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GENERAL INSTABILITY OF RING-STIFFENED CYLINDRICAL SHELLS SUBJECT TO EXTERNAL HYDROSTATIC PRESSURE

by

Thomas E. Reynolds and William F. Blumenberg

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STRUCTURAL MECHANICS LABORATORY

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ABSTRACT

Hydrostatic pressure tests were conducted on four machined ring-stiffened cylinders of different lengths to investigate the influence of end fixity on the elastic general-instability pressure. Each cylinder was tested nondestructively with several different end conditions. Pressure variations of as much as 65 percent were observed for a particular cylinder.

Other tests were conducted with one cylinder reinforced by internal movable bulkheads whereby buckling pressures were obtained as a function of bulkhead spacing. The resulting experimental curve confirms the existence of discontinuities which occur, according to Kendrick's theory, at the transition between two circumferential buckling modes.

INTRODUCTION

A project designated "Deep Dish" was instituted at the David Taylor Model Basin for the study of elastic general instability of ring-stiffened cylindrical shells under external hydrostatic pressure. This work has been reported in References 1 through 7.* A primary objective of the project has been the evaluation of Kendrick's theoretical work[8,9] through tests of small machined cylinders having systematic variations in geometric parameters. This report summarizes background information on this project and presents results of recent studies.

SUMMARY OF EARLIER STUDIES

Three series of cylinders have been studied, the first of which (designated DD-8[3,4]) had frame size as the single variable. The other two series, DD8-1 and DD8-2, had overall length as the variable but were geometrically different. Tests of these two series were described in a previous progress report.[5] All cylinders were equipped with heavy bolting rings, and usually the bottom ring was bolted to a flat closure plate; the top ring was held against the head of a vertical pressure tank. With this arrangement it was believed that a condition of clamped support at the edges of the shell would be approximated.

Although Kendrick's theory assumes the edges to be simply supported, his contention is that clamped ends will have only a local influence on the buckling pattern and that the resulting buckling pressure will not differ appreciably from that obtained for the case of simple support. Kaminsky,[10] in extending Kendrick's analysis for the clamped condition, found this difference to be considerable. The buckled shape which Kaminsky assumed is, however, only one of many possible shapes fulfilling the clamped condition. Moreover, there is no assurance that by using some other (possibly more correct) shape substantially lower pressures could not be obtained. Although Kaminsky's work thus did not refute Kendrick's

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*References are listed on page 11.

contention, it still led investigators at the Model Basin to believe that if clamped ends could be approximated in the laboratory, the experimental pressures would exceed Kendrick's pressures by a considerable margin. The results of the first (DD-8) series agreed well with Kendrick's theory. However, this close correlation did not persist in the second and third series of tests.[5] Some of these cylinders collapsed at pressures well above the Kendrick pressures, although not nearly as high as those given by Kaminsky's theory.

The results of the DD-8, DD8-1, and DD8-2 series appeared to be so inconsistent that no definite conclusions regarding the validity of available theories could be made without additional investigation. Furthermore, it seemed clear that any intelligent interpretation of these or future test results would require some knowledge of the influence of the experimental end conditions on the collapse pressure. In Reference 5 it was proposed that duplicates of of the three cylinders of the DD8-2 series which showed poorest agreement with Kendrick's theory be tested with several different end closure arrangements. The buckling pressures would be determined nondestructively by means of the Southwell method.[6] In this way it was also hoped that some closure arrangement giving close agreement with Kendrick's theory could be found, since it was desirable to use such an arrangement with forthcoming studies of cylinders with intermediate heavy stiffeners.

RECENT STUDIES

So far the three duplicate cylinders have been tested with five different end attachments, and an additional cylinder (DD8-2-6), the shortest of the series, has also been tested. Furthermore, the longest cylinder, 4-A, had been tested using a technique with movable insert disks to provide a test section of varying length. Use of this arrangement should more closely simulate the actual conditions existing in a submarine. Also, since the only rotational restraint at the ends of the test section would come from the continuity of the shell, this condition would appear to approximate that of simple supports more closely than the closure arrangements used previously, thereby providing a more valid test of Kendrick's theory. Since experimental points could be obtained at small intervals of length, a true experimental curve of buckling pressure versus length could be established. Of particular interest was the transition between two and three circumferential lobes, a point where, theoretically, a discontinuity (cusp) should exist.

This method thus ~~provides~~ a rapid and inexpensive means of obtaining much additional data through the use of one existing cylinder. Thus far, tests have been completed with ten different compartment lengths. The results of these tests together with those obtained using the various end closure arrangements are summarized briefly in this third progress report.

TESTS WITH DIFFERENT END CONDITIONS

The four cylinders under consideration designated DD8-2-6, 2A, 3A, and 4A are pictured in Figure 1. In this series the length-to-diameter ratio varies from 1.91 to 6.85.

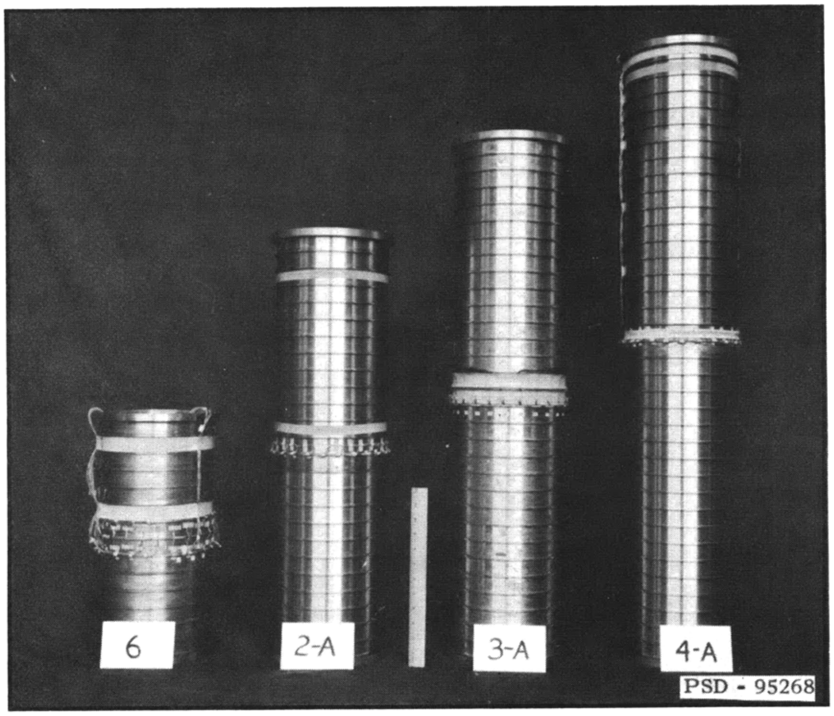

Figure 1 – Test Cylinders with Gages Installed

Cylinder dimensions are shown in Figure 2. As with the original series of cylinders, these cylinders had rectangular stiffening rings and were accurately machined from steel tubing whose high yield strength (85,000 psi) was sufficient to prevent inelastic action prior to the onset of buckling. The various end closure arrangements are illustrated in Figure 3. Case IV, the arrangement used with the original set of cylinders,[4] employed a closure plate approximately ½ in. thick. The 2-in. plates were chosen to find out whether a thicker plate would affect the collapse pressure significantly. Case II closure was used to determine the effect of replacing the distributed pressure over the surface of the thinner plate with a concentrated load around its edge. This was done by means of pressure caps which are shown in Figure 4 along with the ½-in. and 2-in. closure plates. All the closure plates were equipped with machined inserts fitting tightly within the cylinder wall to prevent radial deflections at the ends.

Each cylinder was instrumented with electrical resistance strain gages most of which were confined to the exterior of the centermost stiffener and oriented circumferentially. The buckling pressures were obtained by the Southwell method[6] applied to the strain-pressure plots.

The buckling pressures obtained under the various end conditions are given in Table 1 together with the pressures for the original group of cylinders and the theoretical pressures according to Kendrick Part III* and Kaminsky.

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*Since it has been shown[3,7] that the theory of Kendrick Part I is often unconservative, and sometimes may be unsafe to use, attention is confined to the second solution of Kendrick Part III, which is simply a more rigorous treatment of the same problem.

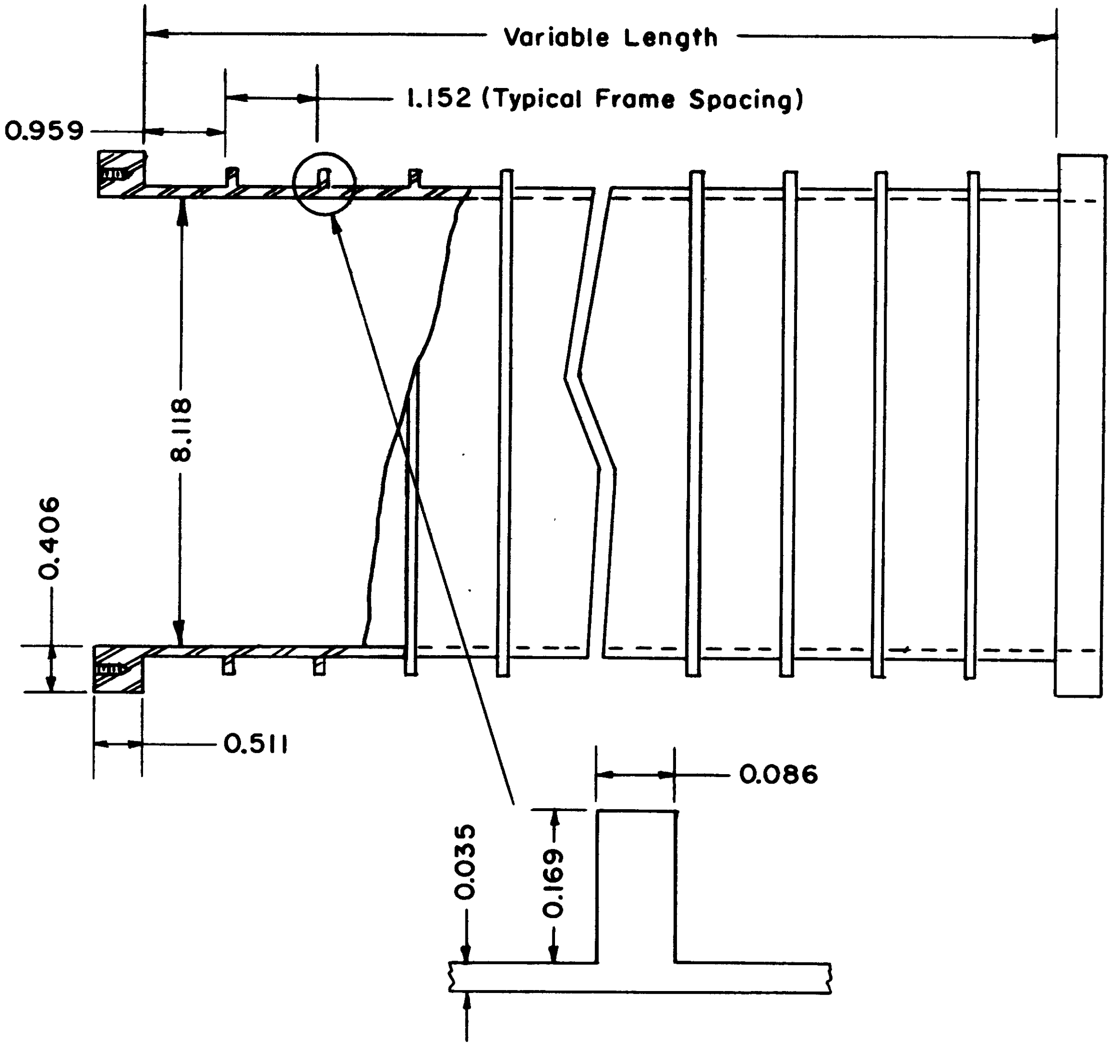

Figure 2 – Schematic Diagram Showing Dimensions of Machined Cylinders

All dimensions are in inches.

The same information is presented graphically in Figure 5. The close agreement between Cylinders 2 and 2-A, 3 and 3-A, and 4 and 4-A under Case IV in Table 1 indicates that the three cylinders were accurately duplicated. Most striking, perhaps, is the very large increase in strength realized in progressing from Case I to Case V. For Cylinder 3-A, this amounted to a pressure increase of 65 percent. Also remarkable is the nearly perfect agreement with Kendrick's theory for Case I. Another interesting result was that, in those cases (I-IV) where one or both ½-in. plates were used, the appearance of the characteristic lobar strain pattern prior to buckling was influenced by the orientation of the end plates. That is, with Case II, Cylinder 3-A, for example, a rotation through 90 deg of the end plates produced a 90 deg rotation of the circumferential strain pattern, along with a slight change in the buckling pressure,

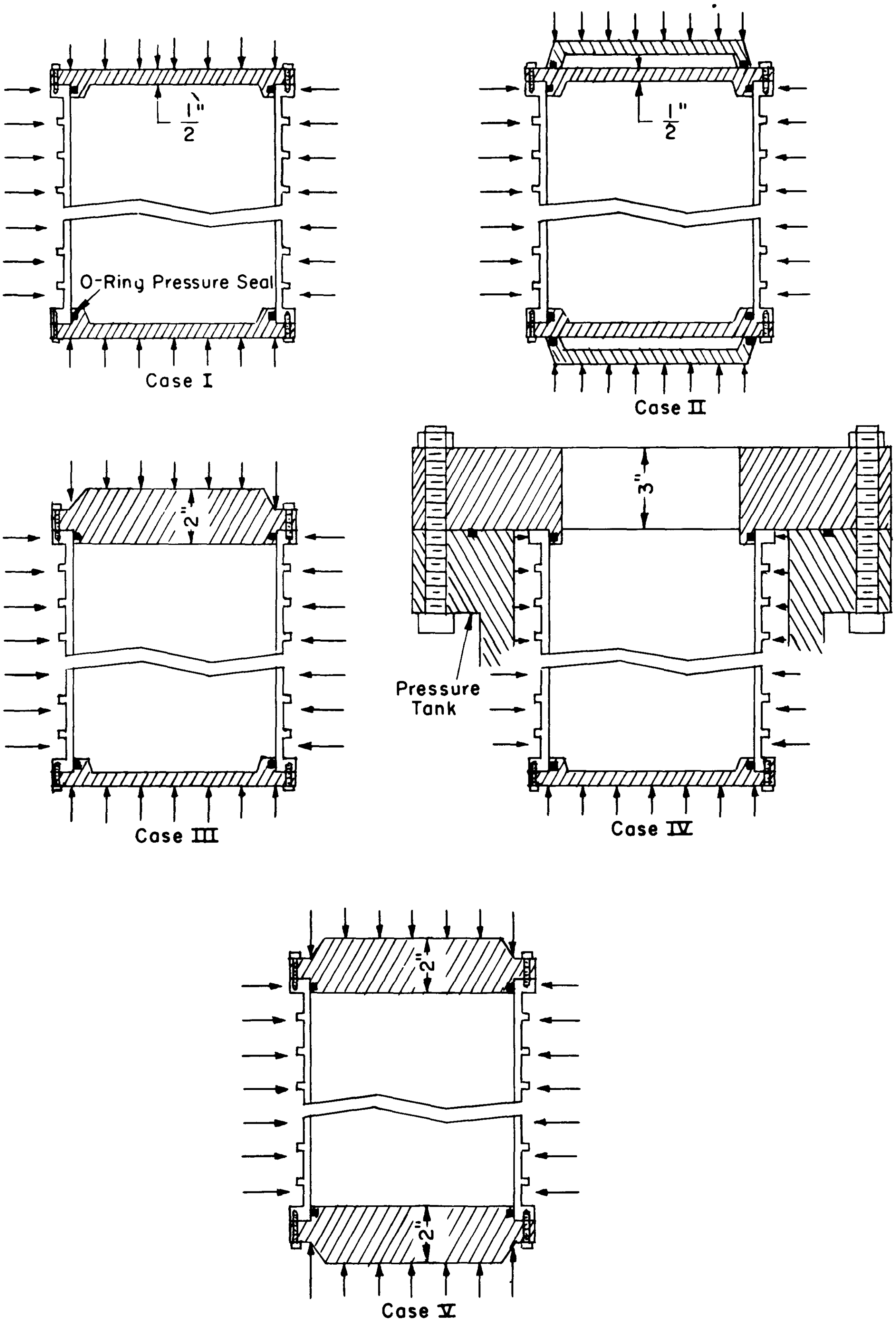

Figure 3 – Schematic Diagram of End Closure Arrangements

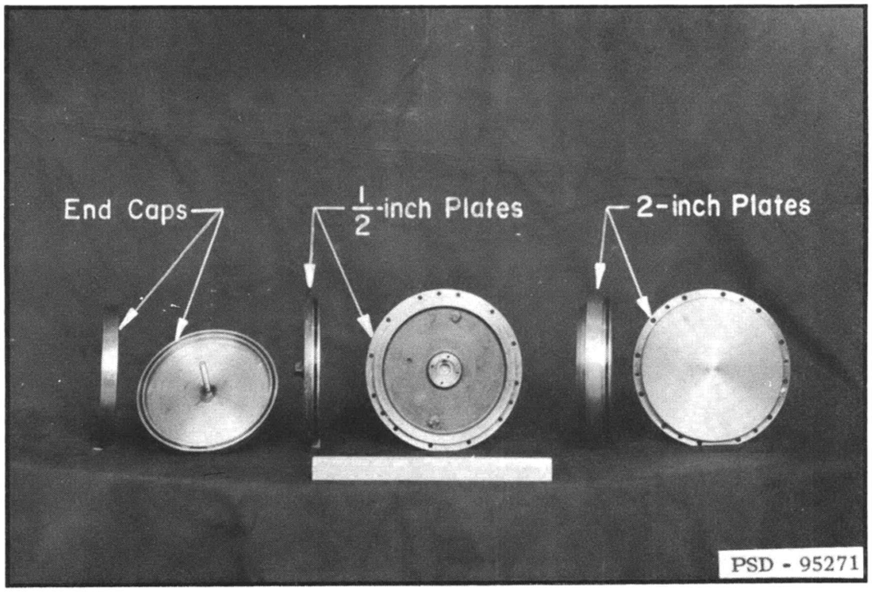

Figure 4 – Closure Plates used in Tests

TABLE 1

Results of Tests with Varied End Conditions

Cylinder Number	Frame Spaces	Length / Diameter	Theoretical Pressures, psi: Kendrick III (Simple Support)	Theoretical Pressures, psi: Kaminsky (Clamped Support)	Experimental Pressures, psi: Case I	Experimental Pressures, psi: Case II	Experimental Pressures, psi: Case III	Experimental Pressures, psi: Case IV	Experimental Pressures, psi: Case V
6	14	1.91	499(3)*	910(3)	504(3)	576(3)	576(3)		700(3)
1	19	2.61	415(3)	702(3)				492(3)	
2	25	3.36	317(2)	575(3)				408(3)	
2-A					339(2)	365(2)	394(3)	398(3)	409(3)
3	31	4.31	216(2)	551(3)				321(2)	
3-A					229(2)	246(2)	298(2)	315(2)	378(2)
						238(2)**			
4	37	5.16	180(2)	494(2)				238(2)	
4-A					178(2)	189(2)	231(2)	228(2)	289(2)
5	49	6.85	159(2)	360(2)				173(2)	

*Numbers in parentheses indicate the number of circumferential lobes.

**End plates rotated 90 deg with a corresponding rotation of lobar strain pattern.

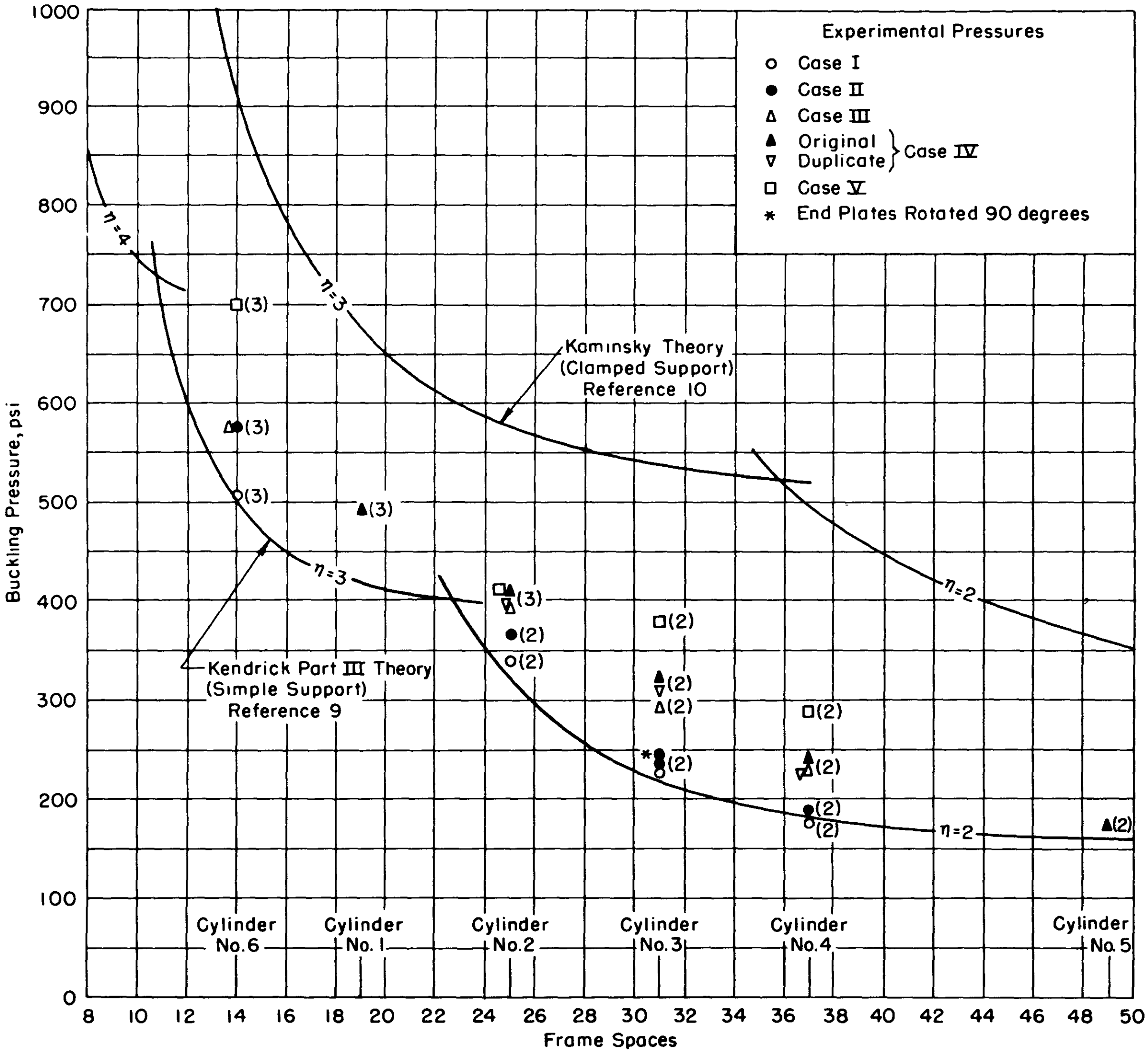

Figure 5 – Experimental Pressures for Different End Conditions with Theoretical Pressures for Simple and Fixed Support

Numbers in parentheses indicate number (*n*) of circumferential lobes.

as shown in Table 1. This cannot be attributed to any lack of circularity in the plate inserts, which were more nearly circular than the cylinders, nor was there any obvious lack of uniformity in the plate cross sections which would indicate a lack of symmetry in the bending rigidity. Rotation of the 2-in. plates, on the other hand, had no effect on the orientation of the strain pattern nor on the buckling pressure.

TESTS WITH MOVABLE INSERT DISKS

Cylinder 4-A, the longest of the duplicate group, was tested with movable insert disks simulating bulkheads. These were circular disks with edges machined to a small radius to

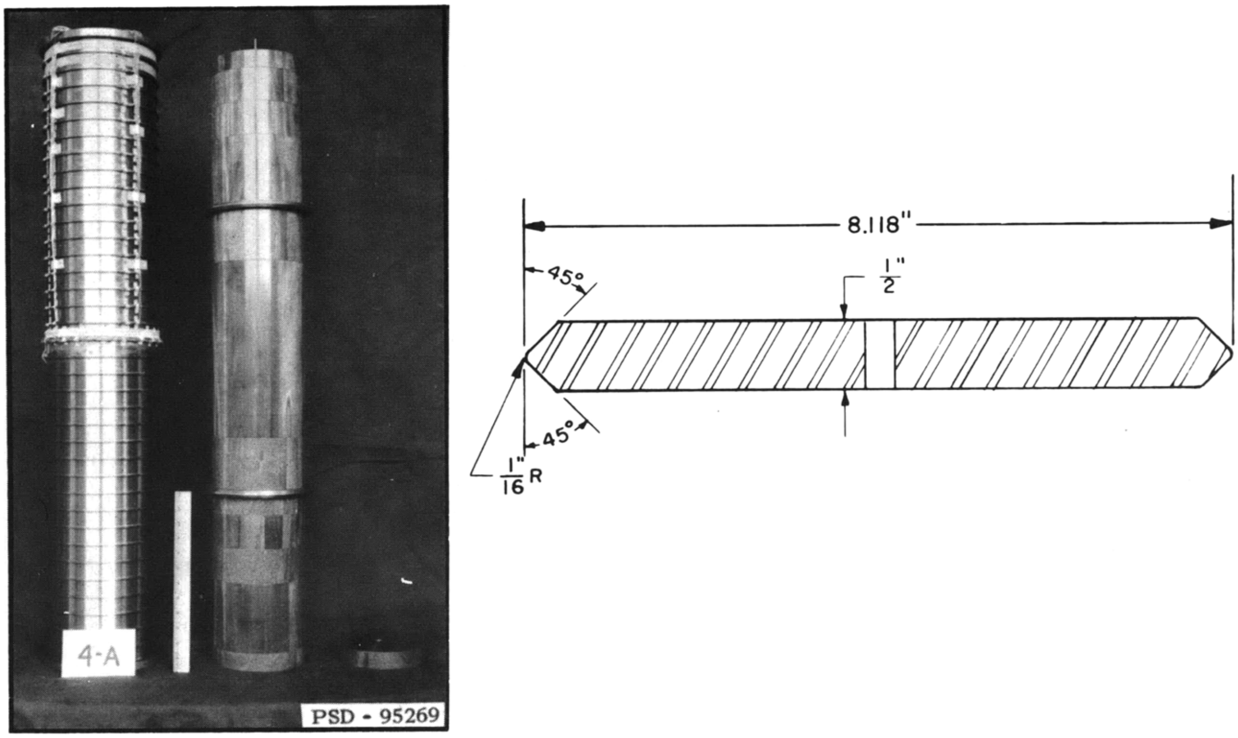

Figure 6 – Arrangement for Tests with Internal Movable Bulkheads

TABLE 2

Results of Tests with Internal Bulkheads

Frame Spaces	Length/Diameter	Kendrick III Pressure, psi	Experimental Pressure, psi
17	2.40	428(3)*	473(3)
21	2.96	404(3)	422(3)
23	3.25	367(2)	412(3)
25	3.53	305(2)	401(3)
26	3.67	281(2)	398(3)
27	3.81	262(2)	394(3)
28	3.96	246(2)	391(3)
29	4.10	233(2)	383(2)
31	4.38	212(2)	329(2)
33	4.66	197(2)	281(2)

*Numbers in parentheses indicate the number of circumferential lobes.

minimize the area of contact with the shell. The disks were held in place by circular wooden spacers of various widths as shown in Figure 6. By separating the disks with different combinations of the spacers, the length of the central test section could be varied from one-third to the full length of the cylinder in increments of one frame space. In each test the disks were located directly beneath a stiffener and the ends of the cylinder were closed as in Case I of Figure 3.

In addition to the strain gages located around the centermost stiffener for the previous tests, additional gages were installed circumferentially along two generators over one half of the cylinder length to measure the longitudinal shape of the deflection pattern. As before, the Southwell method was used to determine the elastic buckling pressures.

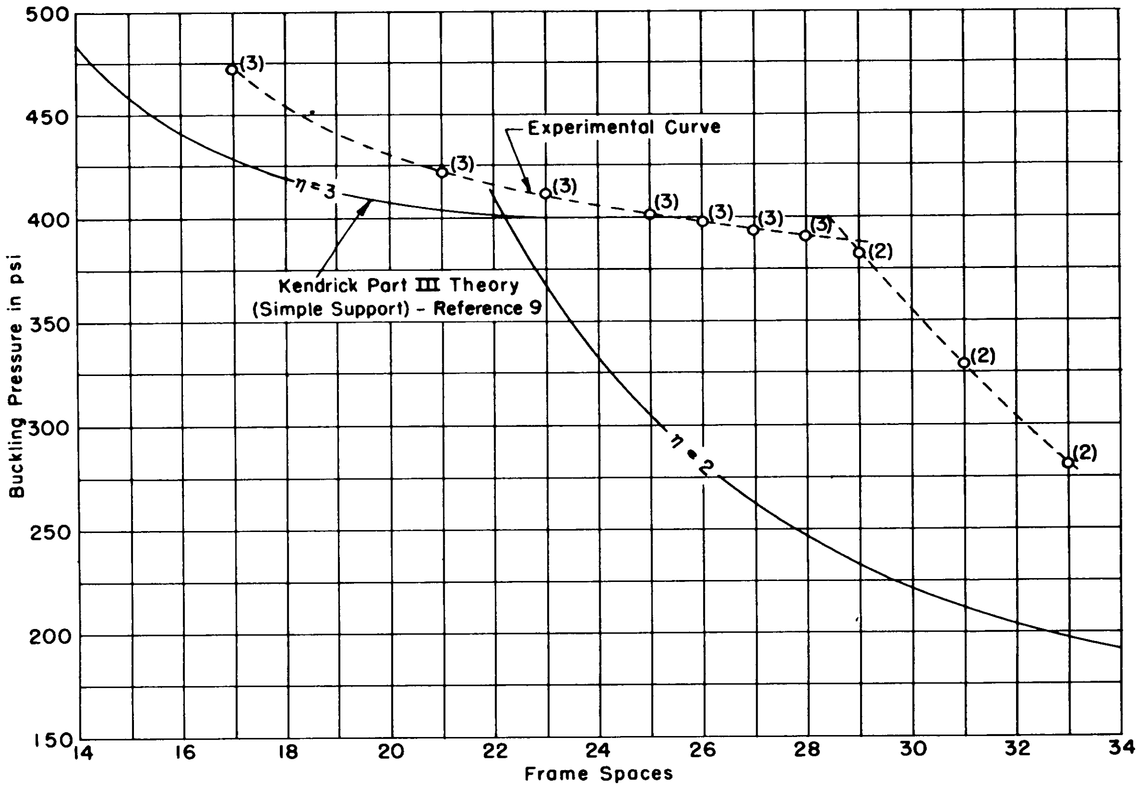

Figure 7 – Experimental Pressures versus Internal Bulkhead Spacing Compared with Kendrick's Theory for Simple Support

Numbers in parentheses indicate number (*n*) of circumferential lobes.

Tests were conducted with ten different compartment length-diameter ratios ranging from 2.40 to 4.66. Table 2 lists the compartment length-diameter ratios, the corresponding experimental pressures, and the pressures given by the Kendrick Part III theory.* These results appear graphically in Figure 7. It is seen that, although the experimental points lie somewhat above the theoretical curve, they follow a similar curve, proving the existence of the discontinuity (cusp) formed at the intersection of the curves for two and three circumferential lobes. This fact, not previously verified experimentally for plate and shell structures, should be of more than mere academic interest, since it indicates that compartment lengths must be chosen with care if the material is to be used efficiently. For example, the experimental curve shows that a compartment comprising 31 frame spaces can be appreciably strengthened if it is shortened by two frame spaces. Beyond that point, however, little benefit is gained unless a reduction of ten spaces or more is made. In comparing these results with the previous tests, good agreement is found between the pressures for 25 and 31 frame spaces and those under Case IV for the cylinders (2 and 3) of corresponding length.

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*The calculated pressures for 25 and 31 frame spaces given in Table 2 do not agree exactly with those given in Table 1 for Cylinders 2 and 3 because, for all cylinders, the end frame spacings are slightly shorter than typical, whereas the test sections in Table 2 had uniform frame spacing.

DISCUSSION AND CONCLUSIONS

Probably the most important conclusion that can be drawn from the results of these tests is that the boundary conditions have an appreciable influence on the buckling pressure. Furthermore, it is significant to note that while Kendrick's theory in some cases seriously underestimates the buckling pressure, in no case does his theory appreciably overestimate it. Tests with Case V, where the ends would seem to be extremely rigid, indicate that the influence of clamped ends may be overemphasized by Kaminsky.

To draw more specific conclusions regarding the effect of boundary conditions would be unwarranted on the basis of the limited data available at this time. However, one possible explanation bears mention. In Reference 5, it was suggested that "the introduction of partial fixity at the ends of a simply supported cylinder is equivalent to shortening its effective length." It is possible that this effect may depend on the simple relationship,

$$L_e = KL$$

where L is the actual length of the cylinder,

L_e is its effective length, and

K is a constant depending on the degree of fixity, being less than one for all cases falling between simple support and clamped ends.

Such a relationship is valid for the buckling of a centrally compressed column and has been utilized with some success by Arnold and Warburton[11] in a study of the effect of boundary conditions on the flexural vibrations of unstiffened tubes.

If this relationship were to hold for the case of general instability, all conditions of edge fixity could be represented by a single curve of pressure versus effective length, provided K is known as a function of edge fixity. From this representation comes the interesting result that the transition from one circumferential mode to another must occur at the same pressure, regardless of the degree of fixity. Figure 7 shows just this sort of correspondence between Kendrick's solution for simple support and the experimental curve obtained from the tests with internal bulkheads. The theoretical transition between two and three lobes occurs at about 401 psi, while experimentally it appears at about 389 psi. This small difference in pressures could result from variations in cylinder dimensions within the specified tolerances or from slight inaccuracies arising from the use of nominal values for Young's modulus (30×10^6 psi) and Poisson's ratio (0.3) in the calculations.

On this basis Kendrick's theory would appear to be very accurate. On the other hand, Kaminsky's theory for clamped ends (Figure 5) shows the transition from two to three lobes at about 520 psi, considerably higher than that found experimentally.

It is emphasized that this concept of effective length is presented merely as one interpretation of the results thus far obtained. At the present time there are insufficient data for a proper evaluation of the idea. It can only be said that none of the tests to date has shown it to be invalid, and that one of the immediate objectives of Project Deep Dish should be to provide the additional data necessary for a proper evaluation.

FUTURE WORK

In order to continue the line of study described in this report and to investigate other related areas of interest, the following work is anticipated:

1. Further tests employing internal bulkheads to provide a central compartment of varying length. In particular, tests are now underway using the shorter cylinder 2-A to investigate the influence of a shorter end compartment on the buckling pressures already obtained as well as to provide additional data in the range of shorter central compartments.

2. Efforts to design a test arrangement that will simulate the simple support condition, assumed by Kendrick in his theoretical treatment.

3. Further tests of cylinders having one intermediate heavy stiffener, to investigate the efficiency of deep frames in breaking up the overall length of a submarine compartment. The size of the stiffener will be systematically reduced, and the collapse pressure will be determined at each stage by nondestructive testing.

4. An investigation of the range of boundary conditions existing in an actual submarine so that future structural model tests of submarine designs would incorporate more realistic end closure bulkheads.

5. An examination of the clamped-end case to find a more realistic theoretical buckling shape which would lead to buckling pressures lower than those given by the Kaminsky analysis.

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3. Cylindrical shells (Stiffened) – Buckling – Test methods (Nondestructive)
4. Submarine hulls – Characteristics – Model tests.
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lydrostatic pressure tests were conducted on four machined
;-stiffened cylinders of different lengths to investigate the in-
ence of end fixity on the elastic general-instability pressure.
:h cylinder was tested nondestructively with several different
conditions. Pressure variations of as much as 65 percent
e observed for a particular cylinder.
)ther tests were conducted with one cylinder reinforced by in-
ial movable bulkheads whereby buckling pressures were ob-
ied as a function of bulkhead spacing. The resulting experi-
ital curve confirms the existence of discontinuities which
ur, according to Kendrick's theory, at the transition between
circumferential buckling modes.

1. Cylindrical shells (Stiffened) – Buckling – Model tests.
2. Cylindrical shells (Stiffened) – Buckling – Graphical analysis.
3. Cylindrical shells (Stiffened) – Buckling – Test methods (Nondestructive)
4. Submarine hulls – Characteristics – Model tests.

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II. Blumenberg, William F.
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