Combined Compression-Torsion Buckling Tests of Laminated Composite Cylindrical Shells

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Results are described for six composite test cylinders constructed of either four or six plies of graphite-epoxy or boron-epoxy and subjected to combined torsion and compressive loading. All the shells were 15 in. in diam and 15 in. long. The stacking sequence effects exhibited by these thin laminates produced significantly different buckling capacities depending on the direction of torque application. A test technique was developed to define the entire range of torsion/compression ratios for each type of shell, and one specimen was used to define each shell type. The interaction between the torsion-compression buckling stresses was shown to be linear except at high percentages of compression stress.

Introduction

FUSELAGE curved panels must be designed to resist combined shear and compression loads without buckling. Assumptions may be made in the design of these panels without considering interaction effects, but the panels will then weigh more than needed. Thus, a need for experimental research was identified.

Some previous work in the area has been performed by Holston, Feldman, and Stang[1] and Tennyson.[2] Holston, Feldman, and Stang's experimental work with glass-fiber-reinforced filament-wound cylinders indicated the torsion-compression interaction to be relatively linear. Their test program included three laminates of four-plies each (pseudo-isotropic, pseudo-orthotropic, and anisotropic) and three radius-to-thickness ratios (90, 200, 360). One combination of torsion and compression was investigated in addition to pure torsion and pure compression. Since each cylinder was used for one data point, 27 cylinders were required. Their analytical predictions were acknowledged to be poor for torsion loadings due to neglect of specific boundary conditions. Tennyson tested 14 three-ply glass-epoxy cylinders, including 9 different laminate arrangements. Four combinations of torsion and compression were obtained. Most of the interaction plots were linear in the high torque part of the curve and nonlinear in the high compression region.

During final review of this paper, two additional pertinent papers came to the authors' attention. Marlowe, Sushinsky, and Dexter[3] tested several boron-epoxy and graphite-epoxy shells in pure torsion. They compared their results with an analysis by Wu,[4] and obtained good agreement.

Test Plan

A test program was planned to investigate two of the important variables affecting the buckling interaction. The first variable was material type. Boron-epoxy and graphite-epoxy were chosen to represent typical composite fuselage applications.

The number and direction of plies in the laminate were chosen as the second test variable. The two configurations consisted of a 4-ply $[\pm 45]_s$ laminate and a 6-ply $[0/\pm 45]_s$ laminate, where the cylinder axis defines the 0° direction. These also are typical of composite fuselage designs. The small number of plies used creates the worst case of stacking sequence effects obtainable for these types of laminates. Since the laminates are both symmetric about the midthickness to prevent bending-stretching coupling, the ply stacking causes a significant bending-twisting coupling to arise. This means that the torsional stiffness for positive torques is not the same as that for negative torques. The direction of torque application placing the outermost 45° plies in compression always results in a higher buckling load. The overall cylinder geometry was maintained constant, with the outside diameter 15 in. and the test section length 15 in.

Specimen Fabrication

A requirement was established to manufacture a 15-in. diam, thin-walled, constant-section composite cylinder approximately 18 in. long with the smooth or tool surface to the outside. In other composite research programs, small

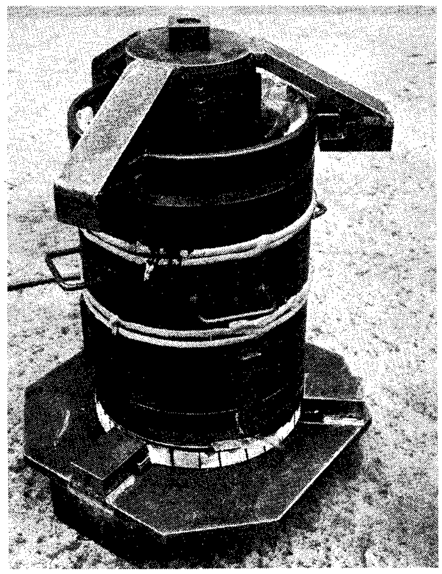

Fig. 1 Large diameter cylinder mold.

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Index categories: Structural Composite Materials, including Coatings); Aircraft Structural Materials; Aircraft Structural Design (including Loads).

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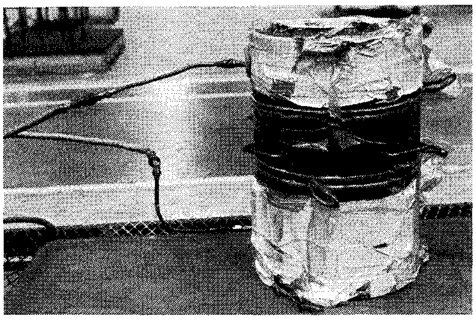

Fig. 2 Mold ready for autoclave.

diameter tubes have been made using an expandable mandrel and a split female tool, but this is expensive. Considering the previous difficulties encountered while building tube test specimens, the method used here was straightforward, inexpensive, and provided excellent quality shells.

A steel tube of the required diameter was cut into two lengths—one 3 in. longer than the specimen and the other about 4 in. long. Three straps were welded to the outer surface of the long section and bolted to the short section. Then, the assembly was bored to the correct diameter, and a band was machined on the outside of each length to provide a reference for realignment in a lathe. In Fig. 1, these details are shown, together with the base plate on which the assembly is mounted, an air cylinder, and the yoke used to separate the two sections.

When a cylinder was fabricated, adhesive was applied to the short section of the mold, release agent was applied to the long section, and the 3-in. prepreg tape was hand-laid inside the cylinder, including about 2 in. of the short section. A vacuum bag was applied to the inside surface. This bag extended over the joint of the two sections, as shown in Fig. 2. Immediately after autoclave cure, and before appreciable cooling took place, the bag was removed from the outside of the mold and the mold was placed over the air cylinder. Because of the difference in thermal expansion of the steel mold and the composite shell, the removal had to be done near the cure temperature. The separation yoke was positioned, and the three strap bolts were removed. Then, air pressure lifted the upper mold section from the composite cylinder, which was held in the lower mold section by the adhesive. This procedure is shown in Figs. 3 and 4. The lower mold section was set up in a lathe for final trimming and cutting. The composite remaining in the short section was then cut out, and the mold was ready for another cycle.

This method of manufacture was used successfully and was economical for making the large-diameter composite cylinders. The properties of the cylinders are shown in Table 1. For testing, the cylinders were potted into grooved, 2-in. thick aluminum end plates.

Cylinders G4-I, B4-I, and B6 were fabricated and tested very early in the program. Subsequently, specimens G4-II, G6, and B4-II were built and tested to complete the series. The cylinder designated G4-I in Table 1 was a poor-quality specimen made from scrap material to try the fabrication approach. It was also used to set up the test and to check the equipment.

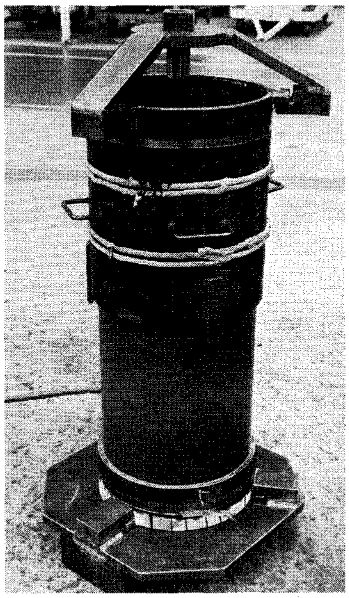

Fig. 3 Mold separation by air cylinder.

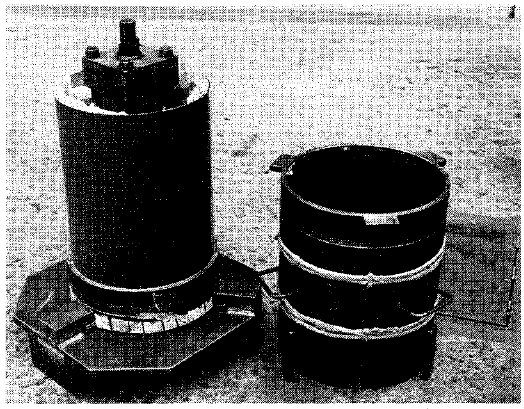

Fig. 4 Molded cylinder retained in lower mold section.

Table 1 Cylinder properties

Cylinder	G4-I	G4-II	G6	B4-I, B4-II	B6
Layup[a]	$[\pm 45]_s$	$[\pm 45]_s$	$[0/\pm 45]_s$	$[\pm 45]_s$	$[0/\pm 45]_s$
Material type	Graphite-epoxy	Graphite-epoxy		Boron-epoxy	
Material	Modmor II/5206	Narmco 5208/T300		Avco 5505	
Ply thickness, mils	6.5	5.6		5.3	
$E_{11}(10^6 \text{psi})$	21.0	21.7		30.0	
$E_{22}(10^6 \text{psi})$	1.7	1.44		2.7	
$G(10^6 \text{psi})$	0.65	0.65		0.65	
ν_{12}	0.28	0.28		0.21	

[a]Subscript s denotes a symmetric laminate.

Test Technique Development

At the outset, the test program was planned around multiple data points from a given specimen. The work of Singer[5] and Tennyson[2] had indicated that much better repeatability could be obtained by using single specimens rather than building several "identical" specimens. Singer repeatedly buckled his conical shell specimens at varying ratios of torsion and external pressure. They buckled gently enough so that relatively little damage occurred. Tennyson made use of an internal mandrel to limit the buckling deflections of his fiberglass cylinder specimens; thus, he obtained torsion-compression interaction results from single specimens.

Previous testing at General Dynamics[6] had shown the utility of the Southwell technique and the Moire grid-shadow method in shell testing. Our original test plan was to use a modified Southwell method that requires strain data as input. Since the Moire method was not practical for a full-cylinder application, an alternate method was thought to be required for finding the location of the first buckle. The alternate method was obtained from Craig,[7] who suggested the use of a

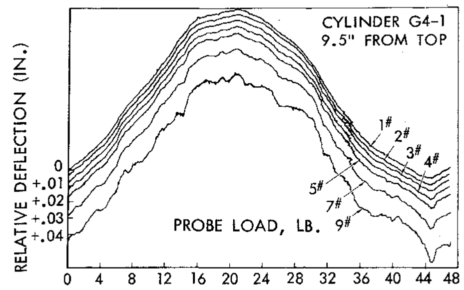

Fig. 5 Deflection vs location for various loads.

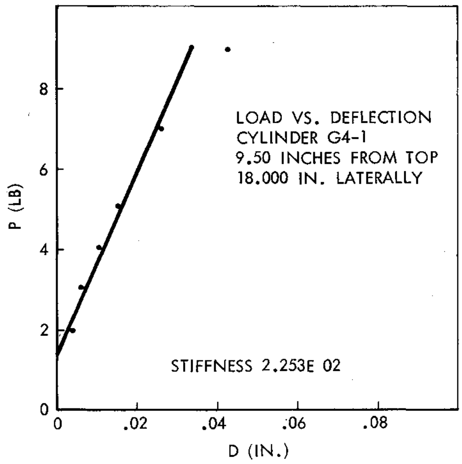

Fig. 6 Load vs deflection at a location.

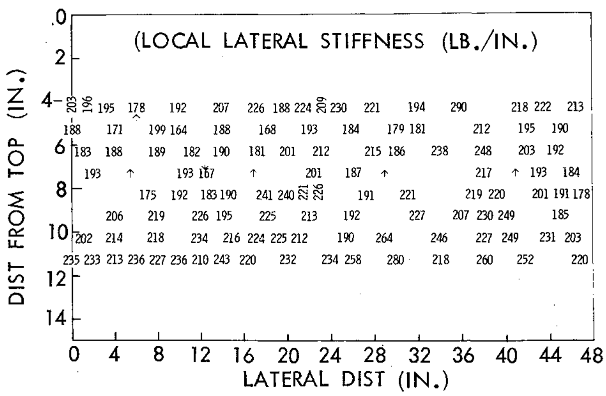

Fig. 7 Stiffness survey, cylinder G4-I.

lateral stiffness survey. The survey was used to define low stiffness areas in a shell where a buckle would be assumed to start.

The equipment used for the stiffness survey included a stiffness probe, which was used to apply normal loads to the shell. The probe was dead loaded through a pulley arrangment, and the resulting cylinder wall displacement was monitored with a linear variable differential transformer. Since the cylinder was end-loaded through spherical balls at the center of each end plate, the cylinder could be rotated under the fixed probe load to take a quick circumferential displacement survey. A thin wire connected a point on the cylinder to a potentiometer and provided an analog measure of circumferential location of the probe. The surveys were taken at 1-in. intervals vertically. A typical probe output is shown in Fig. 5. The raw experimental data were converted to stiffness information very quickly using a Hewlett-Packard 9820A calculator with an attached digitizer and plotter. In one step, the deflection vs circumferential location data were converted to load vs deflection at given locations, a linear regression was performed, and

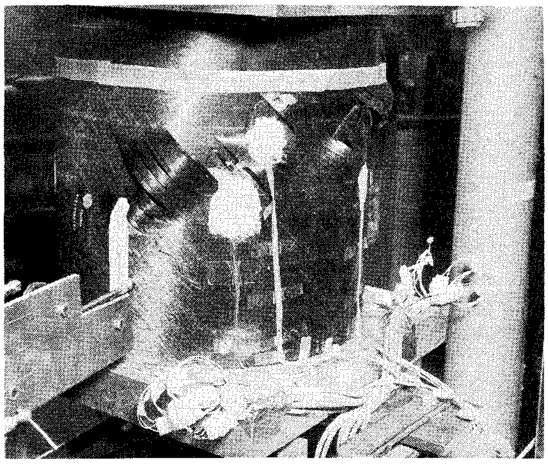
Fig. 8 Failed G4-I cylinder, side view.

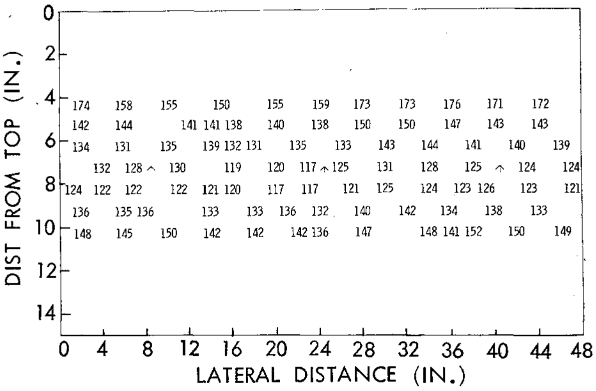

Fig. 9 Stiffness survey, B4 cylinder.

the stiffness was calculated. A typical stiffness plot is shown in Fig. 6. Approximately 100 of these plots were combined to produce the stiffness survey shown in Fig. 7.

A survey of the G4-I cylinder showed two low stiffness areas, one at 4.5 in. down and 6 in. circumferentially and the other at 7.5 in. down and 12.4 in. circumferentially. Strain rosettes were installed at those two locations in addition to the four rosettes already installed at the quarter points on the cylinder midlength.

During the first test of the G4-I cylinder, in pure torsion, strain data were recorded but not plotted as the test progressed. Previous experience indicated that visual observations would indicate the initiation of a slowly forming torsion buckle. That idea was dispelled by the catastrophic snapthrough buckle that occurred. After the disaster, the strain data were plotted and showed the classic hooked shape that indicates that buckling was imminent. The two strain gages at low stiffness locations showed somewhat better sensitivity than those that had been arbitrarily placed in the center of the cylinder, but the arbitrarily placed gages showed clearly that the cylinder could have been saved for further testing, had the strains been plotted during testing. The failed graphite-epoxy cylinder is shown in Fig. 8. Its experimental critical buckling stress τ_{cr} was 3500 psi in the stiff direction of torque.

A stiffness survey of the B4-I cylinder (shown in Fig. 9) was performed using the method described previously. This boron-epoxy cylinder had a much more uniform stiffness distribution than the previous graphite-epoxy one. Three rosettes were arbitrarily placed at 120° intervals at the center of the cylinder. Fourteen points on the torsion-compression interaction curves were obtained by applying a given compression load and then applying the torque incrementally. These tests were conducted by loading the specimen, reading the data, reducing the load, and then plotting the data by

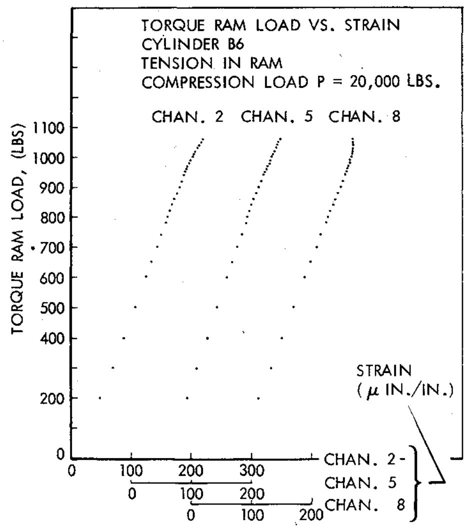

Fig. 10 **Load-strain curve.**

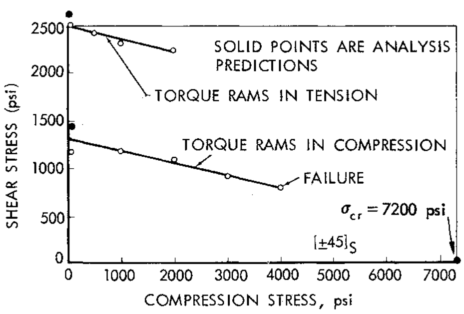

Fig. 11 **Buckling interaction, cylinder G4-II.**

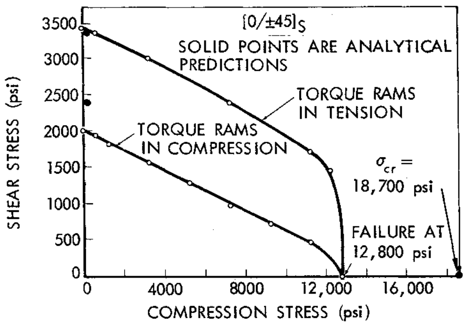

Fig. 12 **Buckling interaction, cylinder G6.**

hand prior to the next load application. The loading was stopped when the hooked shape of the load-strain curve indicated that the next increment of load would cause the specimen to collapse. The results from tests of this cylinder showed that it was not necessary to use strain data from a low stiffness point; it also showed that any of a number of applied strain gages was likely to be in an area where a buckle was active. The cylinder was buckled catastrophically before the interaction curve could be fully developed.

For cylinder B6, no stiffness survey was taken and three equally spaced rosettes were used to obtain strain data. A typical load-strain curve on which the buckling criterion was based is shown in Fig. 10. The strain data were hand-plotted as received from the data acquisition system. Loading was stopped when any of the load-strain curves showed that the next 5- or 10-lb torque load increment would result in a large strain increase.

This method worked very well, although the cylinder was buckled before the pure compression load was reached. For the remaining three specimens (G4-II, G6, and B4-II), the load-strain plots were automated and another rosette was added to the pattern.

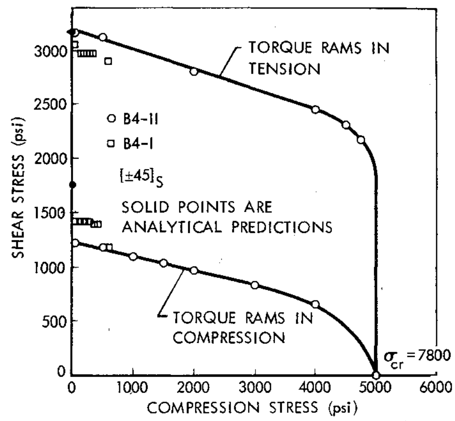

Fig. 13 **Buckling interaction, cylinder B4.**

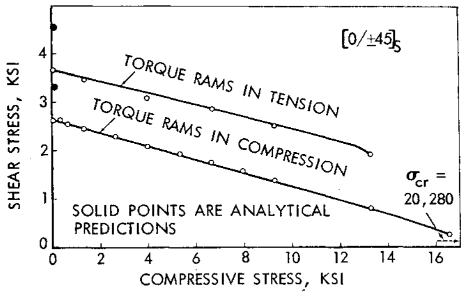

Fig. 14 **Buckling interaction, cylinder B6.**

These three cylinders were each instrumented with four strain gage rosettes equally spaced around the circumference and equidistant from the ends. The time at load and hysteresis effects were reduced by setting up a system to allow direct plotting of the data. This system consisted of d.c. amplifiers plus a manual selector box and an x-y plotter in addition to the existing data acquisition system used for previous tests. All data were plotted as loading progressed and data were occasionally read by the original data acquisition system to check calibration. Subsequent conversion of the load-strain data into Southwell plots was not deemed necessary since the raw data provided a suitable buckling criterion.

Results and Conclusions

The torsion-compression buckling interaction curves for the four cylinder types are shown in Figs. 11-14. Because the shells were designed to buckle, the stresses shown are well below the torsional or compressive strengths for the shells.

Analytical predictions of the classical compressive buckling stress were calculated according to the analysis of Holston, Feldman, and Stang.[1] The fact that the classical analysis is unable to predict the buckling stress accurately is not surprising. The apparent knockdown factor of about 65% could easily be explained by an imperfection sensitivity argument.[6]

The analytical predictions for pure torsion were provided by Tennyson and Booton.[8] (Similar calculations according to Wu's theory[4] provide nearly identical results.) These theories obviously give good agreement with the experiments. Note particularly that the effect of reversed torque is quite adequately predicted. This type of correlation between experiments and a classical linear analysis was to be expected because of previous correlation[9] obtained for laminated curved panels loaded in shear. With these experiments, the stage should be set for generating or verifying an analysis that can accurately predict the torsion-compression performance of these shells up to high percentages of compression load before imperfection sensitivity begins to influence the response.

A most interesting aspect of the test results is the linearity of the interaction exhibited over a wide range of stresses. A

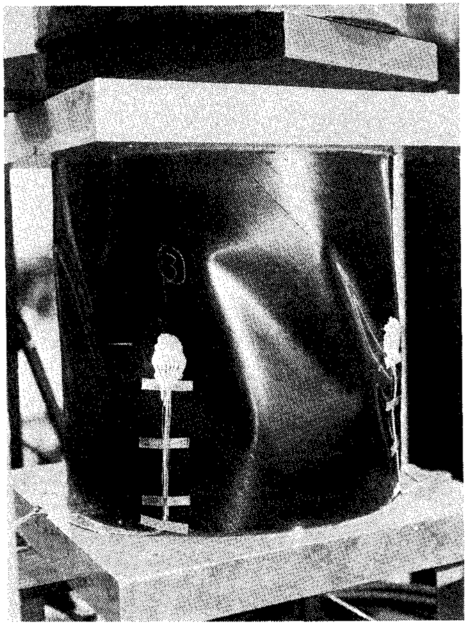

Fig. 15 Buckled G4-II cylinder.

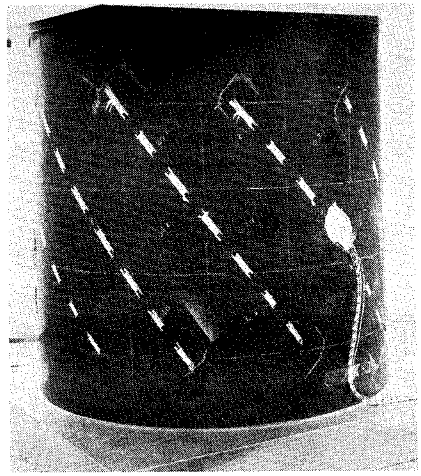

Fig. 16 B4-I cylinder.

calculation of the slopes of all the lines indicates that the torsional buckling stress is reduced by 10-20% of the applied compressive stress within the linear range. This knowledge will prove useful in design, at least for similar materials and laminates.

The buckled G4-II cylinder is shown in Fig. 15; when the load was removed, the cylinder snapped back to its original shape. This specimen was not tested further since the extent of internal damage could not be assessed.

In Fig. 16, the B4-I cylinder is shown after buckling and removal from the fixture. This specimen, which was buckled at a high percentage of the torsional buckling load, was badly damaged. The white lines show where the buckles were located. The broken fibers that could not withstand the buckling deflections can be seen at the ends of each of the white lines.

The B4-II cylinder was successfully tested throughout its interaction range. Subsequently, this cylinder was buckled six times in pure compression. Its buckling load on the sixth time was 95% of its original buckling load with no visible damage.

The program described herein served several important purposes. A novel manufacturing method was developed which could be used again for large-diameter tubes. The program added another nondestructive shell test capability for possible later use and also provided much-needed design information on the buckling interaction of thin-walled advanced composite shells.

References

[1] Holston, A., Feldman, A., and Stang, D. A., "Stability of Filament-Wound Cylinders under Combined Loading," AFFDL-TR-67-55, May, 1967. Air Force Flight Dynamics Lab., Wright-Patterson Air Force Base, Ohio.

[2] Tennyson, R. C., "Buckling of Laminated Composite Cylinders," to appear in *Composites Journal.*

[3] Marlowe, D. E., Sushinsky, G. F., and Dexter, H. B., "Elastic Torsional Buckling of Thin-Walled Composite Cylinders," ASTM STP 546, 1974, American Society for Testing and Materials, Philadelphia, Pa.

[4] Wu, C., *Buckling of Anisotropic Circular Cylindrical Shells,* Ph.D dissertation, Case-Western Reserve University, Cleveland, Ohio, June 1971.

[5] Singer, J., "On Experimental Technique for Interaction Curves of Buckling of Shells," *Experimental Mechanics,* 1964, pp. 279-280.

[6] Wilkins, D. J., "Compression Buckling Tests of Laminated Graphite-Epoxy Curved Panels," *AIAA Journal,* Vol. 13, 1975, pp. 465-470.

[7] Craig, J. I. and Duggan, M. F., "Nondestructive Shell Stability Estimation by a Combined Loading Technique," *Experimental Mechanics,* Vol. 13, Sept. 1973, p. 381.

[8] Tennyson, R. C. and Booton, M., private communication, 1974.

[9] Wilkins, D. J. and Olson, F., "Shear Buckling of Advanced Composite Curved Panels," *Experimental Mechanics,* Vol. 14, Aug. 1974, pp. 326-330.