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Axial Passive Damping Testing of Mass-Produced Stress Coupled, Cocured Damped Composites

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Lightweight, dynamically stiff composite structures with in-plane damping levels one and one-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. 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.

Nomenclature

E	=	material modulus
e	=	nondimensional modulus ratio
H	=	thickness
h	=	nondimensional thickness ratio
S_0	=	positive wave for the fiber
S_{180}	=	180-deg out-of-phase wave for fiber
$T/2$	=	half of the period
W	=	width of the material
ζ	=	modal damping ratio
η	=	loss factor
θ	=	angle of the wave pattern

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, predominately for 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 SCAD passive damping technology uses the stress coupling effect of anisotropic materials, such as fiber-reinforced composites oriented in wavelike 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) (Fig. 1) when the material is strained by in-plane stresses. By the control of 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. Because the primary load path through the part remains in the composite layers, the part retains high stiffness. More important, 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]

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.

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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 on existing rocket payload fairing structure requirements. The objective was to not only determine whether 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. Note 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, where amp represents the amplitude of the wave pattern. Table 1 summarizes the two different composite material specifications.

Four different panels were constructed using a variety of stacking sequences and two different viscoelastic interply orientations. Table 2 summarizes the layups used in each specimen. The designation S_0 is a positive wave, whereas S_{180} is 180-deg out-of-phase. VEM indicates a 0.25-mm thickness of ISD 112 VEM.[5] All panels were 30.5 cm wide and 119.4 cm 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. A weight was attached to the free end of the panel. 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.

Test Setup

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

Four accelerometers (1–4 in Fig. 3) were placed on the weight to measure acceleration in the vertical direction. By the use of four accelerometers, it could be determined whether the weight was rocking or only moving vertically. Four accelerometers (5–8 in Fig. 3) were placed on the panel, two at approximately one-third span and two at two-thirds span. These accelerometers measured out-of-plane accelerations and were used to identify bending modes. Two accelerometers (9 and 10 in Fig. 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 in Fig. 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 Fig. 4. Accelerance is the complex ratio of output acceleration to input forces (gravitational acceleration per newton). Plotting various forms of accelerance (for example, magnitude, phase, and 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 $\eta/2$.

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

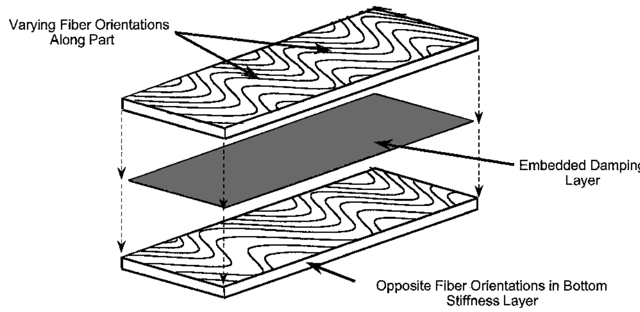


Fig. 1 Damping concept.[2]

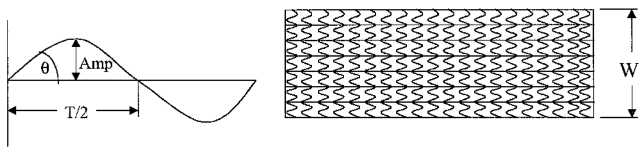


Fig. 2 Characterization parameters.

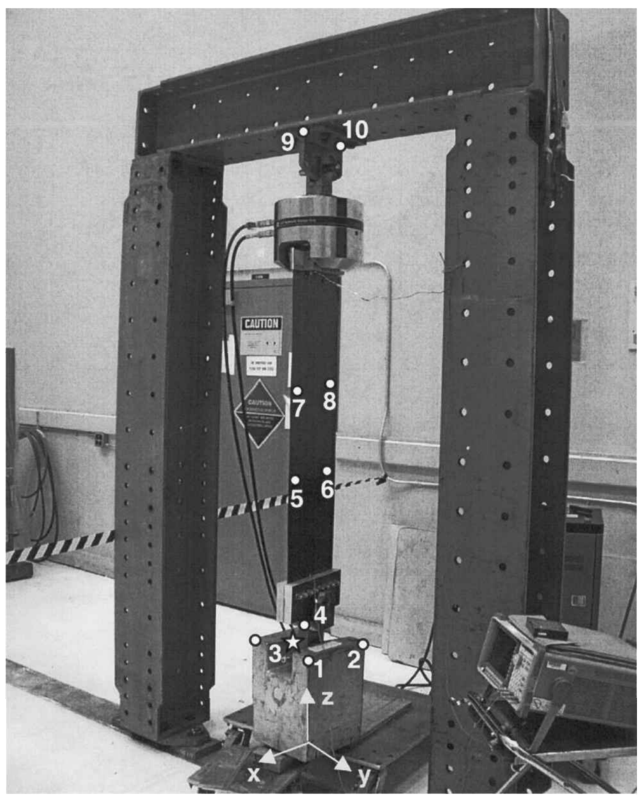


Fig. 3 Test setup.

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	$(S_0/S_0/\mathrm{VEM}/S_{180}/S_{180}/S_{180}/S_{180}/\mathrm{VEM}/S_0/S_0)$	10.54
A2	A	$(S_0/S_0/S_0/S_0/\mathrm{VEM}/\mathrm{VEM}/S_{180}/S_{180}/S_{180}/S_{180})$	10.50
B3	B	$(S_0/S_0/\mathrm{VEM}/S_{180}/S_{180}/S_{180}/S_{180}/\mathrm{VEM}/S_0/S_0)$	9.87
B4	B	$(S_0/S_0/S_0/S_0/\mathrm{VEM}/\mathrm{VEM}/S_{180}/S_{180}/S_{180}/S_{180})$	9.92

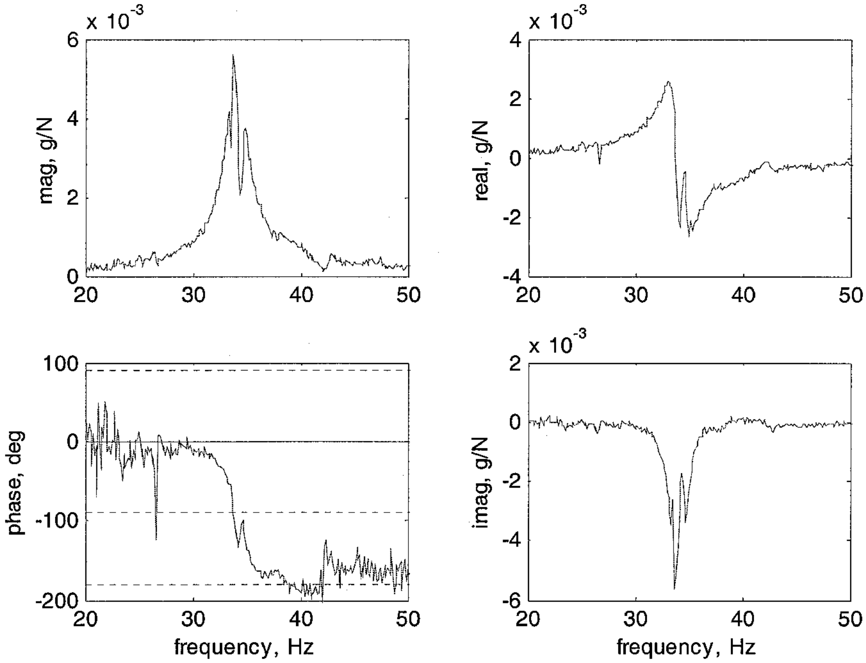


Fig. 4 Sample accelerance FRF magnitude, phase, real part, and imaginary part for control panel.

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, for example, composites, where the material is highly anisotropic. In cases where several modes are closely spaced (as shown in Fig. 4), it is necessary to compare several accelerance measurements to identify accurately a particular mode and calculate its appropriate natural frequency and modal damping ratio.

Results

Two sets of tests were performed on each panel. The first set of tests was conducted using a 2.22-kN weight attached to each panel, whereas the second set of tests was conducted using a 3.11-kN 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 ±0.5% critical damping can be expected.

Figure 5 shows the modal damping in each of the panels' longitudinal extension modes 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, whereas 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.

The average modal damping for each of the damped panels with 2.22 kN attached was roughly 4% (a more than threefold 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

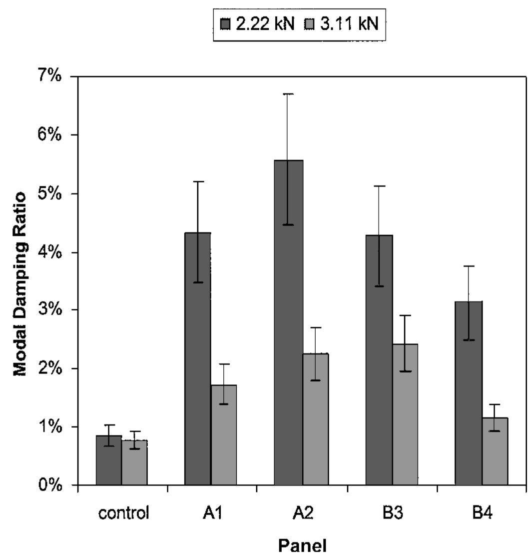


Fig. 5 Longitudinal extension modes modal damping ratios.

is expected because the loss factor of the VEM is slightly higher at higher frequencies (for a given temperature) and loss factor tends to decrease slightly with increasing static preload.[6]

As a point of reference, Fig. 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, the extension–twist modes exhibit less damping than longitudinal extension modes. Although 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.

Recall that, in general, it is more difficult to estimate modal damping parameters from closely spaced modes than from well-spaced modes. Figure 7a shows that the frequency spacing between the extension and extension–twist modes with 2.22 kN, and Fig. 7b

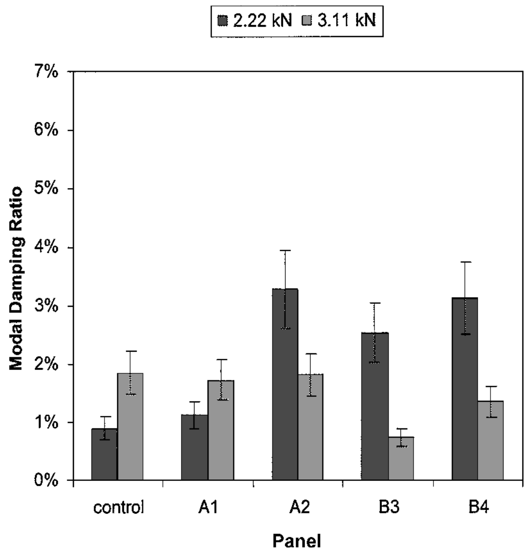
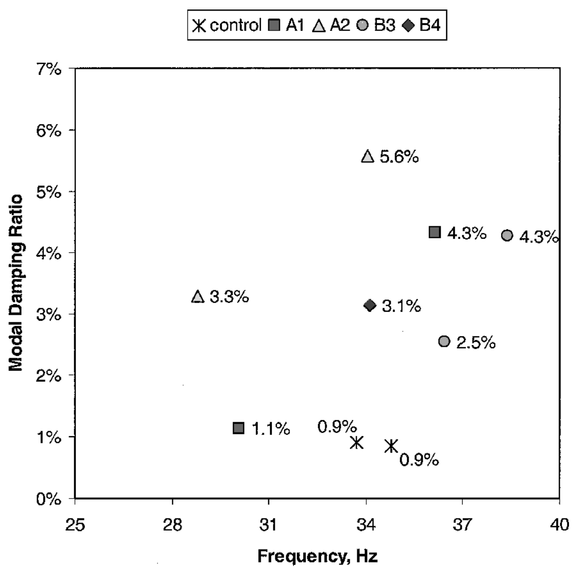
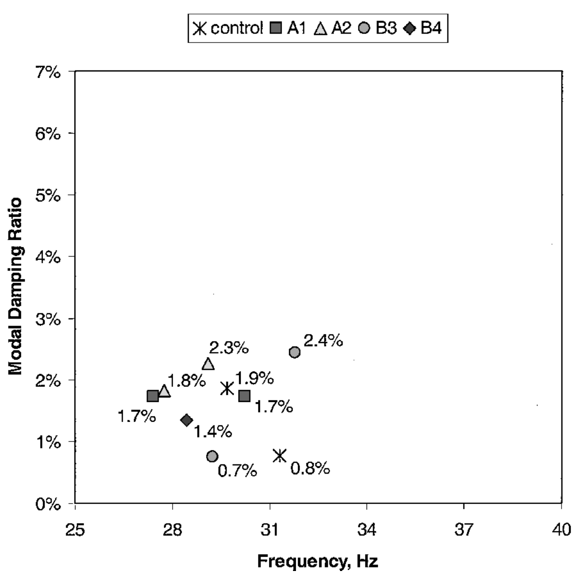


Fig. 6 Extension–twist modes modal damping ratios.



a)



b)

Fig. 7 Extension and extension–twist frequencies and modal damping ratios with a) 2.22 kN and b) 3.11 kN.

illustrates the frequency spacing for the 3.11-kN case. Clearly, the 2.22-kN data set shows more widely spaced modes than the 3.11-kN case. The data shown in Figs. 7a and 7b provide 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, when it is assumed that the same analysis technique is applied to each data set.

Comparison

One of the more common passive damping technologies is CLD. Inherent to the design of CLD is that CLD is not as effective for in-plane vibrations. Hence, 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 SCAD 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 cannot be made.

One can, however, theoretically compare the SCAD technology vs 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.,[6] start by defining

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

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

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

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:

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

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 ISD112 were used in each panel. Thus, given that $e_2 = 3.6e{-}5$ and $h_2 = 0.25$ and that at room temperature in the frequency range of approximately 30–40 Hz loss factor η_2 is approximately 1.0, one can calculate a system loss factor using Eq. (3) of $\eta = 4.3e{-}5$.

Thus, the free-layer viscoelastic treatment would provide approximately 0.002% added modal damping, whereas the SCAD technology provides roughly 0.5–3.0% 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 34–38 Hz, had more than a threefold increase in modal damping compared to the control panel. The panels with 3.11 kN attached, in the frequency range of 28–32 Hz, had a one and one-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%, whereas the average modal damping with 3.11 kN attached was roughly 1.5%. The decrease in damping was expected because the loss factor of the VEM 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.

Acknowledgments

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