VIBRATION DAMPING CHARACTERISTICS OF GRAPHITE/EPOXY COMPOSITES FOR LARGE SPACE STRUCTURES

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INTRODUCTION

Characterization of the vibration damping properties of fiber reinforced composites is important for several reasons. If the dynamic behavior of composite material components is to be estimated during the design stage, it is necessary to know something about the damping in the materials. Indeed, one of the advantages of fiber reinforced composites over conventional structural metals is their enhanced damping characteristics. Another advantage is the potential for designing materials which have predetermined damping, strength and stiffness properties. It is also known that damping is sensitive to microstructural detail, so that the potential exists for using damping measurements to study such things as microstructural damage and fiber-matrix bond integrity. A logical extension of such studies would be the development of non-destructive damping measurement techniques for use in quality control and in-service inspection of composite components.

Previous work by researchers at NASA Langley has resulted in the development of nestable tapered columns and tetrahedral truss elements for orbital assembly of large space structures (refs. 1 and 2). In order to satisfy constraints on Space Shuttle payloads and on deflections of the assembled structures due to physical loading and solar heating, a $[90/0_3/90]$ graphite fiber reinforced epoxy laminate has been selected as the material to be used in the columns. Although the design of the columns has been based primarily on static loading, it is now desirable to investigate the dynamic behavior as well. Dynamic response of such components is governed by their stiffness, mass and damping properties. On earth, such a structure would be subjected to significant damping due to aerodynamic drag and/or acoustic radiation. In the vacuum of space, however, vibrational energy must be dissipated by either internal damping in the structural material, by friction in the connecting joints, or by active control (refs. 3 and 4). The intent of this investigation was to measure the internal damping and dynamic stiffness of the graphite/epoxy composite material used in the columns, and to study the relationships between dynamic properties derived from tests of small specimens and those of full-scale columns. Such relationships are important because, if it can be shown that small specimens are representative of full-scale components, the effects of loading and environmental conditions can be assessed more readily by testing small specimens.

Based on previous experience with structural vibrations of truss networks and experiments with small models, it is expected that transient vibration of a large space structure will cause both extensional and flexural vibrations of the column elements. Extensional vibrations of the columns would occur primarily at the lowest natural frequency of the structure, which is expected to be well below 1 Hz. The dominant flexural modes have been found to occur in the range 10 - 1000 Hz. Accordingly, the techniques used to measure material damping involve both extensional and flexural vibration in the appropriate frequency ranges.

Flexural damping was measured by using a forced vibration technique developed and used previously by the author (refs. 5 and 6). The technique is based on resonant flexural vibration of shaker excited double cantilever specimens.

Extensional damping was measured by subjecting similar specimens to low frequency sinusoidal oscillation in a servohydraulic tensile testing machine while plotting load versus extensional strain. The size of the resulting hysteresis loop is directly related to the damping in the material. The reason for using such a technique is that the vibration exciters such as those used in the flexural damping tests will not operate at the low frequencies required for extensional vibration testing.

DESCRIPTION OF MATERIALS

The current version of the basic graphite/epoxy nestable column element is a tapered tube 2.6 m (102 in.) long with a 0.635 mm (0.025 in.) thick $[90/0_3/90]$ laminated wall. The large end of the tube has a diameter of 102 mm (4 in.) while the small end has a diameter of 51 mm (2 in.). The 90° plies are of 0.127 mm (0.005 in.) thick T-300 fiber/epoxy, while the 0° plies are 0.127 mm (0.005 in.) thick VSB-32 pitch fiber/epoxy. Two types of epoxies have been used. Early tubes were made using prepreg tape with Hexcel F-263 (or equivalent) epoxy resin, while current tubes are made using a dry fiber resin impregnation procedure with Hysol ADX-16/Epotuf 37-619 resin. The aim of this investigation was to test small flat samples of both types of composites and both types of resins. Resin data is needed for the micromechanics analysis of the composite in terms of constituent material properties (ref. 6). At this writing, only the graphite/F-263 composites and the ADX-16 neat resin samples have been fabricated and tested. All specimens were fabricated by Lockheed Missiles and Space Company, Sunnyvale, California (ref. 7). Test specimens 300 mm long x 25 mm wide x 0.635 mm thick were cut from cured 300 mm x 300 mm panels. Neat resin samples were typically 300 mm x 25 mm x 2.5 mm.

EXTENSIONAL VIBRATION

Global motions of large space structures are expected to occur at extremely low frequencies (< 1 Hz). These motions will generate corresponding low frequency extensional oscillations of the column elements. This type of deformation was simulated on small strain-gaged specimens (Fig. 1) by using a MTS servohydraulic tensile testing machine, and damping was found from the resulting load-strain (or stress-strain) hysteresis loops, on an x-y plotter. Loading was applied through

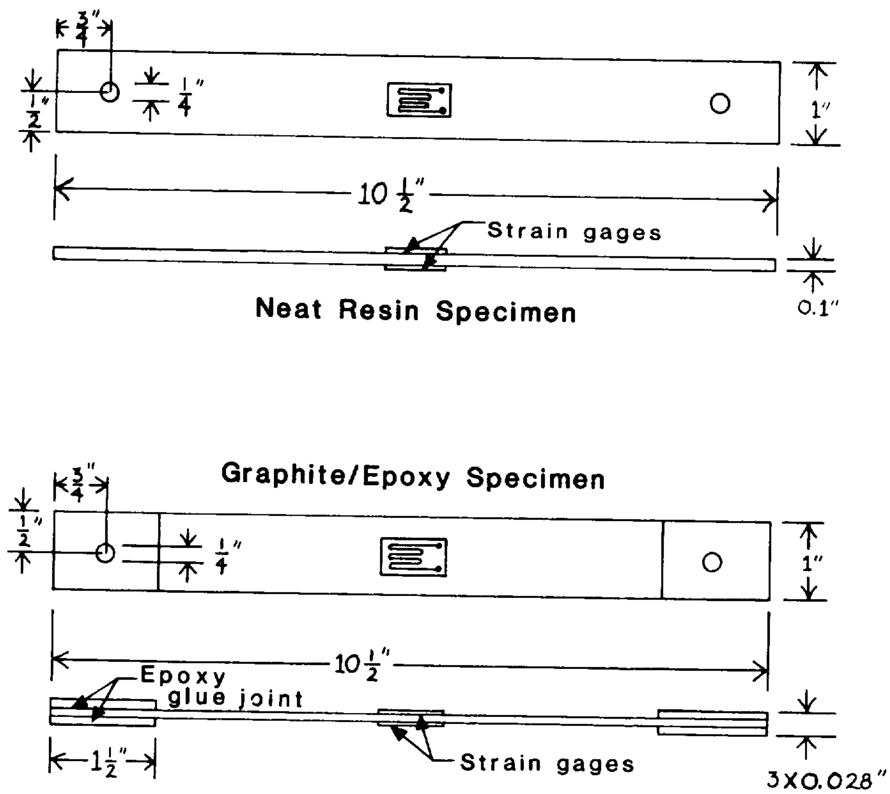

Figure 1

6.35 mm (0.25 in.) diameter steel clevis pins which were inserted through the holes in the ends of the specimens, as shown in Fig. 1. Clevis type loading fixtures were used rather than flat friction grips in order to minimize parasitic energy losses due to clamping friction. A tensile mean stress was applied so that the specimen-loading fixture assembly was always in tension and possible nonlinearities associated with load reversal were avoided. The amplitude of the alternating load was 60-80% of the mean load. The two active gages on each specimen were wired into the bridge circuit on opposite sides, so that bending strains were cancelled out and the resulting bridge output was twice the extensional strain.

Data from the measured hysteresis loops was converted to damping in terms of the loss factor, or loss coefficient (refs. 8 and 9). The loss factor is defined as

$$\eta = \frac{D}{2\pi U} \tag{1}$$

where D = energy dissipated per cycle

U = energy stored at maximum vibratory displacement

For a single-degree-of-freedom model (which is generally acceptable for lightly damped systems near resonance) the loss factor is related to other damping definitions as follows:

$$\eta = \frac{E''}{E'} = \frac{\delta}{\pi} = \frac{\Delta f}{f_o} = \frac{1}{Q} = 2\frac{c}{c_{cr}} \tag{2}$$

where

E'' = loss modulus

E' = storage modulus

δ = logarithmic decrement

Δf = bandwidth of the 3 db down or half-power points

f_o = undamped natural frequency

Q = quality factor

$\frac{c}{c_{cr}}$ = damping ratio

Referring to Fig. 2, the energy dissipated per cycle is simply the area enclosed by the elliptical hysteresis loop

$$D = \pi cd = \pi \left(\frac{a}{2} \cos\theta \right) d \qquad (3)$$

The energy stored at maximum displacement may be approximated by the triangular area

$$U = \frac{1}{2} \left(\frac{b}{2} \right) d \cos\theta \qquad (4)$$

Note that U is the energy associated with the vibration only, not the total area under the curve (i.e., the energy due to the mean stress is not included). According to Eq. (1) the loss factor is then

$$\eta = \frac{D}{2\pi U} = \frac{\pi a d \cos\theta}{2(2\pi)\left(\frac{1}{2}\right)\left(\frac{b}{2}\right) d \cos\theta} = \frac{a}{b} \qquad (5)$$

Specimens were subjected to several cycles of loading to stabilize the hysteresis loops; then the load cell output was plotted versus the strain gage bridge output on an x-y recorder and the dimensions a and b were used to find the loss factor, η , according to Eq. (5). Although the MTS controller has a frequency range of 0.01-100 Hz, the HP 7045A x-y plotter introduced its own phase lag at frequencies above approximately 0.1 Hz. This was checked by plotting load versus load for various frequencies. The resulting plot was a straight line (indicating no phase lag) for 0.01 - 0.1 Hz, but a narrow ellipse of steadily increasing size for frequencies above 0.1 Hz. Thus, with the equipment used, the frequency range for the damping measurements was limited to 0.01 - 0.1 Hz.

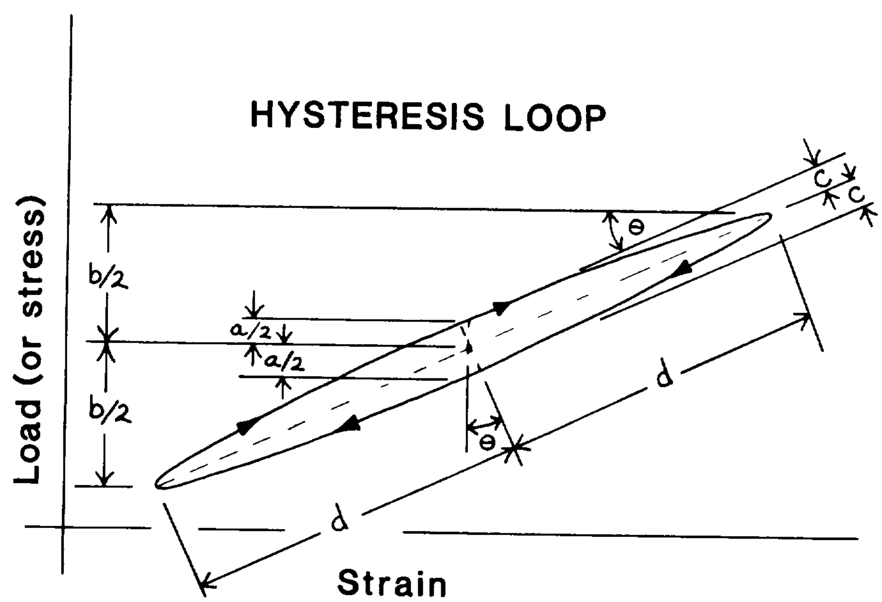

Figure 2

Loss factor data for the previously described materials are shown in Figs. 3 and 4, along with data from flexural vibration tests (to be described in the next section). It was expected that damping would approach zero as the frequency

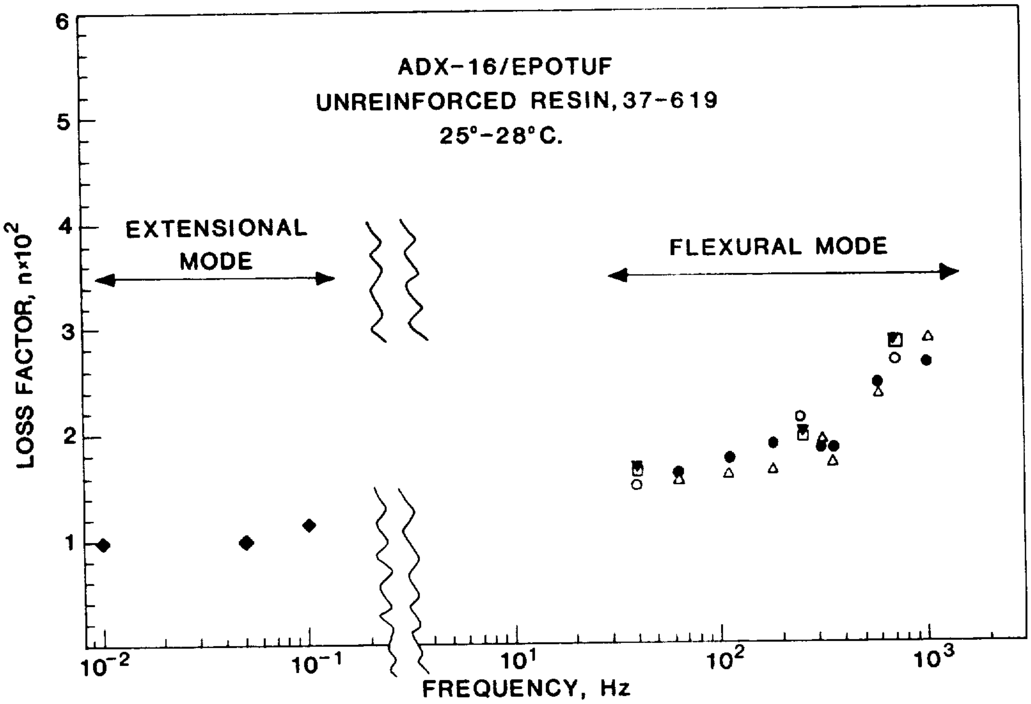

Figure 3

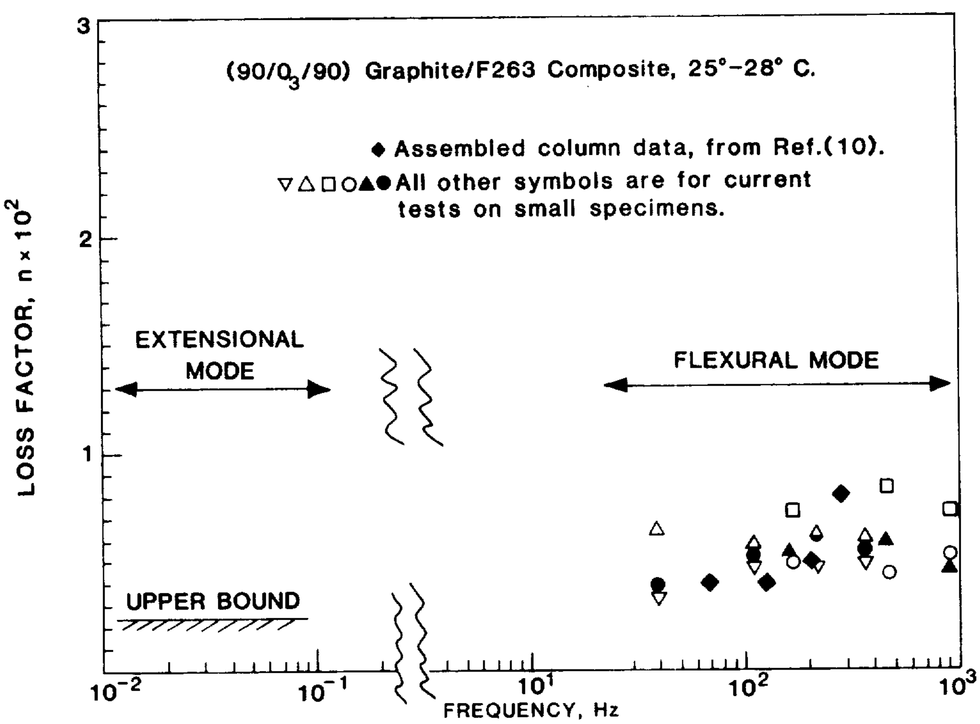

Figure 4

approaches zero, but the rate of decrease appears to be very low. Neat resin damping is greater than composite damping, as expected from micromechanics analysis (i.e., most of the energy dissipation occurs in the resin, and the fibers contribute little damping). Since the neat resin is essentially isotropic, the extensional damping is consistent with flexural damping, as it should be. The composite is anisotropic, however, and the flexural damping depends on the laminate stacking sequence, whereas the extensional damping should not (e.g., a high damping layer is more effective on the outer surface than on the neutral axis in flexure). Thus, the flexural damping for the $[90/0_3/90]$ layup should be greater than the extensional damping. Only the upper bound is shown for the composite extensional damping, because the hysteresis loop was very "noisy." This was due to the fact that the composite extensional modulus was much higher (and the resulting strains were much lower) than those for the neat resin, so the strain signals for the composite samples were more susceptible to electrical noise.

FLEXURAL VIBRATION

Flexural vibrations of column elements are expected to occur in the intermediate frequency range 10-1000 Hz. Experiments with small structural models under transient excitation show that flexural vibrations of the columns continue long after the global motion of the structure has been damped out. Thus, flexural damping is needed in order to minimize flexural fatigue problems.

Flexural damping of the previously described specimens was found by using a forced vibration, resonant dwell technique (ref. 5). As shown in Figs. 5 and 6, the specimen is mounted in a double cantilever arrangement on an electrodynamic shaker. Damping is found by measuring the ratio of base amplitude to tip amplitude at resonance. Unlike other techniques, this method allows precise control over both frequency and amplitude, so that the effects of these parameters can be studied. Considerable effort has been made to minimize parasitic losses in the apparatus which could cause errors in measured damping. For example, the balanced double cantilever arrangement serves to minimize frictional losses at the specimen clamping surfaces, and the specimen is tested at very low amplitude to minimize aerodynamic damping.

It is shown in ref. 5 that the loss factor is related to the resonant amplitude ratio of the cantilever specimen by

$$\eta = C_r \frac{a_b}{a_t} \qquad (6)$$

where C_r = dimensionless coefficient which depends on resonant mode number, r

a_b = resonant amplitude at base of specimen (x=0)

a_t = resonant amplitude at tip of specimen (x=L)

If the base acceleration and the displacement at some arbitrary point, x_o, are measured, it is shown in ref. 5 that Eq. (6) reduces to

$$\eta = \frac{C_r \phi_r(x_o)\ddot{a}(o)}{\phi_r(L)\omega_r^2 a(x_o)} \qquad (7)$$

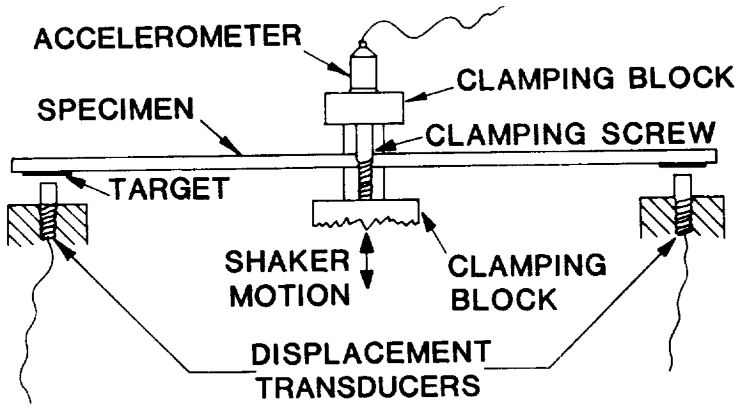

Figure 5

Figure 6

where $\phi_r(x_o)$, $\phi_r(L)$ = mode shape function evaluated at $x = x_o$ and $x = L$, respectively

$\ddot{a}(o)$ = base acceleration amplitude ($x = 0$)

ω_r = resonant frequency of rth mode

$a(x_o)$ = displacement amplitude at $x = x_o$

The present apparatus differed from the one reported in ref. 5, in that tracking filters were used for both the accelerometer and displacement probe signals, and data was generated in digital form by using a digital RMS voltmeter and a computer-printer system.

Flexural loss factors are shown in Figs. 3 and 4, along with previously described extensional data. Specimens of several lengths were tested at frequencies up through the fifth mode. Beyond the fifth mode, the displacement probe signals dropped to the noise level, making the data questionable. The rate of increase in the loss factor with increasing frequency is not as great for the composite as for the neat resin, although it should be pointed out that the resin tested is not the same as the resin in the composite tested. It is hoped that testing of the F263 resin and the ADX-16 composite at a future date will shed further light on this matter. Along with small specimen data in Fig. 4, some limited data (unpublished) for assembled full-scale columns are shown. S. A. Leadbetter (Structural Mechanics Branch, Langley Research Center) obtained this data in 1976 by using the Kennedy-Pancu method on free-free columns vibrated in flexure with an electrodynamic exciter. Due to nonlinear behavior in the connecting joint, no column damping values were found for the first mode, and the data plotted in Fig. 4 is for modes 2-5. Comparison of small specimen loss factors with column loss factors in Fig. 4 shows that there is no significant difference. It is believed that the graphite/F263 composite is representative of the early model columns tested by Leadbetter. Thus, it appears that for the frequencies and amplitudes used in these tests, the material damping is much greater than the joint damping. This contradicts the widely held notion (based on accumulated experience with metallic structures) that joint damping is more important than material damping. Additional data on both columns and small specimens is needed before definite conclusions can be drawn, however.

CONCLUSIONS

Limited data on extensional and flexural damping of small specimens of graphite/epoxy and unreinforced epoxy resin have been obtained. The experimental techniques and data reduction techniques have been described, and the limitations explained. Damping was found to vary slowly and continuously over the frequency range 0.01 - 1000 Hz, and no drastic transitions were observed. Composite damping was found to be less than neat resin damping, as expected from micromechanics theory. Comparison of small specimen damping values with assembled column damping values seems to indicate that, for these materials, material damping is more important than joint damping. Additional tests of current model columns and current specimens with the ADX-16 resin are needed before definite conclusions can be drawn, however. Additional extensional testing is also needed in order to examine frequency and amplitude effects beyond the limited range reported here. The data reported here was limited not by the test apparatus, but by signal conditioning and data acquisition. It is believed that filtering of the strain gage signals and the use of digital storage with slow playback will make it possible to extend the frequency and amplitude ranges significantly.

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REFERENCES

1. Bush, H.G., Mikulas, M.M., and Heard, W.L. Jr.: Some Design Considerations for Large Space Structures. AIAA Journal, 16 (4), April 1978, 352-359.

2. Bush, H.G., and Mikulas, M.M.: A Nestable Tapered Column Concept for Large Space Structures. NASA TMX -73927, 1976.

3. Trudell, R.W., Curley, R.C., and Rogers, L.C.: Passive Damping in Large Precision Space Structures, Proceedings of 21st Structures, Structural Dynamics and Materials Conference, Seattle, Washington, May 1980, Paper No. 80-0677.

4. Structural Dynamics and Control of Large Space Structures, NASA Conference Publication 2187, June 1981

5. Gibson, R.F., Yau, A., and Riegner, D.A.: An Improved Forced Vibration Technique for Measurement of Material Damping, Proceedings of SESA 1981 Spring Meeting, Dearborn, Michigan, June 1981.

6. Gibson, R.F., and Yau, A.: Complex Moduli of Chopped Fiber and Continuous Fiber Composites: Comparison of Measurements with Estimated Bounds, J. of Composite Materials, 14, April 1980, 155-167.

7. Fabrication of Damping Test Specimens, LMSC Report No. D757128, September 1981, Lockheed Missiles and Space Company, Sunnyvale, California.

8. Lazan, B.J.: Damping of Materials and Members in Structural Mechanics, Pergamon Press, 1968.

9. Ungar, E.E., and Kerwin, E.M., Jr.: Loss Factors of Viscoelastic Systems in Terms of Energy Concepts. J. Acoustical Society of America, 34, (7), July 1962, 954-957.