiated sound power for point force excitation.

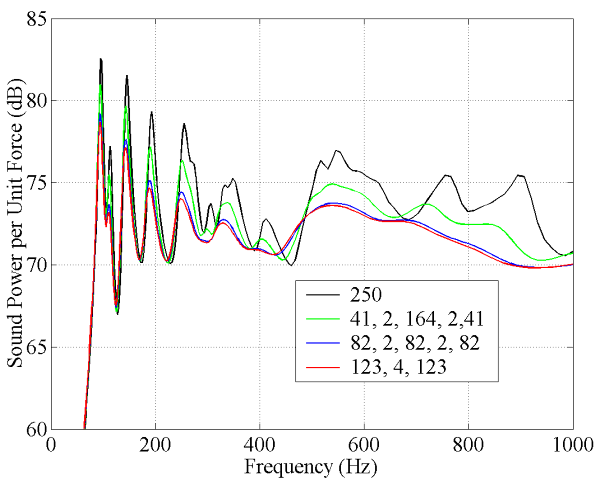

Figure 10. Results of parametric study showing the effect of window layup.

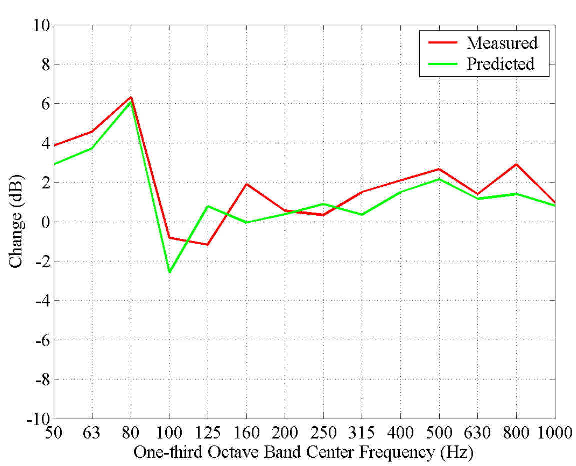

Figure 11. Comparison of the change in transmission loss due to the constrained layer damping.

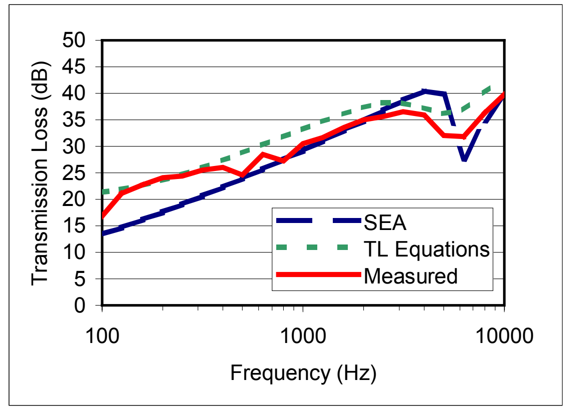

Figure 12. Measured and predicted sound power transmission loss versus frequency for uniform (216) window.

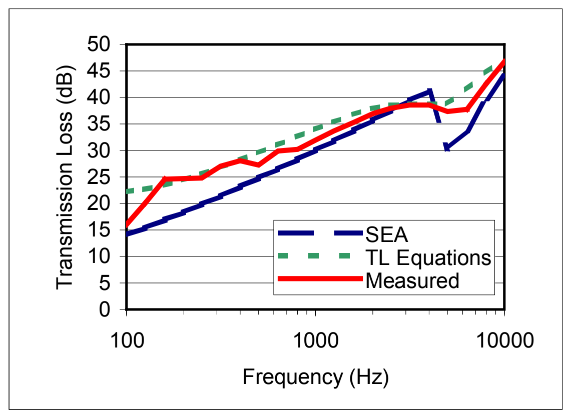

Figure 13. Measured and predicted sound power transmission loss versus frequency for damped (60,2,114,2,60) window.

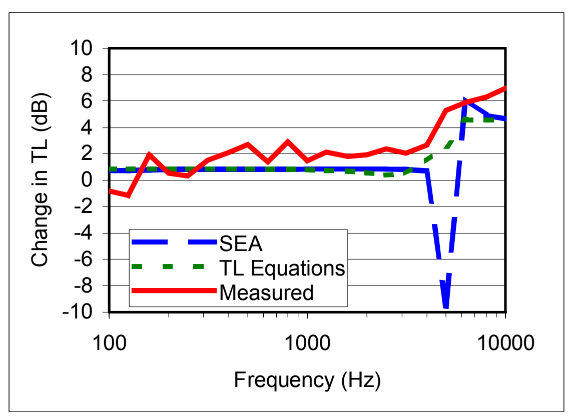

Figure 14. Comparison of the change in transmission loss due to the constrained layer damping.