Fabrication and Full-Scale Structural Evaluation of Glass-Fabric Reinforced Plastic Shells

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This paper reports the initial investigations on full-size sandwich shell structures consisting of cylindrically curved panels and full cylinders, although some previous work in fabrication and basic buckling tests on small flat panels is presented as background information. The large thin-walled sandwich shells were fabricated by bonding 2-ply epoxy-fiber glass laminate facings to aluminum honeycomb cores. The curved panels were tested in axial compression and in torsional shear. The complete cylinders were loaded in a special torsion-bending testing frame capable of applying bending moments and torques of 3,000,000 in.-lb. each. In general, the buckling test results compared as well with calculated values as do those for buckling of homogeneous isotropic shells.

1. Introduction

THE use of glass-reinforced plastic (GRP) in aircraft is not new[1] although it has not been used to any appreciable extent as material for primary load-carrying members except on an experimental basis.[2] Ease of repair makes this material attractive for primary load-carrying members of limited-warfare aircraft. However, its inherently low stiffness compared to that of most metals requires that for effective use, it must be incorporated in the form of sandwich construction. Sandwich construction, even with GRP facings, is not new.[1,2] However, most of the applications have been radomes and other nonprimary structures. Thus, only limited experimental data are available on the failure characteristics of GRP-facing sandwich shell structures, especially in the buckling mode. Such data are necessary for successful design of primary structures of this type, since, as mentioned by Professor Hoff,[3] the effect of imperfections always present in any practical structure is not known for this type of structure. There is some basis for expecting that the effect would be less than for monolithic shells, and thus less scatter in buckling stresses would be expected.

The Forest Products Lab.(FPL) has conducted the most extensive tests (compression and shear) on curved sandwich panels, including facings of plywood, aluminum alloy, and GRP and cores of various materials (but not honeycomb).[4,5] The most extensive experiments on buckling of sandwich-type, complete cylindrical shells under loadings typical of aircraft fuselages were conducted by Gerard.[6,7] His cylinders were quite different in that they 1) were subscale in size, 2) had cores which were impractical due to their weakness in shear (foam plastic and balsa wood), and 3) had facings of aluminum alloy that is isotropic, of high modulus, and of sufficient ductility to permit extensive plastic deformation prior to final failure.

More recent tests on cylindrical shells include those reported by North[8] on 20-in.-diam cylinders with GRP facings and hexagonal-cell honeycomb (HCH) cores and by Peterson and Anderson[9] on 77-in.-diam cylinders with aluminum-alloy facings and HCH cores. The tests reported here include cylindrically curved sandwich panels as large as 36 in. square and loaded in axial compression and shear, and sandwich-type complete cylindrical shells (44 in. diam × 72 in. long) loaded in bending and torsion.

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2. Materials

The shell specimens were fabricated from fiber glass-reinforced plastic laminate facings separately bonded to an aluminum honeycomb core (see Ref. 10 for details). Two significant advances which influenced the program are 1) development of a resin-impregnating machine that minimizes air voids and produces a multilayer prepreg of more uniform resin content and 2) empirical determination of the relationships between important process variables and laminate strength properties.

The multilayer prepreg machine (Fig. 1) used is the one described in Ref. 11, but modified to handle fabric up to 56 in. wide. The machine has produced 2-ply prepreg of Epon 828-Z epoxy resin and 181-style fiber glass fabric in lengths greater than 700 in. while maintaining the previously achieved uniformity of ±1% resin content by weight.

Table 1 lists the properties of the laminate used as the shell facing material. The sampling from the autoclave cured laminates, although not large, was sufficient to establish the properties needed in the shell analyses. Test coupons were obtained from flat laminates fabricated identically to the curved facing laminates. Curing was accomplished at 160°F

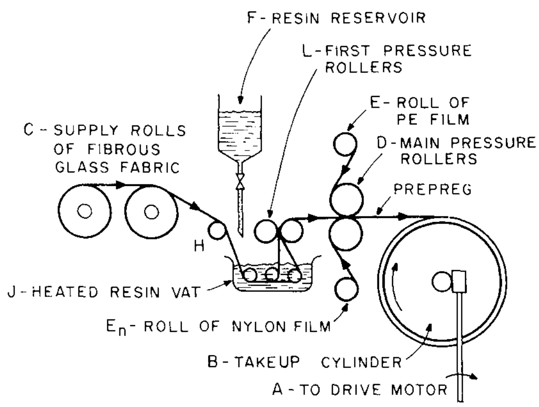

Fig. 1 Multilayer prepreg machine.

Table 1 Shell constituent properties

Designation	Facing[a] Resin content,[b] %	Facing[a] Thickness,[b] in.	Moduli, psi × 10^{-6} E_x Initial	E_x Secondary	E_y Initial	E_y Secondary	G_{xy} Secondary	G_{xy} Tertiary
A	35.1	0.022	4.14	3.23	3.96	3.09	...	...
B	34.6	0.021	4.19	3.28	4.01	3.14	0.562[d]	0.416[d]
	Core[c]		G_{xz}		G_{yz}		E_z	
C	...	0.30	0.0320		0.0183		0.052[e]	
D	...	0.30	0.0306		0.0179		0.052[e]	

[a] Two-ply, E-glass, 181-style fabric, 828-Z epoxy. All laminates were parallel plied with the warp oriented in the longitudinal direction of the shell. Facing moduli at each of the two stress levels were obtained from curves such as Fig. 2.
[b] Average values obtained from each of the laminate types tested. Poisson's ratio ν obtained from Ref. 15 ($\simeq 0.13$).
[c] Nonperforated 5052 aluminum honeycomb core, $\frac{1}{4}$-in. cell, 1-mil foil.
[d] Values obtained from torsional loading of shell.
[e] Values supplied by the core manufacturer.

laminate temperature for 100 min at a pressure of 70 psi with a zero-pressure precure of 23 min. Postcure usually took place during the bonding cycle. This cure closely approximates that used in an electrically heated press at which optimum surface finish and laminate uniformity were obtained previously.[12]

Of the major process variables, resin content seems to be the most usable one for relating laminate fabrication to the final strength properties. Numerous test data showing the effect of resin content on strength properties were determined previously for 3-ply laminates.[12,13] Figures 2 and 3 show some of the compressive property data along with limited test data for 2-ply laminates. In comparing the 2- and 3-ply laminate data, the usual thickness effect on laminate strength is evident. The modulus data suggest the same trend in relation to resin content, but the thickness effect is less pronounced than for strength. These newly generated 2-ply data together with the average resin contents for the shell facings were used to determine the properties listed in Table 1 along with core properties.

3. Tooling

Tooling (Fig. 4) for the shells consisted of 1) a main plaster mold, 2) a concave plaster mold, and 3) a GRP working tool. The main mold was splash cast on a steel framework and swept to the correct diameter (see Figure 5). The master contour was reproduced by back-casting from the main mold using an epoxy surface coat to obtain a concave mold (Fig. 6) on which the GRP tool was laid up by hand. The $\frac{3}{8}$-in.-thick tool laminates were fabricated using a high-temperature surface coat, one layer each of 120- and 181-style fiber glass fabric, and approximately 30 layers of 10-oz fiber glass tooling fabric saturated with high-temperature epoxy.

The tool laminates were reinforced with rings and longerons fabricated by wrapping $\frac{3}{4}$-in.-thick plywood forms with tooling fabric and resin to about $\frac{3}{8}$-in. thickness. A Teflon spray-on parting agent insured separation of each half of the tool from the concave mold. The halves were pinned and bolted together with thin aluminum wedges to permit tool collapse for shell removal (Fig. 4). Finally, the completed tool was heat treated to the maximum anticipated bonding temperature before being used.

Zinc chromate tape was used to seal the wedges, the pressure blanket, and the vent tubes at the ends. The vent tubes were of felt-padded copper tubing with drilled vent holes.

4. Fabrication

Fabrication of the cylinders on the working tool was accomplished in three phases: facing lamination, core rolling, and sandwich bonding (see Ref. 10 for details). Both the inside and the outside facings were cured on the same tool. This was possible because the facings were so flexible that an inside facing could be warped into an outside facing contour with no adverse effects.

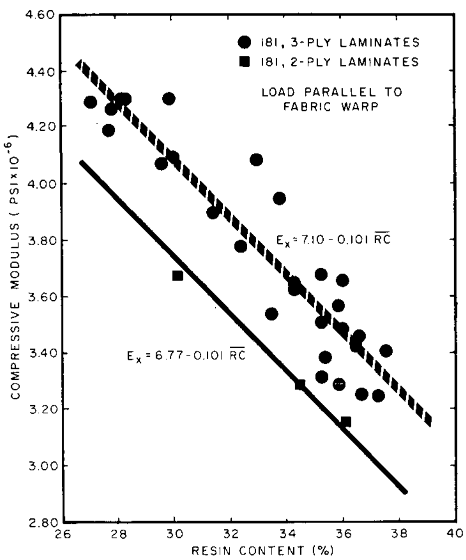

Fig. 2 Effect of resin content on laminate compressive modulus.

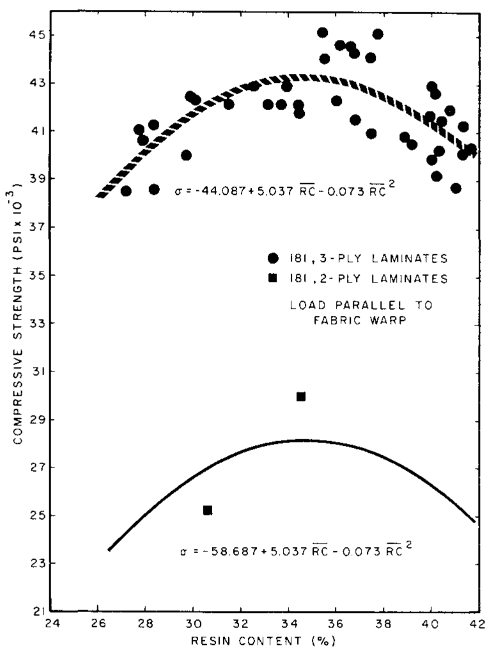

Fig. 3 Effect of resin content on laminate compressive strength.

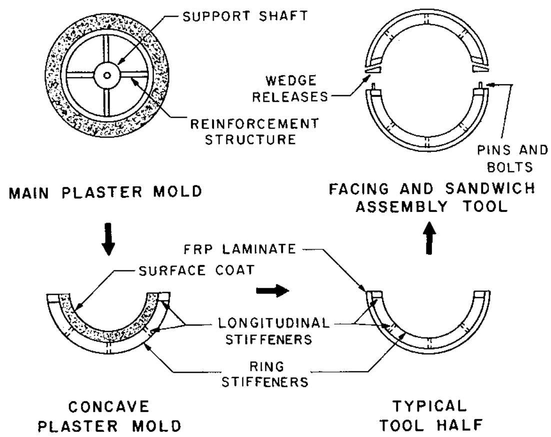

Fig. 4 Schematic of tooling components for sandwich shells.

Facing Lamination

A dry layer of 181-style fabric was used as a bleeder to improve resin-content uniformity and to insure uniform venting during vacuum application. $\frac{1}{2}$-mil-thick Teflon (FEP) perforated film was used as a parting agent between the bleeder and prepreg.

Another innovation was prepreg curing with one cover film (Fig. 1) left in place. The film, 3-mil-thick heat-stabilized nylon, provided support for handling the prepreg and prevented glass-fiber misalinement.

Layup procedure for the shells consisted of butting precut pieces of prepreg together, and connecting them by 4-in.-wide single-shear lap splices of prepreg. The perforated Teflon film was placed on the sticky prepreg which then had B-staged about 7 hr. For an inside facing, the prepreg was wrapped onto the tool with the cover film toward the tool, using the bleeder wrapped under tension. For an outside facing, the prepreg was wound over the bleeder with the Teflon side against the tool and a closely woven fabric was used to apply hold-down force.

A longitudinal bar clamp with a sponge-rubber face held the bleeder, prepreg, tension wrap material, and vacuum blanket at one end during wrapping. Another such clamp was used to hold the tension while the closing prepreg joint was made and the vacuum blanket was sealed (Fig. 7). The outside facing layup was closed by overlapping the prepreg and preventing laminate bonding during cure. A 10-psi vacuum was applied, the clamps were removed, and the layups were permitted to set under the vacuum at room temperature for a 10-hr B-stage. Layup of the curved panels was similar but much simpler.

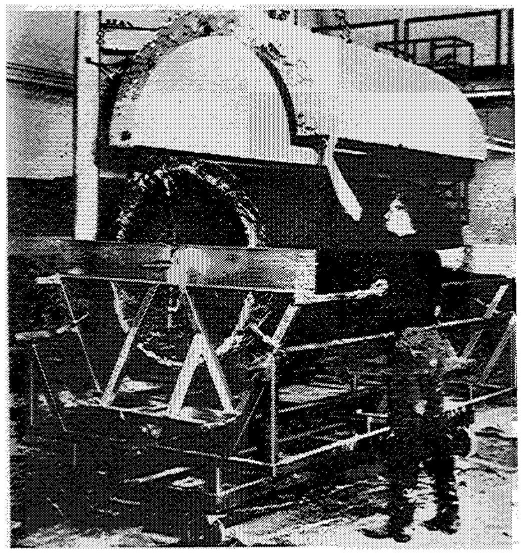

Fig. 6 Parting of concave and main plaster molds.

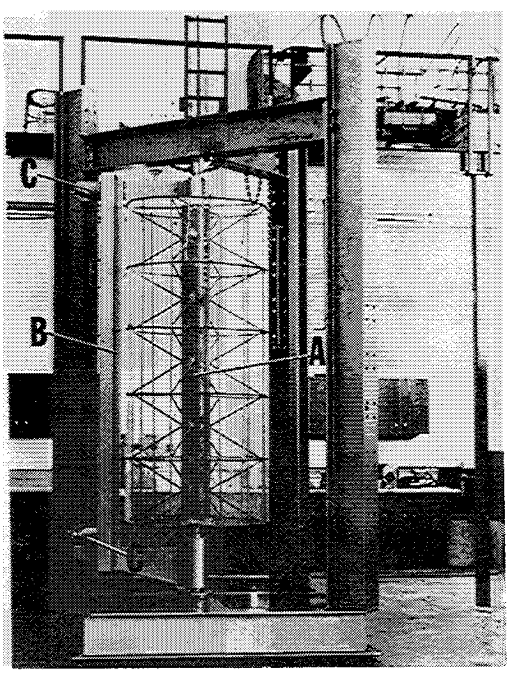

Fig. 5 Plaster sweeping device: A, plaster support framework; B, sweep and screed; C, precision slides for radius adjustment.

Each layup was rolled into the preheated autoclave,[10] suspended in the same dolly as used for making the layup. After curing, cooling, and removing from the autoclave the facing laminates were stripped of bleeder and films and the bonding surface was lightly sanded. The outside facing was removed from the tool and stored; the inside facing was kept on the tool for buildup of the sandwich.

Core Rolling

Four slabs of honeycomb core were used for each cylinder and two slabs for each curved panel. The core slabs were formed with single curvature (no anticlastic curvature) by rolling between protective sheets, in a sheet-metal roll-forming machine. The fully expanded core was sandwiched between sheets of heavy Kraft paper and thin sheets of soft aluminum.

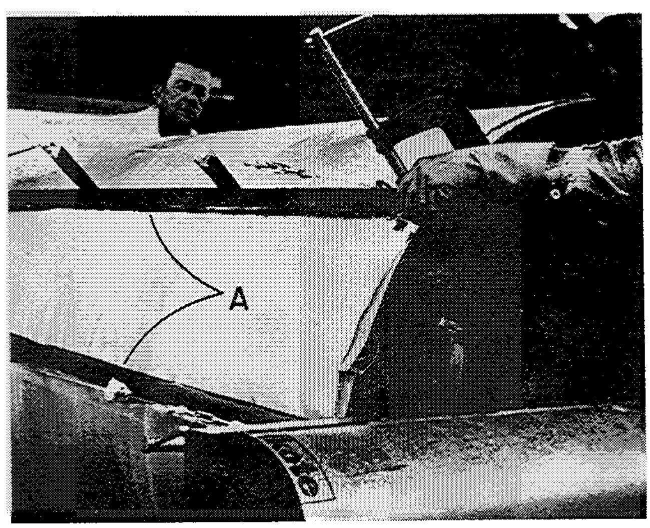

Fig. 7 Layup of cylinder facing; A, clamps used in tension wrap.

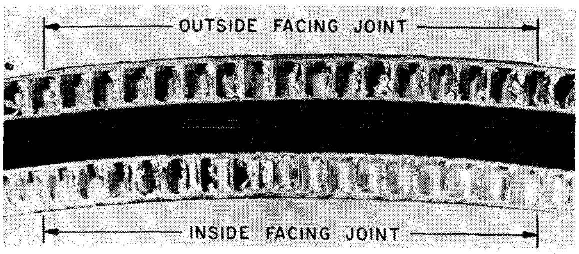

Fig. 8 Cross-sectional pieces of cylinder sandwich showing laminated facing joints.

Sandwich Bonding

A 3-ft.-wide, film-supported adhesive (AF-110B) was wrapped onto the inner facing while it was still on the tool. Then the core slabs were pressed into this adhesive layer by hand and connected using bonded compressive butt splices. The outer adhesive layer was wrapped over the core, and the outside facing was installed so that the three joints of each facing were staggered symmetrically. The final cylinder joint was made with a 4-in.-wide precured strip and the same adhesive. A layer of fiber glass fabric was wrapped over the sandwich layup to insure uniform vacuum. As the final steps, the vacuum blanket was wrapped on and sealed with zinc chromate tape. The sandwich layup was cured in the preheated autoclave at 320°F average temperature under approximately 10-psi vacuum for a total of 150 min, including 70 min of warm up.

After slow cooling, the wedges were extracted from the tool and the shells were removed while the tool was suspended vertically. Figure 8 shows inside and outside facing splices as they appear in the sandwich construction; the good thickness transition is apparent.

5. Curved-Panel Tests

The curved-panel compressive specimens were reinforced on the loaded ends with an epoxy-ceramic potting compound and, in some cases, with thin, narrow strips of laminate bonded to the facings. The panels were positioned in a hydraulic testing machine so that the centroids of the cross-sectional arcs were on the loading axis. The sides of the panels were simply supported by knife-edge grips, or grooved blocks in the case of the smaller specimens. Side-deflection measurements were made for each specimen tested to determine occurrence of buckling.

The shear panels were bonded in a special gripping device to simulate the type of loading expected in an aircraft panel bordered by spars or stringers. The device is analogous to the picture-frame grip-and-linkage often used for flat panels. Figure 9 shows the system and how it was attached to the torsion-bending testing machine described in Sec. 8.

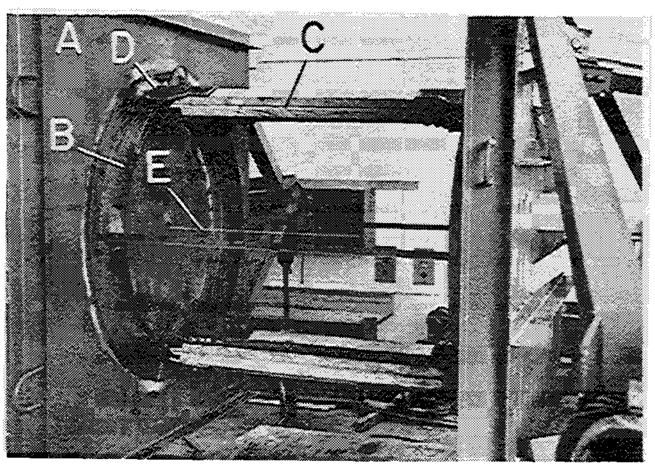

Fig. 9 Gripping device for curved-panel shear: A, torque plate; B, ring end-grip; C, grooved edge-grip; D, tilting pins; E, alinement axis.

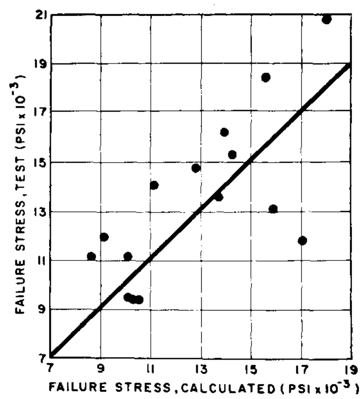

Fig. 10 General buckling of flat sandwich panels with simply supported edges.

6. Discussion of Curved-Panel Results

The structural behavior of GRP-facing sandwich panels was initially studied with small flat panels sized to buckle in a single half-wave. Figure 10 shows test results[14] for flat panels which were about $\frac{1}{8}$ the size of the present curved panels and constructed of similar constituents (0.2-in.-thick cores with $\frac{3}{16}$- or $\frac{3}{8}$-in. cells; 2-, 3-, and 4-ply facings). The analysis of Ref. 15 was conservative for most of these panels.

Comparison of theory with test is not as clear for the curved-panel tests. None of the theoretical developments available to the authors for analysis of compressive buckling in curved sandwich panels take into consideration the orthotropic nature of the GRP facings. Thus, it is necessary to improvise either by using semiempirical analyses or by combining several rigorous analyses.

For curved panels smaller than the theoretical buckle size of the cylinder, FPL recommends[3] using the complete-cylinder postbuckling analysis of Ref. 16 and then adding empirically the stress carried by the panel considered as a flat plate.[15] The ratio of the experimental results to those predicted by this method are shown in Table 2, column 5. This approach has been questioned recently because it does not seem reasonable to use a large-deflection theory when it is observed that sandwich cylinders in the practical range of values of $R/h < 100$, where R is the nominal cylindrical radius and h is the distance between the centerlines of the facings, do not undergo large deflections prior to failure. This is especially true for the present tests because the GRP material fractures instead of undergoing large deflections.

Another approach which appears to be more logical is to adapt the small-deflection, isotropic, sandwich-panel buckling analysis of Stein and Mayers[17] to the case of orthotropic cores and orthotropic facings. This adaptation is carried out using some results of a recent analysis by Dow and Rosen,[18] which was an extension of the Ref. 17 complete-cylinder analysis to the case of orthotropic facings but with core transverse-shear deformation effects neglected completely. The following effective isotropic moduli were substituted for the corresponding isotropic ones:

for the core: $$G_c = (G_{xz}G_{yz})^{1/2} \tag{1}$$

for the facings: $$E_f = (E_xE_y)^{1/2} \tag{2}$$

where G_{xz} and G_{yz} are the actual core transverse-shear moduli in the axial and hoop directions, and E_x and E_y are the Young's moduli for the axial and hoop directions. These substitutions were suggested directly by the nature of the presence of

Table 2 Results of cylindrical-panel tests and comparison with calculations

Panel no.	Facing and core designation[a]	Square panel size, in.	Compression loading: Experimental buckling stress, psi	Ratio of exp. to calc. buckling stress: Large-defl. anal.[c]	Ratio of exp. to calc. buckling stress: Small-defl. anal.[d]
1	A,D	36	26,600	1.03	0.90
1A[b]	A,D	34	20,000	0.76	0.66
2	A,D	36	23,700	0.92	0.80
3	B,C	36	11,600	0.60	0.34
5A[b]	B,C	34	17,800	0.91	0.62

Panel no.	Facing and core designation[a]	Square panel size, in.	Shear loading: Experimental buckling stress, psi	and angle (°)	Calculated buckling stress, psi	and angle, γ(°)[c]
4	B,C	49	5,400	49	16,300	27
5[b]	B,C	49	2,600	...	16,300	27

[a] See Table 1. All facings were parallel plied with the warp in the longitudinal direction on the shell. Average central radii of panels are 21.99 and 21.94 for Types A–D and B–C, respectively.
[b] These panels loaded previously as numbers indicate. Panels 4 and 5 loaded simultaneously on first test.
[c] Sum of values from complete-cylinder analysis programmed for computer from equations of Ref. 16 and flat-plate analysis of Ref. 15. References 29 and 30 used for shear-loading case.
[d] Analysis of Ref. 17, corrected for facing orthotropy as described in text.

E_x and E_y in Ref. 18, i.e., square root of the product rather than simple arithmetic mean. The effect of the facing shear modulus G_{xy} is more subtle. It primarily affects a factor ϕ which is given by Eq. (26) in Ref. 18, except that if the calculated value exceeds unity, a value of unity is used for ϕ. This effect is used to obtain a correction factor to apply to the Ref.17 isotropic analysis. If the facings were isotropic, the following familiar relation of isotropic elasticity theory would be valid:

$$G_i = E_f/[2(1 + \nu)] \tag{3}$$

where ν is the Poisson's ratio of the facing material. In Ref. 18, the factor ϕ would be the same for the isotropic material as for the orthotropic one, except that it would be proportional to the square root of the appropriate shear modulus. Thus, provided that $\phi \leqq 1$, the correction factor C_0 is given by:

$$C_0 = (G_{xy}/G_i)^{1/2} \tag{4}$$

Although a 181-style fiber glass-cloth laminate facing is nearly isotropic in regard to its normal-stress stiffness, it has a very low in-plane shear stiffness compared to an isotropic material. In fact, the ratio G_{xy}/G_i did not exceed unity; thus, use of Eq. (4) is valid. The ratio of the experimental buckling stresses to those calculated by this second approach is presented in column 6 at the top of Table 2.

Since the effect of the core shear deformation was quite small in these specimens, the alternative procedure of using the small-deflection analysis of Ref. 18 directly results in buckling stresses only very slightly higher than the procedure described in the preceding paragraph. Thus, the corresponding ratios would be only slightly lower than those in column 6 at the top of Table 2.

In comparing columns 5 and 6 at the top of Table 2, it is seen that the ratios for the small-deflection analysis are lower than the ratios for the large-deflection analysis, which is reasonable conceptually. However, the large-deflection ratios in every case but one are smaller than unity, even before the inclusion of the flat-plate stress; this seems unusual. The small-deflection ratios are all less than unity; this is reasonable.

The considerations mentioned in the preceding paragraph suggest that the small-deflection analysis is an upper bound, since the calculated values in all cases were higher than the experimental ones. Even though some of the facings contained small fabricational imperfections, no failures occurred near them. Nevertheless, in only one case was the experimental value higher than the large-deflection-analysis prediction, which is supposedly a lower bound.

In the one exception in which the experimental value was exceptionally low, failure occurred at a localized dimple which formed just before failure (Fig. 11). This suggests that failure may have been due to face wrinkling. However, small sandwich compression specimens, 4 in. square, cut from the panels failed in face wrinkling at an average stress of 35,000 psi.

The torsional shear panels were analyzed by the semiempirical method suggested by Kuenzi[5] in which the buckling stress for a complete cylinder[29] is added to that for the same size flat panel.[30] An original estimate of 8700 psi (buckle angle $\gamma = 30°$) was obtained using secondary properties of the facings, those at a stress level above 11,000 psi, and this value compared reasonably with the first test value. However, a second calculation using facing properties corresponding to that failure stress gave a much higher value (column 6, bottom of Table 2).

This large difference between theory and test suggests a shortcoming in the analysis or the possibility of extraneous loading at the longitudinal boundary during the test. Figure 12 shows the buckle that occurred on the first shear panel (No. 4). Buckle formation started on the second panel, but did not completely develop before the panel failed by tearing at a loading ring.

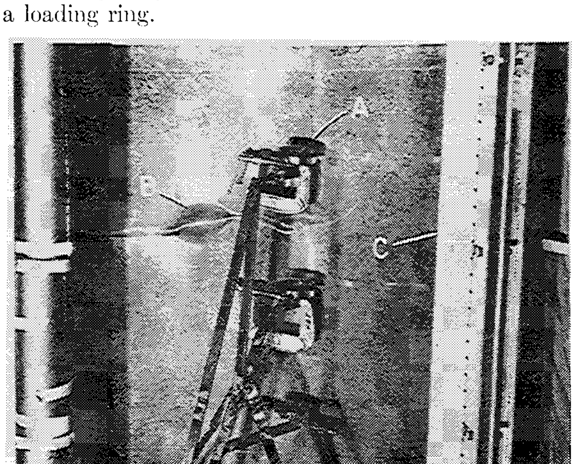

Fig. 11 Compressive buckling failure of thin curved panels: A, dial gage; B, buckle after panel collapse (panel 3); C, side grip.

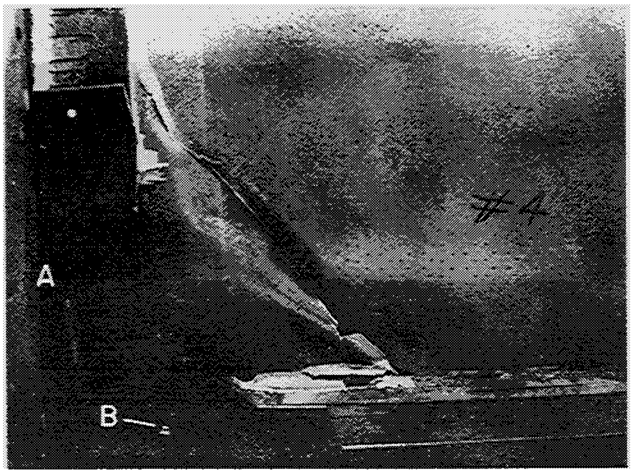

Fig. 12 Curved-panel shear failure: A, torque plate and ring end-grip; B, tilting pin.

7. Design of Cylinders

Design of sandwich shells for buckling tests is almost the converse of design of a successful sandwich shell structure. This is because in designing a successful sandwich structure, one wants to prevent general buckling from occurring before the facing and core materials fail locally, whereas to assure a "successful" buckling test, one wants to be certain that general buckling does not occur before local failure of the constituent materials. Thus, in sandwich structural design, the lower bound of the buckling stress is most important, whereas in setting up a buckling test program, the upper bound is most important. Here the term bound is understood to pertain mainly to the scatter of experimental values from theory (as much as $\pm 33\%$ in monolithic shells).

The basic criteria used in designing the shells are:

1) To assure buckling, the ratio of the estimated buckling stress to the strength of the materials should not exceed 75%.

2) There should be a limit on shell size, to minimize the cost of tooling and autoclave curing equipment. An effective shell length of 72 in. and a nominal diameter of 44 in. were chosen.

3) The facing and core thicknesses and as many other geometrical parameters as possible should be as typical of potential army aircraft applications as is consistent with the other requirements.

4) All specimens should be identical (actual center radius was 21.94 in.) to reduce manufacturing costs and to permit direct comparison of data for the various loading conditions.

In view of conflicts in the requirements, many design compromises had to be made. Preliminary calculations showed that to meet requirement 1 within the constraints of 2, it was necessary to use 2-ply facings, each approximately 0.02 in. thick (rather than 3-ply which would be more typical of good design), and a core with a thickness of 0.3 and 0.25-in., hexagonal cells of 1-mil-thick foil (same constituents as curved panels 3, 4, and 5).

8. Cylinder Tests

The cylinders were mounted in the University's 3-million in.-lb. torsion-bending testing machine via 4-in.-deep concentric steel rings (B in Fig. 13). The specimens were fitted and alined in the machine and then tack bonded. The mounting rings were subsequently connected by steel columns to maintain alinement, whereas the assembly was removed for final bonding. After sealing the backs of the rings with tape, the specimens were cast in place using 828 epoxy and a polyamide curing agent with a ceramic thickener.

As in the curved-panel shear tests (Sec. 5), torque was applied by means of a hydraulic jack and a hand-operated pump (Fig. 13). Pure bending moment was applied by the cable and hydraulic jack system also shown in Fig. 13. Rollers prevented any significant axial load from being applied to the specimens.

9. Discussion of Cylinder Results for Bending Loading

Calculation of buckling due to pure bending moment in sandwich cylindrical shells with orthotropic facings and orthotropic core is severely limited because such an analysis does not exist on a rigorous basis. In fact, there are very few analyses for this loading even in the isotropic case. Because of the paucity of appropriate analyses that are directly applicable, it is usual practice to relate the maximum fiber stress at which buckling occurs in a cylinder subject to pure bending $(\sigma_{cr})_b$ to that of an identical cylinder subject to axial compression $(\sigma_{cr})_a$. This raises the question as to what is the correct relationship of the buckling stresses for the two kinds of loading.

Early tests by Donnell and others (see Ref. 19 for discussion) indicated that for homogeneous, isotropic cylinders $(\sigma_{cr})_b \simeq 1.4\ (\sigma_{cr})_a$. According to small-deflection theory of homogeneous, isotropic cylinders,[20-22] $(\sigma_{cr})_b \simeq (\sigma_{cr})_a$; thus, the factor 1.4 has been attributed to the nonuniformity in circumferential distribution of axial stress and to large deflections and imperfections. In fact, Yao's analysis[23] indicates that the ratio $(\sigma_{cr})_b/(\sigma_{cr})_a$ increases from a value of 1.00 at radius/thickness (R/t) ratio of 45 to a value of 1.3 at R/t of 200 approximately. However, recently a very careful statistical analysis of tests on more than 300 cylinders[24] indicates that this ratio is essentially equal to unity.

For sandwich-type cylinders, small-deflection analyses[25,26] indicate that just as in the homogeneous, isotropic case, $(\sigma_{cr})_b/(\sigma_{cr})_a \simeq 1$. Experimental results seem to be inconclusive: Gerard[6] found a ratio of approximately 1.32; North[8] obtained 1.99; and Peterson and Anderson[9] achieved 1.00. Thus, apparently more tests are necessary to definitely resolve this question for sandwich cylinders.

The situation regarding calculation of buckling for complete sandwich cylinders in axial compression is similar to that discussed in Sec. 6 for curved panels, except that here there are more theories available.

Use of the large-deflection theory of Ref. 16 gives a value of 18,300 psi for $(\sigma_{cr})_a$. However, as discussed previously, this theory may not be valid here because large deflections do not take place physically and the assumed form of the buckle waveform shape does not permit the equations to reduce to the correct form for small deflections.

All of the small-deflection analyses available at present are for the case of isotropic facings and give essentially the same results. The infinite-length analyses of Refs. 9, 27, and 28 all

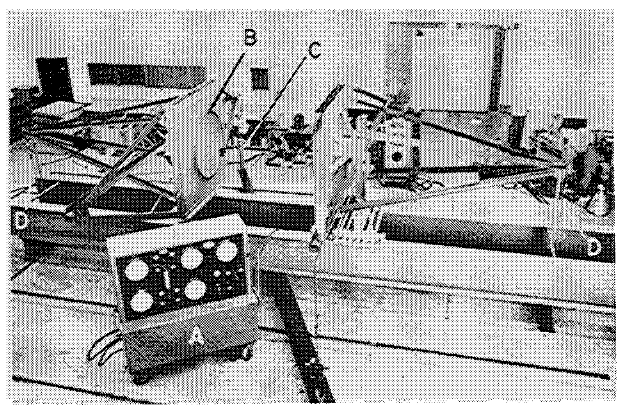

Fig. 13 Torsion-bending testing machine: A, hydraulic control panel; B, ring end-grip; C, torsion-arm loading jack; D, bending-arm cables connecting with common jack.

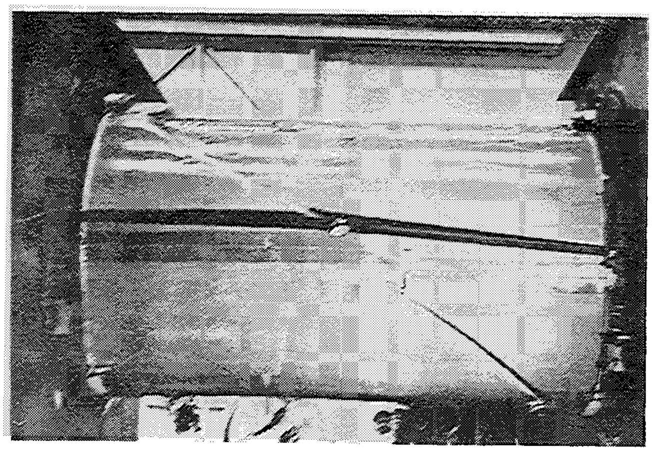

Fig. 14 Torsional buckling in progress.

give a buckling stress of 45,300 psi. The finite-length analysis of Stein and Mayers[17] yields a value of 40,100 psi.

To incorporate orthotropic facings, in particular the effect of the low in-plane shear stiffness, the correction factor of 0.545, derived in Sec. 6 on the basis of Ref. 18, can be applied to the values listed in the preceding paragraph with the following results, respectively: 24,700 psi for the infinite-length analyses; 21,900 psi for the finite-length analysis. The experimentally measured value of 14,100 psi is even lower than the large-deflection prediction (18,300 psi); this was probably due to specimen imperfections, since this was the first shell fabricated. The particular mode of buckling could not be identified in the present test, as the failure was explosive in nature. A new theory[31] considers face wrinkling as well as general instability and thus would give more accurate predictions if it were modified to consider orthotropic facings.

10. Discussion of Cylinder Results for Torsional Loading

The small-deflection theory of Ref. 29 was used for analysis of the sandwich cylinder loaded in torsion. As explained in Sec. 6 for the curved-panel shear case, an initial estimate of the buckling stress was made using the secondary values of the facing elastic properties. This calculated facing stress of 5390 psi was lower than the test value of 8560 psi; hence, in an effort to evaluate the theory more thoroughly, a calculation was made using facing tangent moduli that actually existed at the failure stress. This calculation gave a buckling stress of 10,700 psi.

Observations indicate that the failure of the specimen may have started near a poorly bonded spot on the outside facing; this suggests that the test value may be slightly low. Should this be the case, the gap between theory and test would be reduced. This would lend support to the hypothesis that a linear theory gives an accurate prediction of the buckling stress, provided the material properties at the failure stress are used. Of course, an iteration procedure would have to be used to apply this to design.

Figure 14 is an action shot of the specimen collapse showing the huge buckle that developed across the cylinder. The buckle angle was 29° measured from the cylinder axis. Calculated values of this angle corresponding to stress levels of 5390 and 10,700 psi were 30° and 23°, respectively.

11. Concluding Remarks

This paper has described some unique fabricational aspects of sandwich cylindrical panels and shells with GRP facings, and the results of buckling tests which agreed reasonably well with existing theories or simple modifications of them. The next steps in the over-all research program, of which the work reported here is a part, are buckling tests of cylindrical shells under combined bending and torsion and buckling testing of conical shells.

References

[1] "Application of Glass Fiber Laminates in Aircraft," Advisory Circular AC 20-21, 1964, Federal Aviation Agency, Washington, D.C.

[2] Hackman, L. E., Stoller, C. L., and Worthington, D. G., "Development Program for a Structural Fiber Glass Aircraft Component," NAEC-ASL-1097, June 1966, U.S. Naval Air Engineering Center, Philadelphia, Pa.

[3] Hoff, N. J., "Thin Shells in Aerospace Structures," (Fourth von Kármán lecture), *Astronautics & Aeronautics*, Vol. 5, No. 2, Feb. 1967, pp. 26–45.

[4] Kuenzi, E. W., "Design Criteria for Long Curved Panels of Sandwich Construction in Axial Compression," Rept. 1558, Dec. 1946, Forest Products Lab., Madison, Wis.

[5] Kuenzi, E. W., "Stability of a Few Curved Panels Subjected to Shear," Rept. 1571, May 1947, Forest Products Lab., Madison, Wis.

[6] Gerard, G., "Compressive and Torsional Instability of Sandwich Cylinders," *Symposium on Structural Sandwich Construction*, American Society for Testing and Materials, Special Technical Publication No. 118, June 1951.

[7] Gerard, G., "Bending Tests of Thin-Walled Sandwich Cylinders," *Journal of the Aeronautical Sciences*, Vol. 20, 1953, pp. 639–641.

[8] North, L. O., "Design Data on High Temperature Resistant Reinforced Plastics & the Design, Fabrication, and Evaluation of a Reinforced Plastic Missile Component," WADC TR 56-355, Part I, Nov. 1956, Wright Air Development Center, Wright-Patterson Air Force Base, Ohio.

[9] Peterson, J. P. and Anderson, J. K., "Structural Behavior and Buckling Strength of Honeycomb Sandwich Cylinders Subjected to Bending," TN D-2926, Aug. 1965, NASA Langley Research Center, Hampton, Va.

[10] Nordby, G. M., Crisman, W. C., and Bert, C. W., "Fabrication and Full-Scale Structural Evaluation of Sandwich Shells of Revolution Composed of Fiber Glass Reinforced Plastic Facings and Honeycomb Cores," USAAVLABS TR 67-65, 1967, U.S. Army Aviation Materiel Lab., Ft. Eustis, Va.

[11] Crisman, W. C., Nordby, G. M., and Bert, C. W., "Automated Multilayer Prepreg Production," *Modern Plastics*, Vol. 42, No. 12, Aug. 1965, pp. 118, 121.

[12] Nordby, G. M., Crisman, W. C., and Bert, C. W., "The Effect of Resin Content and Voids on the Strength of Fiberglass-Reinforced Plastics for Airframe Use," USAAVLABS TR 65-66, Nov. 1965, U.S. Army Aviation Materiel Lab., Ft. Eustis, Va.

[13] Nordby, G. M. and Crisman, W. C., "Fabricating Laminates for Specific Properties," *Modern Plastics*, Vol. 42, No. 2, Oct. 1966, pp. 135ff.

[14] Nordby, G. M. and Crisman, W. C., "Strength Properties and Relationships Associated with Various Types of Fiberglass-Reinforced Facing Sandwich Structure," USAAVLABS TR 65-15, Aug. 1965, U.S. Army Aviation Materiel Lab., Ft. Eustis, Va.

[15] Norris, C. B., "Compressive Buckling Curves for Simply Supported Sandwich Panels with Glass-Fabric-Laminate Facings and Honeycomb Cores," Rept. 1867, Dec. 1958, Forest Products Lab., Madison, Wis.

[16] March, H. W. and Kuenzi, E. W., "Buckling of Cylinders of Sandwich Construction in Axial Compression," Rept. 1830, June 1952; revised Dec. 1957, Forest Products Lab., Madison, Wis.

[17] Stein, M. and Mayers, J., "Compressive Buckling of Simply Supported Curved Plates and Cylinders of Sandwich Construction," TN 2601, Jan. 1952, NACA Langley Aeronautical Lab., Hampton, Va.

[18] Dow, N. F. and Rosen, B. W., "Structural Efficiency of Orthotropic Cylindrical Shells Subjected to Axial Compression," *AIAA Journal*, Vol. 4, No. 3, March 1966, pp. 481–485.

[19] Timoshenko, S. P. and Gere, J. M., *Theory of Elastic Stability*, 2nd ed., McGraw-Hill, New York, 1961, pp. 484–485.

[20] Abir, D. and Nardo, S. V., "Thermal Buckling of Circular Cylindrical Shells under Circumferential Temperature Gradients," *Journal of Aerospace Sciences*, Vol. 26, 1959, pp. 803–808.

[21] Bijlaard, P. P. and Gallagher, R. H., "Elastic Instability of a Cylindrical Shell under Arbitrary Circumferential Variation of Axial Stress," *Journal of Aerospace Sciences*, Vol. 27, 1960, pp. 854–859.

[22] Seide, P. and Weingarten, V. I., "On the Buckling of Circular Cylindrical Shells under Pure Bending," *Journal of Applied Mechanics*, Vol. 28, 1961, pp. 112–116.

[23] Yao, J. C., "Large-Deflection Analysis of Buckling of Cylinder under Bending," *Journal of Applied Mechanics*, Vol. 29, 1962, pp. 708–714.

[24] Horton, W. H. and Cox, J. W., "The Stability of Thin-Walled Unstiffened Circular Shells under Non-Uniformly Distributed Axial Load," SUDAER 220, Feb. 1965, Stanford Univ., Dept. of Aeronautics & Astronautics, Stanford, Calif.

[25] Wang, C. T. and Sullivan, D. P., "Buckling of Sandwich Cylinders under Bending and Combined Bending and Axial Compression," *Journal of the Aeronautical Sciences*, Vol. 19, 1952, pp. 468–470.

[26] Gellatly, R. A. and Gallagher, R. H., "Sandwich Cylinder Instability under Nonuniform Axial Stress," *AIAA Journal*, Vol. 2, No. 2, Feb. 1964, pp. 398–400.

[27] Zahn, J. J. and Kuenzi, E. W., "Classical Buckling of Cylinders of Sandwich Construction in Axial Compression—Orthotropic Cores," Rept. FPL-018, Nov. 1963, Forest Products Lab., Madison, Wis.

[28] Almroth, B. O., "Buckling of Axially Compressed Sandwich Cylinders," TR 6-62-64-9, July 1964, Lockheed Missiles & Space Co., Sunnyvale, Calif.

[29] March, H. W. and Kuenzi, E. W., "Buckling of Sandwich Cylinders in Torsion," Rept. 1840, June 1953; revised Jan. 1958, Forest Products Lab., Madison, Wis.

[30] Kuenzi, E. W., Ericksen, W. S., and Zahn, J. J., "Shear Stability of Flat Panels of Sandwich Construction," Rept. 1560, May 1947; revised May 1962, Forest Products Lab., Madison, Wis.

[31] Bartelds, G. and Mayers, J., "Unified Theory for the Bending and Buckling of Sandwich Shells—Application to Axially Compressed Cylindrical Shells," SUDAER 287, Nov. 1966, Stanford Univ., Dept. of Aeronautics and Astronautics, Stanford, Calif.; also *AIAA/ASME 8th Structures, Structural Dynamics and Materials Conference*, American Institute of Aeronautics and Astronautics, March 1967, pp. 619–637.