

**MEASURED DAMPING AND MODULUS OF COMPOSITE CYLINDERS**

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**ABSTRACT**

Damping and moduli are measured on graphite and glass fiber/thermoset, thermoplastic, and metal matrix composite cylinders. Measurements are made by the impact-hammer modal-test method with the specimen suspended in the near free-free boundary condition. The axial and in-plane shear moduli are determined from the free-free axial and torsional vibration modes of the cylinder. Damping-loss factors for each of the associated modes are determined from the frequency-response function by the half-power point method or may also be determined by other curve-fitting methods. The effects on damping are presented for a number of different fiber/resin composites and for different lay-ups of the same fiber/resin material.

Cylinders are unique specimen configurations that permit simplified measurements of both the axial and in-plane shear (torsional) material properties from the same sample. The impact-hammer modal-test method provides a quick, inexpensive, small deformation, nondestructive estimate of the moduli and damping for the as-fabricated cylinders. After tests are completed, the cylinder can subsequently be used for its intended purpose. Details of the test method and procedure are described.

\*Based on work performed at the Oak Ridge National Laboratory, operated for the U.S. Department of Energy, under contract DE-AC05-84OR21400 with Martin Marietta Energy Systems, Inc.

## INTRODUCTION

Many composite structures are cylindrical shaped. When designing and fabricating composites, it is highly desirable to know the composition and material properties of the as-fabricated structure. It is common practice to cut coupon samples from an unneeded portion of the structure to submit to the laboratory for physical and mechanical property tests. These tests may include fiber volume fraction, matrix fraction, void, density, strength, modulus, and damping properties. Cutting and testing coupons can be expensive and time consuming. Also, the coupons are usually cut from one end of the composite structure and may not represent what the material properties are throughout the main structure.

Cylinders are unique specimen configurations that permit simplified measurements of both the axial and in-plane shear (torsional) material properties from the same sample. The impact hammer modal test method provides a quick, inexpensive, small-deformation, nondestructive estimate of the moduli and damping for the as-fabricated cylinders. This technique also applies to bar or rod samples to obtain axial modulus and associated damping. Damping values associated with cylindrical tube bending or rod bending are also provided by the impact-hammer modal-test method.

The cylinder and bar specimens are tested in the near free-free boundary condition. Therefore, no special specimen mount fixtures are needed to conduct the test. Specimens are simply suspended in soft rubber elastic or "bungee" cord. The instrumentation is minimal consisting of a force-gaged striker hammer, lightweight accelerometer, and a lightweight tangential striker block.

Knowing the specimen density, the axial,  $E_z$ , and shear,  $G_{\theta z}$ , moduli can be obtained from the axial and torsional resonances using the frequency formula for vibration of continuous media. The resonances are determined from a fast Fourier transform (FFT) analyzer using the frequency-response function. Damping associated with each mode is also determined from the frequency-response function by the half-power point method or some other suitable method.

This paper describes the test setup and the test method, including the use of the exponential window to avoid leakage errors, correcting damping for the exponential window, the effect of mass loading of light specimens due to the striker block and accelerometer, and the use of a noncontact microphone to avoid mass loading problems. The test method provides a pure estimate of axial modulus and torsional modulus because the axial resonance of the specimen is dominated primarily by the axial modulus and the torsional resonance of the specimen is dominated primarily by the in-plane shear modulus,  $G_{\theta z}$ . Likewise, axial and torsional damping estimates are also unencumbered by possible complex combined loading and deformation mechanisms. Other advantages of the test method are that (1) it is nondestructive on as-fabricated cylinders; (2) no coupons are necessary; (3) boundary conditions and air damping effects are minimized with free-free axial and torsional modes; and (4) the test method is quick and inexpensive.

## APPLICABLE SPECIMENS

The axial and in-plane shear moduli are determined from the cylinder's free-free axial and torsional vibration modes, respectively. Any uniform cylindrical, tubular, or bar-shaped (axial only) specimen that approximates the long, slender, continuous media assumptions will provide reasonable data from the impact modal test method without conducting a complete modal test. Usually, it is easy to extract the modes of interest from specimens that are long, slender, thick walled, heavy, and have a high length-to-diameter (L/D) ratio. It is more difficult to identify the modes of interest of specimens that are short, of large diameter, thin walled, and lightweight because of the high modal density of the shell modes in the proximity of the axial and torsional modes.

The test method employed in these studies can accommodate quite a range in size, shape, and configuration of specimens. Examples of specimens tested are pictured in Fig. 1. Note the 1-ft rule in the lower lefthand corner of the photograph. Ranges of weight and dimensions of specimens tested are listed.

Weight	15 g to 210 lb
Outside diameter	1 to 17 in.
Wall thickness	0.030 to 1.5 in.
Length	5.0 to 109.0 in.
L/D	2.2 to 31.2

## TEST DESCRIPTION

The test method employs standard modal test techniques using a force-gaged impact hammer for input and an accelerometer for response. Specimens are tested under near free-free boundary conditions for axial and torsional vibration modes to determine modulus of elasticity and damping-loss factor. The cylindrical or bar-shaped specimens are soft mounted using elastic bands, surgical tubing, or bungee cord depending on their weight. The axial and torsional modes of vibration are excited by impacting the specimen in the axial and tangential directions, respectively. Data are acquired on a dual channel FFT signal analyzer. The data are subsequently postprocessed, stored on disk, and plotted or printed using a microcomputer. A schematic of the test setup is shown in Fig. 2. A typical test is pictured in Fig. 3. Details of the axial and torsional test input and response are shown in Figs. 4 and 5, respectively.

The axial impact point at the edge of the cylinder, as indicated in Fig. 4, is eccentric to the cylinder's neutral axis and therefore excites bending as well as axial modes of vibration. The axially mounted accelerometer can respond to bending as well as axial modes. It is necessary to take enough data to be able to distinguish between modes. In some cases, it may be necessary to perform a full modal test. Some shortcut alternative mode identification techniques are discussed in the Results section.

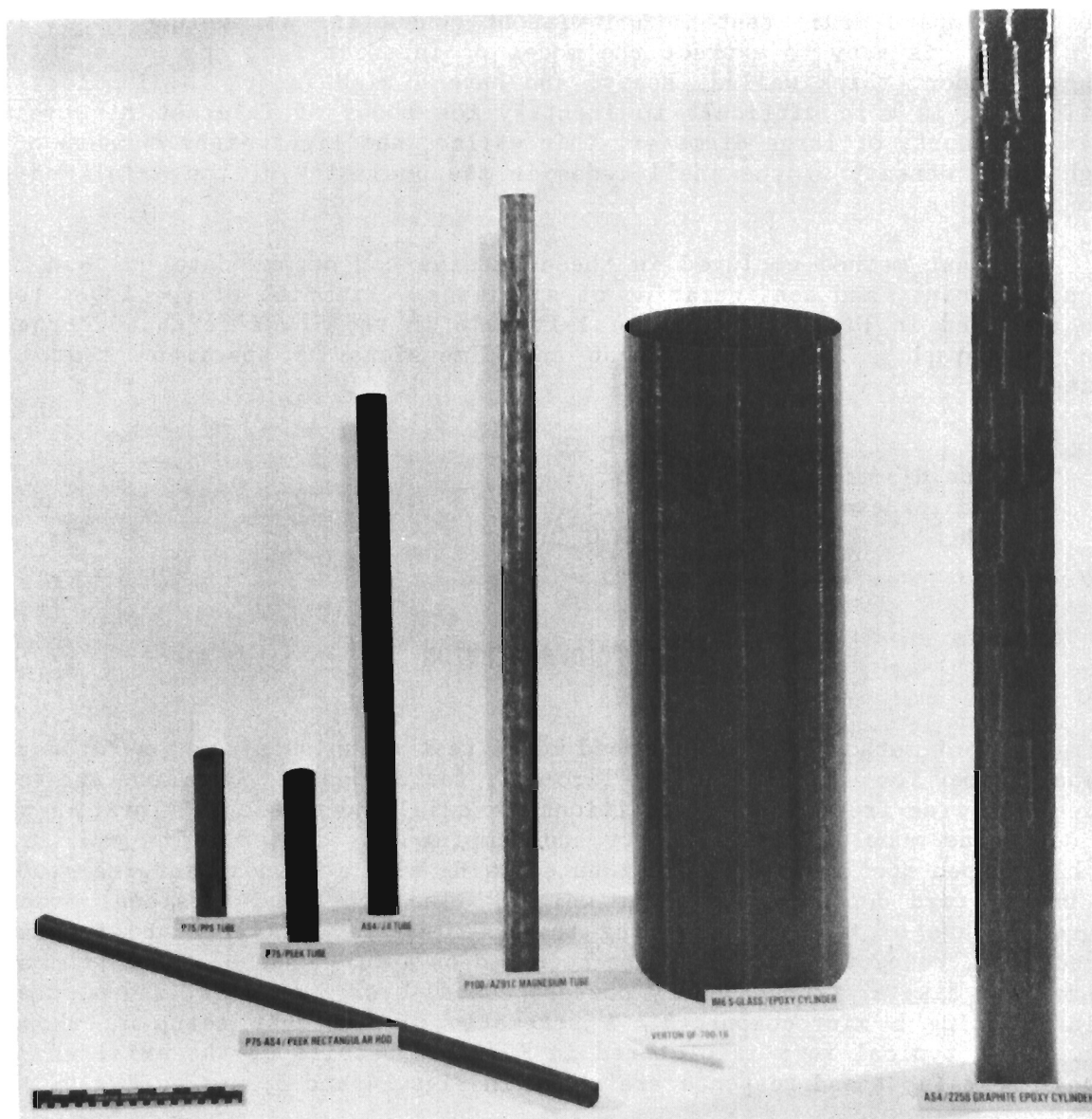


Fig. 1. Examples of cylindrical, tubular, and bar specimens.

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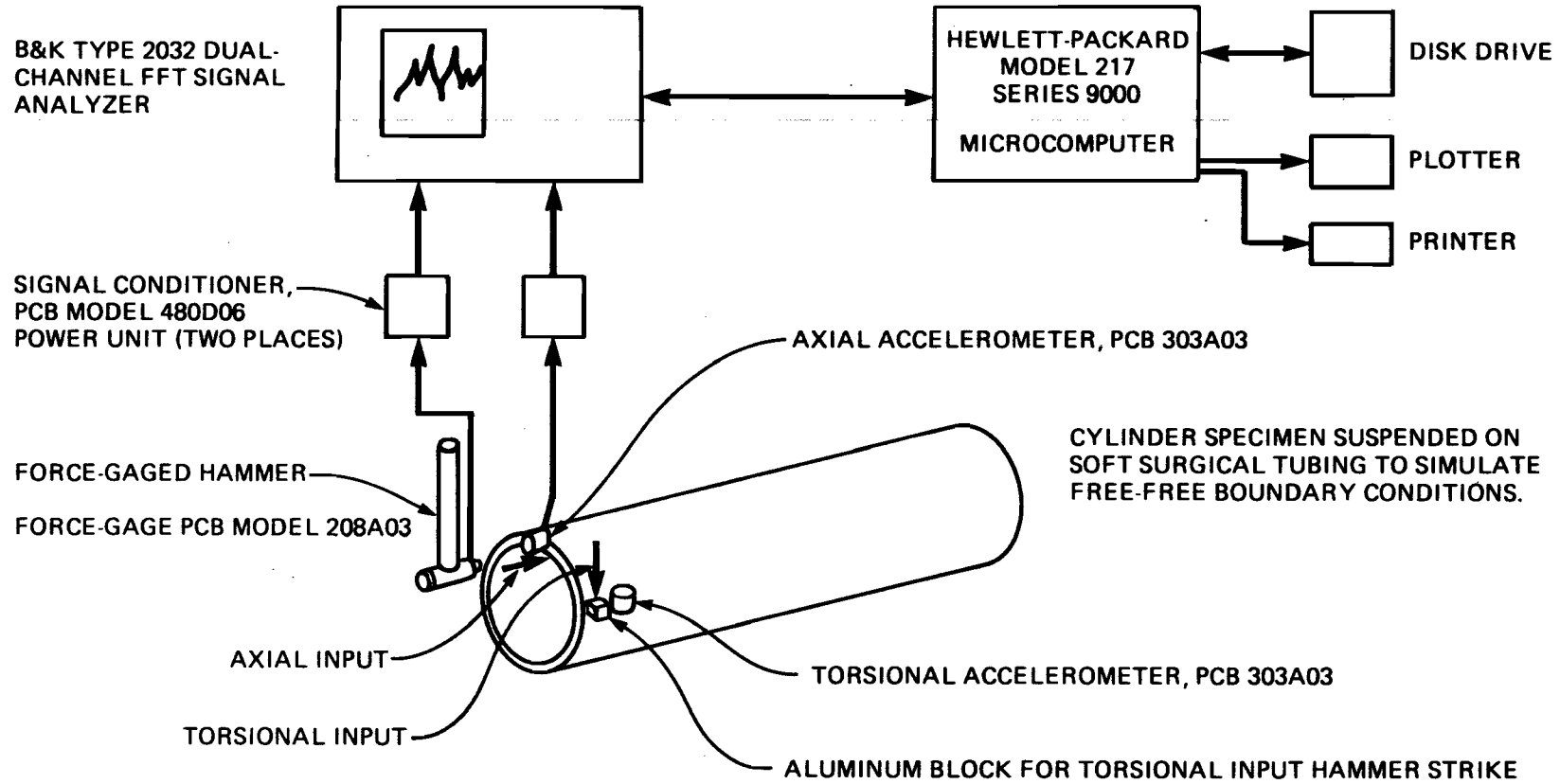


Fig. 2. Modal test schematic for axial and torsional free-free vibrations to determine modulus of elasticity and damping loss factor.



Fig. 3. Typical test setup.

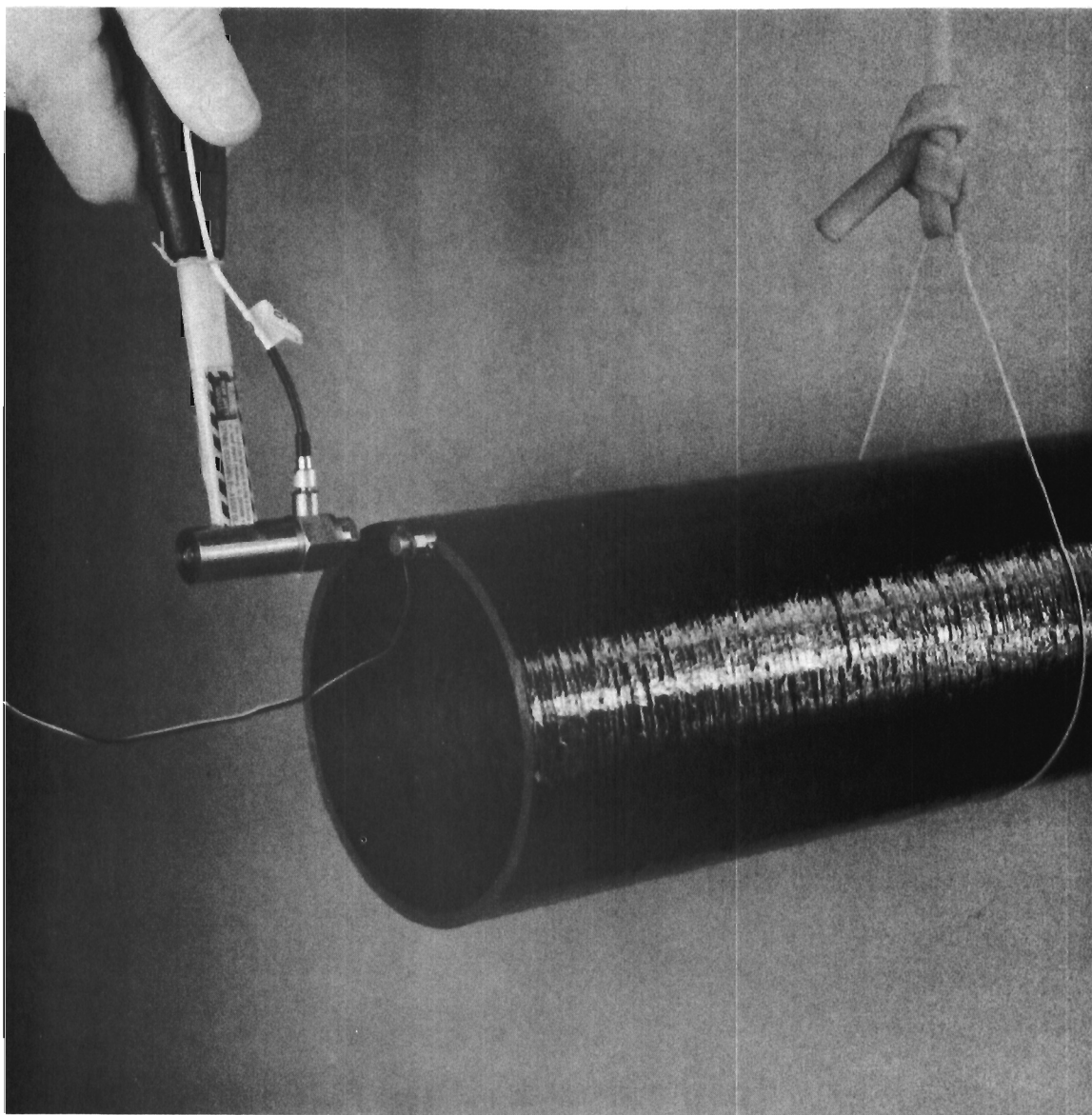


Fig. 4. Axial input and response.

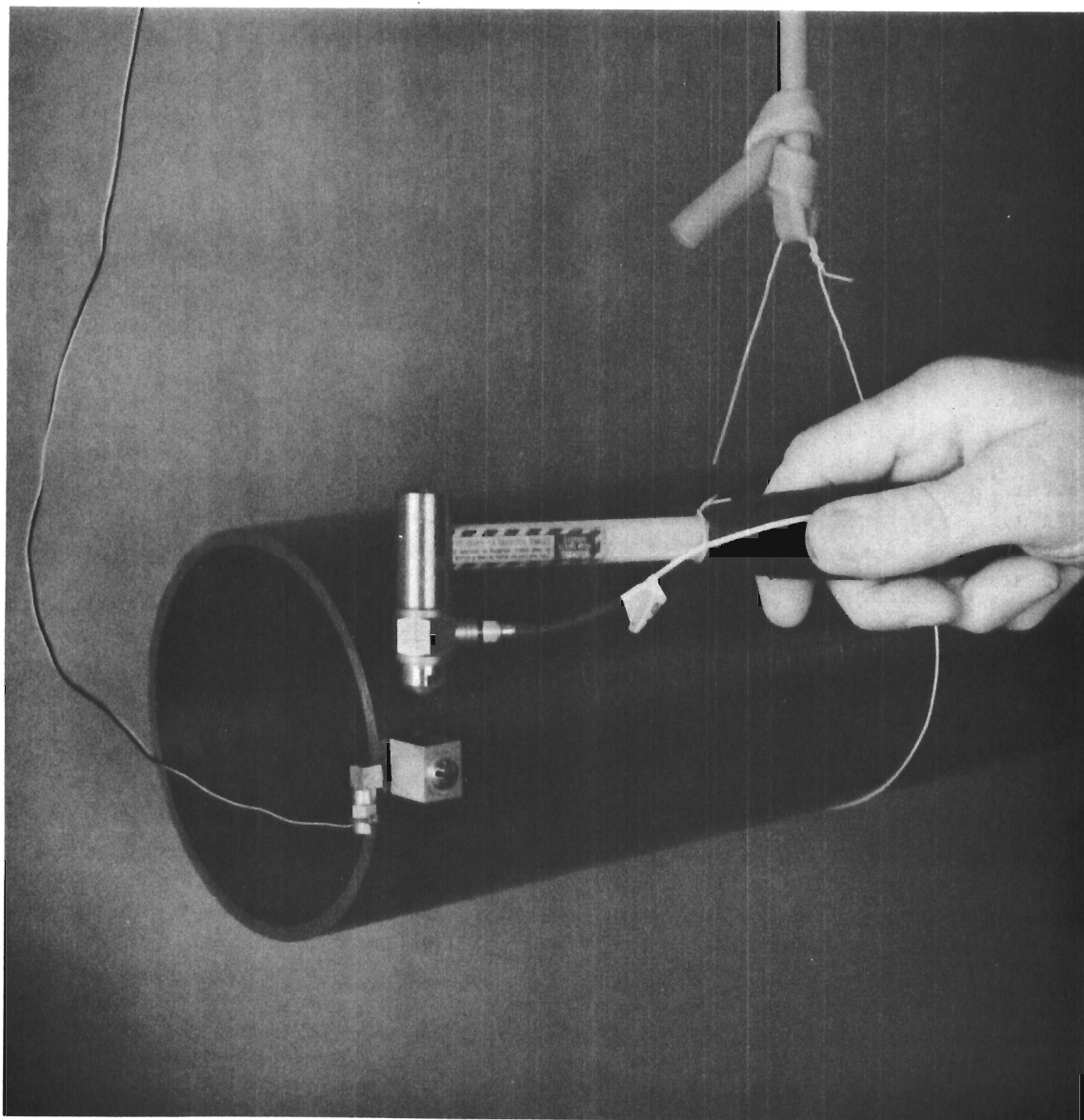


Fig. 5. Tangential (torsional) input and response.



Tangential force is transmitted into the specimen through a lightweight aluminum striker block bolted to the wall of the cylinder, as shown in Fig. 5. The block should be just big enough to apply the hammer blow and of low weight to minimize mass loading. Some effort should be made to match the striker block with the curvature of the cylinder. An exact match is not necessary. The same block has been used on cylinders from 2 to 20 in. in diameter, provided as the bolt is sufficiently tight to prevent rattle or slippage. In addition to the bolt, bee's wax has been used at the striker block/cylinder interface to compensate for slight differences in curvature and surface irregularities.

The tangential accelerometer is mounted as close as possible to the neutral axis of the shell surface, as shown in Fig. 5, to minimize sensitivity of the accelerometer to circumferential ring and shell modes. An alternative is to mount the accelerometer to the bottom of the striker block. On thick-wall specimens, this may be acceptable, but on thin-wall specimens, this alternative is not advisable because of prevalence and sensitivity to shell modes in proximity to the primary torsional modes of interest.

The axial modulus,  $E_z$ , and the shear modulus,  $G_{\theta z}$ , were determined from the free-free axial and torsional vibration modes using the frequency equation for prisms of continuous media, as shown in Fig. 6. Damping was measured using the half-power point method and corrected, as shown in Fig. 7, for artificially added damping due to the exponential window used to acquire the response accelerometer data. The exponential window was used to acquire response data to ensure that the assumption of periodicity employed in the FFT analyzer for the sample time history was not violated. Violating the FFT analyzer periodicity assumption results in leakage errors. Leakage errors introduce an "apparent" damping of an unknown quantity (wider peaks) in the frequency-response function. The exponential window can eliminate leakage errors, but it introduces damping of a known quantity. The decaying exponential window is a weighting function applied to the sampled response time history. The FFT analyzer processes the product of the measure-time history and the exponential window. The measured damping was subsequently corrected (see Fig. 7) for damping artificially added by the exponential window. A rule of thumb for selecting the time constant of the exponential window is that it should be about one-fourth of the sample time. For moderately damped materials, the response-time history decays to a negligible value within most test sample times. Therefore, an exponential window may not be necessary. For specimens with very high resonant frequency of interest and very low damping, the response signal may not decay to near-zero values within the test sample time. Therefore, the use of the exponential window is advisable.

The impact-modal method is applicable to composite rings as well as cylinders and rods. But one needs to use a combination of testing and finite element analysis to adjust the hoop modulus,  $E_{\theta}$ , and shear modulus,  $G_{R\theta}$ , to match the first circumferential or ring resonance, as well as higher circumferential harmonics. For thin rings, adjusting for hoop modulus,  $E_{\theta}$ , may be all that is necessary to match predicted and tested resonances. For thicker rings, it is necessary to adjust the hoop modulus,  $E_{\theta}$ , and the shear modulus,  $G_{R\theta}$ , to obtain a reasonable match between tested and finite element-predicted resonances and harmonics.

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AXIAL MODULUS  $E_{z,n} = \frac{4\ell^2 f_{A,n}^2 \rho}{n^2 g}$   $E_z$  = AXIAL MODULUS, lb/in.<sup>2</sup>

$G_{\theta z}$  = SHEAR MODULUS, lb/in.<sup>2</sup>

$\ell$  = CYLINDER LENGTH, in.

SHEAR MODULUS  $G_{\theta z,n} = \frac{4\ell^2 f_{T,n}^2 \rho}{n^2 g}$   $f_{A,n}$  = nth AXIAL FREE-FREE RESONANCE, Hz

$f_{T,n}$  = nth TORSIONAL FREE-FREE RESONANCE, Hz

$n$  = MODE NUMBER, 1, 2, ...

$\rho$  = MATERIAL DENSITY, lb/in.<sup>3</sup>

$g$  = GRAVITATIONAL CONSTANT, 386.1 in./s<sup>2</sup>

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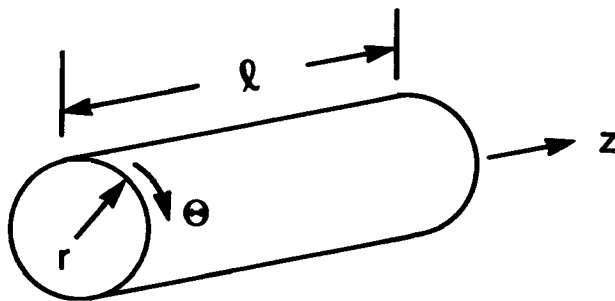


Fig. 6. Moduli are determined from free-free axial and torsional vibration modes.

## HALF-POWER POINT OR 3-dB-DOWN METHOD TO COMPUTE LOSS FACTOR

MEASURED LOSS FACTOR  $\eta_M = \frac{\Delta f_M}{f_M}$  INCLUDES ARTIFICIAL  
EXPONENTIAL WINDOW  
DAMPING

CORRECTED LOSS FACTOR  $\eta = \frac{\Delta f_M - \frac{1}{\pi\tau}}{f_M}$  CORRECTS FOR  
EXPONENTIAL WINDOW  
DAMPING

WHERE

$\Delta f_M$  = MEASURED HALF-POWER POINT BANDWIDTH, Hz  
FROM FREQUENCY RESPONSE FUNCTION

$f_M$  = MEASURED RESONANT FREQUENCY, Hz

$\tau$  = EXPONENTIAL WINDOW TIME CONSTANT s/RADIAN; TIME AT  
WHICH AMPLITUDE IS REDUCED BY THE FACTOR,  $1/e$

Fig. 7. Damping is measured by the half-power point method and corrected for artificially added damping by the exponential window.

**EXAMPLE RESULTS****COMPOSITE DRIVE SHAFT (A LARGE SPECIMEN)**

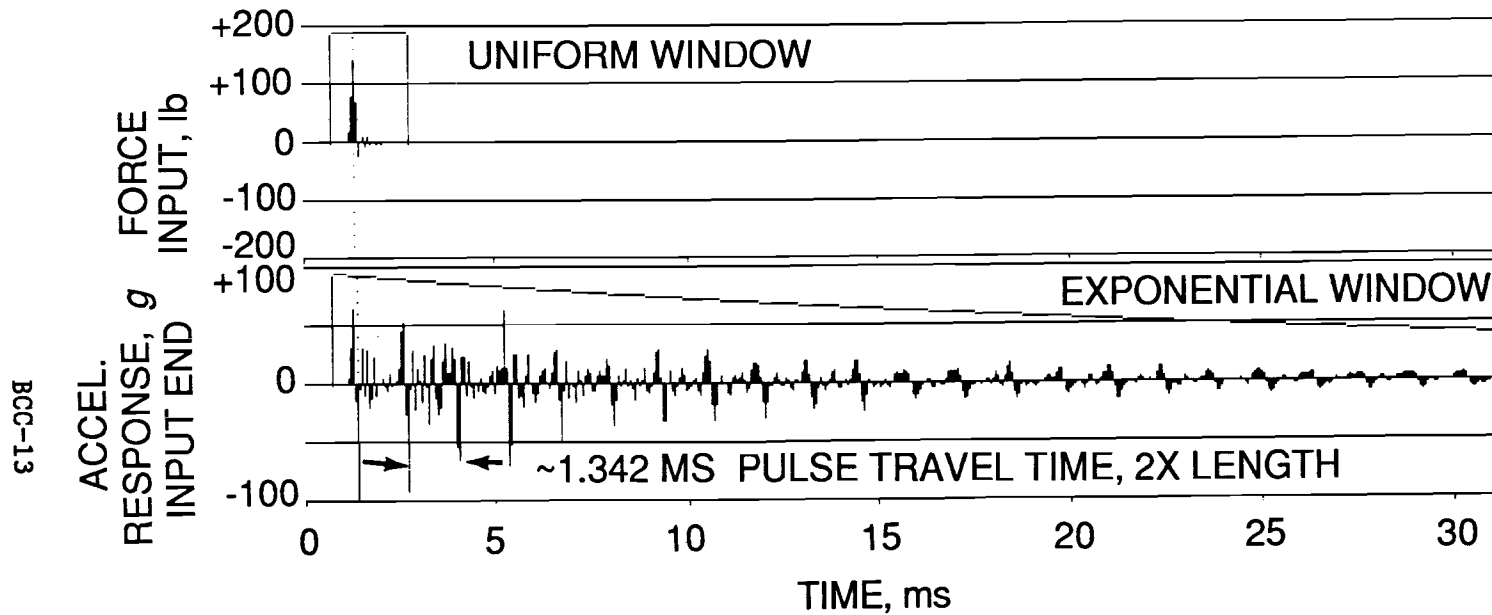
The composite drive shaft (shown as a typical test setup in Fig. 3) is described as follows.

Outside diameter	3.515 in.
Inside diameter	2.698 in.
Length	109.66 in.
Weight	27.14 lb
Density	~0.0621 lb/in. <sup>3</sup> , ~1.719 specific gravity
Material	T-700 graphite fibers, OCF S-Glass, and 2258/m-phenylenediamine resin system

<u>Material</u>	<u>Orientation</u>	<u>Fraction of thickness</u>
S-Glass	Hoop	0.203
S-Glass	Helical $\pm 45^\circ$	0.203
T-700	Helical $\pm 45^\circ$	<u>0.594</u>
		1.000

Typical time histories of the axial forced input and acceleration response are shown in Fig. 8. About 140-lb force lasting a fraction of a millisecond is shown as input. The acceleration response at the input end of the cylinder is shown on the lower plot. In this example, the total sample time was 125 ms, about 31 ms of that sample is shown. The axial hammer input and the accelerometer installation are shown in Fig. 4. The 1.342-ms acceleration pulse time increment shown in Fig. 8, is the time required for the acceleration pulse to travel from the input end, reflect off the opposite end and return to the input end. The inverse of this time (~145 Hz) is approximately equal to the first axial resonance of the system (157 Hz) as determined from the frequency-response function (see Fig. 9). If observed, the acceleration pulse-time increment provides a good first estimate of the axial-resonant frequency, which aids in distinguishing this mode of interest from the other modes. Integrating the acceleration response time history of Fig. 8 twice yields a typical maximum axial displacement of 0.1 to 0.2 mils single amplitude (SA).

The first two axial modes are shown in the axial frequency-response function of Fig. 9. The first seven bending modes can also be observed in the axial frequency-response function because the input and response is eccentric to the neutral axis of the cylinder. The first axial mode of 756 Hz yields an axial modulus of ~4.43E6 psi. The loss factors for the first and second axial modes is 0.01022 and 0.01125, respectively. Note that the first two axial modes have about the same loss factor. The classical laminate code prediction for axial modulus for this cylinder is ~4.4E6 psi, which is very close to the measured value.

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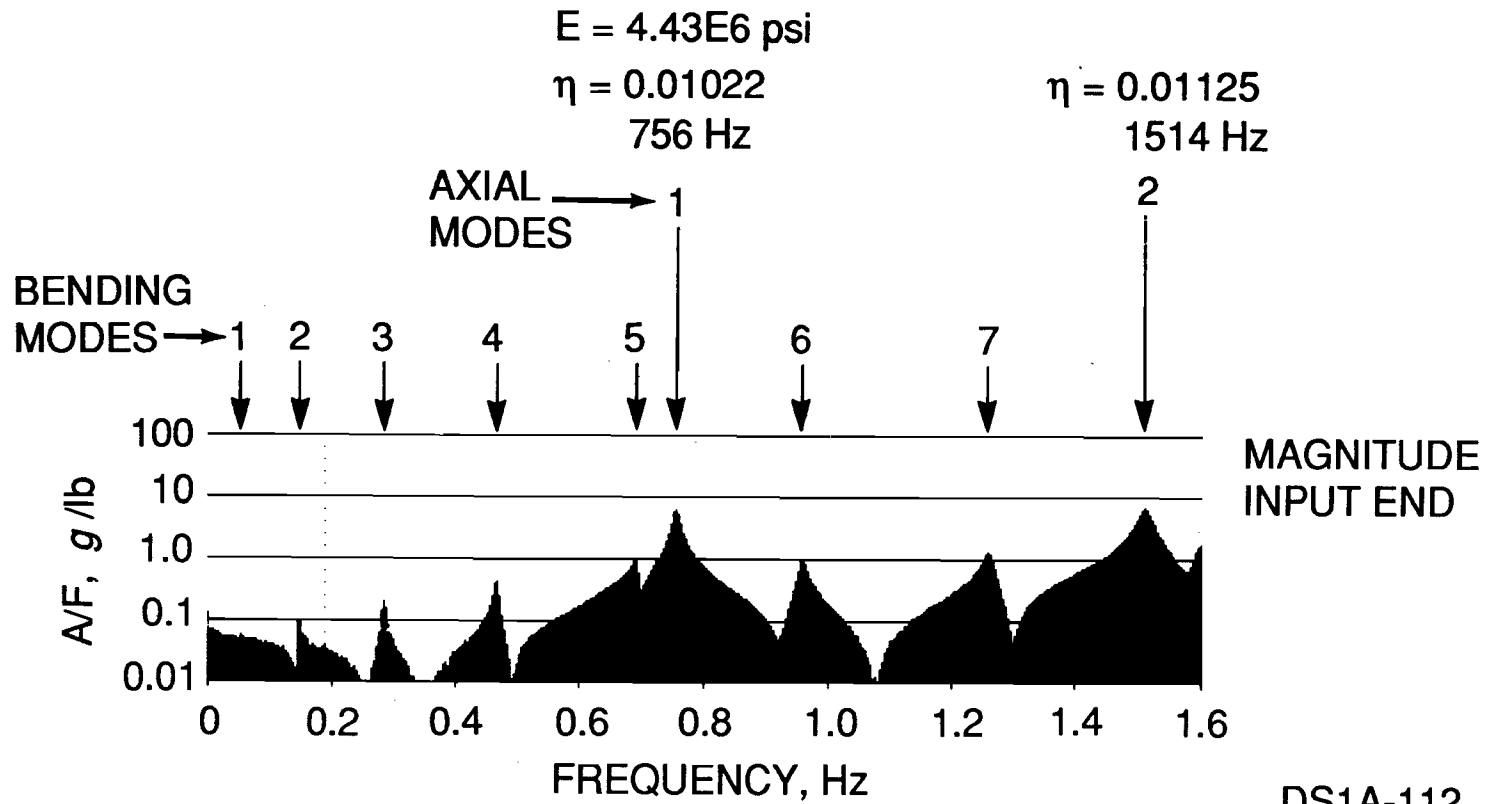
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NOTE: TYPICAL MAXIMUM AXIAL DISPLACEMENT = 0.1 TO 0.2 mils SA.

Fig. 8. Composite drive shaft, typical time histories for axial input and response.

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BGC-14



LAMINATE CODE PREDICTION  $E = 4.40E6 \text{ psi}$

TYPICAL MAXIMUM AXIAL DISPLACEMENT = 0.1 TO 0.2 mils SA.

Fig. 9. Frequency-response function, A/F, measured with axial input and response, for a composite drive shaft.

Similarly, for tangential input and response, the first two torsional modes are observed as shown in the frequency-response curve of Fig. 10. The first torsional mode of 767 Hz yields a shear modulus of  $\sim 4.55E6$  psi. Loss factor for the first and second modes is 0.00403 at 767 Hz and 0.00306 at 1537 Hz, respectively. The classical laminate code prediction for shear modulus of this specimen is  $5.09E6$  psi. Typical tangential displacement in this test was 3 to 4 mils SA.

The very small axial displacement of from 0.1 to 0.2 mils SA and tangential displacement of 3 to 4 mils SA are indications that the air damping for this specimen is probably negligible. According to Gibson's work,\* where small flexure beam specimens were tested in both air and vacuum, it was found that air damping was negligible if beam displacements were equal to or less than the smallest cross-sectional dimension (thickness) of the flexure beam specimen. Consequently, air damping on the cylindrical specimens is expected to be very small because the response displacements are small compared with other dimensions of the specimen. Also, for axial motion, the edge surface area of the specimen doing the air pumping is extremely small. Likewise, resistance to torsional motion is drag of the air along the circumference of the cylinder, which is expected to be negligible.

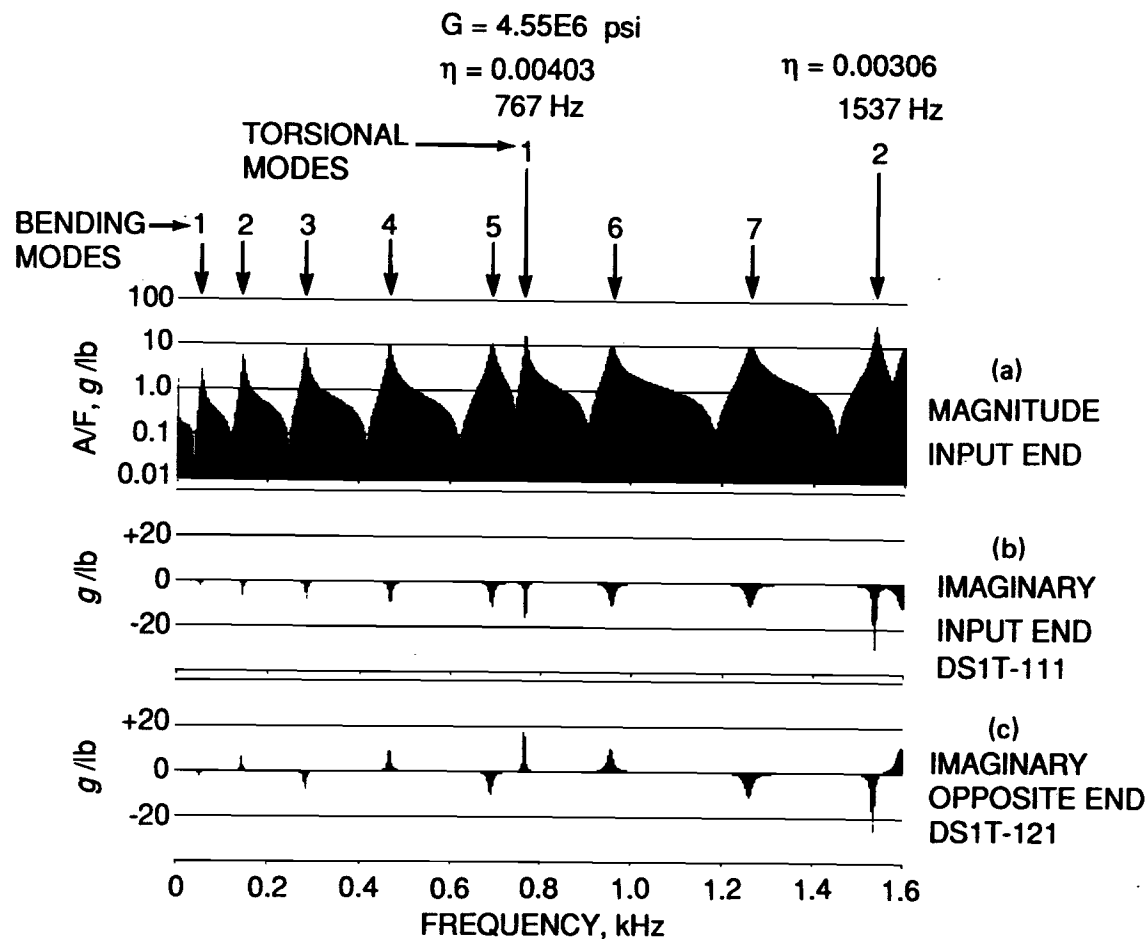
One way to identify free-free axial and torsional modes of vibration from other modes is that axial and torsional harmonics are interger multiples of the first mode. This distinction helps to pick out these modes from the magnitude of the frequency-response function.

Another shortcut in determining vibration modes from modal test data is also illustrated in Fig. 10. The imaginary part of the frequency-response functions for tangential accelerometers mounted at opposite ends of the cylinder are shown in Fig. 10b and c. Because the real part of the free-free response function is zero at resonance, the imaginary part is essentially the total response of the specimen at resonance. The phase relationship of the ends of the cylinder can be determined from the imaginary part of the frequency-response function at resonance. For example, with the first torsional mode at 767 Hz, the imaginary part of the frequency-response function at the input and opposite ends have opposite signs indicating that the ends are out of phase, as would be expected for the first torsional mode. Likewise, for the second torsional mode at 1537 Hz, the imaginary part of the input and opposite ends have the same sign, indicating that the ends are in-phase as would be expected for the second free-free torsional mode shape. The same information can be determined for the bending modes. Note that the first bending mode has opposite ends of the cylinder in-phase, the second bending mode has opposite ends of the cylinder out of phase, and so on, as would be expected for the free-free bending mode shapes.

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\*R. F. Gibson et al., *Vibration Characteristics of Automotive Composite Materials*, ASTM STP 772, 133-150 (1982).

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LAMINATE CODE PREDICTED  $G = 5.09E6 \text{ psi}$   
 AVG  $\eta$  FOR BENDING MODES 2 TO 5  $\sim 0.0094$   
 TYPICAL MAX TANGENTIAL DISPLACEMENT = 3 TO 4 mils SA

Fig. 10. Frequency-response function, A/F, measured with tangential (torsional) input and response, for a composite drive shaft.

BCC-16



## INJECTION-MOLDED BAR (A SMALL SPECIMEN)

Another example of a specimen tested is shown in Fig. 11. It is a standard 5-in.-long flex-beam specimen made from long glass fiber-reinforced injection-molded nylon. The specimen weighs just over 15 g, and the accelerometer shown in the figure weighs 1 g. This example is shown to illustrate the effect of mass loading of very light specimens with the accelerometer and to illustrate the use of a microphone to measure the axial response and thereby eliminate the need and the effect of the accelerometer mass. In Fig. 12a, the axial acceleration frequency-response function shows a resonance at 11,000 Hz with a damping loss factor of 0.00870. In Fig. 12b, the frequency response is measured by a microphone with the accelerometer still attached to the specimen. It shows a resonance of ~10,995 Hz and a loss factor of 0.00880. Fig. 12c shows the frequency-response function measured by the microphone without the accelerometer attached, giving a resonance of 11,840 Hz and a loss factor of 0.0071. The resonance is higher because the accelerometer mass is removed from the specimen. The 11,840-Hz resonance yields an estimated modulus of 2.01E6 psi, which compares favorably with the vendor specification axial modulus of 2.22E6 psi from tensile tests and 2.00E6 psi from flexure tests. Note that the microphone shows more resonances than does the accelerometer. The reason is that the microphone is a omni-directional transducer and can measure noise resulting from many of the specimen modes, whereas the accelerometer is a single-direction transducer, and it will pick up motion, primarily in the direction that the accelerometer is oriented.

## FREQUENCY RESOLUTION EFFECTS

A comparison of frequency-resolution effects on estimating damping is shown in Fig. 13. It is desirable to have the smallest frequency resolution in order to make a good estimate of damping using the half-power-point method. For the example trial-fabrication thin-walled composite cylinder, the axial resonance is determined by using base-band frequency analysis with a resolution of 8 Hz. A loss factor of 0.00231 was estimated at 4546-Hz axial resonance. By using zoom with frequency resolution of 2 Hz (see Fig. 13b), the loss factor for the first axial mode was 0.00219 at 4546 Hz. The axial modulus was estimated to be 24E6 psi, which compared well with the classical laminate code prediction of 27.3E6 psi. This example illustrates that there is very little difference in estimating the loss factor whether one uses zoom or a reasonable base band analysis as long as interpolation is used to estimate the half-power frequency width. When conducting tests on a large number of specimens, it is much easier and faster to conduct base-band frequency analyses than it is to conduct zoom analyses. As an initial test setup, it is recommended that one select a base-band frequency range so that the resonance of interest falls in the upper half of the frequency analysis range. Also, it is recommended that one use an interpolation method to estimate the half-power bandwidth.

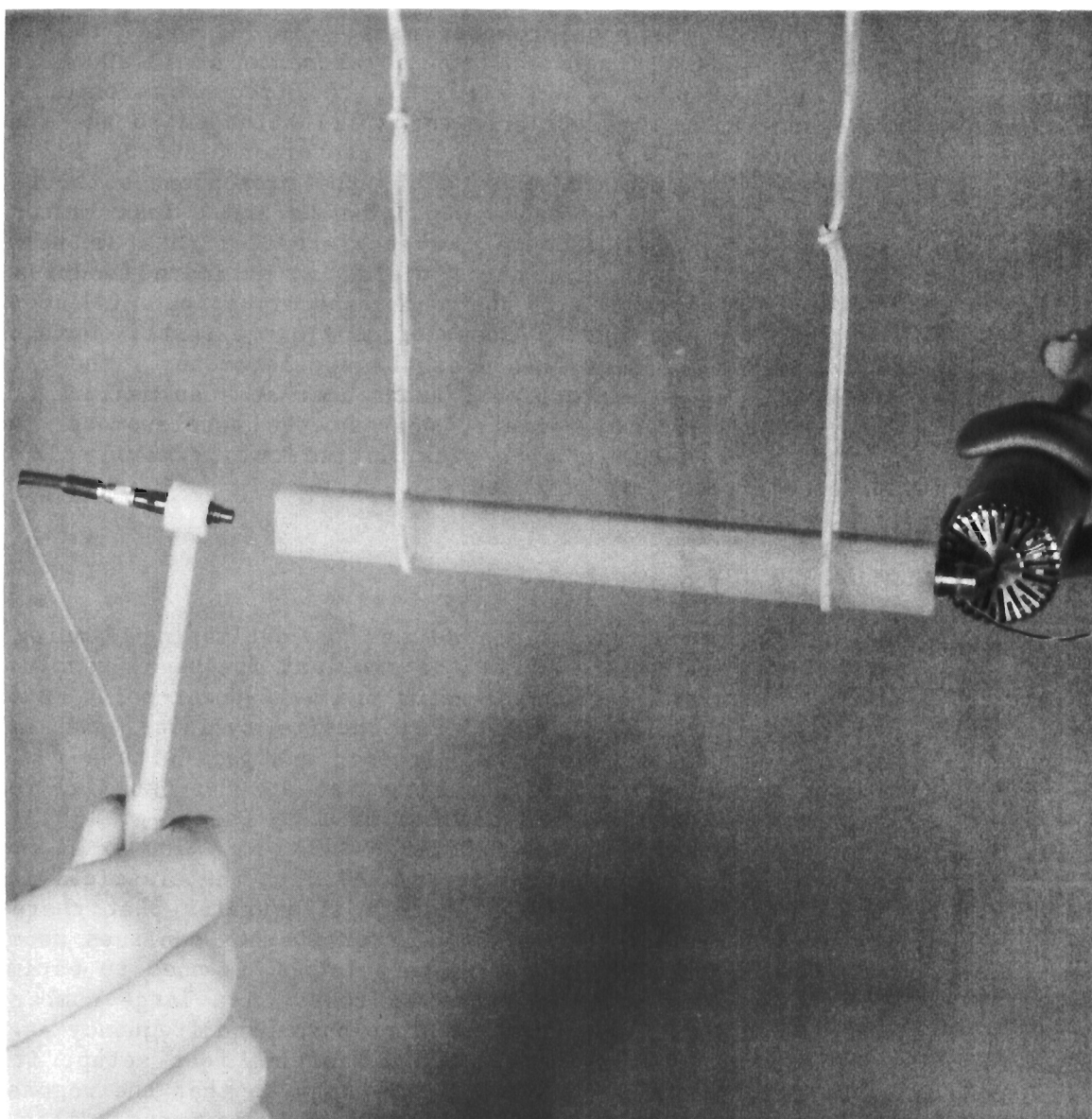
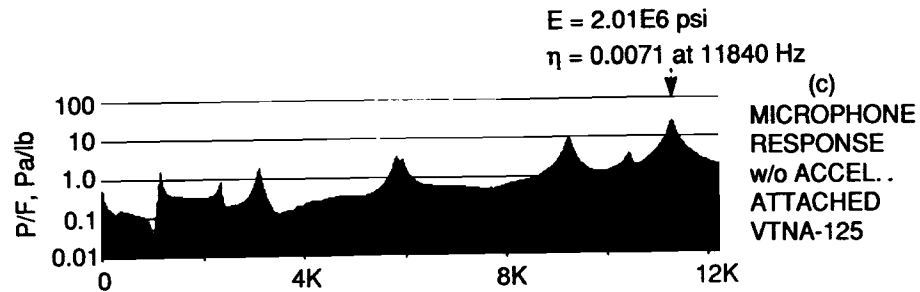
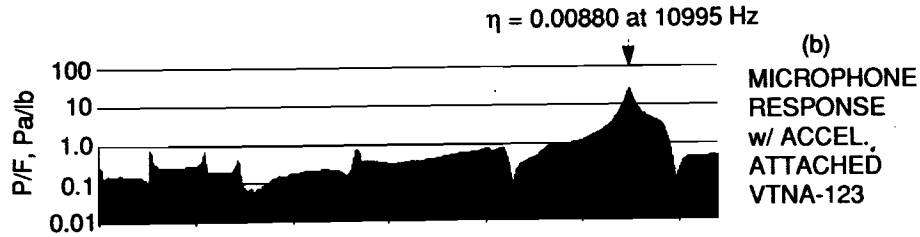
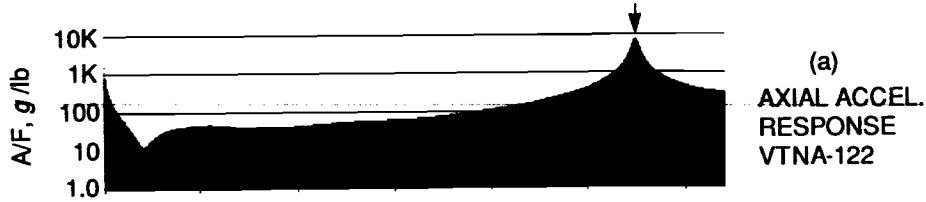


Fig. 11. Axial test of a small bar specimen.

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**INJECTION MOLDED FLEX BEAM**  
**VERTON QF-700-10 0.25 x 0.50 x 5.00 in.**  
**LONG-FIBER-REINFORCED NYLON 6/10 ICI/LNP**

**AXIAL INPUT AND RESPONSE**      **SPECIMEN MASS 15.2 g**  
**FREQUENCY RESPONSE FUNCTION**      **ACCEL MASS 1.0 g**  
 $\eta = 0.00870$  at 11000 Hz



VENDOR SPEC.      E TENSILE = 2.22E6 psi  
 E FLEXURAL = 2.00E6 psi

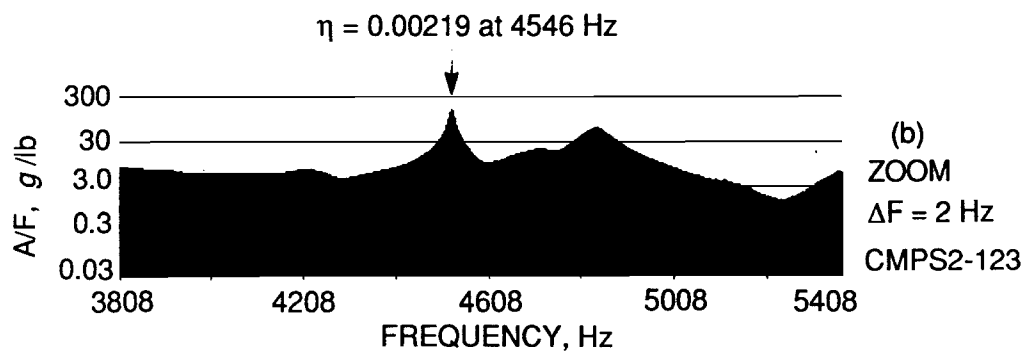
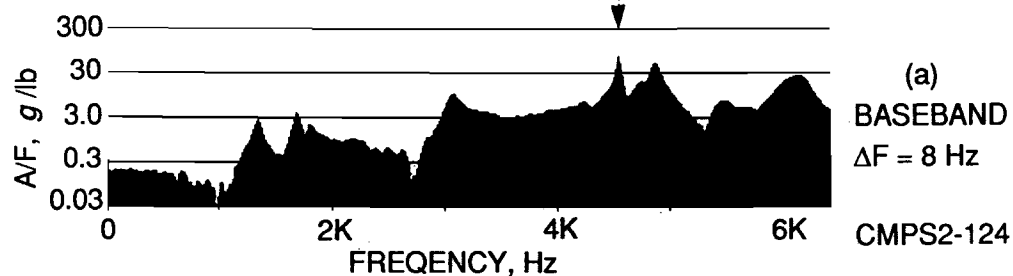
**Fig. 12. Accelerometer mass loading can be avoided.**

BCC-19

14.02 in. OD, 41.73 in. LONG, ~0.062 in. WALL  
79% P55 GRAPHITE AXIAL, 9% S-GLASS AXIAL, 13% S-GLASS HOOP  
ERL 2258 EPOXY RESIN

AXIAL INPUT AND RESPONSE  
FREQUENCY RESPONSE FUNCTION A/F g/lb

E = 24.0E6 psi  
 $\eta = 0.00231$  at 4546 Hz



LAMINATE PREDICTED E = 27.3E6 psi

Fig. 13. Effect of zoom on damping measurement compulsator trial fabrication cylinder.

BCC-20

**AN EXAMPLE OF THE EFFECT OF COMPOSITE LAY-UP ON DAMPING**

An interesting set of axial and torsional damping loss factor data were taken on a dimensionally consistent set of cylinders made from various wet-wound composite lay-ups of AS4 graphite fiber/ERL-2258 epoxy matrix composite. Trial specimens were fabricated consisting of four different lay-ups, each with three different wall thicknesses. The inside diameter was ~4.40 in, the cylinder length varied from ~ 61 to 67 in., and the three nominal wall thicknesses were 0.125, 0.188, and 0.25 in. Table 1 shows the lay-up angle, the fraction of the total thickness at that angle, and the damping-loss factors for axial, torsional, and bending resonances. The 0° angle direction is axial, and the 90° direction is hoop. Because there was no significant difference in damping for various wall thicknesses, the average damping value for all three thickness is reported in Table 1.

**Table 1. Axial and torsional damping loss factors for various cylinder lay-ups of AS4/2258 epoxy composite**

Lay-up angle, <sup>a</sup> fraction	Loss factor (normalized loss factor)		
	Axial	Torsional	Bending
0°, 0.5/90°, 0.5	0.00245 (1.00)	0.01749 (7.14)	0.00726 (2.96)
0°, 0.33/90°, 0.67	0.00355 (1.45)	0.01740 (7.10)	0.00688 (2.81)
0°, 0.25/90°, 0.25/±45°, 0.5	0.00456 (1.86)	0.00448 (1.83)	0.00744 (3.04)
±54° Helix	0.01653 (6.75)	0.00327 (1.34)	0.01632 (6.66)

<sup>a</sup>0° = axial, 90° = hoop

Inside diameter, in. = 4.40

Nominal wall, in. = 0.125, 0.188, 0.25

Tube length, in. = 61.78 to 66.88

In the 0°/90° lay-up with the same amount of materials in the axial and hoop directions, the torsional loss factor is over seven times higher than that of the axial direction. This is because the torsional mode is dominated by the matrix material which provides a much higher damping than the fiber, whereas the axial mode is dominated by the fiber material. In the 0°/90° lay-up with twice as many hoops as axial, the torsional loss factor is likewise six to seven times higher than that of the axial loss factor. Again, this is due to the matrix dominance of the torsional motion and the fiber dominance of the axial motion.

For the  $0^\circ/90^\circ/\pm 45^\circ$  quasi-isotropic composite lay-up, the torsional and axial damping loss factors are approximately equal as would be expected for a quasi-isotropic lay-up. Because the damping value for the quasi-isotropic lay-up is very close to that of the axial damping for the  $0^\circ/90^\circ$  lay-up, one would conclude that both the torsional and axial modes of vibration for the quasi-isotropic lay-ups are fiber dominated.

For the  $\pm 54^\circ$  helix lay-up, the axial loss factor is about five times greater than the torsional loss factor. This is true because, for the  $\pm 54^\circ$  helix, the axial resonance is dominated by the matrix, and the torsional resonance is dominated by the fiber, which is opposite the case for the  $0^\circ/90^\circ$  lay-up, as would be expected. Also, note that for the  $\pm 54^\circ$  helix lay-up, the loss factor for bending is approximately the same as for axial. This should be expected because bending vibration involves the same axial strain energy loss mechanism as axial vibration.

In these data, it is also observed that the loss factor associated with bending for the  $0^\circ/90^\circ$  lay-ups and the quasi-isotropic lay-ups are approximately two to three times higher than the axial loss factor. It is speculated that there could be two reasons for this. One is that beam bending involves some shear deformation. Consequently, matrix-dominated shear deformation in beam-bending vibration provides a supplemental damping mechanism to the fiber-dominated axial strain in beam bending. A second possibility is that additional apparent damping could be added to the free-free bending modes by the soft bungee cord or surgical tube suspension system. Additional work is planned to evaluate the significance of possible added damping from the soft suspension system by testing standard aluminum and steel (low damping material) in the same test setup under various test conditions including bending.

A similar set of data and conclusions can be drawn for a pair of specimens shown in Table 2. These data are for 7-in.-inside-diameter IM6 graphite/ERL-2258 epoxy composite cylinders--one with  $0^\circ/90^\circ$  lay-up and one with quasi-isotropic  $0^\circ/90^\circ/\pm 45^\circ$  lay-up. For the  $0^\circ/90^\circ$  specimen, there is about eight times more damping for the matrix dominated torsional vibration than for the axial vibration. For the quasi-isotropic lay-up, there is about the same amount of damping in both the torsional and axial vibration modes.

#### ADVANTAGES OF THE TEST METHOD

The advantages of the impact modal test method are described as follows.

1. The modal impact test method provides nondestructive modulus and damping data on as-fabricated cylinders. There is no need to cut, fabricate, and test coupon samples.
2. The method applies to a wide variety of cylindrical or bar specimen sizes, weight, and configurations.

Table 2. Axial and shear moduli and damping loss factors for shear stiffened cylinders

Cylinder (material)	Fiber angle %	Fiber vol fraction	Modulus		Loss factor (frequency, Hz)		Normalized loss factor	
			Axial, $E_z$ E6 psi	Shear, $G_{\theta z}$ E6 psi	Axial	Torsional	Axial	Torsional
SSC1 <sup>a</sup> (IM6/2258 epoxy)	0°/33.5	0.6659	8.90 <sup>b</sup>	1.07 <sup>b</sup>	0.0035	0.0293	1.00	8.37
	90°/66.5		[7.89] <sup>c</sup> (9.68) <sup>d</sup>	(0.82) <sup>d</sup>	(4034)	(1397)		
SSC2 <sup>e</sup> (IM6/2258 epoxy)	0°/25	0.6217	8.21 <sup>b</sup>	3.42 <sup>b</sup>	0.0053	0.0051	1.51	1.48
	90°/25 ±45°/50		[7.83] <sup>c</sup> (9.18) <sup>d</sup>	(3.50) <sup>d</sup>	(4468)	(2883)		

<sup>a</sup>SSC1 (SSC-2) 8.20 in. OD × 7.02 in. ID × 30.53 in. long, 11644 g, 25.67 lb.

<sup>b</sup>Data from modal tests.

<sup>c</sup>Data in [ ] are from strain gaged external pressure tests (i.e., compressive stress).

<sup>d</sup>Data in ( ) are from CLASS laminate code prediction.

<sup>e</sup>SSC2 (SSC-7) 8.31 in. OD × 7.02 in. ID × 26.65 in. long, 10311 g, 22.73 lb.

3. The near free-free boundary test condition requires no special fixturing. Also, the free-free test condition eliminates uncertainty in the resonant frequency and added damping which other constraining support mechanisms may introduce.
4. The pure axial and torsional modes provide data that are free of complex loading issues as might be encountered in a bending specimen that has both axial and shear deformation associated with bending vibration. Both axial and shear moduli and damping are obtained from the same specimen sample.
5. Air-damping effects are expected to be minimal because (a) displacements are small for axial and torsional modes compared with other dimensions of the specimen and (b) the pumping action normal to the moving surface area is negligible for axial modes of vibration and zero for torsional modes of vibration. For example, surface shear drag on the cylinder in torsional motion is expected to be negligible.
6. A noncontacting microphone can be used to detect resonances to avoid accelerometer mass loading of lightweight specimens. The use of a microphone has been demonstrated for detecting axial resonances. It is questionable as to whether it would detect torsional resonances.
7. Analyzer leakage errors can be eliminated by using exponential windows to acquire response time history data.
8. Tests are simple, quick, and inexpensive.

#### DISADVANTAGES OF THE TEST METHOD

The disadvantages of the impact modal test method include the following.

1. Data are limited to small deformations.
2. The modulus is associated with oscillatory stress--not one direction loading.
3. Modulus and damping are global--not local properties.
4. Data are associated with only specific specimen resonant frequencies unless steps are taken to alter them.
5. A variable, nonstandard specimen size and configuration may make it difficult to compare modulus and damping properties of similar material and composite lay-up.
6. Data may be difficult to obtain from specimens with low L/D ratios and with thin walls. High shell modal density or bending modes may obscure axial and torsional modes of interest. Acquiring meaningful data in a short period of time may be difficult or impossible under these conditions.



7. The accelerometer and tangential striker block can mass load and significantly alter resonances of relatively lightweight specimens. This is expected to affect the modulus measurement because of the lower resonant frequency but not to affect the damping measurement.
8. A noncontact displacement probe may be difficult to use on a free-free mounted specimen that is lightweight because high displacements after impact may move the specimen outside the range of the probe. This is less of a problem with heavy specimens that undergo small displacement under impact conditions.
9. There may be a question of the effect on axial resonance due to point excitation at the edge of a large-diameter cylinder instead of uniform loading distributed over the entire circumference.

#### ISSUES NEEDING ADDITIONAL WORK

To resolve some questions about the test methods, additional work in the following areas is recommended.

1. Investigate the effect on damping of the soft near free-free suspension system especially for vibration modes associated with beam bending. Dr. D. I. G. Jones of the Wright Research Development Center/Materials Laboratory recommended that this could be evaluated by testing specimens of known low-damping value, such as aluminum or steel, with the same near free-free soft suspension system.
2. Eliminate or compensate for the striker block and accelerometer mass loading affects on lightweight specimens in torsion.
3. Evaluate the use of a microphone as a noncontact transducer with other specimens.
4. Evaluate the affects of point vs uniform excitation on axial resonants of low L/D ratio cylindrical specimens.

#### SUMMARY

The impact modal test method on cylinders will provide a "pure" measurement of axial and in-plane shear moduli and associated damping. Since these are nondestructive tests on as-fabricated cylinders, no coupon tests are necessary. Boundary conditions and air-damping effects are minimized with free-free axial and torsional modes. For most specimens, the impact modal test method is a quick, inexpensive and nondestructive means of measuring moduli and damping of as-fabricated cylinders and rods.

Examples of damping data and modulus were shown for several sizes, configurations, and lay-ups of composites. Composite damping was shown to be

very sensitive to the applied principal stress relative to the fiber direction, even in the same specimen. In the composites illustrated, matrix-dominated deformation or motion provides (6) to (8) times more damping than fiber-dominated motion.

Axial and in-plane shear moduli determined from the impact modal test method of free-free cylinders compared well with expected values for most lay-ups.

Some helpful hints (short of a full-modal test) were described to distinguish axial and torsional modes of interest without performing a time-consuming, more costly full modal analyses including measured frequencies and mode shapes.

Careful selection of base-band analyses to ensure that resonances of interest are in the upper half of the frequency band, in many cases, will provide reasonable estimates of half-power points and damping loss factors. For many specimens, the use of time-consuming zoom analyses to increase frequency resolution can be avoided.

The effect on the bending-mode damping due to the soft, near free-free suspension system needs to be investigated. Difficulties may be encountered with low L/D, thin-walled, lightweight specimens because of the possible existence of high shell modal density in the vicinity of the primary axial or torsional modes of interest. Striker block and accelerometer mass loading of lightweight specimens may affect the estimate of modulus but is not expected to affect damping measurements.