Axial Passive Damping Testing of Mass Produced

Stress Coupled, Co-cured Damped Fiber Reinforced Composites

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ABSTRACT

Lightweight, dynamically stiff composite structures with in-plane damping levels one and a half to four times greater than the control panels have been demonstrated. The structures were also compared to analytical designs using the constrained layer damping theory. The structures are created using viscoelastic material sandwiched between orthotropic composite layers. The composite layers have different orientation angles, purposefully unsymmetric. Stress coupling between the stiffness layers when excited by in-plane and / or out-of-plane vibrations produces hysteresis losses that are distributed throughout the viscoelastic layers, resulting in vibrational damping with weight savings over the constrained layer damping, free layer treatment and various active damping approaches. Previous testing has shown that these structures improve damping [2,3,5]. Commercial scale production of the desired fiber patterned material has not been available. It was determined that modification of a weaving process could produce a product with the patterns shown to be effective by earlier testing and analysis [2,3,5].

Nomenclature

Amp	the amplitude of the wave pattern
W	the width of the material
T/2	half of the period
θ	theta, the angle of the wave pattern
S_0	positive wave for the fiber
S_{180}	180 degrees out of phase wave for fiber
VEM	a 0.25mm thickness of 3M ISD 112 viscoelastic material [6]
A, B	Material specifications with various wave patterns
A#, B#	Specimen layup designation
E	material modulus
e	nondimensional modulus ratio
H	thickness
h	nondimensional thickness ratio
ζ	modal damping ratio
η	loss factor

INTRODUCTION

Vibrations caused by rotating parts and air turbulence affect equipment in all industries. Uncontrolled vibrations can cause such problems as fatigue damage, structural failure, and noise in sensitive electronic equipment on air and space vehicles. Either active or passive damping methods are usually used to reduce vibrations. Active damping consists of measuring the structure's output or response to determine the applied force necessary to obtain the desired response [1]. Passive damping can be accomplished relatively inexpensively. It uses geometric and material changes to reduce vibrations inherently by converting kinetic energy (movement) to thermal energy.

Two passive damping techniques are Constrained Layer Damping (CLD) and Stress Coupling Activated Damping (SCAD©). CLD is in wide use today but is limited to damping out-of-plane vibrations. On the other hand SCAD© technology, using a combination of wavy patterned composites and viscoelastic materials, has the capability to increase damping in both in-plane and out-of-plane modes. Previous papers [2,3] have demonstrated damping benefits and manufacturing methods for using SCAD©

technology. Current research presented in this paper has demonstrated large-scale production of wavy patterned composite prepreg. This paper outlines the axial modal testing of several panels using this wavy prepreg and SCAD© technology compared to analytical CLD designs and also to analytical design results for panels using the free-extensional layer damping treatment.

BACKGROUND

The passive damping technology called Stress Coupling Activated Damping (SCAD©) uses the stress coupling effect of anisotropic materials, such as fiber reinforced composites oriented in wave-like patterns, to distribute damping through the entire volume of embedded viscoelastic layers [4]. The key to SCAD© is that the fiber orientation angle in two adjacent composite layers is altered many times down the length of the structure with a viscoelastic damping layer between them. The fibers in the first layer have orientations opposite to those in the adjacent layer. At each of the direction changes, the opposing fiber orientations generate a region of high shear stress in the viscoelastic damping layer(s) (see figure 1) when the material is strained by in-plane stresses. By controlling parameters such as the orientation angle, thickness, periods and moduli as well as the viscoelastic material, significant shearing will now occur through most of the structure. Since the primary load path through the part remains in the composite layers, the part retains high stiffness. More importantly, SCAD© provides damping for both in-plane as well as out-of-plane vibrations, which makes it applicable to a wide range of structures and geometries, including, for example, tubes, plates, beams, and panels. [2].

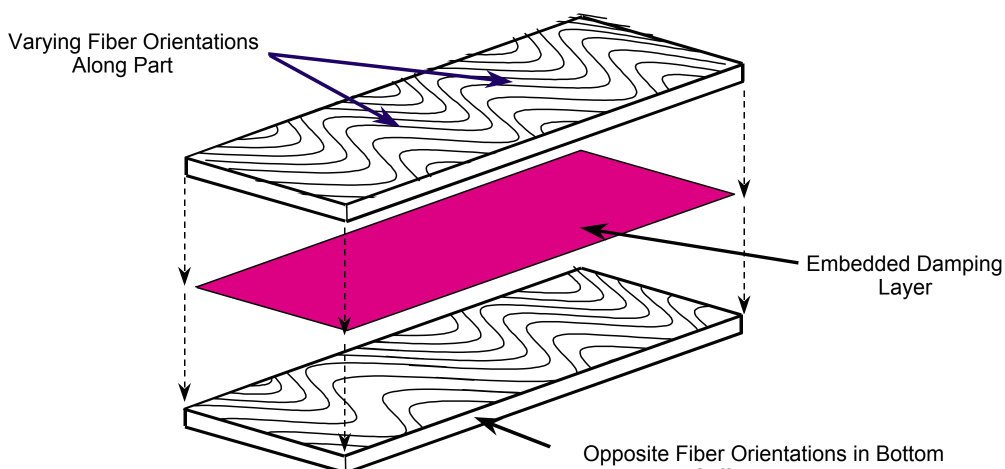


Figure 1. Olcott's Damping Concept.

The first demonstrations of this damping technique had a chevron type pattern rather than the wavy pattern shown here because processes to manufacture the wavy patterns were unavailable [2]. For this research it was demonstrated that a wave patterned fabric could be produced in commercial scale quantities using textile weaving processes. The fabric was then made into an epoxy prepreg using standard prepreg methods.

An optimum wavy pattern was not produced in this first weaving test. As a result, maximum damping was not achieved in the damping test samples made from this material. Other work has demonstrated that damping and acoustic properties of various basic structural elements [2, 3] can be predicted and optimized.

Objective

A series of tests were designed to quantify the damping added to a panel in an extensional vibration mode using the commercially woven wave patterned prepreg. The sample layup was defined based upon existing rocket payload fairing structure requirements. The objective, was to not only determine if the prepreg could be mass produced, but to also explore the effects of viscoelastic material (VEM) placement within the layups and to compare the SCAD© technology to existing CLD analytical predictions. It was noted that the design was not optimized for damping due to the choice of patterns generated in this initial proof of concept weaving test. . Figure 2 defines the characterization parameters for the material and layup specifications. Amp represents the amplitude of the wave pattern, W is the width of the material, T/2 is half of the period and θ, theta is the angle of the wave pattern.

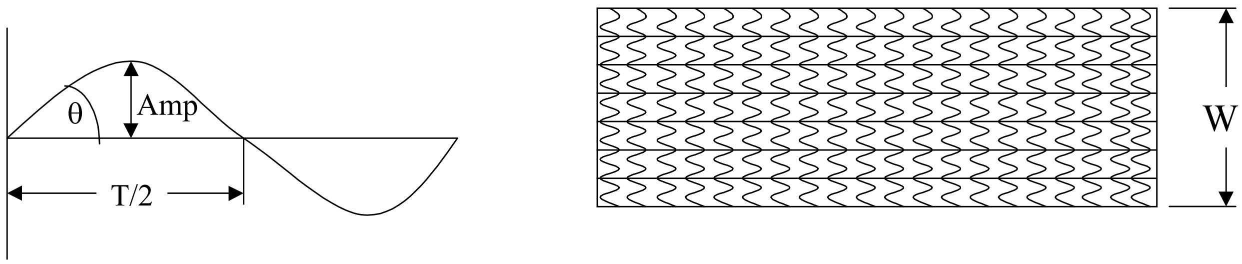


Figure 2. Characterization Parameters.

Table 1 summarizes the two different composite material specifications.

Table 1. Prepreg Specifications

Material Type	Width (cm)	Angle (deg)	Amplitude (cm)	T/2 (cm)
A	122.9	15.4	2.8	20.3
B	63.5	20.1	2.8	15.2

Four different panels were constructed using a variety of stacking sequences and two different viscoelastic interply orientations. Table 2 below summarizes the layups used in each specimen. The designation S_0 is a positive wave while S_{180} is 180 degrees out of phase. VEM indicates a 0.25mm thickness of ISD 112 viscoelastic material.

Table 2. Specimen Layups

Designation	Material Type	Stacking Sequence	Weight (N)
Control	A	(S_0 / S_0 / S_0 / S_0 / S_{180} / S_{180} / S_{180} / S_{180})	8.76
A1	A	(S_0 / S_0 / VEM / S_{180} / S_{180} / S_{180} / S_{180} / VEM / S_0 / S_0)	10.54
A2	A	(S_0 / S_0 / S_0 / S_0 / VEM / VEM / S_{180} / S_{180} / S_{180} / S_{180})	10.50
B3	B	(S_0 / S_0 / VEM / S_{180} / S_{180} / S_{180} / S_{180} / VEM / S_0 / S_0)	9.87
B4	B	(S_0 / S_0 / S_0 / S_0 / VEM / VEM / S_{180} / S_{180} / S_{180} / S_{180})	9.92

All panels were 30.5cm wide and 119.4cm long (thickness varied according to stacking sequence). The method of testing the panels was designed to maximize the in-plane motion of the panel. To do this, each panel was suspended vertically. To the free end of the panel a weight was attached. This setup creates a simple mass-spring system where the longitudinal stiffness and damping of the panel defines the complex stiffness of the "spring". Changing the size of the weight would alter the frequency of the longitudinal mode.

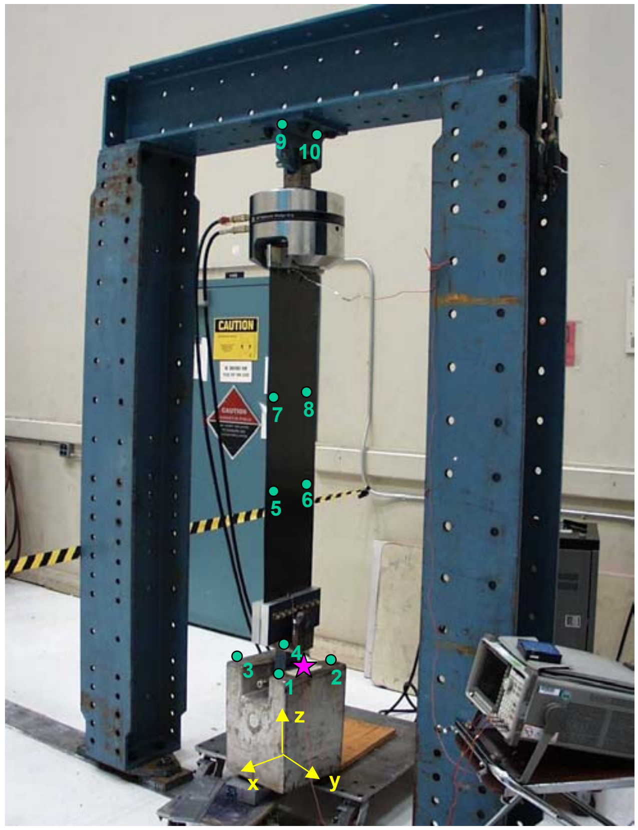


Figure 3. Test Setup

Test Setup

A test frame was constructed consisting of three large I-beams as shown in Figure 3. To the top I-beam a 11.3 kNm hydraulic grip was mounted. A friction clamp was constructed from 2.54cm thick aluminum. A 2.22kN weight was attached to the bottom of the panel using a 2.54cm thick aluminum friction plate and a large clevis. This reduced the bending loads produced by the mass rocking.

Four accelerometers (labeled 1 through 4 in Figure 3) were placed on the weight to measure acceleration in the vertical direction. By using four accelerometers, it could be determined whether the weight was rocking or only moving vertically. Four accelerometers (labeled 5 through 8 in Figure 3) were placed on the panel, two at approximately 1/3 span and two at 2/3 span. These accelerometers measured out of plane accelerations and were used to identify bending modes. Two accelerometers (labeled 9 and 10 in Figure 3) were placed above the hydraulic grip to measure the motion of the support structure. A force hammer was used to excite the system predominantly in the vertical direction at the location denoted by a star symbol in Figure 3. Several measurements were taken and averaged to reduce overall error. All data were recorded with an LMS* data acquisition system and LMS* modal analysis software was used to analyze the response data.

Data

For each accelerometer, an accelerance frequency response function (FRF) was generated. An example of which is shown in Figure 4. Accelerance is the complex ratio of output acceleration to input forces (units of g/N). Plotting various forms of accelerance (*e.g.,* magnitude, phase, real and imaginary parts) against frequency provides insight into the natural frequencies (or modes) of the system and the damping in each mode. One measure of damping is the modal damping ratio, ζ. Simply stated, the modal damping ratio is the ratio of viscous damping in the system to the critical viscous damping of the system (usually presented in terms of percent critical damping). A similar measure of hysteretic system damping is loss factor, η. For $\eta \ll 1$, the corresponding modal damping ratio is approximately equal to $\eta/2$.

Several curve fitting methods both in the time and frequency domains were used to generate modal parameters such as natural frequency and modal damping ratio. High degrees of accuracy in frequency and damping estimates occur when single modes exist that are well spaced with respect to frequency. Errors can arise, however, when modes are closely spaced. This is especially important for complex systems (e.g., composites) where the material is highly anisotropic. In cases where several modes are closely spaced (as shown in Figure 4), it is necessary to compare several accelerance measurements to accurately identify a particular mode and calculate its appropriate natural frequency and modal damping ratio.

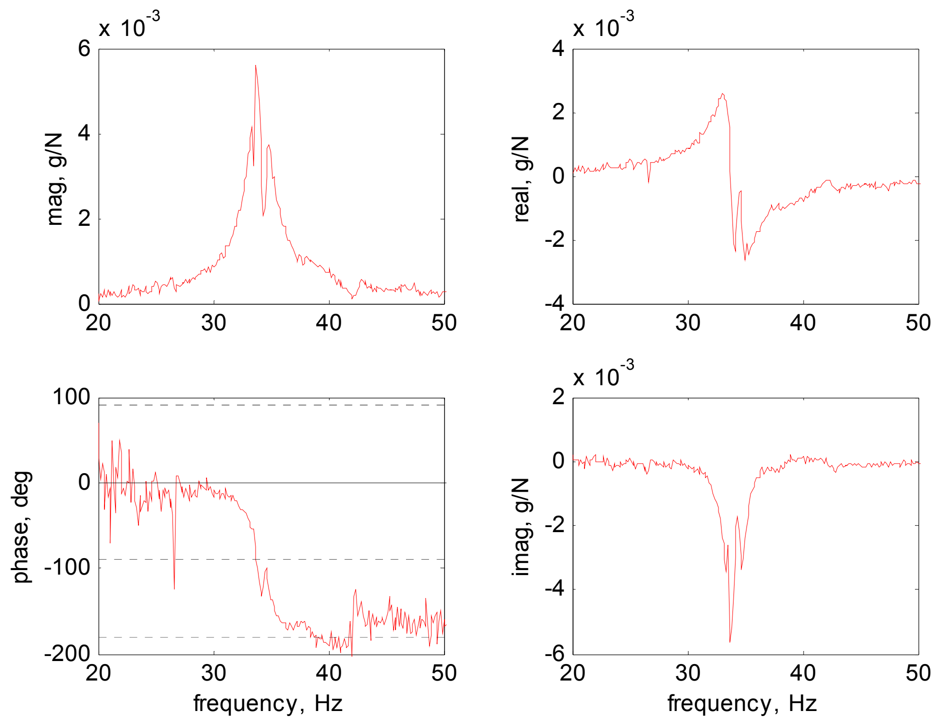


Figure 4. Sample Accelerance Frequency Response Function Magnitude, Phase, Real Part and Imaginary Part for the Control Panel.

RESULTS

Two sets of tests were performed on each panel. The first set of tests were conducted using a 2.22kN weight attached to each panel, while the second set of tests were conducted using a 3.11kN weight. Using two different weights changed the frequency of the longitudinal modes providing additional damping results for the same test specimen and a second opportunity to separate modes. In each set of tests for all panels, the longitudinal extension mode was strongly coupled to a longitudinal extension-twist mode. Careful analysis of the response of accelerance functions collected from both in-plane and out-of-plane accelerometers enabled verification of the appropriate mode. However, the numerical accuracy of the damping estimates suffered slightly. For all modal damping results reported, errors of approximately $\pm$ 0.5% critical damping can be expected.

Figure 5 graphically illustrates the modal damping in each of the panel's longitudinal extension mode with both 2.22 and 3.11 kN of attached weight. The natural frequency of the longitudinal extension mode with 2.22 kN was between 34 and 38 Hz for all panels while the corresponding natural frequencies for the panel with 3.11 kN were between 28 and 32 Hz respectively. Modal damping ratios for the control panel longitudinal modes with 2.22 and 3.11 kN were approximately 0.9% and 0.8% respectively.

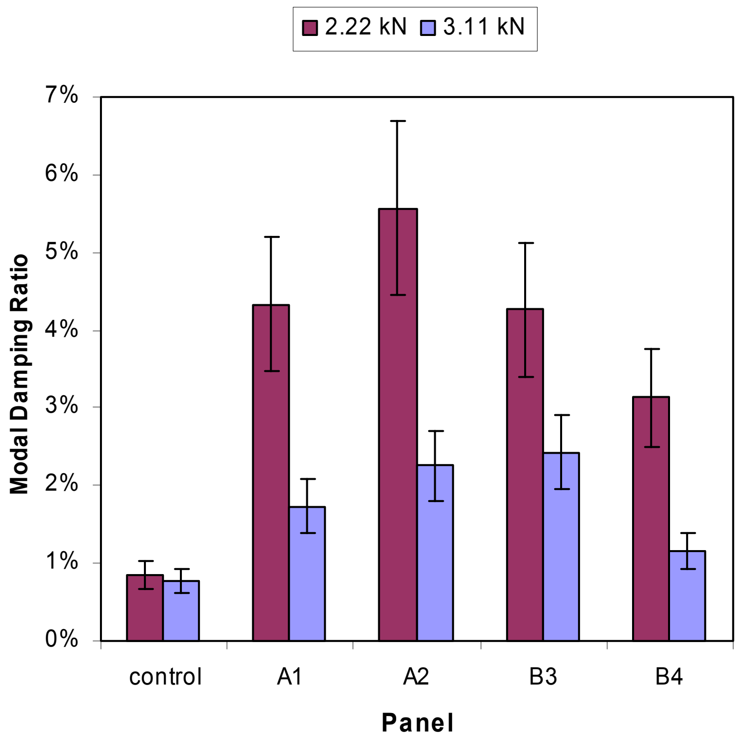


Figure 5. Longitudinal Extension Modes Modal Damping Ratios

The average modal damping for each of the damped panels with 2.22 kN attached was roughly 4% (a more than 3 fold increase in modal damping compared to the control panel). The average modal damping for each of the damped panels with 3.11 kN attached was roughly 1.5%. No conclusions could be made about the differences between specimen layups and material types (A or B) due to the high error percentages that occurred because of the coupling of the modes.

Panel damping with 2.22 kN added weight was, in all cases, higher than the respective modes with 3.11 kN added weight. This result is expected since the loss factor of the viscoelastic material is slightly higher at higher frequencies (for a given temperature) and loss factor tends to decrease slightly with increasing static preload [5].

As a point of reference, Figure 6 illustrates the modal damping ratios for the extension-twist modes of each of the five panels with both 2.22 and 3.11 kN respectively. Generally speaking, the extension-twist modes exhibit less damping than longitudinal extension modes. While measuring damping due to extension-twist coupling was not the main goal of these experiments, the close coupling of these modes to the purely longitudinal modes warrants their study. With the exception of panel B4, the 2.22 kN extension-twist modes do follow roughly the same pattern. The 3.11 kN extension-twist modes, however, do not exhibit this relationship. This discrepancy can, perhaps, be answered by examining the relative frequency spacing of the longitudinal and extension-twist modes for both the 2.22 and 3.11 kN cases.

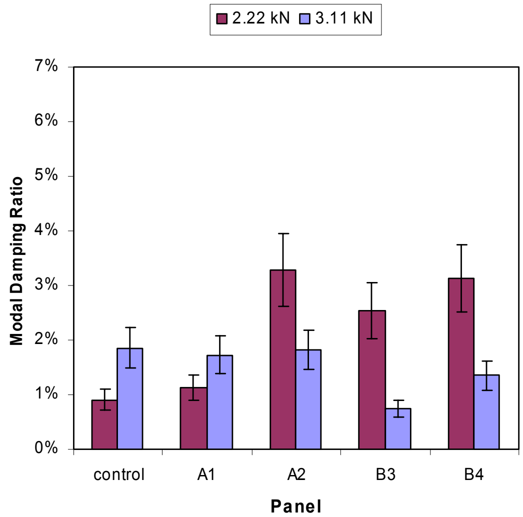
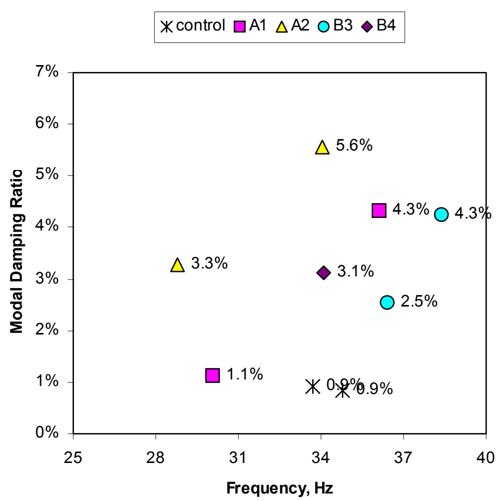
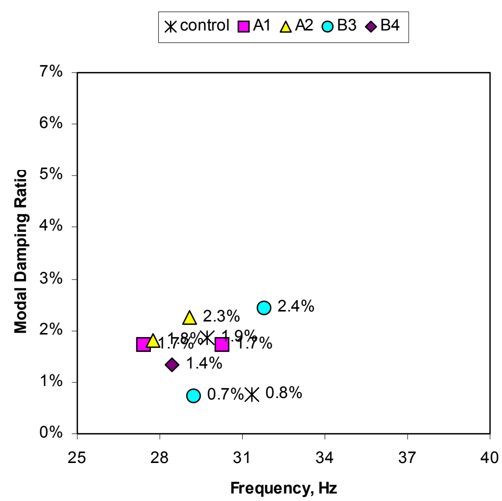


Figure 6. Extension-Twist Modes Modal Damping Ratios

Recall that, in general, it is more difficult to estimate modal damping parameters from closely spaced modes than from well-spaced modes. Figure 7(a) shows that the frequency spacing between the extension and extension-twist modes with 2.22 kN and Figure 6(b) illustrates the frequency spacing for the 3.11 kN case. Clearly, the 2.22 kN set of data shows more widely spaced modes than the 3.11 kN case. The data shown in Figure 7 provides insight into the relative accuracy of each modal damping ratio measurement due to the frequency proximity of the coupled extension-twist mode to the longitudinal mode. For example, the mode spacing for panel A1 with 2.22 kN suggests that the resulting 4.3% modal damping ratio estimate is more accurate than the 2.3% estimate of panel A2 with 3.11 kN (assuming the same analysis technique is applied to each data set).



(a)



(b)

Figure 7. (a) Extension and Extension-Twist Frequencies and Modal Damping Ratios with 2.22 kN, (b) Extension and Extension-Twist Frequencies and Modal Damping Ratios with 3.11 kN

COMPARISON

One of the more common passive damping technologies is Constrained Layer Damping or CLD. Inherent to the design of CLD is the fact that CLD is not effective for out of plane vibrations. An additional treatment must be applied to the structure to dampen extensional modes. CLD must also be applied to the surface of the structure. The wave patterned Stress Coupling Activated Damping method, on the other hand, is not only effective for out of plane vibrations, but it can also be integrated structurally into the component. Because of these issues a direct comparison for the extensional modes between CLD and SCAD© can not be made.

One can, however, theoretically compare the SCAD© technology versus a free layer viscoelastic treatment. Analytically applying an ISD112 viscoelastic layer onto the control panels equal to the amount of material applied onto the experimental panels gives a direct comparison between the two damping methods. Following the derivation in Nashif, *et al.* [5], start by defining

$$e_2 = \frac{E_2}{E_1} \tag{1}$$

where E_1 and E_2 are the modulii of the structure and damping material respectively. In addition, let

$$h_2 = \frac{H_2}{H_1} \tag{2}$$

where H_1 and H_2 are the thicknesses of the structure and damping material respectively.

The system loss factor of the structure, η, can be expressed as a function of e_2, h_2 and the loss factor of the damping material, η_2, as follows:

$$\eta = \eta_2 \left(\frac{e_2 h_2 \left(3 + 6h_2 + 4h_2^2 + 2e_2 h_2^3 + e_2^2 h_2^4\right)}{\left(1 + e_2 h_2\right)\left(1 + 4e_2 h_2 + 6e_2 h_2^2 + 4e_2 h_2^3 + e_2^2 h_2^4\right)} \right) \tag{3}$$

Note that the loss factor of the structure, η_1, is assumed to be small compared to the damping material loss factor, η_2.

The average longitudinal modulus of the panels tested was approximately 58.6 GPa and the thickness (excluding the viscoelastic material) was on the order of 2.03 mm. At room temperature and in the frequency range of interest, ISD112 has a modulus of approximately 2.1 MPa. In addition, two layers of 0.25 cm inch ISD112 were used in each panel. Thus, given that e_2 = 3.6e-5, h_2 = 0.25, and that at room temperature in the frequency range of approximately 30 to 40Hz η_2 is approximately 1.0, one can calculate a system loss factor using equation (3) of η = 4.3e-5.

Thus, the free layer viscoelastic treatment would provide approximately 0.002% added modal damping while the SCAD© technology provides roughly 0.5 to 3.0% added modal damping. Even within the accuracy of the test results there is clearly a tremendous improvement for adding structural modal damping while maintaining equal added weight.

CONCLUSIONS

It was shown that the general pattern necessary to achieve an increase in damping properties could be produced in commercial fabric weaving processes.

A series of tests were designed to quantify the damping added to a panel in an extensional mode using the wave patterned prepreg produced on a commercial scale. Interpretation of test data was difficult due to the closely spaced vibration modes. As a result the damping results are accurate to only approximately $\pm$ 0.5% of critical damping.

The panels with 2.22 kN attached, in the frequency range of 34 to 38 Hz, had more than a three-fold increase in modal damping compared to the control panel. The panels with 3.11 kN attached, in the frequency range of 28 to 32 Hz, had a one and a half times increase in modal damping compared to the control panel. The average modal damping for the damped panels with 2.22 kN attached was roughly 4% while the average modal damping with 3.11 kN attached was roughly 1.5%. The decrease in damping was expected since the loss factor of the viscoelastic material is slightly higher at higher frequencies and the loss factor tends to decrease slightly with an increasing preload.

The experimental results shown here illustrate the benefits of using SCAD© technology for extension mode damping of planar structures. For these types of panel damping problems, SCAD© technology adds significant modal damping without increasing system weight as compared to conventional free layer viscoelastic treatments which have little or no effect on modal damping.

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