Progress Report

for

A RESEARCH PROGRAM TO REDUCE INTERIOR NOISE IN GENERAL AVIATION AIRPLANES

KU-FRL-317-5

NASA Grant NSG 1301

Jan Roskam
Principal Investigator

Vincent U. Muirhead
Co-Investigator

Howard W. Smith
Co-Investigator

Prepared by: Ton Peschier
Don Durenberger
Kees van Dam
Tzy-Chuan Shu

October 1977
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<th>Definition</th>
<th>Dimension</th>
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<tr>
<td>(a)</td>
<td>Panel dimension</td>
<td>m</td>
</tr>
<tr>
<td>(b)</td>
<td>Panel dimension</td>
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<tr>
<td>(c)</td>
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<td>(f)</td>
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<tr>
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<tr>
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<td>(N)</td>
<td>Internal membrane force per unit length</td>
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<tr>
<td>(P)</td>
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<td>(s)</td>
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<tr>
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<td>Transmission loss</td>
<td>dB</td>
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<tr>
<td>(t)</td>
<td>Thickness of panel</td>
<td>m</td>
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### Greek Symbols

<table>
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<tr>
<th>Symbol</th>
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<tr>
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<tr>
<td>(\rho)</td>
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<tr>
<td>(\nu)</td>
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<tr>
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<tr>
<td>(\theta)</td>
<td>Angle of incidence of sound</td>
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1. **Introduction**

The objective of this NASA-sponsored KU-FRL noise project (NSG 1301) is to investigate experimentally and analytically the transmission of sound through isolated panels. The purpose of this report is to give the first test results of the panels tested up to mid-September.

In the month of August the KU-FRL noise research team completed the construction and the calibration of the test facility and started testing flat panels. During this time the team also continued predictions of behavior of the panels according to pertinent analytical methods. The first results have permitted improvement of semi-empirical methods of predicting panel transmission loss.

In this report the influence of stiffness and mass of the plate will be discussed. Also the effect of pressurization and the usefulness of damping materials will be described.

Gratitude is expressed to Beech, Cessna and Grumman American Aircraft Corporations for providing most of the panels, and some of the panel treatments, tested for this report. Appreciation is also expressed to Insul-Coustics Corporation and Specialty Composites Corporation for providing some of the vibration damping materials tested.
2. Description of the Test Facility

The KU-FRL test facility for measurement of sound transmission through panels is described in detail in Reference 1. This chapter will give only a brief description. A sketch of the test facility is shown in Figure 2.1, and a photograph in Figure 2.2.

The panel to be tested is mounted between the two chambers A and B. The source chamber A, consisting of a massive brick wall, concrete collar and steel box, contains nine evenly spaced loudspeakers. This chamber can be considered a speaker box. Its purpose is to support the speakers and to prevent radiation of sound to the rear and side. It contains sound absorbing materials to minimize standing waves that could induce undesired speaker sound radiation characteristics.

The panel under test is separated from the front side of the speaker baffle by a small distance, about one inch. This arrangement prevents standing waves between the baffle and specimen at frequencies in the range of interest, i.e. 20-5000 Hz. Other standing waves, parallel to the panel and the speaker baffle, would disturb the desired uniformity of excitation at the panel surface. The strength of these standing waves is reduced by sound absorbing material, which nearly fills the space between baffle and panel.

The receiving chamber (B in Figure 2.1) is an acoustic termination which absorbs very nearly all the sound passing through the panel. It also significantly improves the noise environment in which research personnel work; and at the same time, it influences the transmission of sound through the panel in the same way as an infinite space of air. To facilitate the installation of test specimens between this termination and the speaker box, the receiving chamber is mounted on wheels and rests on a steel table.

The loudspeakers can be driven by the amplified signal of either a white noise generator, a pure tone generator, or a tape recording of in-flight boundary layer pressure fluctuations. An equalizer is included in this noise generating system to obtain a reasonably flat noise spectrum.
The noise measuring system includes two microphones, one on each side of the panel under test. The output signals of these microphones are fed, one at a time, into a real time analyzer. The resulting spectra are plotted by an X-Y recorder. Next, these curves are read into a desktop calculator having curve digitizing capabilities, which subtracts one spectrum from the other, applies corrections and plots the final test results. The corrections account for sound energy reflected from the panel and for the different acoustic impedances on each side of pressurized panels.

For testing under simulated pressurized-airplane conditions, there is a depressurizing system to reduce the pressure in chamber A below the atmospheric pressure of chamber B.

Figure 2.3 shows details and dimensions of the test section and the clamping system. Each test panel is 20 in. x 20 in. However, the effective panel size is reduced to 18 in. x 18 in. by the one inch clamping margin on all edges. Each of the eight clamps is tightened with a torque wrench to 25 in. lb. torque. This corresponds to approximately 390 lb. force in each clamp, or a total clamping force of 3100 lb. on the panel edges.
Figure 2.1: The plane wave tube
Figure 2.2: The plane wave tube test facility
Figure 3.3: Detail of the panel clamping system.
3. The Influence of Stiffness

This chapter contains a discussion of the influence of the stiffness of a panel on its vibrations and transmission loss. Analytical and semi-empirical methods are included, and experimental results are given and compared with theory.

3.1 Analytical Consideration

Below the fundamental frequency the panel motion and noise transmission are controlled by stiffness. In general aviation aircraft, fundamental panel frequencies range from about 50 to 200 Hz, depending on panel size, material and edge conditions.

Reference 2 gives the following equation to estimate the transmission loss (TL) at frequency $\frac{f_{1,1}}{4}$:

$$TL (\frac{f_{1,1}}{4}, \text{stiffness}) = TL (f_{1,1}, 45^\circ, \text{mass law}) + 10 \log s^2 + 15 \ [\text{dB}]$$

(3.1)

where $f_{1,1} =$ the fundamental resonance frequency

$s =$ fraction of the total surface mass of the plate which participates fully in the motion at resonance ($s = 0.2$)

$TL (f_{1,1}, 45^\circ, \text{mass law}) =$ the mass law TL for $45^\circ$ incidence at the resonance frequency, considering the total surface mass.

The KU-FRL noise research team derived the following semi-empirical equation to predict the transmission loss at a certain frequency, $f$, below the fundamental resonance frequency:

$$TL = 20 \log \frac{\bar{m} (f_{1,1}^2 - f^2)}{f} - 39 \ [\text{dB}]$$

(3.2)

where $\bar{m} =$ the panel surface mass (kg/m$^2$)

For the derivation of this equation, see Reference 3.

One possible way to get a high transmission loss in the sub-resonance region is to raise the resonance frequency by rigidification of the panel. This effect can also be achieved by
decreasing the panel density \( \bar{m} \) and size or increasing the bending stiffness:

\[
D = \frac{Et^3}{12(1-\nu^2)} \quad [Nm]
\]  

where \( E \) = Young's modulus \( \frac{N}{m^2} \)

\( t \) = thickness (m)

\( \nu \) = Poisson's ratio

3.2 Experimental Results

Tests have been carried out on aluminum, plexiglass and steel panels of various thicknesses. A honeycomb panel and aluminum panels with two and four stiffeners have also been tested. The transmission loss curves of these different panels can be found in Chapter 7.

In Figure 3.1 it is shown how the bending stiffness influences the transmission characteristics. If the bending stiffness of a panel is increased, the transmission loss also increases. Note especially the high transmission loss of the honeycomb panel at lower frequencies.

The fundamental resonance frequencies of the test panels can easily be located as the point of lowest transmission loss.

3.3 Comparison

Figure 3.2 compares the actual transmission loss and the transmission loss predicted by Equation 3.2, in the stiffness controlled region. The predicted values are in fair agreement with the experimental results.

The experimental resonance frequencies differ slightly from the predicted frequencies. The discrepancies vary with the exact edge clamping conditions, as described in Reference 3.
Figure 3.1: The influence of bending stiffness on the sound transmission characteristics of 18" x 18" panels.

Test Conditions

- Temperature: 71° F.
- Remarks:
- Angle of Incidence: 90°
- Material: 023" thick aluminum (weight 79 lbs)
- Noise Source: Pure tone
- Pressure Differential: 0 psi

Calculation check

App.:

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Figure 3.2: Panel behavior in stiffness-controlled frequency region.

Transmission Loss ~ dB

FREQUENCY ~ Hz

0.01

0.1

1.0

10

100

1.0

10

100

0.01

0.1

1.0

10

100

0.01

0.1

1.0

10

100
4. The Influence of Mass

The mass of a panel has a significant influence on its vibrations and transmission loss. The influence of mass will be shown by analytical equations and by experimental results.

4.1 Analytical Considerations

Above the frequency range of the lower resonance modes, panels of finite dimensions behave approximately like infinite panels. Consequently, panel transmission loss above this range obeys the "mass law," which can be stated in theoretical form as:

\[
TL = 10 \log \left[ 1 + \left( \frac{\omega m}{2 \rho c} \cos \theta \right)^2 \right] \text{[dB]} \quad (4.1)
\]

where \( \theta \) = angle of incidence of the sound (incidence normal to the panel is defined as zero degrees)

\[
\omega = \frac{2 \pi f}{s} \quad \text{[rad]}\]

\( \rho = \text{density of air (kg/m}^3) \)

\( c = \text{speed of sound (m/s)} \)

This expression can be found in Reference 4. If the surface mass is doubled, the transmission loss increases by 6 dB. Equation 4.1 and other similar equations show that the transmission loss in this region is independent of stiffness and damping. From this equation it also follows that the transmission loss increases by 6 dB for each doubling of frequency.

4.2 Experimental Results

Chapter 7 contains the experimental results obtained up to mid-September. These results include the transmission loss curves of aluminum, steel and plexiglass panels of different thickness and with certain "sound reduction" treatments.

Figures 4.1 and 4.2 show the effect of thickness on transmission loss. Increased thickness gives higher transmission losses in both the mass controlled and the stiffness controlled frequency regions. Transmission loss increases with thickness in the mass law region because of the higher surface density of the panels, while the TL increase in the stiffness region is due to the higher bending
stiffness of the thicker panels. The same shift is demonstrated when comparing the transmission loss curves of aluminum panels with different thicknesses.

4.3 **Comparison**

The theory predicts that the transmission loss in the mass controlled frequency region will increase 6 dB per doubling of frequency. The straight line which is drawn in the figures of Chapter 7 represents the least-squares curve fit of the data of that particular frequency region. The slope of this line, in almost all the figures, falls between 5 and 7 dB per octave, showing fair agreement between theory and experiments. When the weight is doubled, the transmission loss is predicted to increase by 6 dB. In Figures 4.1 and 4.2 there is an increase of 6 dB at low frequencies. However, the slope of the curve decreases, so the difference between the two least-squares curve fit lines is smaller than 6 dB at higher frequencies. This difference may be due to coincidence phenomena which reduce the transmission loss below mass law at frequencies near the coincidence frequency. Such an effect would indeed be expected to be stronger for thicker panels. Figures 4.3 and 4.4 compare the mass law transmission loss (Equation 4.1) with the average (least-squares curve fit) of the experimental data. The difference of 3 dB is typical for the KU-FRL test facility. Other test facilities have reported similar variations between theory and experimental results, as stated in Reference 3.
Figure 4.1: The effect of thickness on the sound transmission characteristics of an 18" x 18" panel.
Figure 4.2: The effect of thickness on the sound transmission character.

Transmission Loss ~ dB

Frequency ~ Hz

Test Conditions
Pressure Differential: 0 psig
Panel Weight Differential: 100 lb/ft²
Material: 2024-T3 Aluminum
Noise Source: Far Field

Remarks: 0.125" thick (weight 1.0 lb/ft²)

Remarks: 0.49" thick (weight 0.49 lb/ft²)
Figure 4.4: Comparison between predicted and measured influence of mass on the transmission loss.

**Test Conditions**

- Temperature: 71°F
- Angle of Incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: 0.032" (0.00032 m)
- Noise Source: Pure Tone

---

**Graph Details**

- **FREQUENCY (Hz)**: Range from 2 Hz to 5 kHz.
- **TRANSMISSION LOSS (dB)**: Range from -10 dB to 50 dB.

---

**Line Descriptions**

- Least-squares curve fit of measured data
- Hase-law (Equation 4.1)
5. The Influence of Pressurization

A pressure differential across a panel has a stiffening effect which tends to increase transmission loss below the major resonance frequency. This will be demonstrated by using an expression derived by the KU-FRL noise research team and by showing experimental results.

5.1 Analytical Considerations

Pressurization raises the transmission loss curve at lower frequencies by raising the fundamental resonance frequency several octaves. Applying a pressure differential across the panel has no effect on panel surface density, so the "mass law" transmission loss above the new fundamental resonance frequency is not changed significantly.

Several methods are available to compute the vibrations of a panel subjected to an acoustic excitation, but including the effect of a pressure differential across the panel appears more difficult. Assuming the pressure differential across the plate results in an internal membrane stress, which is constant throughout the plate, the fundamental resonance frequency of the panel is (Reference 3):

\[ f_{1,1} = \frac{K}{Z} \sqrt{\frac{D}{m}} \sqrt{\frac{2\pi}{\left(\frac{1}{a} + \frac{1}{b}\right)^2}} + \frac{N}{D} \left[\frac{1}{a^2} + \frac{1}{b^2}\right] [Hz] \] (5.1)

where:
- \( D \) = the bending stiffness of the plate (Nm)
- \( m = \) the panel surface mass (kg/m²)
- \( a, b = \) the dimensions of the panel (m)
- \( K = 1.81 \) for a panel with clamped edges

and the internal membrane force per unit length is:

\[ N = 0.21 t \left[ \frac{\Delta P x a^2}{t} \right]^{1/3} \left[ \frac{N}{m} \right] \] (5.2)

where \( \Delta P = \) pressure difference across panel (Nm²)

Using Equation 3.2 the transmission loss for frequencies below major resonance can be calculated.

5.2 Experimental Results

To investigate the influence of pressurization the KU-FRL noise research team measured the transmission loss of several panels with
pressure differences across them. The experimental transmission loss curves of several panels at different pressure differentials can be found in Chapter 7.

Figure 5.1 shows the experimental influence of a pressure differential across an aluminum panel. This figure indicates that the first 0.5 psi of pressure differential greatly increases transmission loss at the lower frequencies. The transmission loss in the mass controlled region changes hardly at all. The same result can be seen in Figures 5.2 and 5.3.

5.3 Comparison

Figure 5.4 compares measured and predicted influence of pressure on the fundamental resonance frequency. The predicted frequencies were calculated by Equation 5.1. This figure demonstrates a fair agreement between both predicted and measured data. The comparison between measured and predicted influence of pressure on the transmission loss is shown in Figure 5.5. Here also both lines show fair agreement. Notice the large transmission loss increase resulting from a small pressure difference.
Transmission Loss ~ dB

Frequency ~ Hz

Temperature: 70°F
Angle of Incidence: 90°
Panel Treatment: none
Material: 2024-T3 Aluminum
Test Conditions:
- 1.0 psi pressure differential
- 2.0 psi pressure differential
- 0.5 psi pressure differential
- 1.0 psi pressure differential

Figure 5.1: The influence of a
Figure 5.2: The influence of a pressure differential across an 18" x 18" panel on the sound transmission characteristics.

**Test Conditions**
- Temperature: 70° F.
- Angle of Incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: .020""
- Noise Source: Pure Tone

**Pressure Differential** (see "Remarks")
- Panel Weight (18"x18") .62 lbs

**Remarks**
- 0 psi pressure differential
- 0.5 psi pressure differential
- 1.0 psi pressure differential

**Graph**: Transmission loss vs. frequency (in Hz).
Figure 5.3: The effect of a pressure differential across an 18" x 18" panel on the sound transmission characteristics.

Test Conditions

Temperature: 70°F
Angle of Incidence: 90°
Material: Steel
Thickness: .020"
Noise Source: Pure Tone

Pressure Differential: (see "Remarks")
Panel Weight (18"x18") 1.92 lbs.

Remarks
- 0 psi pressure differential
- .5 psi pressure differential
- 1.0 psi pressure differential
Figure 5.4: Comparison between measured and predicted influence of pressure on the fundamental resonance frequency.
Figure 5.5: Comparison between measured and predicted influence of pressure on the transmission loss.
6. **The Influence of Vibration Damping Material**

The dynamic responses and sound barrier properties of a structure are governed primarily by mass, stiffness and damping. The influence of stiffness and mass have already been discussed in Chapters 3 and 4. Damping is the most complicated property and is difficult to predict. When a structure deforms, it dissipates some energy. It is this energy dissipation—this conversion of mechanical energy to thermal energy—that is called damping.

The primary effect of increased panel damping, under the conditions of KU-FRL noise research, is the reduction of vibration amplitudes at resonances. According to Reference 4, care must be exercised in damping measurements to prevent extraneous energy dissipation. To minimize support-related damping in damping measurements, it is usually necessary to suspend the test panel from two long strings. In the KU-FRL situation, however, the edges are clamped. Thus, the prediction of the total damping experienced by a test panel will be very difficult.

However, by comparing the transmission loss of a bare panel with the transmission loss of a panel treated with a viscoelastic material, the influence of the viscoelastic layer can be readily seen. To investigate possible weight savings, some panels were only partially covered with a viscoelastic layer.

In the frequency region well below the fundamental resonance, the transmission loss is independent of damping, as shown in Chapter 4. However, when a panel has been treated with a damping material, its mass will be greater, so the transmission loss in the (high frequency) mass law region will increase. Such an increase in TL is due to the greater weight of the treated panel, not to the damping material itself. The increase in transmission loss due to the increase in weight can be predicted with Equation 4.1.

6.1 **Experimental Results**

The KU-FRL noise research team is continuing to test the influence of vibration damping materials. Figures 6.1 shows how vibration damping materials affect the sound transmission loss of an aluminum panel. The average transmission loss is hardly changed by adding a layer of
vibration damping material; however, the resonance peaks become smaller. It can be noted that the shift of the least-squares curve fit to a higher transmission loss is due to the higher surface density of the treated panel. Figure 6.2 demonstrates the influence of partial coverage. The potential user himself must weigh the lower weight of the partially covered panel against reduced sound blockage of the resonance frequencies. Chapter 7 contains additional transmission loss curves of panels treated with vibration damping materials.
Figure 6.1: The effect of vibration damping materials on the sound transmission characteristics of 18" x 18" panels.

Test Conditions:
- Temperature: 71°F
- Angle of Incidence: 90°
- Panel Treatment:
  - None (weight 1.29 lbs)
  - 100% LD-400 (weight 1.63 lbs)
  - 100% Y-370 (weight 1.64 lbs)
- Material: 2024-T3 Aluminum
- Thickness: 0.032 in.
- Noise Source: Pure tone

* see Appendix A

Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.
Test Conditions

Temperature: 71° F.
Pressure Differential: 0 psi
Angle of Incidence: 90°
Panel Weight (18"x18"): (see "Remarks")
Material: 2024-T3 Aluminum

Remarks:

Thickness: .032"
Noise Source: Pure Tona

40% Y-370" (weight 1.28 lbs.)
100% Y-370X (weight 1.64 lbs.)

* see Appendix A

Figure 6.2: The effect of vibration damping materials on the sound transmission characteristics of 18" x 18" panels.

Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.
7. Test Results

The tests in the following table have been carried out up to mid-September. The figures referred to in the table can be found on the pages following. Numbers in parentheses refer to the manufacturers of the acoustic materials. See Table 7.2.

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<th>Figure</th>
<th>Material</th>
<th>Thickness (in.)</th>
<th>Test Condition or Treatment</th>
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<td>.016</td>
<td>bare</td>
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<td>.020</td>
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<td>.032</td>
<td>3.0 psi</td>
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<tr>
<td>7.38</td>
<td>Honeycomb</td>
<td>.50</td>
<td>bare</td>
</tr>
<tr>
<td>7.39</td>
<td>Honeycomb</td>
<td>.50</td>
<td>100 % Y-370 (1)</td>
</tr>
<tr>
<td>7.40</td>
<td>&quot;Noiseless Steel&quot;</td>
<td>≈ .050</td>
<td>2 sheets of steel with vibration damping material between them</td>
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Table 7.2 List of Manufacturers of Acoustic Materials

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<tr>
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<tr>
<td>1</td>
<td>3-M, Minneapolis, Mn.</td>
</tr>
<tr>
<td>2</td>
<td>Lord Corp., Erie, Pa.</td>
</tr>
<tr>
<td>3</td>
<td>Insul-Couotic/Burma Corp., Sayville, N.Y.</td>
</tr>
<tr>
<td>4</td>
<td>Specialty Composites Corp., Newark, Del.</td>
</tr>
</tbody>
</table>
Figure 7.1: Experimental sound transmission loss of an aluminum panel.

Test Conditions

Temperature: 70°F
Pressure Differential: 0 psi
Angle of Incidence: 90°
Panel Weight (18"x18") .49 lbs.
Material: 2024-T3 Aluminum
Remarks:

Thickness: .016"
Noise Source: Pure Tone
Test Conditions
Temperature: 70° F
Pressure Differential: 0 psi
Angle of Incidence: 90°
Panel Weight (18"x18")': .62 lbs.
Material: 2024-T3 Aluminum
Remarks:
Thickness: .020"
Noise Source: Pure Tone

Transmission Loss

FREQUENCY ~ Hz
Figure 7.3: Experimental sound transmission loss of an aluminum panel.

Test Conditions:
- Temperature: 71° F.
- Pressure Differential: 0 psi
- Angle of Incidence: 90°
- Panel Weight: (18"x18") .79 lbs.
- Material: 2024-T3 Aluminum
- Thickness: .025"
- Noise Source: Pure Tone

FREQUENCY ~ Hz

TRANSMISSION LOSS

-10 0 10 20 30 40 50

 mascara
Test Conditions

Temperature: 71° F.
Pressure Differential: 0 psi

Angle of Incidence: 90°
Panel Weight (18"x18") : 1.04 lbs.

Material: 2024-T3 Aluminum
Remarks:

Thickness: .032"
Noise Source: Pure Tone
Figure 7.5: Experimental sound transmission loss of an aluminum panel.

Test Conditions
- Temperature: 71°F
- Pressure Differential: 0 psi
- Angle of Incidence: 90°
- Panel Weight (18"x18") A: 1.26 lbs
- Material: 2024-T3 Aluminum
- Remarks:
- Thickness: 0.040"
- Noise Source: Pure Tone
Figure 7.6: Experimental sound transmission loss of a steel panel.

Test Conditions
- Temperature: 71°F
- Pressure Differential: 0 psi
- Angle of Incidence: 90°
- Panel Weight (18"x18"): 1.44 lbs.
- Material: MIL-S-5059 Steel
- Thickness: .016"
- Noise Source: Pure Tone

Transmission Loss vs. Frequency
Test Conditions

Temperature: 71° F.
Pressure Differential: 0 psi
Angle of Incidence: 90°
Panel Weight (18"x18") : 1.92 lbs.
Material: Steel
Thickness: .020"
Remarks:
Noise Source: Pure Tone

Figure 7.7: Experimental sound transmission loss of a steel panel.
Figure 7.9: Experimental sound transmission loss of a plexiglass panel.

**Test Conditions**
- Temperature: 73°F
- Angle of incidence: 90°
- Material: Plexiglass
- Thickness: .1875"
- Noise Source: Pure Tone

- Pressure Differential: 0 psi
- Panel Weight (18"x18"): 2.46 lbs.

**Remarks**
Figure 7.10: Experimental sound transmission loss of a plexiglass panel.

Test Conditions:
- Temperature: 71°F
- Angle of Incidence: 90°
- Material: Plexiglass
- Thickness: 0.250"
- Noise Source: Pure Tone

Pressure Differential: 0 psi
Panel Weight (18"x18"): 3.37 lbs.
Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

Panel treatment application pattern.

**Figure 7.11:** Experimental sound transmission loss of an aluminum panel.

**Test Conditions**

- Temperature: 71° F
- Pressure Differential: 0 psi
- Angle of incidence: 90°
- Panel Weight (18"x18") : 1.28 lbs.
- Material: 2024-T3 Aluminum
- Remarks: *  
- Thickness: .032"
- Noise Source: Pure Tone
- +40% Y-370 (.25")

* see Appendix A
Panel treatment application pattern.

Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

Test Conditions

- Temperature: 71°F
- Pressure Differential: 0 psi
- Angle of Incidence: 90°
- Panel Weight (18"x18''): 1.44 lbs.
- Material: 2024-T3 Aluminum
- Remarks: *
- Thickness: .032"
- Noise Source: Pure Tone
  + 70% Y-370 (1.25"

* see Appendix A
Figure 7.13: Experimental sound transmission loss of an aluminum panel.

Panel treatment application pattern.

Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

Test Conditions
- Temperature: 71°F
- Angle of Incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: .032"
- Noise Source: Pure Tone
- Pressure Differential: 0 psi
- Panel Weight (18"x18") 1.64 lbs.
- Remarks: * + 100% Y-370 (.25"
- Test Conditions: see Appendix A
Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

Panel treatment application pattern.

Test Conditions

- Temperature: 70°F
- Angle of Incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: .032" (0.81 mm)
- Noise Source: Pure Tone
- Panel Weight (18"x18"): 1.27 lbs.
- Pressure Differential: 0 psi
- Remarks: +39.5% LD-400

* see Appendix A
Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

Panel treatment application pattern.

Test Conditions
Temperature: 70° F.
Pressure Differential: 0 psi
Angle of Incidence: 90°
Panel Weight (18"x18") 1/41 lbs.
Material: 2024-T3 Aluminum
Remarks: + 63.2% LD-400
Thickness: .032"
Noise Source: Pure Tone

* see Appendix A
Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

Panel treatment application pattern.
TRANSMISSION LOSS ~ db

FREQUENCY ~ Hz

Temperature: 70°F
Angle of Incidence: 90°
Material: .032" Aluminum
Noisy Source: Pure Tone

Test Conditions
Pressure Differential: 0 psi
Remarks: +100% LD-400
* see Appendix A

Sound energy dissipation may be significant near the fundamental resonance frequency. This effect was assumed to be nonexistent in this case.
Figure 7.18: Experimental sound transmission loss of an aluminum panel.

Test Conditions:
- Temperature: 70°F
- Angle of incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: 0.090" (0.016" sheets with IC-998° between them)
- Noise Source: Pure Tone
- Pressure Differential: 0 psi
- Panel Weight (18"x18"), 1.01 lbs.

Remarks:
- 2 x 0.016" sheets with IC-998° between them

* See Appendix A
Figure 7.19: Experimental sound transmission loss of an aluminum panel.

Test Conditions

- Temperature: 73°F
- Pressure Differential: 0 psi
- Angle of Incidence: 90°
- Panel Weight (18"x18"): 0.98 lbs.
- Material: 2024-T3 Aluminum
- Remarks: + 2 Stiffeners
- Thickness: 0.025"
- Noise Source: Pure Tone

FREQUENCY ~ Hz

Transmission Loss

Panel-stiffener geometry.
Transmission Loss

Test Conditions

- Pressure Differential: 0.051
- Panel Weight: 19.4 lb/ft²
- Remarks: +4 Stiffeners

Panel: 6061 Aluminum

Thickness: 0.025 in

Noise Source: Pure Tone

Angle of Incidence: 30°

Temperature: 72° F

Figure 7.20: Experimental sound transmission loss of an aluminum panel.
Figure 7.21: Experimental sound transmission loss of an aluminum panel.

Test Conditions

- Temperature: 70°F
- Angle of incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: .025" (0.64 mm)
- Noise Source: Pure Tone
- Pressure Differential: 1.0 psi
- Panel Weight (18"x18"): 1.07 lbs.

Remarks:
- +4 stiffeners

Panel-stiffener geometry.

Stiffener cross-section.
Figure 7.22: Experimental sound transmission loss of an aluminum panel.

Test Conditions

Temperature: 69°F
Angle of Incidence: 90°
Material: 2024-T3 Aluminum
Thickness: 0.025"

Pressure Differential: 2.0 psi
Panel Weight (18"x18") : 1.07 lbs.
Remarks
+ 4 stiffeners

Noise Source: Pure Tone

Panel-stiffener geometry.

Stiffener Cross Section
Transmission loss vs. frequency graph.

Figure 7.23: Experimental sound transmission loss of an aluminum panel.

Test Conditions:
- Temperature: 69°F
- Pressure Differential: 3.0 psi
- Angle of Incidence: 30°
- Panel Weight: (18" x 18") 1.07 lbs.
- Material: 2024-T3 Aluminum
- Remarks: +4 stiffeners
- Noise Source: Pure Tone

Panel-stiffener geometry:
- 10" 18"
- 125°
- 1/8" 0.032"
Test Conditions

Temperature 69°F
Angle of incidence 90°
Pressure Differential 0 psf
Panel Weight (18"x18") 1.19 lbs.
Material 2024-T3 Aluminum
Remarks +4 stiffeners Y-370 in the middle
Thickness .025"
Noise Source Pure Tone

Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

* see Appendix A

TRANSMISSION LOSS

FREQUENCY — Hz

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Figure 7.25: Experimental sound transmission loss of an aluminum panel.

Panel-stiffener geometry.

Stiffener Cross Section

Transmission Loss vs Frequency

Test Conditions
- Temperature: 65°F
- Angle of Incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: .025" (0.250"
- Noise Source: Pure Tone
- Panel Weight (16"x18") = 19 lbs.
- Pressure Differential: 1.0 psig

Remarks:
- + 4 stiffeners + Y-370 in the middle

Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.
Test Conditions

Temperature: 70°F
Angle of Incidence: 90°
Material: Steel
Thickness: .020"
Noise Source: Pure Tone

Pressure Differential: 0.5 psi
Panel Weight (10' x 10'): 1.92 lbs.
Remarks:

Figure 7.26: Experimental sound transmission loss of a steel panel.
**Figure 7.27:** Experimental sound transmission loss of a steel panel.

Transmission Loss $\Delta L$ vs Frequency $f$.

- **Temperature:** 70°F
- **Material:** Steel
- **Thickness:** 0.020”
- **Angle of Incidence:** 90°
- **Test Conditions:**
  - Sound Source: Tone
  - Noise Source: Fix
  - Panel Weight (18 x 18”): 1.22 lbs

Remarks:

- Differential Pressure: 1.0 psi
Test Conditions

Temperature 70° F.
Pressure Differential: 2.0 psi
Angle of Incidence 90°
Panel Weight (18"x18"): 1.92 lbs.
Material: Steel
Remarks

Transmission Loss of a steel panel.

Frequency ~ Hz
Figure 7.30: Experimental sound transmission loss of an aluminum panel.

Transmission Loss ~ dB

FREQUENCY ~ Hz

-100 -80 -60 -40 -20 0 20 40 60

-10 -8 -6 -4 -2 0 2 4 6 8 10 12 14 16 18 20

Temperature: 71°F
Angle of Incidence: 90°
Material: 2024-T3 Aluminum
Panel Weight (18x18in): 19 lbs
Noise Source: Pure Tone
Remarks:

Test Conditions:
Pressure Differential: 1.0 psi

Panel: 0.6in
Figure 7.31: Experimental sound transmission loss of an aluminum panel.

Test Conditions

Temperature: 71° F.
Pressure Differential: 1.5 psi
Angle of Incidence: 90°
Panel Weight (18" x 18") : .49 lbs.
Material: 2024-T3 Aluminum
Remarks:
Thickness: .016".
Noise Source: Pure Tone.
Figure 7.32: Experimental sound transmission loss of an aluminum panel.

Test Conditions

- Temperature: 71° F.
- Angle of incidence: 90°
- Material: 2024-T3 Aluminum
- Pressure Differential: 0.5 psi
- Panel Weight (18"x18") - 0.62 lbs.
- Thickness: .020"
- Noise Source: Pure Tone

Remarks:
Test Conditions

Temperature: 71° F.
Pressure Differential: 1.0 psi
Angle of Incidence: 90°
Panel Weight (18''x18''): .62 lbs.
Material: 2024-T3 Aluminum
Thickness: .020''
Remarks.
Noise Source: Pure Tone

Figure 7.33: Experimental sound transmission loss of an aluminum panel.
Figure 7.34: Experimental sound transmission loss of an aluminum panel.

Test Conditions

Temperature: 71°F
Angle of Incidence: 90°
Material: 2024-T3 Aluminum
Thickness: 0.020"
Noise Source: Pure Tone

Pressure Differential: 1.5 psi
Panel Weight (18" x 18") : 0.62 lbs.
Remarks:
Figure 7.35: Experimental sound transmission loss of an aluminum panel.

Test Conditions

- Temperature: 70°F
- Angle of Incidence: 90°
- Material: 2024-T3 Aluminum
- Thickness: .032" 
- Panel Weight (18"x18") 1.04 lbs.
- Noise Source: Pure Tone
- Pressure Differential: 1.0 psi

Remarks:
Figure 7.36: Experimental sound transmission loss of an aluminum panel.

Test Conditions

Temperature: 70°F
Angle of Incidence: 90°
Material: 2024-T3 Aluminum
Pressure Differential: 2.0 psi
Panel Weight (18"x10") = 1.04 lbs
Thickness: 0.032"
Noise Source: Pure Tone
Figure 7.37: Experimental sound transmission loss of aluminum panel.

Test Conditions
- Temperature: 70°F
- Pressure Differential: 3.0 psig
- Angle of Incidence: 90°
- Panel Weight (18" x 18"), 1.04 lbs.
- Material: 2024-T3 Aluminum
- Thickness: .032"
- Noise Source: Pure Tone
Figure 7.38: Experimental sound transmission loss of a honeycomb aluminum panel.

Test Conditions:
- Temperature: 70°F
- Angle of Incidence: 90°
- Pressure Differential: 0 psi
- Material: Honeycomb (Aluminum)
- Panel Weight (18"x18") = 1.64 lbs.
- Remarks:
  - Noise Source: Pure Tone
  - Thickness: .50"
Sound energy dissipation by vibration damping material may be significant near the fundamental resonance frequency. This effect was assumed to be negligible and is still being investigated.

**Test Conditions**
- Temperature: 72° F.
- Angle of Incidence: 90°
- Material: Honeycomb (Aluminum)
- Thickness: .50"
- Noise Source: Pure Tone
- Pressure Differential: 0 psi
- Panel Weight (18"x18") : 2.30 lbs.
- Remarks: +100% Y-370 (.25"")

Panel treatment application pattern.
Figure 7.40: Experimental sound transmission loss of a "noiseless steel."
8. Conclusions

The panel test results presented in this report follow predicted trends and generally agree with predicted values of sound transmission loss as well. That is, the well known mass law gives results close to experiment, in the applicable frequency range. An improved stiffness law, which agrees with experiment more closely than do previous stiffness laws, is given in the report.

Slight errors in resonant frequency predictions are thought due to less-than-ideal experimental edge conditions. This is not considered a serious drawback, since actual aircraft panels are likewise "imperfectly" clamped.

Pressurization of panels is found to dramatically increase transmission loss in the low frequency stiffness controlled region. The mechanism responsible is thought to be increased internal membrane stresses; a semi-empirical equation is presented which predicts low frequency transmission loss reasonably well.
References


Appendix A. Vibration Damping Materials

In this appendix some data of the materials applied to panels to prevent vibration will be given.

IC-998

This product is made by Insul-Coustic/Birma Corporation. It is a visco-elastic bonding compound and is designed for bonding of secondary sheet metal panels as treatment for resonant vibration. IC-998 does not require full coverage, but 1/8 in. beads of material applied in a specific design pattern onto resonating surfaces.

- Color: brown
- Viscosity: thick paste
- Application: pneumatic or hand extrusion equipment
- Surface temperature limits: -40° F. to 250° F.
- Safety: wet - non-flammable
  - dry - fire resistant
  - Conforms to NFDA 90 A.

LD-400

This vibration damping material is produced by Lord Corporation. It is a visco-elastic damping material, and the thickness is = .030 in. The material has to be glued to the panel.

Y-370

This constrained layer visco-elastic damping tape is made by 3M. It is available in several thicknesses; the present research is conducted with the generally used 1/4 in. thickness.

Noiseless Steel

Noiseless steel is a steel/visco-elastic damper/steel sandwich panel produced by Specialty Composites. The total thickness of this panel is = .050 inch.
CRINC LABORATORIES

Chemical Engineering Low Temperature Laboratory
Remote Sensing Laboratory
Flight Research Laboratory
Chemical Engineering Heat Transfer Laboratory
Nuclear Engineering Laboratory
Environmental Health Engineering Laboratory
Information Processing Laboratory
Water Resources Institute
Technical Transfer Laboratory
Air Pollution Laboratory
Satellite Applications Laboratory