Location effects of passive damping material in cross-ply laminates on natural frequency and mode shape

E.I. Elghandour & F.A. Kolkailah
Aeronautical Engineering Department, California Polytechnic State University, San Luis Obispo, California 93407, USA
Email: eelghand@calpoly.edu, fkolkail@calpoly.edu

Abstract

This study presents an experimental investigation of the free vibration of cantilevered composite laminated plates with embedded passive damping material at different stages. A total of five composite laminated plates are considered. The lay-up sequences for the five composite laminated plates with and without two embedded layers of passive damping material are $[90°/0°/90°/0°]_s$, $[90°/0°/90°/0°/d]_s$, $[90°/0°/90°/d/0°]_s$, $[90°/0°/d/90°/0°]_s$, and $[90°/d/0°/90°/0°]_s$. The passive damping material employed is a 3M material, SJ-2015 ISD 112, with peak damping properties in the ambient temperature range of 32°F to 140°F. The composite material used is a carbon fiber (977-2)/epoxy resin (IM7). The effect of the passive damping system employed in this study for the composite plates are discussed. Modal testing is performed on these plates to determine resonant frequencies, amplitude and mode shape information. The study included white noise and sinusoidal dynamic testing techniques, a PC computer based data acquisition system, and a virtual instrument dynamic analysis.
The different locations of the passive damping material in the cross-ply laminated plates resulted in degradation effects on the natural frequency, damping and mode of shape.

Introduction

Conventional structural designs are often unacceptable in coping with modern problems of structural resonance caused by the complex nature of the dynamic

environments. Current interest in large flexible space structures provides new motivation and requirements of structural damping enhancement for vibration control. The objective of enhancing damping in structural elements is to control the response of the elements in order to prevent catastrophic failure due to excessive deformation.

In this work, passive damping is used to enhance the damping of a carbon/epoxy cantilever plate under free vibration by adding a viscoelastic-damping layer to the plate. Cantilevered flat plates are convenient simulations for structural components, such as wings, horizontal and vertical stabilizers, and compressor fan blades for aircraft engines.

A number of papers have appeared which explore the effects of interlaminar damping of beam and plate structures. Ditaranto and Mcgraw [1] investigated the dissipation of vibratory energy in sandwich plates with a viscoelastic core. Only transverse inertial effects were included in the analysis. The solution for the damping was given for simply supported edges, and a relation between modal frequency and loss factor was obtained. Khatua and Cheung [2] used a finite element technique for the study of elastic multi-layer beams and plates. Orthotropy was included in the analysis, but rotatory and translatory inertia effects were neglected. Barrett [3] developed a comprehensive model to predict the damping of composite laminated plates with a viscoelastic layer. The effect of stress coupling and compliant layering was examined. Saravanos and Pereira [4] developed a discrete layer laminate theory for composite laminates with damping layers by incorporating a piecewise continuous displacement field through the thickness. Nadella and Rao [5] applied the modal strain energy method proposed by Johnson and Kienholz [6] to estimate the modal parameters of the multi-damping layer anisotropic laminated composite beams. Gerst, Rao and He [7] presented experimental results for composite beams with single and double damping layers, and demonstrated that co-curing is an effective way of fabricating highly damped composite structural components.

The ability to detect modes of vibration using surface mounted piezoelectric ceramics (PZT) as sensors was investigated [8]. The objective of this study was to investigate the effect of passive damping on composite plates. Three tasks were included in this study: 1) constructing carbon composite plates with different lay-up sequences and location of the embedded viscoelastic layers, 2) mounting piezoelectric sensors, 3) determining the modal parameters of the plates.

Experimental Procedure

The following is a review of the experimental procedure and materials used in the fabrication and dynamic testing of the cantilever composite plates. Fabrication and testing of these plates were conducted in the Aerospace Composite and Structural Laboratory (ACSL) at California Polytechnic State University.

Fabrication of Composite Plates

The composite plates were fabricated using carbon fiber/epoxy resin and two layers of viscoelastic material. The passive damping material provides high damping at room temperature but low stiffness and strength. The property of the materials used are listed in table 1.

Table 1. Material properties for 977-2 carbon fiber/IM7 epoxy resin.

Properties	Symbol	Units	carbon/epoxy
Youngs Modulus 0°	E_L	psi	2.5e7
Youngs Modulus 90°	E_T	psi	1.1e6
Density	ρ	Lb*s^2/in^4	1.43e-4
Poisson's Ratio	v		0.36
Thickness	t	inches	0.00625

The damped composite plates consist of eight layers of carbon/epoxy and two layers of passive damping material. Several different lay-ups of composite material with and without damping were fabricated. An example of a typical lay-up is $[90°/0°/90°/0°/d]_s$, which represents a symmetric lay-up of 90°, 0°, 90°, 0° composite material and a damping layer, d, symmetric about the center of the plate.

The lay-ups consisted of 8" x 9" composite plates. The composite lay-up was placed in a composite air press in three steps. The first two steps consisted of curing separately the top and bottom lay-up of carbon/epoxy with temperature and pressure cycles as recommended by the manufacturer of the prepreg material. The third step was to bond them together with the embedded viscoelastic layers, as recommended by the manufacturer of the 3M material. Once these plates were fabricated, they were cut into plate samples of 4" width and 9" length. Two 0.97" x 4" x 0.25" aluminum bars were secured with two bolts sandwich clamped on one edge of the plate. A third bolt was placed in the center of the aluminum bars, in order to secure the test specimen to the shaking table as shown in figure 1.

Piezoelectric ceramic sensors (0.75" x 0.5") were bonded with the positive poling axis oriented perpendicular to the top surface. The conductive epoxy allows the surface of the plate to be used as a common negative pole for the sensor. During the bonding process the Air Press applied 80 lbs. of force at room temperature for 20 minutes.
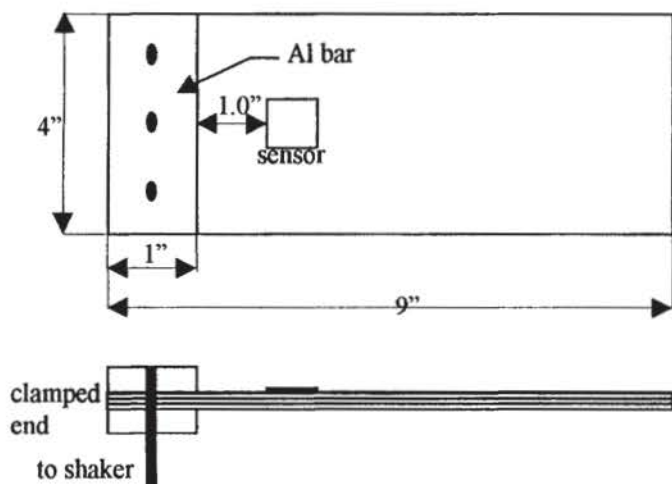
Figure 1. Cantilever Plate

Identifying Resonant Node Lines

To identify resonant node lines of the test specimen, the carbon fiber plate was mounted on the shaking table and a function generator was used to scan through the frequencies ranging from 0 to 500Hz. Resonant frequencies were identified by sharp rises in audible amplitudes, visual plate displacements, and the formation of node lines using sugar traces on the plate surface.

Dynamic Testing

A schematic of the experimental setup for dynamic testing of the cantilever plates is presented in Figure 2
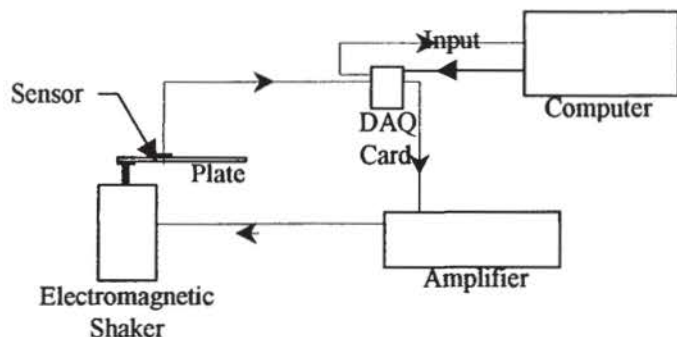

Figure 2. Schematic of Dynamic Testing

Time response and frequency response data was acquired using National Instruments AT-MIO-16F-5 Data Acquisition (DAQ) card in conjunction with National Instruments Labview for Windows Network Analyzer Virtual Instrument (VI). A cantilever plate was mounted on a shaker and excited using a chirp signal from Labview. A power amplifier was used to boost the signal before entering the shaker. The DAQ card acquires the data signals directly from the sensor, then conditions, digitizes the data, and enters the data into the PC's bus.

Chirp Signal Tests

The chirp signal is an impulsive type of signal that can have excitation over a wide range of frequencies and avoids impulse loading problems. The Network Analyzer provided the chirp signal for the shaking table and sampled the data provided by the sensors. The following Network analyzer parameters were used for the chirp tests:1) Sample rate (1,000 samples/sec), 2) Frame size (1,024 samples), 3) Windowing (Hann) and 4) Averaging (3).

Results and Discussion

Experimental Results

Figures 3 and 4 present the free vibration frequency response curves and the time response curves, respectively, for plates $[90°/0°/90°/0°]_s$ and $[90°/d/0°/90°/0°]_s$. Plate $[90°/d/0°/90°/0°]_s$ has two damping layers located near the top and bottom between the 90o and 0o degree orientations. The decrease in frequency and the increase in damping can be seen in both figures. To reduce the frequency response, the damping material is more effective at higher modes rather than the lower modes. One can see that the second and third modes are completely damped out. The first resonant frequency and amplitude decrease with the addition of damping layers near the top and bottom of the plate. These figures also show that the embedded damping layer has a more significant effect on the time response curve. The plate without damping layer $[90°/0°/90°/0°]_s$ reached steady state at about 0.75 second, whereas by adding the damping layers to the plate $[90°/d/0°/90°/0°]_s$ the steady state time response was reached in 0.2 second. Figures 5 and 6 present the free vibration frequency response curves and the time response curves, respectively, for plates $[90°/0°/90°/0°/d]_s$ and $[90°/d/0°/90°/0°]_s$. The decrease in frequency and the increase in damping can be seen in both figures. The damping material is significant effective at higher modes rather than the lower modes, resulting in frequency response reduction. One can see that the third mode is completely damped out and the first resonant frequency and amplitude decrease for plate $[90°/d/0°/90°/0°]_s$ rather than the plate $[90°/0°/90°/0°/d]_s$. The embedded damping layer located near the top and bottom has a more significant effect on the time response curve than the plate with an embedded layer located at the mid-plane. Plate $[90°/0°/90°/0°/d]_s$ reached steady state at about 0.234 second, whereas the plate $[90°/d/0°/90°/0°]_s$ reached a steady state time response in 0.2 second.
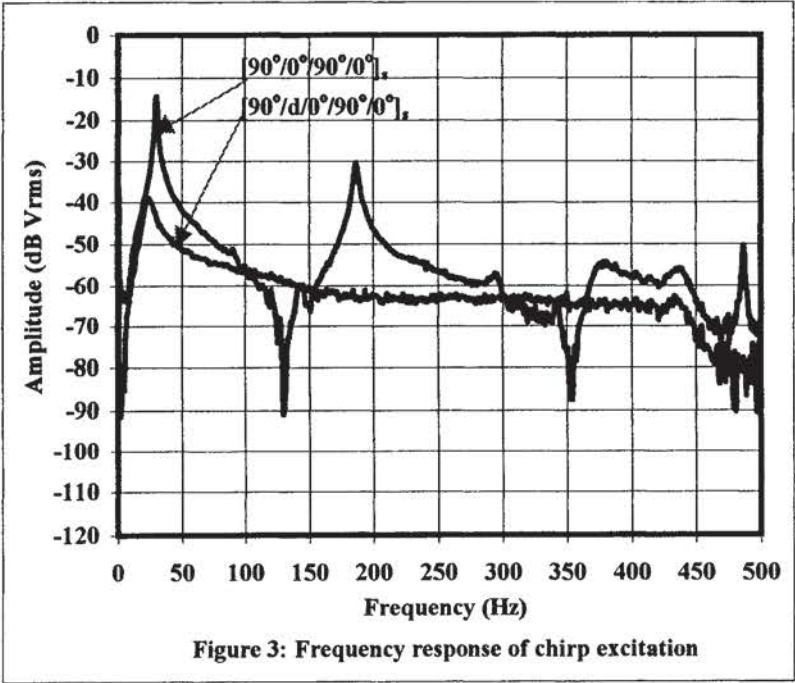
Figure 3: Frequency response of chirp excitation

Figure 4 : Voltage of sensor output Vs time

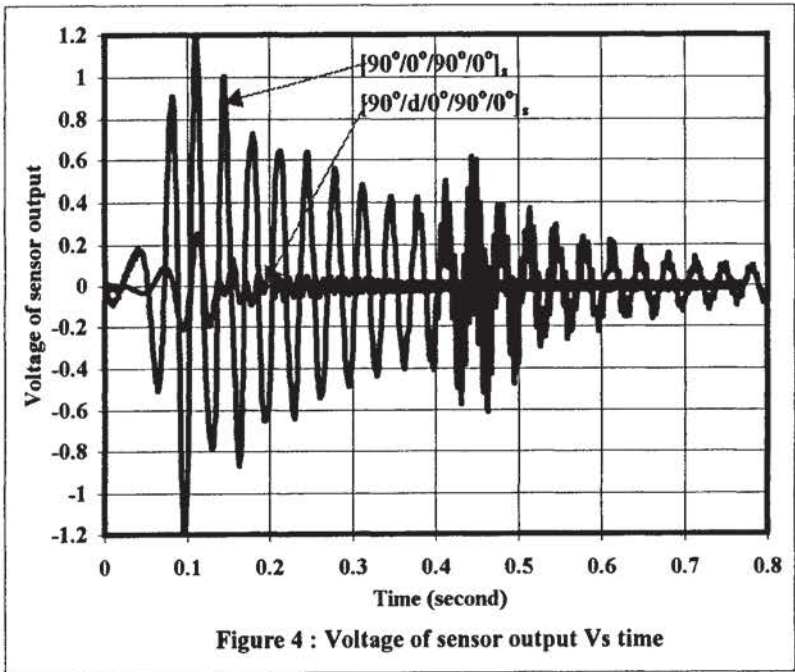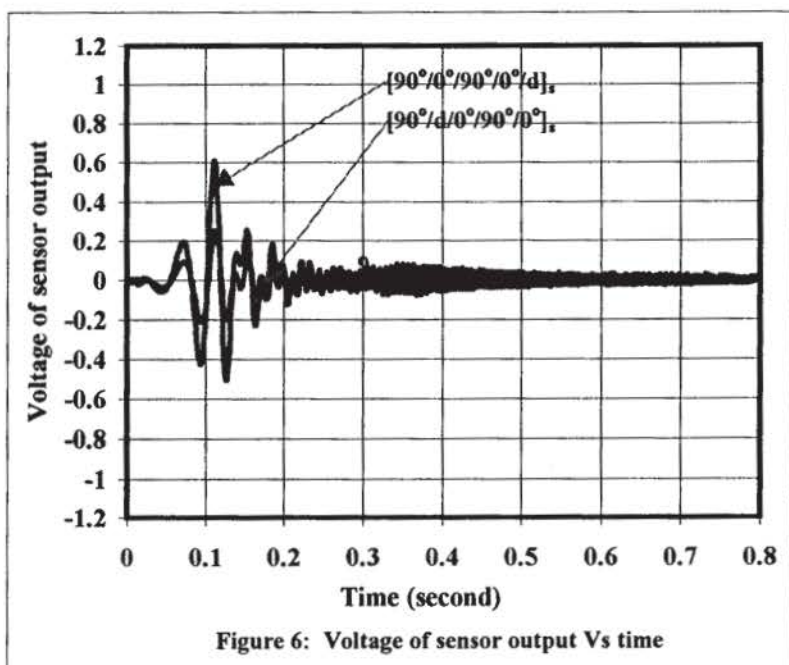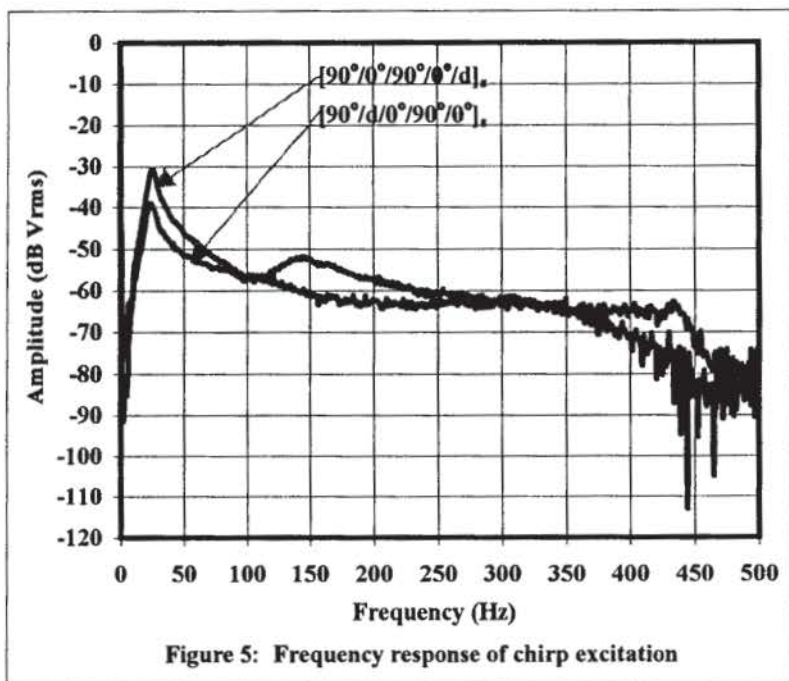**Figure 5: Frequency response of chirp excitation**

Figure 6: Voltage of sensor output Vs time

Figures 7 and 8 present the free vibration frequency response curves and the time response curves, respectively, for plates $[90°/0°/90°/d/0°]_s$, $[90°/d/0°/d/90°/0°]$, and $[90°/d/0°/90°/0°]_s$. One can see that the above orientation is not significantly effective on the resonant frequencies and amplitudes. Also, as can be seen from these figures, the location of the embedded damping layer in the above orientations has no significant effect on the time response curve.

Whenever the plate is subjected to cyclic bending, the composite constraining layers will constrain the viscoelastic material and force it to deform in shear, which is how the vibrational energy is dissipated. This is why the plates with embedded viscoelastic layers show an increase in damping. The obvious decrease in natural frequencies of the plates with damping is caused by the decrease in stiffness introduced with the addition of the damping layers.

One can see that the shear damping is clearly a function of constraining layer thickness. The variation of the damping amplitude is due to the change in location of the embedded damping layer within the plates.

Table 2 lists the natural frequencies obtained from the chirp tests for the first three bending modes for all plates.

Table 2. Natural Frequencies for [90/0/90/0]s with and without embedded layers.

Sequence	ω_1 (Hz)	ω_2 (Hz)	ω_3 (Hz)
$[90°/0°/90°/0°]_s$	30.27	186.52	486.3
$[90°/0°/90°/0°/d]_s$	25.39	142.57	--
$[90°/0°/90°/d/0°]_s$	20.51	--	--
$[90°/0°/d/90°/0°]_s$	23.44	--	--
$[90°/d/0°/90°/0°]_s$	23.44	--	--

Conclusion

This study has wide applications where vibration and noise reduction are of main concern. It has shown an effective way of damping composite structure components and different laminated structures. The addition of the damping layer (3M) material increased the damping ratio and decreased the stiffness/mass ratio. Careful selection of embedded damping layer location is necessary to optimize the damping benefits desirable and stiffness reductions that can be tolerated. The addition of viscoelastic material in that location of the plate $[90°/d/0°/90°/0°]_s$ further reduces the amplitude of vibration in all the fundamental bending mode frequencies more effectively than the other orientations plates. This is because the stiffness/mass ratio for the first plate is greater than that for the second plate. The optimum location of the embedded passive damping material in the cross ply laminated plate $[90°/d/0°/90°/0°]_s$ resulted in degradation effects on the natural frequency, damping and mode of shape.

Figure 7: Frequency response of chirp excitation

Figure 8: Voltage of sensor output Vs time
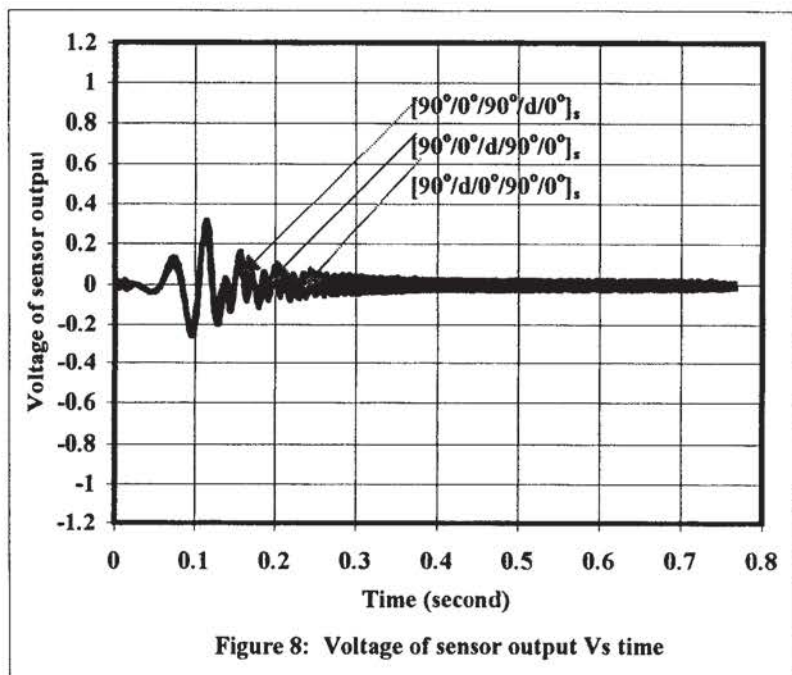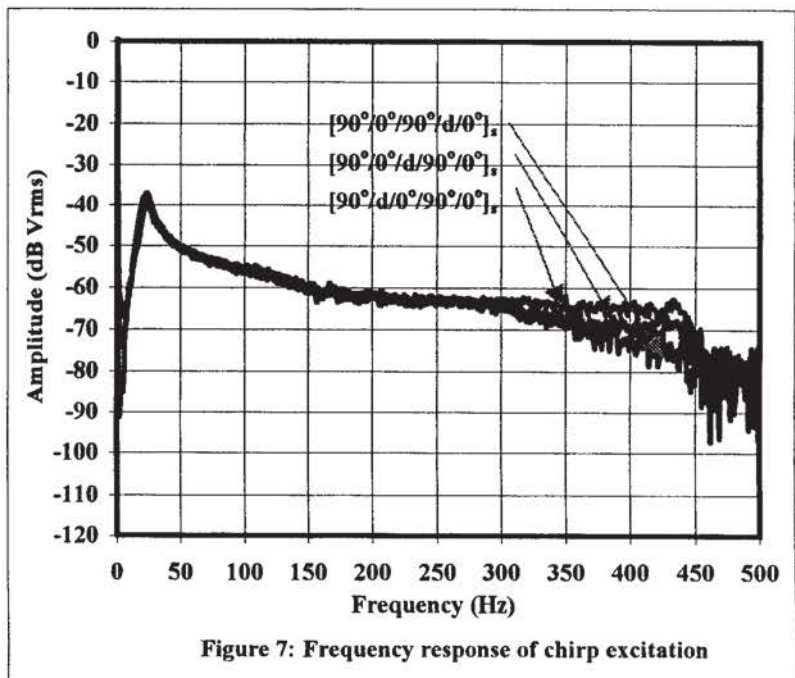
References

1. Ditaranto, R. A., and McGraw, Jr., Vibratory bending of damped laminated plates. *Journal of Engineering for Industry* 91,1081-1090, 1969.

2. Khatua, T. P., and Cheung, Y. K., Bending and vibration of multilayer sandwich beams and plates. *International Journal of Numerical Methods for Engineers* 6(1), 11-24, 1973.

3. Barrett, D. J., An anisotropic laminated damped plate theory. *Journal of Sound and Vibration* 154(3), 453-465, 1992.

4. Saravanos, D.A., and Pereira, J.M., Effects of interply damping layers on the dynamic characteristics of composite plates. *AIAA Journal* 30(12), 2906-2913, 1992.

6. Nadella, S., and Rao, M.D., Damping of composite structures using embedded viscoelastic layers. *Proceedings of the 13th International Modal Analysis Conference, Nashville, Tennessee* V1, 233-239, 1995.

7. Johnson, C.D., and Kienhotz, D.A., Finite element prediction of damping in structures with constrained viscoelastic elements. *AIAA Journal* 20(9), 1284-1290,1981.

8. Gerst, D., Rao, M.D., and He, S., Damping of co-cured composite structures incorporating viscoelastic materials. *Proceedings of the Damping of Multiphase Inorganic Materials Symposium,* 85-93, Chicago, Illinois, 2-5 November 1992.

9. Cavalli, J.R., Elghandour, E.I. and Kolkailah, F.A, Hybrid damping system for an electronic equipment mounting shelf. NASA-Ames Research Center, Ca, NCA2-779, 1997.