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Noise Transmission Characteristics of Advanced Composite Structural Materials
L.A. Roussos and C.A. Powell, NASA Langley Research Center, Hampton, VA; and F.W. Grosveld, The Bionetics Corp., Hampton, VA; and L.R. Koval, Univ. of Missouri-Rolla, Rolla, MO
NOISE TRANSMISSION CHARACTERISTICS OF ADVANCED COMPOSITE STRUCTURAL MATERIALS

Louis A. Roussos
NASA Langley Research Center
Hampton, Va.

Ferdinand W. Grosveld*
The Rimetics Corporation
Hampton, Va.

Leslie R. Koval*
University of Missouri-Rolla
Rolla, Mo.

Clemans A. Powell
NASA Langley Research Center
Hampton, Va.

Abstract

An experimental and theoretical research program has begun to develop an understanding of the noise transmission characteristics of composite materials. Such an understanding will ensure that the weight advantage of composites in aircraft fuselage design is not compromised by high noise transmission or heavy acoustic treatments. Noise transmission tests have been conducted on large unstiffened panels representative of the outer skin or inner trim panels of aircraft fuselages. Also, an analytical model based on infinite panel theory has been developed which allows for exact modeling of the anisotropic properties of the panels.

In the mass-controlled and coincidence frequency regions, agreement between the measured and analytical noise transmission loss was quite good. A theoretical design comparison between aluminum and composite general-aviation type panels based on equal critical shear load showed that graphite/epoxy and Kevlar/epoxy panels allowed 3 to 4 dB more noise transmission than an aluminum panel over most of the frequency range due to their lighter weight and 6 to 12 dB less transmission loss at high frequency because of their lower critical frequencies.

A finite panel field incidence transmission loss theory has also been developed. Preliminary calculations for oblique incidence transmission loss indicate that improved low frequency transmission loss may be possible with composites relative to conventional aluminum panels.

Nomenclature

- \( f_{\text{coinc}} \): coincidence frequency, Hz
- \( f_{\text{crit}} \): critical frequency, Hz
- \( f_L \): lower frequency of a one-third octave band, Hz
- \( f_U \): upper frequency of a one-third octave band, Hz
- \( F/E \): fiberglass/epoxy
- \( G \): isotropic plate shear modulus, Pa
- \( G_{12} \): shear modulus for composite tape ply, Pa
- \( G/E \): graphite/epoxy
- \( h \): panel thickness, cm
- \( i \): integer designating the \( k \)-th layer of a composite panel
- \( k_x \): wave number in \( x \) direction = \((2\pi f \sin \theta_1 \cos \phi_1)/c, \ m^{-1}\)
- \( k_y \): wave number in \( y \) direction = \((2\pi f \sin \theta_1 \sin \phi_1)/c, \ m^{-1}\)
- \( K/E \): Kevlar/epoxy
- \( \bar{w} \): mass per unit area, kg/m²
- \( N \): number of layers in composite panel
- \( N_R \): noise reduction, dB
- \( p \): pressure, Pa
- \( P_t \): amplitude of an incident pressure wave, Pa
- \( P_t \): amplitude of a transmitted pressure wave, Pa
- \( P_t \): the reduced stiffnesses relating stress to strain in a composite ply, Pa
- \( q_{ij} \): surface area, m²
- \( S \): power spectral density of the incident pressure at frequency \( f \), Pa²/Hz
- \( T \): transmission loss, dB
- \( T_{\text{ML}} \): field incidence mass law transmission loss, dB
- \( w \): plate displacement, m
- \( X, Y, Z \): panel coordinate axes
- \( x, y, z \): displacement along respective axes, m
- \( z_k \): \( z \)-direction distance from panel middle surface to bottom of the \( k \)-th layer (see fig. 4), m
- \( \Delta f \): narrow frequency bandwidth, Hz
- \( \eta \): damping loss factor
- \( \theta \): angle of fiber direction relative to a specified boundary axis of the panel, deg
- \( \theta_1 \): the angle of incident pressure wave relative to the \( Z \) axis (the axis normal to the panel), deg
- \( \nu \): isotropic plate Poisson's ratio

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1Kevlar is a registered trademark for an aramid fiber produced by E. I. DuPont de Nemours & Co.

Number AI 328
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composite tape ply Poisson's ratio
mass density of air, kg/m³
transmission coefficient
field incidence transmission coefficient
angle of incident pressure wave relative to the X axis of the panel when projected into the plane of the panel.

Introduction

The noise transmission characteristics of composite materials have been identified as a principal design consideration for aircraft fuselages. These characteristics must be taken into account early in the design process to ensure that the weight-saving advantage of composite construction is not compromised by high noise transmission which would necessitate heavy add-on acoustic treatments. Studies of noise transmission of composites, either experimental or theoretical, have been limited in both number and scope. The experimental study of Yang and Tsu considered only three panels and was conducted in a small facility with a usable frequency range of 400 Hz to 10 kHz. The theoretical study by Revell et al. investigated the effects of high-stiffness, low-mass isotropic materials on noise transmission rather than effects of actual composite type materials. The theoretical study by Kova, which provided the first model for noise transmission loss of composite constructions, was for an infinite monocoque cylindrical shell. This study was limited to a few configurations, was not substantiated by experimental data, and compared aluminum and composite configurations only for conditions of equal mass.

Many factors must be considered in the design of a fuselage structure from the standpoint of noise transmission. The first consideration is the level and frequency characteristics of the incident sound. The design of the fuselage structure to meet the required interior noise criteria must then consider those factors which affect the frequency characteristics of the transmission loss provided by the structure. These include the outer wall (skin, stringers, and ring frames) as well as applied damping materials, additional stiffening treatments such as honeycomb, sound absorbing materials, and the spacing and materials for interior trim panels. Composites in particular are expected to play an increasing role in outer load bearing walls as well as interior trim panels in future aircraft designs.

The NASA Langley Research Center has begun a theoretical and experimental program to provide the necessary noise transmission information for composite structures so that these characteristics can be incorporated early in design phases for weight-efficient aircraft structures. The objective of the first phase of this program is to determine how composite structures will affect fuselage noise transmission relative to current aluminum structures. This paper presents the results of a theoretical and experimental study of noise transmission of large unstiffened panels representative of aircraft outer skins and interior trim which was conducted to meet this objective.

The experimental part of the study will be described in the first few sections of the paper.

The first section describes how the tests were conducted using the "two room" transmission loss measurement method in a facility providing acceptable results that include one-third octave bands with center frequencies greater than or equal to 100 Hz. Details of the construction (materials, lay-up, and properties) of the 14 composite panels which were tested will then be presented. The development of the theory for the transmission loss of the panels is then described. This theory is an application of the infinite composite cylinder analysis of Kova to infinite flat anisotropic panels. Measured transmission loss of the panels is compared with both field incidence mass law theory and the newly developed infinite panel theory. Using the infinite panel theory, design comparisons for composite and aluminum panels are presented based on equal critical shear load carrying capability.

The final section of the paper will discuss a finite panel theory which has been developed. Calculations natural from structural and effects of fiber orientation for the test panels are also presented. In addition, design comparisons based on this finite panel theory are discussed for the same types of panels used in the design comparisons based on infinite panel theory.

Description of Experimental Method

To experimentally establish the noise transmission characteristics of the composite test panels, they were mounted as partitions in between two adjacent rooms which are designated source room and receiving room. Top and side views of the transmission loss apparatus are shown in Fig. 1. In the source room, which measures 3.35 m by 3.66 m by 3.94 m, a diffuse field is established by two reference sound power sources. Sound is transmitted from the source room into the receiving room only by way of the test panel, which has a sound exposed vibrating area of 0.85 m by 1.46 m. The test specimen is mounted in a steel frame, which is designed for minimum structural flanking. The receiving room, with dimensions of 3.36 m by 3.36 m by 2.90 m, is acoustically and structurally isolated from the rest of the building. A space and time average of the sound pressure levels is taken in
each of the rooms by a microphone mounted at the end of a rotating boom. The noise reduction (NR), defined as the difference between the measured averaged sound pressure levels in the source and receiving rooms, includes characteristics of the test specimen as well as room characteristics. The Classical Method of measuring transmission loss assumes that the sound pressure levels are measured in a diffuse field in both the source and receiving rooms. By correcting for the room characteristics using the measured absorption area $A$ of the receiving room, the noise transmission loss (TL), which is a function of the properties of only the test specimen, can be calculated by

$$ TL = NR + 10 \log (S/A) \tag{1} $$

where $S$ is the surface area of the test specimen. The underlying assumptions for this technique are not applicable for the present setup in the low frequency region (below 500 Hz) due to the large wavelength of sound relative to the dimensions of the rooms and microphone booms. In reference 6 the "plate reference method" is suggested to more accurately measure the TL in a frequency range from 100 Hz to 10 kHz. This method not only corrects the noise reduction for the absorption in the receiving room but also corrects for the nondiffusivity of both rooms by assuming a mass law of behavior of the reference panel.

With the plate reference method, equation (1) is re-written as,

$$ TL = NR + 10 \log (S/A_{eq}) \tag{2} $$

where $A_{eq}$ is an equivalent absorption area. Assuming that TL follows field incidence mass law, equation (2) can be solved for $A_{eq}$

$$ A_{eq} = S \times 10^{(\log TL - \log NR)/10} \tag{3} $$

$TL_{ML}$ is the field incidence mass law transmission loss given by,

$$ TL_{ML} = 10 \log \left[ \frac{0.978 f^2}{pc} \right] \left[ \frac{1 + \frac{\sin f}{pc}}{1 + \frac{\sin f}{pc}} \right] \tag{4} $$

where $m$ is the mass per unit area, $f$ is frequency, and $pc$ is the characteristic impedance of air. The derivation of equation (4) is given in reference 7 although the final equation in that reference has a typographical error which has been corrected here. The equivalent absorption area $A_{eq}$ is dependent on frequency and may strictly be used as a correction factor for only one particular test setup and room configuration. Practically, the $A_{eq}$ measured for the reference panel is good for a variety of test panels since the panel surface area is small compared to the total area of the receiving room. The repeatability of transmission loss tests using this method is very good as the accuracy between tests is within a tolerance of a dB. A 3.18 cm thick rubber reference panel was used to determine $A_{eq}$ for this test series. It is expected to follow the field incidence mass law over the frequency range of interest (100 Hz to 10 kHz) because it has a calculated resonant frequency of 0.2 Hz (simply supported edge conditions) and a critical frequency of approximately 3 x $10^5$ Hz, both of which are far outside the frequency region of interest. The data acquisition system incorporates measuring microphones, power supplies, amplifiers, a digital one-third octave band analyzer, and a desk top computer. The entire system is under control of the computer using a relay actuator and several remote relays to assure a high repeatability of the tests.

**Description of Test Panels**

A total of 14 fiber-reinforced composite panels were tested. Ten of these panels were of tape construction, two were of fabric construction, and two were of sandwich construction with fabric composite skins and microballoon-filled epoxy cores. The panels of tape construction were made by bonding several plies of unidirectional fibers. Each ply (see Fig. 2) consists of bundles of high strength fibers all lying in one direction and held together by an epoxy resin giving the appearance of a strip of tape (hence, "tape construction"). The plies were formed by cutting the tape strips so that each ply has the desired fiber direction with respect to the boundaries of the panel. For example, a 0° or 90° ply has its fiber direction parallel to one of the panel boundaries while a 45° ply has its fibers running in a direction that forms a 45° angle with one of the boundaries. The composite panels were then formed by bonding together several plies of various angles usually in a manner called "balanced symmetric" where the ply angles are symmetric about the mid-plane of the panel and every ply angle is balanced by another ply at the negative of that angle. All the tape panels in the present tests were balanced symmetric. Three different types of fiber/epoxy panels were tested: graphite/epoxy (G/E), Kevlar/epoxy (K/E), and fiberglass/epoxy (F/E). The panels were 0.91 m long and 1.52 m wide and had thicknesses of about 0.1 cm and 0.2 cm. The boundary along the long dimension of the panel was chosen as the 0° direction. The designation given to each tape panel for future reference and details on the ply angles, thickness, and surface density for these panels are presented in Table 1. The fiber orientations listed in the table are for one-half of the panel thickness, i.e., from one surface to the mid-plane. The remaining plies are in the reverse order from the mid-plane to the other surface. The panels with fabric construction were similarly made by bonding several plies together. However, instead of having unidirectional fibers in a resin matrix, a fabric ply consists of bundles of fibers woven together perpendicular to each other. In what are called the "fill direction," the fibers bend up and down as they weave around the fill direction fibers (see Fig. 3). As with the tape panels, the fabric panels were all

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**Fig. 2** Details of tape ply construction.
balanced symmetric. Only G/E fabric panels were constructed in time for the present study. All the fabric plies were cut so that the warp direction was parallel to the long dimension of the panel. The designation, thickness, and surface density for the fabric panels and the sandwich panels (since they had fabric skins) are presented in Table 2.

\[ w_{xxxx} + 4 D_{16} w_{xxxy} + 2(D_{12} + 2 D_{66}) w_{xxyy} + 4 D_{26} w_{xyyy} + D_{22} w_{yyyy} + \overline{m} w_{tt} = p(x,y,t) \]  

where a comma denotes the partial differentiation with respect to the subscript, \( w \) is the plate displacement, \( p(x,y,t) \) is the pressure acting on the plate, and the \( D_{ij} \) terms are the anisotropic plate rigidity values that relate the internal bending and twisting moments of the plate to the twists and curvatures they induce. For an isotropic plate, \( D_{11} = D_{22} = \frac{Eh^2}{12(1 - \nu^2)} \), \( D_{12} = \nu D_{11} \), \( D_{66} = \frac{Eh}{12} \), and, because twisting behavior is uncoupled from bending behavior, \( D_{16} = D_{26} = 0 \). For an orthotropic plate, \( D_{11} \) no longer equals \( D_{22} \), but again, \( D_{16} = D_{26} = 0 \). Many composite panels, however, are governed by anisotropic behavior where \( D_{16} \) and \( D_{26} \) are non-zero which means that the bending and twisting moments are coupled to the twists and curvatures, respectively.

**D** for **Tape Panels**

The theory for calculating the flexural rigidities \( D_{ij} \) for tape panels is well established and documented. Each ply is modeled as an orthotropic layer with the following properties:

\[ E_{11} = \text{modulus of elasticity in direction parallel to fibers} \]
\[ E_{22} = \text{modulus of elasticity in direction perpendicular to fibers} \]
\[ \nu_{12} = \text{ratio of strains perpendicular and parallel to stress where the stress is parallel to fibers} \]
\[ G_{12} = \text{shear modulus} \]
\[ v_{21} = \frac{E_{22}}{E_{11}} \nu_{12} \]

**Table 1** Description of panels with tape construction

<table>
<thead>
<tr>
<th>Panel designation</th>
<th>Tape material</th>
<th>Fiber orientation</th>
<th>Number of plies</th>
<th>Thickness, cm</th>
<th>Surface density, kg/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>Graphite</td>
<td>45/-45/45/-45</td>
<td>8</td>
<td>0.102</td>
<td>1.59</td>
</tr>
<tr>
<td>GT2</td>
<td>Graphite</td>
<td>0/90/0/90</td>
<td>8</td>
<td>0.102</td>
<td>1.59</td>
</tr>
<tr>
<td>GT3</td>
<td>Graphite</td>
<td>45/-45/45/-45</td>
<td>16</td>
<td>0.185</td>
<td>3.05</td>
</tr>
<tr>
<td></td>
<td></td>
<td>45/-45/45/-45</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>KT1</td>
<td>Kevlar*</td>
<td>45/-45/45/-45</td>
<td>8</td>
<td>0.102</td>
<td>1.37</td>
</tr>
<tr>
<td>KT2</td>
<td>Kevlar*</td>
<td>0/90/0/90</td>
<td>8</td>
<td>0.102</td>
<td>1.37</td>
</tr>
<tr>
<td>KT3</td>
<td>Kevlar*</td>
<td>45/-45/45/-45</td>
<td>16</td>
<td>0.203</td>
<td>2.79</td>
</tr>
<tr>
<td>KT4</td>
<td>Kevlar*</td>
<td>0/90/45/-45</td>
<td>16</td>
<td>0.203</td>
<td>2.79</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0/90/45/-45</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>FT1</td>
<td>Fiberglass</td>
<td>45/-45/45/-45</td>
<td>8</td>
<td>0.102</td>
<td>2.21</td>
</tr>
<tr>
<td>FT2</td>
<td>Fiberglass</td>
<td>0/90/0/90</td>
<td>8</td>
<td>0.102</td>
<td>2.18</td>
</tr>
<tr>
<td>FT3</td>
<td>Fiberglass</td>
<td>45/-45/45/-45</td>
<td>16</td>
<td>0.201</td>
<td>4.70</td>
</tr>
<tr>
<td></td>
<td></td>
<td>45/-45/45/-45</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Table 2** Description of panels with graphite fabric construction

<table>
<thead>
<tr>
<th>Panel designation</th>
<th>Core thickness, cm</th>
<th>Number of fabric plies</th>
<th>Total thickness, cm</th>
<th>Surface density, kg/m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>GF1</td>
<td>N.A.</td>
<td>3</td>
<td>0.109</td>
<td>1.64</td>
</tr>
<tr>
<td>GF2</td>
<td>N.A.</td>
<td>6</td>
<td>0.211</td>
<td>3.31</td>
</tr>
<tr>
<td>GF3</td>
<td>0.10</td>
<td>6</td>
<td>0.318</td>
<td>4.16</td>
</tr>
<tr>
<td>GF4</td>
<td>0.20</td>
<td>6</td>
<td>0.418</td>
<td>4.82</td>
</tr>
</tbody>
</table>
The properties of each ply and the angle of orientation \( \theta \) of the fibers in each ply are used to calculate the flexural rigidities from the following equation:

\[
\mathbf{D}_{ij} = \frac{1}{3} \sum_{k=1}^{N} \left( \mathbf{Q}_{ij}^{k} \right) \left( z_k^3 - z_{k-1}^3 \right)
\]

where \( N \) is the number of layers, \( k \) designates the \( k \)-th layer, \( z_k \) is the z-direction distance from the middle surface to the bottom of the \( k \)-th layer (see Fig. 4), and \( \mathbf{Q}_{ij}^{k} \) are the reduced stiffnesses for the \( k \)-th layer that relate the stresses in that layer to the strains. The \( \mathbf{Q}_{ij} \) are a function of \( E_{11}, E_{22}, v_{12}, G_{12}, \) and \( \theta \). The derivation of the equations for \( \mathbf{Q}_{ij} \) is straightforward and given in reference 8. A ply that has its fibers parallel to one of the panel boundaries, either \( \theta = 0^\circ \) or \( 90^\circ \), will behave orthotropically. However, for all other \( \theta \) values the ply will behave anisotropically. Therefore, unless the panel is made up of plies that are either all \( 0^\circ \), all \( 90^\circ \), or all \( 0^\circ \) and \( 90^\circ \), the panel will behave anisotropically. From Table I, most of the panels are seen to be anisotropic. For the case of "many" layers, reference 8 indicates that \( D_{16} \) and \( D_{26} \) are small compared to the other rigidities so that orthotropic analysis may be used; but care should be taken in calculating critical loads because in that case even small values of \( D_{16} \) and \( D_{26} \) can have a significant effect.

The measured material properties of the tape panels were not available in time for the publication of the present paper, so estimated stiffness properties from design guides, company brochures, etc. were used in the analytical model predictions. Table 3 presents the estimated ply properties, i.e., \( E_{11}, E_{22}, v_{12}, \) and \( G_{12} \), that were used for each material. In Table 4, the calculated \( D_{ij} \) values are given for each of the 10 tape panels. The angle lay-up is seen to have a significant effect on \( D_{ij} \). Thus, as will be discussed later, angle lay-up has an important effect on the resonant frequency of a panel. Also note that judging whether \( D_{16} \) and \( D_{26} \) are small compared to the other rigidities is not necessarily a simple decision.

\( D_{ij} \) for fabric panels

The theory for calculating the flexural rigidities \( D_{ij} \) for fabric panels is neither well-established nor documented. Because each ply consists of a weave of fibers, each ply may be anisotropic in behavior whereas a ply of a tape panel is orthotropic in behavior when the principal axes are appropriately chosen. However, the only data available were estimates of equivalent orthotropic moduli. Since measured data on the fabric test panels were not available, the orthotropic estimates were used, and are presented in Table 5.

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**Table 3** Estimated material properties of tape plies

<table>
<thead>
<tr>
<th>Material</th>
<th>( E_{11}, \text{Pa} )</th>
<th>( E_{22}, \text{Pa} )</th>
<th>( v_{12} )</th>
<th>( G_{12}, \text{Pa} )</th>
<th>Density, kg/m\text{\textsuperscript{3}}</th>
</tr>
</thead>
<tbody>
<tr>
<td>Graphite</td>
<td>( 13.7 \times 10^9 )</td>
<td>( 9.65 \times 10^9 )</td>
<td>0.30</td>
<td>( 4.8 \times 10^9 )</td>
<td>1.55 \times 10\text{\textsuperscript{3}}</td>
</tr>
<tr>
<td>Kevlar</td>
<td>7.6</td>
<td>5.5</td>
<td>.34</td>
<td>2.1</td>
<td>1.36</td>
</tr>
<tr>
<td>Fiberglass</td>
<td>3.9</td>
<td>9.0</td>
<td>.30</td>
<td>2.4</td>
<td>2.19</td>
</tr>
</tbody>
</table>

**Table 4** Rigidity values for tape panels

<table>
<thead>
<tr>
<th>Panel</th>
<th>( D_{11}, \text{N} )</th>
<th>( D_{12}, \text{N} )</th>
<th>( D_{16}, \text{N} )</th>
<th>( D_{22}, \text{N} )</th>
<th>( D_{26}, \text{N} )</th>
<th>( D_{66}, \text{N} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>3.79</td>
<td>2.95</td>
<td>1.06</td>
<td>3.79</td>
<td>1.06</td>
<td>3.12</td>
</tr>
<tr>
<td>GT2</td>
<td>8.60</td>
<td>.75</td>
<td>.00</td>
<td>4.37</td>
<td>.00</td>
<td>.42</td>
</tr>
<tr>
<td>GT3</td>
<td>21.06</td>
<td>12.93</td>
<td>2.31</td>
<td>23.06</td>
<td>2.21</td>
<td>18.95</td>
</tr>
<tr>
<td>KT1</td>
<td>2.04</td>
<td>1.68</td>
<td>.58</td>
<td>2.04</td>
<td>.58</td>
<td>1.71</td>
</tr>
<tr>
<td>KT2</td>
<td>4.73</td>
<td>1.54</td>
<td>.00</td>
<td>2.40</td>
<td>.00</td>
<td>.18</td>
</tr>
<tr>
<td>KT3</td>
<td>16.34</td>
<td>13.45</td>
<td>2.33</td>
<td>16.34</td>
<td>2.33</td>
<td>13.65</td>
</tr>
<tr>
<td>KT4</td>
<td>27.65</td>
<td>5.05</td>
<td>.87</td>
<td>24.83</td>
<td>.87</td>
<td>5.26</td>
</tr>
<tr>
<td>FT1</td>
<td>2.58</td>
<td>1.02</td>
<td>.30</td>
<td>2.58</td>
<td>.30</td>
<td>1.40</td>
</tr>
<tr>
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<td>1.40</td>
<td>.00</td>
<td>2.60</td>
<td>.00</td>
<td>.78</td>
</tr>
<tr>
<td>FT3</td>
<td>18.90</td>
<td>7.83</td>
<td>1.16</td>
<td>18.90</td>
<td>1.16</td>
<td>10.76</td>
</tr>
</tbody>
</table>
Also given in Table 5 are the estimated material properties of the microballoon-filled epoxy core used in the sandwich panels. In Table 6, the calculated $D_{ij}$ values are given for the fabric panels. If the actual ply properties are anisotropic, then the equations for the reduced stiffnesses $\bar{D}_{ij}$ given in reference 8 are not applicable and new equations for $\bar{D}_{ij}$ should be derived.

Transmission Loss Calculation

Transmission loss (TL) is given by,

$$TL = 10 \log \left( \frac{1}{\tau} \right)$$  \hspace{1cm} (6)

where $\tau$ is the transmission coefficient and defined by

$$\tau = \frac{\text{transmitted acoustic intensity}}{\text{incident acoustic intensity}}$$  \hspace{1cm} (7)

The pressure $p(x,y,t)$ acting on the panel is the sum of the incident, reflected, and transmitted pressures. These pressures along with the displacement of the plate $w(x,y,t)$ are assumed to be harmonic travelling waves. Because the pressures are modeled as plane waves, the equation for the transmission coefficient (eq.(7)) reduces to

$$\tau = \left| \frac{P_t}{P_i} \right|^2 / \left| \frac{P_t}{P_i} \right|^2$$  \hspace{1cm} (8)

where $P_i$ and $P_t$ are the amplitudes of the transmitted and incident pressures. Forcing velocity to be continuous through the plate provides the two necessary boundary conditions so that equation (5) can be solved for the ratio of incident to transmitted pressure.

$$\frac{P_t}{P_i} = \frac{1}{2} \left[ 1 + \frac{\cos \theta_1}{\sin \phi_1} \left( 1 + \sin \phi_1 \right) \right] \left[ k_x^2 + 2(D_{12} + 2D_{66})k_y^2 \right]$$

$$\times \left[ k_x^2 + 2(D_{12} + 2D_{66})k_y^2 \right]$$

$$+ \left( 2D_{12} + 8D_{66} \right) k_y^2 k_x^3 + 4D_{26} k_x k_y^3 + D_{22} k_y^4$$

where

$$k_x = \frac{(2\pi f \cos \phi_1)}{c}$$

$$k_y = \frac{(2\pi f \sin \phi_1)}{c}$$

$\phi_1$ = angle the incident pressure wave makes with the Z axis when the wave is projected into the plane of the panel and

$\theta_1$ = angle incident pressure wave makes with the X axis when the wave is projected into the plane of the panel.

Substituting (9) into (8) gives the transmission coefficient for oblique incidence transmission at a single frequency whereas the tests are for field incidence transmission in one-third octave bands. To calculate the field incidence transmission coefficient $T$, the incident and transmitted intensities are each integrated over a hemispherical solid angle defined by $\theta_1$ and $\phi_1$. Thus, writing $\tau(f)$ in terms of $\tau(\theta_1, \phi_1, f)$ results in

$$\tau(f) = \int_0^{\theta_{\text{lim}}} d\theta_1 \int_0^{\phi_{\text{lim}}} d\phi_1$$

$$\times \int_0^{\frac{2\pi}{f_{\text{lim}}}} \tau(\theta_1, \phi_1, f) \cos \theta_1 \sin \phi_1 d\theta_1 d\phi_1$$

$$\times \int_0^{\frac{2\pi}{f_{U}}} \tau(\theta_1, \phi_1, f) \cos \theta_1 \sin \phi_1 d\theta_1 d\phi_1$$

where $\theta_{\text{lim}}$ is commonly equated to 78° for field incidence transmission.

For predicting $TL$ in one-third octave bands, the incident and transmitted intensities must be summed over the bands. Thus, the $TL$ for one-third octave band is given by

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

$$= \frac{1}{\tau} \int_{f_L}^{f_U} S(f) \Delta f$$

$$= \int_{f_L}^{f_U} \tau(f) S(f) \Delta f$$

where $S(f)$ is the power spectral density of the incident pressure at frequency $f$ with narrow frequency bandwidth $\Delta f$, $f_L$ is the lower frequency of the one-third octave band, and $f_U$ is the upper frequency of the band.

Table 5: Estimated properties of materials used in fabric panels

<table>
<thead>
<tr>
<th>Material</th>
<th>$E_{11}$, Pa</th>
<th>$E_{22}$, Pa</th>
<th>$\nu_{12}$</th>
<th>$G_{12}$, Pa</th>
<th>Density, kg/m$^3$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Graphite fabric</td>
<td>$6.9 \times 10^{10}$</td>
<td>$6.95 \times 10^{10}$</td>
<td>0.30</td>
<td>26.5 $\times 10^{10}$</td>
<td>1.55 $\times 10^{13}$</td>
</tr>
<tr>
<td>Core</td>
<td>.7</td>
<td>.7</td>
<td>.30</td>
<td>4.1</td>
<td>.89</td>
</tr>
</tbody>
</table>

Table 6: Rigidity values for fabric panels

<table>
<thead>
<tr>
<th>Panel</th>
<th>$D_{11}$, N.m</th>
<th>$D_{12}$, N.m</th>
<th>$D_{16}$, N.m</th>
<th>$D_{22}$, N.m</th>
<th>$D_{26}$, N.m</th>
<th>$D_{66}$, N.m</th>
</tr>
</thead>
<tbody>
<tr>
<td>GP1</td>
<td>8.2</td>
<td>2.5</td>
<td>0.0</td>
<td>8.1</td>
<td>0.0</td>
<td>2.9</td>
</tr>
<tr>
<td>GP2</td>
<td>59.2</td>
<td>17.8</td>
<td>.0</td>
<td>59.2</td>
<td>.0</td>
<td>20.7</td>
</tr>
<tr>
<td>GP3</td>
<td>196.6</td>
<td>59.0</td>
<td>.0</td>
<td>196.6</td>
<td>.0</td>
<td>69.0</td>
</tr>
<tr>
<td>GP4</td>
<td>297.8</td>
<td>89.4</td>
<td>.0</td>
<td>297.8</td>
<td>.0</td>
<td>106.0</td>
</tr>
</tbody>
</table>
frequency the term in brackets will be zero. At this frequency \( f(f_1, f_2, f) \) will equal unity and TL will be zero, meaning that all the sound will be transmitted. This frequency is called "coincidence frequency" and is so named because at this frequency the trace wavelength of the sound wave on the panel is equal to the free bending wavelength of the panel. The actual TL at this frequency depends on how much damping is present in the panel. The equation for coincidence frequency is thus calculated to be,

\[
f_{\text{coinc}} = \frac{c^2}{2n \sin^2\theta_i} \left\{ \frac{1}{D_{11}} \cos^2\phi_i + \frac{4D_{16}}{D_{22}} \cos^2\phi_i \sin^2\phi_i + \frac{4D_{26}}{D_{22}} \cos^3\phi_i \sin^2\phi_i + 2(D_{12} + 2D_{66}) \cos^2\phi_i \sin^2\phi_i \right\}^{1/2}
\]

The critical frequency, \( f_{\text{crit}} \), is the lowest possible value of the coincidence frequency. From the above, the lowest value of \( f_{\text{crit}} \) relative to \( \theta_i \) occurs where \( \sin \theta_i \) is a maximum, that is, at \( \theta_i = 78^\circ \) (maximum value of \( \theta_i \) in field incidence TL calculation). However, the value of \( \phi_i \) for minimizing \( f_{\text{coinc}} \) cannot be calculated apriori, therefore \( f_{\text{crit}} \) was found iteratively. Table 7 presents the calculated field incidence critical frequencies for all the composite test panels. The comparatively low critical frequencies for the sandwich panels indicate that highly stiffened panels can have dramatically reduced critical frequencies.

**Results and Discussion**

In Figs. 5 to 7 the measured TL of G/E, K/E, and V/E tape panels are compared with field incidence mass law and with the infinite panel theory developed in the previous section. In Figs. 8 and 9 the measured TL of the sandwich panels are compared with infinite panel theory. These figures are discussed below in relation to comparing the measured data with analysis and comparing the measured data with each other for different composites and different thicknesses.

**Mass Controlled Frequency Region Comparison**

In Figs. 5 to 7 field incidence mass law is seen to be in good agreement with the tape panel data (within 1 dB) over a wide frequency range. The thicker (and stiffer) the panel, the lower the frequency at which the data deviates from mass law because of the lower critical frequency. For example, in Fig. 5 for G/E, mass law agrees well with data up to 6300 Hz for the 0.102 cm panel (panel GT2 in Table 1) but only up to 2500 Hz for the 0.185 cm panel (GT3). In all cases, agreement was good down to 163 Hz. As expected, infinite panel theory agrees with mass law from the lowest frequency up to where coincidence effects become important, that is, where both the data and the infinite panel theory diverge from mass law.

Because the panels behaved in a mass law manner in the mass controlled region, the stiffness and, thus, the fiber orientations of the panel did not affect TL in this region. Therefore, comparing any two panels, it is simply observed that the panel with the larger surface density has the higher TL.

**Coincidence Frequency Region Comparison**

Even though the stiffness properties of the composites were only rough estimates, the agreement between infinite panel theory and experimental data is quite good. In Fig. 5, the theory follows the slope of the coincidence dips for both panels and is within 1 to 3 dB of the levels. In Fig. 6, the coincidence region of the heavier Kevlar panel (KT3 in Table 1) is predicted within less than 1 dB for all but one frequency, but the theory seems to dip at slightly too low a frequency for the lighter panel (KT2). In Fig. 7 the theory for...
Because of this, at high frequencies where the thicker panel enters its coincidence frequency region, the thinner, lighter panel has significantly greater (5 to 9 dB) transmission loss. At even higher frequencies, the thicker panel again would have greater TL.

\[ \text{Fig. 7 Transmission loss characteristics of fiberglass/epoxy panels, FT2 and FT3.} \]

The sandwich panels, which had G/E fabric skins and microballoon-filled epoxy cores, had the lowest predicted critical frequencies and had the estimated stiffness values with the least level of confidence since they were assumed isotropic (see Tables 6 and 7). Still in Figs. 8 and 9 the theory does predict the shape and trend of the measured transmission loss in the coincidence region. Because the theory does so well in predicting the trend of TL in this region, it is reasonable to conjecture that accurately measured stiffness and damping properties would improve the prediction.

In comparing composite panels with equal-thicknesses, the stiffer panel had the lower critical frequency. Thus, the G/E panels have lower critical frequencies than the K/E panels which in turn have lower critical frequencies than the F/E panels. Comparing panels with different thicknesses of the same composite material, the increased thickness causes a greater increase in panel stiffness than in panel surface density so that the thicker panel has a lower critical frequency. Because of this, at high frequencies where the thicker panel enters its coincidence frequency region, the thinner, lighter panel has significantly greater (5 to 9 dB) transmission loss. At even higher frequencies, the thicker panel again would have greater TL.

\[ \text{Fig. 8 Transmission loss characteristics of graphite/epoxy fabric sandwich panel, GF3, with 0.1 cm microballoon-filled epoxy core.} \]

\[ \text{Fig. 9 Transmission loss characteristics of graphite/epoxy fabric sandwich panel, GF4, with 0.2 cm microballoon-filled epoxy core.} \]

\[ \text{Table 7 Calculated field incidence critical frequencies} \]

<table>
<thead>
<tr>
<th>Panel</th>
<th>Critical frequency, Hz</th>
<th>Panel</th>
<th>Critical frequency, Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>GT1</td>
<td>8391</td>
<td>FT1</td>
<td>14902</td>
</tr>
<tr>
<td>GT2</td>
<td>8391</td>
<td>FT2</td>
<td>14902</td>
</tr>
<tr>
<td>GT3</td>
<td>4409</td>
<td>FT3</td>
<td>7862</td>
</tr>
<tr>
<td>KT1</td>
<td>10523</td>
<td>GF1</td>
<td>8835</td>
</tr>
<tr>
<td>KT2</td>
<td>10523</td>
<td>GF2</td>
<td>4621</td>
</tr>
<tr>
<td>KT3</td>
<td>5618</td>
<td>GF3</td>
<td>2867</td>
</tr>
<tr>
<td>KT4</td>
<td>6144</td>
<td>GF4</td>
<td>2173</td>
</tr>
</tbody>
</table>
Although the ply angle lay-up had no effect on TL in the mass controlled region, it may affect the value of the critical frequency and the TL in the coincidence region. Based on infinite panel theory, a panel that is made of all 0°/90° plies has the same coincidence region characteristics as a panel made of all 45°/−45° plies. Therefore, the calculated frequencies (Table 7) are identical for both panels comprised by the pairs G/E/G/E, K/E/K/E and F/E/F/E. However, the 16 ply Kevlar composite KT3/KT4 have different calculated critical frequencies because KT3 has all plies at 45°/−45° whereas KT4 has a mixture of 45°/−45° and 0°/90° plies. This difference in ply lay-up increased the calculated critical frequency for KT4 by 9 percent over that of KT3. This difference, as expected, was not detectable in the measured one-third octave band transmission loss data.

**Design Comparison of Composites and Aluminum**

In this section, the effect on noise transmission of replacing a typical general-aviation type aluminum skin with either G/E, K/E, or F/E skins is investigated. In designing a fuselage skin, the thickness of the skin panel is influenced by several design considerations, namely, impact damage tolerance, fatigue damage resistance, and its fail-safe capability to maintain flight safety in the event of structural damage. The design comparison presented in this section is based on fatigue damage resistance. The criterion for resistance to fatigue damage essentially amounts to a restriction on the initial shear buckling strength of the skin. The design loads of composite material skins are not expected to be any different than those for an aluminum skin; therefore, the design comparison presented here is based on equal critical shear load for the composite and aluminum panels. The panels were assumed to be sized 20.3 cm by 35.6 cm with simply supported boundary conditions. The composite panels were assumed to be of tape construction with each ply restricted to be 0.013 cm (0.005 in.) thick, which was the nominal ply thickness of the test panels. The critical shear load was first calculated for the aluminum panel which was assumed to have a 0.101 cm (0.040 in.) thick skin. Then a number of ply angle configurations was investigated for each composite material so that the minimum-thickness composite panel was found whose critical shear load either met or exceeded that of the aluminum panel. The critical shear loads were calculated using orthotropic theory. The ply angles and calculated rigidities of the design panels are given in Table 8 where it can be seen for the composite panels that $D_{16}$ and $D_{26}$ are about an order of magnitude less than the other rigidity values. With the panels thus designed, the infinite panel noise transmission theory developed earlier in this paper was used to calculate the transmission loss of the panels. These results are presented in Fig. 10.

**Table 8 Description of design comparison panels**

<table>
<thead>
<tr>
<th>Material</th>
<th>Fiber orientations</th>
<th>$D_{ij}$</th>
<th>Rigidity values, $N.m$</th>
<th>Surface density, $kg/m^2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum</td>
<td>N.A.</td>
<td>6.78</td>
<td>2.26 0.00 6.78 0.00 2.32 1.81</td>
<td></td>
</tr>
<tr>
<td>Graphite/epoxy</td>
<td>45/−45/0/90/0/90/0/45</td>
<td>6.86</td>
<td>3.54 0.93 5.26 0.93 3.78 1.78</td>
<td></td>
</tr>
<tr>
<td>Kevlar/epoxy</td>
<td>45/−45/0/90/0/90/0/45/45</td>
<td>5.45</td>
<td>3.35 0.66 7.21 0.66 3.41 1.89</td>
<td></td>
</tr>
<tr>
<td>Fiberglass/epoxy</td>
<td>45/−45/0/90/0/90/0/45/45</td>
<td>6.43</td>
<td>2.15 0.33 7.31 0.33 3.11 3.00</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 10 Infinite panel theory transmission loss for design comparison of aluminum and composite panels.

The F/E panel has slightly greater TL than the aluminum panel because its design weight is slightly greater. The G/E and K/E panels have design weights about 35 percent and 33 percent, respectively, lighter than aluminum and thus have about 3 to 4 dB less transmission loss over their mass controlled regions. At the highest frequencies of comparison, the G/E and K/E have 6 to 12 dB less TL than the aluminum panel because the G/E and K/E have lower critical frequencies.

In the low frequency stiffness controlled transmission loss region, high-stiffness composites might provide increased transmission loss compared with aluminum. This topic is discussed in the following analytical section.

**Stiffness Controlled TL and Finite Panel Analysis**

To investigate the effects of high-strength/lightweight composites in the low frequency region of transmission loss, experimental studies are planned for small panels whose fundamental frequencies will typically be 100 Hz or more. In order to calculate the transmission loss of these panels at
or near resonance, the analytical model must take into account the boundary conditions of the panel. Thus, a finite panel transmission loss theory is needed. Such a theory has been developed and is currently being implemented. The theory models the test panel as a rectangular plate simply supported in an infinite baffle. Field incidence plane waves are assumed to impinge upon one side of the panel. The resulting panel vibrations are calculated by a normal-mode approach with the plate properties assumed to be orthotropic. A Green's function integral equation is used to link the panel vibrations to the transmitted spherical sound waves. The incident and transmitted acoustic powers are calculated by integrating the incident and transmitted intensities over their appropriate areas, and transmission loss is calculated from the ratio of transmitted to incident acoustic power. The model has been implemented to the point of calculating oblique incidence transmission loss for a single plane acoustic wave impinging on the panel as compared to field incidence which is for many plane waves.

Calculations of fundamental frequencies and oblique incidence transmission loss have been performed to obtain preliminary data on the effects of composites on low frequency noise transmission. The equations for the fundamental frequencies of anisotropic simply supported and clamped plates have been presented by Bert and have been used here to calculate the fundamental frequencies of panels used in the present tests. These results are tabulated in Table 9. The angle orientation of the plies is seen to have a strong effect on the simply supported fundamental frequency for G/E and K/E tape panels. The +45°/−45° configurations (GT1 or KT1) had about a 40 percent increase in fundamental frequency relative to the 0°/90° configurations (GT2 or KT2). In comparing the fundamental frequencies of composite and aluminum design comparison panels, the G/E and K/E panels had about 40 percent higher frequencies than the aluminum panels (109 Hz compared to 79 Hz). In Fig. 11 narrow band oblique incidence transmission loss, calculated with finite plate theory, has been plotted for the design comparison panels. Because of their higher fundamental frequencies, the G/E and K/E panels have over 12 dB more TL at the aluminum panel's resonance (79 Hz). The increase in TL over the aluminum panel is about 4 dB at the lowest frequencies plotted. Such transmission loss characteristics indicate that composite materials may be beneficial for low frequency noise transmission problems at or below the resonance of conventional aluminum panels. For frequencies above the fundamental resonance region, the heavier aluminum panel has, in general, higher TL than the G/E or K/E panels; though at particular frequencies panel resonances cause the TL of the composite and aluminum panels to be about the same. Thus, panel resonances can have an effect on frequencies normally considered to be in the mass controlled region. This resonance effect may explain an interesting phenomenon found when comparing the TL of the two sandwich panels in Figs. 6 and 9. The stiffer panel is seen to have significant deviations from theory in the mass controlled region at frequencies (200 Hz to 2 kHz) where the less stiff panel agrees fairly well. This might be due to panel resonant behavior. Modal frequency calculations were performed for both panels which showed the less stiff panel to have, on the average, 33 percent more modes responding to the one-third octave bands from 200 to 500 Hz than the stiffer panel. This should result in the lighter panel being closer to mass law in behavior.

Concluding Remarks

An experimental and theoretical research program has been initiated to develop an improved understanding of the noise transmission characteristics of composite materials. Such understanding is needed to ensure that the weight-saving advantages of using composite materials in aircraft.
fuselage design are not compromised by high noise transmission. As a first step to see how composite structures will affect fuselage noise transmission relative to current aluminum structures, noise transmission tests have been conducted on large unstiffened panels representative of the outer skin or inner trim panels. Concurrently, an analytical model for predicting the transmission loss of the test panels was developed which allows for exact modeling of the anisotropic properties of the composite panels. The model is based on infinite panel theory and is applicable in the mass controlled and coincidence frequency regions, the regions of interest for the test panels. In comparing theory with measured data in the mass controlled region, good agreement, within 1 dB, was obtained. In the coincidence frequency region, agreement was also quite good with respect to the trend of transmission loss even though the elastic properties used in the analysis were only rough estimates. In comparing composites with each other, as expected, the heavier panels had the higher transmission loss in the mass controlled region, and the panels with the higher stiffness to mass ratio had the lower critical frequencies. The ply angle lay-up had no effect in the mass controlled region. Although the angle lay-up can affect the critical frequency, insufficient data was measured or calculated to determine the potential magnitude of the effect.

A theoretical design comparison between aluminum and composite panels based on equal critical shear load was also conducted. This comparison indicated graphite/epoxy and Kevlar/epoxy panels to have 3 to 4 dB less transmission loss over most of the frequency range of interest due to their lighter weight and 6 to 12 dB less transmission loss at the highest frequencies of interest (up to 10 kHz) because of their lower critical frequencies.

In preparation for future tests to investigate the stiffness controlled frequency region, a finite panel field incidence transmission loss theory has been developed. The theory so far has been implemented for oblique incidence transmission loss. Preliminary calculations indicate that improved low frequency transmission loss might be possible with composite panels relative to conventional aluminum panels.

References


