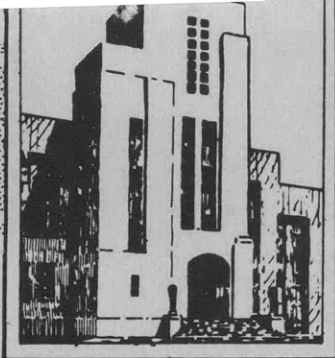DEPARTMENT OF THE NAVY
DAVID TAYLOR MODEL BASIN

HYDROMECHANICS
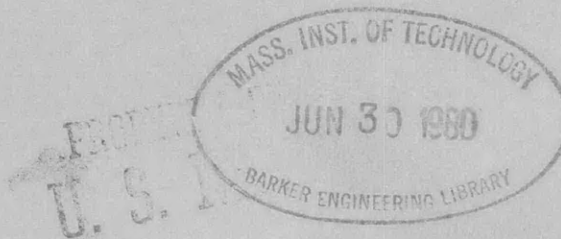
○

AERODYNAMICS

○

STRUCTURAL
MECHANICS

○

APPLIED
MATHEMATICS

HYDROSTATIC TESTS OF INELASTIC AND ELASTIC STABILITY

OF RING-STIFFENED CYLINDRICAL SHELLS

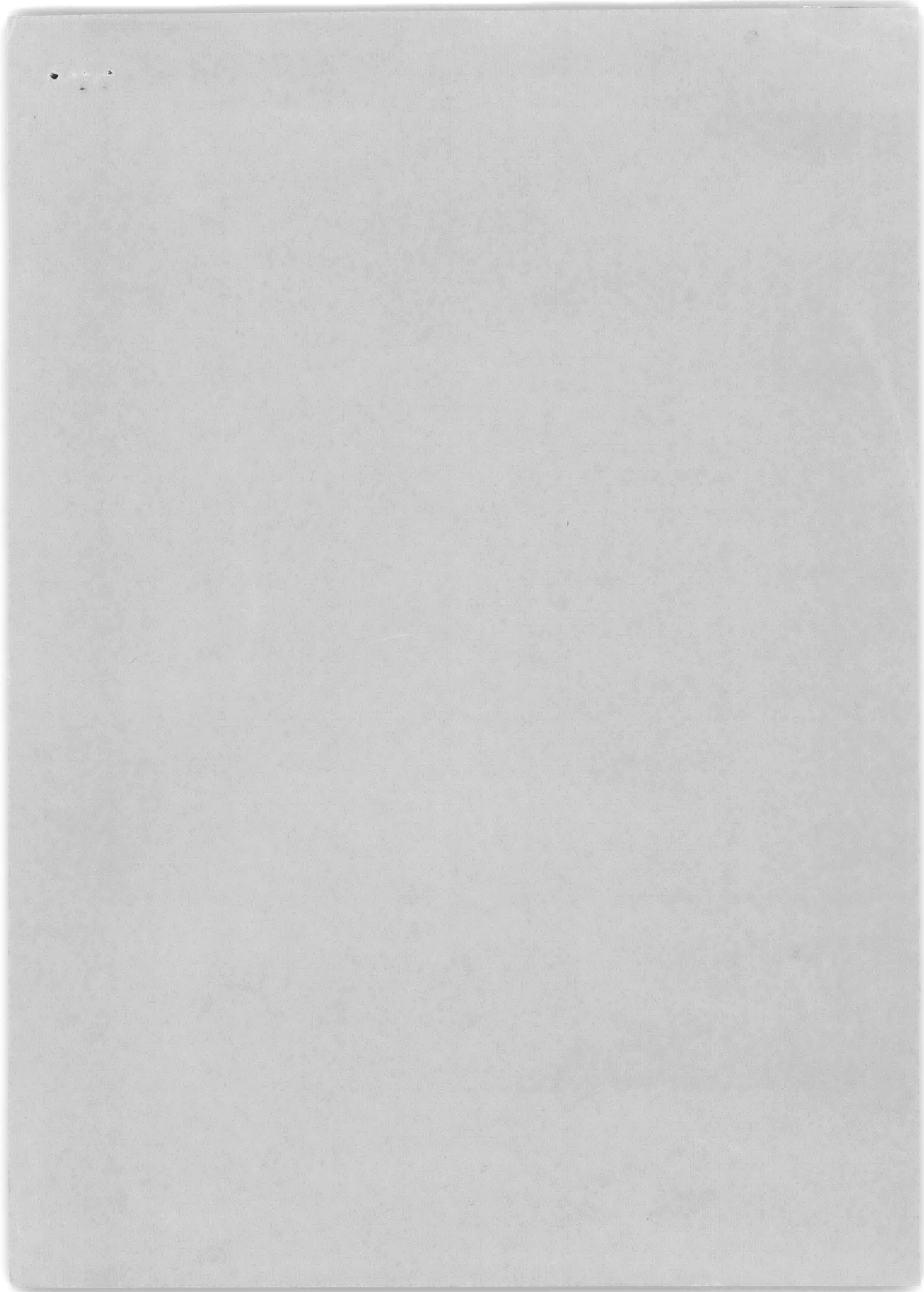MACHINED FROM STRAIN-HARDENING STEEL

by

Kenneth Hom and William P. Couch

STRUCTURAL MECHANICS LABORATORY

RESEARCH AND DEVELOPMENT REPORT

December 1961   Report 1501

From: Commanding Officer and Director, David Taylor Model Basin
To: Chief, Bureau of Ships (442) (in duplicate)

Subj: Experimental investigation of inelastic and elastic
        stability of ring-stiffened cylindrical shells machined
        from strain-hardening steel; forwarding of report on

Encl: (1) DATMOBAS Report 1501 entitled "Hydrostatic Tests of
            Inelastic and Elastic Stability of Ring-Stiffened
            Cylindrical Shells Machined from Strain-Hardening
            Steel"   2 copies

1. The results of an experimental investigation of ten ring-
stiffened cylindrical shells machined from strain-hardening
steel having a range of "thinness ratio" λ from 0.85 to 1.67
are presented in enclosure (1).

2. Agreement within 4 percent was obtained between the exper-
imental collapse pressure and the theoretical pressures com-
puted from Reynolds' inelastic lobar buckling theory for those
cylinders with thinness ratios of 0.85 to 1.10 and from Reynold's
elastic lobar buckling theory for those cylinders with thinness
ratios greater than 1.40. However, agreement was only fair
between experimental pressures and the pressures computed from
Reynolds' inelastic theory for those cylinders with thinness
ratios between 1.10 and 1.40; these cylinders were still in
the inelastic range but tending toward the elastic range.

E.E. JOHNSON
By direction

Copy to:
BUSHIPS (335) with 3 copies of encl (1)
        (320) with 1 copy of encl (1)
        (341A) with 1 copy of encl (1)
        (420) with 1 copy of encl (1)
        (421) with 1 copy of encl (1)
        (443) with 1 copy of encl (1)

Copy to:
CHONR (439) with 1 copy of encl (1)
CDR, USNOL, with 1 copy of encl (1)
DIR, USNRL (2027) with 1 copy of encl (1)
NAVSHIPYD PTSMH, with 2 copies of encl (1)
NAVSHIPYD MARE, with 2 copies of encl (1)
NAVSHIPYD CHASN, with 1 copy of encl (1)
SUPSHIP, Groton, with 1 copy of encl (1)
Elec Boat Div, General Dynamics Corp, with 1 copy of encl (1)
SUPSHIP, Newport News, with 1 copy of encl (1)
NNSB and DD Co, with 1 copy of encl (1)
SUPSHIP, Pascagoula, with 1 copy of encl (1)
Ingalls Shipbldg Corp, with 1 copy of encl (1)
CO, USNROTC and NAVADMINU, MIT, with 1 copy of encl (1)
DIR DEF R and E, Attn: Tech Library, with 1 copy of encl (1)
O in C, PGSCOL, Webb, with 1 copy of encl (1)
Prof, N.J. Hoff, Stanford Univ, with 1 copy of encl (1)
Prof. J. Kempner, Bklyn Polytech Inst, with 1 copy of encl (1)
Dr. E. Wenk, Jr., White House, with 1 copy of encl (1)
Dr. R.C. DeHart, Southwest Res Inst, with 1 copy of encl (1)
Dr. V.L. Salerno, Fairleigh Dickinson Univ, with 1 copy of
   encl (1)
Prof. G. Gerard, College of Eng, New York Univ, with 1 copy
   of encl (1)
Prof. H. Becker, College of Eng, New York Univ, with 1 copy
   of encl (1)

2

HYDROSTATIC TESTS OF INELASTIC AND ELASTIC STABILITY

OF RING-STIFFENED CYLINDRICAL SHELLS

MACHINED FROM STRAIN-HARDENING STEEL

by

Kenneth Hom and William P. Couch

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ABSTRACT

Ten ring-stiffened cylindrical shells machined from strain-hardening steels with yield strengths of 70,000 psi and 100,000 psi were tested to collapse under external hydrostatic pressure. The cylinders were designed to cover a range of "thinness ratio" λ from 0.85 to 1.67. Four of the cylinders collapsed in a combined asymmetric (lobar) and axisymmetric mode; the other six cylinders failed only in the lobar mode.

Agreement within 4 percent was obtained between the experimental collapse pressures and the theoretical pressures computed from Reynolds' _inelastic_ lobar buckling theory for those cylinders with thinness ratios of 0.85 to 1.10 and from Reynolds' _elastic_ lobar buckling theory for those cylinders with thinness ratios greater than 1.40. However, agreement was only fair between experimental pressures and the pressures computed from Reynolds' inelastic theory for those cylinders with thinness ratios between 1.10 and 1.40; these cylinders were still in the inelastic range but tending toward the elastic range.

INTRODUCTION

As part of a test program designed to provide information on the resistance of ring-stiffened cylindrical shells to explosive loading, ten machined externally-framed cylinders were tested to collapse under hydrostatic pressure. The cylinders were designed with geometries in the lobar buckling range and were machined from strain-hardening steel. The static tests were conducted primarily to provide data on static strength for use in evaluating the subsequent results of dynamic tests of similar cylinders. In addition, the static results also afforded an excellent opportunity to compare experimental collapse pressures with the pressures calculated from theories recently developed by Reynolds for inelastic and elastic lobar buckling of stiffened cylinders under hydrostatic pressure.

This report deals only with the static phase of the test program. The design, instrumentation, and testing of the cylinders that were collapsed under hydrostatic pressure are described; experimental and theoretical strains are compared; and the experimental collapse pressures are discussed and compared with theoretical pressures.

BACKGROUND

Two basic _inelastic_ modes of shell failure can occur when ring-stiffened cylinders fabricated from _strain-hardening_ material are subjected to external hydrostatic pressure. These are: axisymmetric buckling, a phenomenon in which circumferential corrugations develop in the shell, and asymmetric or lobar buckling in which alternate inward and outward lobes appear around the circumference of the shell. Recently at the David Taylor Model Basin, analyses[1,2] based on stability considerations were developed for determining the buckling pressures of a stiffened cylinder collapsing in either of these modes. The buckling pressures were found to be a function of the cylinder geometry and the secant and tangent moduli of the shell material as determined from a uniaxial stress-strain curve.

Prior to these recent developments,[*] collapse of the axisymmetric type was usually treated theoretically as failure due to yielding precipitated by high stress rather than as a buckling phenomenon. These past analyses were based on the assumption of an idealized material which, at a certain stress level depending upon the failure criterion used, underwent an abrupt transition to the perfectly plastic state. Analyses of this type are applicable for stiffened cylinders fabricated from material with an ideally plastic (plateau-type) stress-strain curve.
For a structure made of a strain-hardening material, these past analyses have been proven inadequate since the determination of collapse pressure was predicated on the state-of-equilibrium stresses and was not based on stability considerations. Investigations of inelastic lobar buckling have been rather limited in the past and empirical in nature.

Another possible mode of failure which could be analyzed on the basis of stability considerations is that termed "_elastic_ lobar buckling."

---

[1] References are listed on page 15.

[*] A recent Japanese paper dealing with the axisymmetric collapse problem has come to the attention of the Model Basin. It will soon be available as TMB Translation T-298 (Reference 3).

2

The theoretical buckling pressures for this case are independent of yield strength and secant and tangent moduli of the material and can be determined from the cylinder geometry, Poisson's ratio, and Young's modulus of elasticity for the shell material. In 1929, Von Mises[4] developed a rigorous solution for the elastic buckling of a thin-walled cylinder of finite length under hydrostatic pressure, assuming simple support conditions at the ends. He further assumed that the prebuckling deformations of the stiffened cylinder could be approximated by the more simple ones corresponding to an unstiffened cylinder of infinite length. In 1932, Von Sanden and Tölke[5] extended this work; they also assumed simple support conditions at the frames but considered the exact prebuckling deformations of Von Sanden and Günther[6] for a stiffened cylinder in an approximate manner to make the mathematics tractable. Recently at the Model Basin, Reynolds[7] approached the problem more rigorously than these past investigators by considering more realistic boundary conditions at the frames and describing the prebuckling deformations in the form of a Fourier series so as to permit a better approximation to the actual state.

Since the cylinders presented in this report were designed with geometries in the lobar buckling range and were machined from strain-hardening steel, the test results are used to evaluate the theories that have recently been developed by Reynolds for inelastic[2] and elastic[7] lobar buckling.

DESCRIPTION OF MODELS

The models were machined from AISI-4140 seamless steel tubing, heat treated to a yield strength of approximately 70,000 psi for two models and 100,000 psi for the other eight models. Stress-strain curves obtained from uniaxial compression specimens cut from the heat-treated tubing indicated that the material was of the strain-hardening type. Typical stress-strain curves of the materials with yield strengths of about 70,000 psi and about 100,000 psi are plotted in Figure 1; the curves shown represent specimens of the material used for Models 12 and 55.
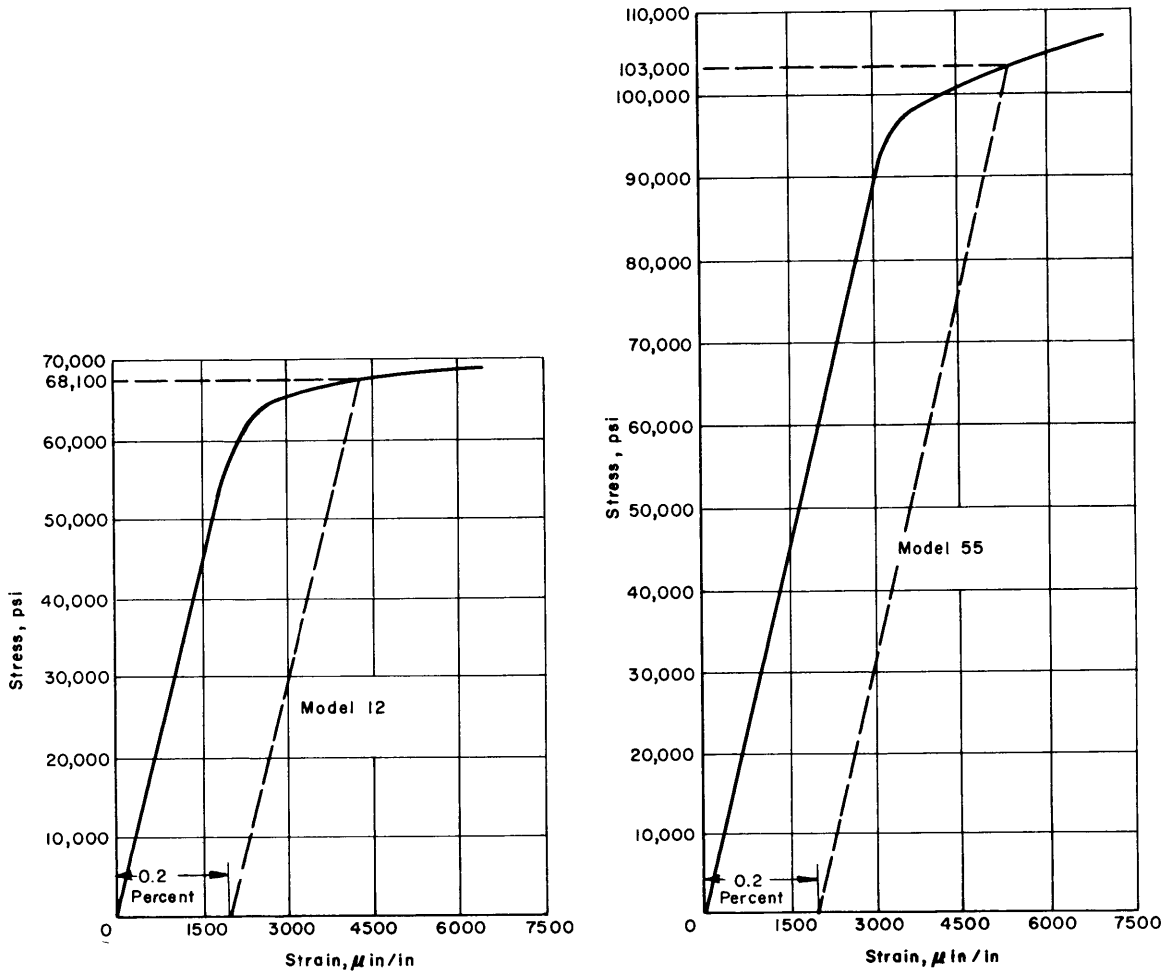
3

Figure 1 - Stress-Strain Curves of Materials Used in Models

The geometry and material yield strength (based on a 0.2 percent offset) for each of the models are summarized in Table 1. All models were designed with an end arrangement as shown in the schematic diagram of Table 1 in order to preclude premature failure near the ends. The models were made pressure-tight by integral heavy rings at both ends, of which one was attached to a closure bulkhead and the other was bolted to the inside surface of the pressure tank head.

TABLE 1

Geometrical and Material Properties of Models

A nominal value of $E = 30.0 \times 10^6$ psi was assumed as Young's modulus of elasticity for the models.
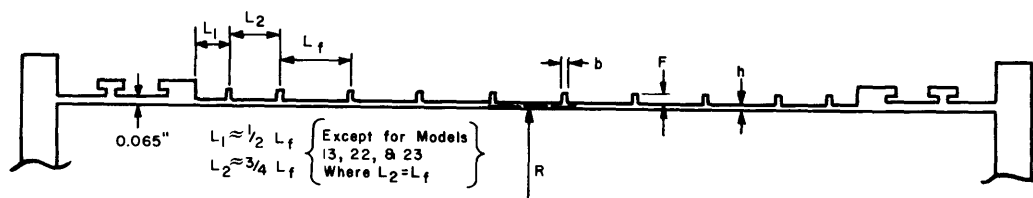
Model	Radius R in.	Shell Thickness h in.	Frame Spacing L_f in.	Frame Width b in.	Frame Depth F in.	Number of Typical Frame Spacings	Yield Strength σ_y psi
12	4.0813	0.0446	1.140	0.079	0.314	8	68,100
13	4.0788	0.0396	0.912	0.078	0.311	13	96,500
22	4.0768	0.0356	1.140	0.068	0.272	10	70,500
23	4.0745	0.0310	1.140	0,061	0.242	10	97,500
54	4.0704	0.0229	0.684	0.057	0.228	15	104,400
55	4.0727	0.0274	0.912	0.059	0.237	11	103,000
56	4.0758	0.0337	1.368	0.063	0.251	6	103,700
62	4.0827	0.0475	1.368	0.078	0.311	7	104,000
65	4.0827	0.0475	0.912	0.078	0.311	11	104,000
73	4.0727	0.0274	1.368	0.078	0.312	7	103,200

INSTRUMENTATION AND TEST PROCEDURE

All models were instrumented with SR-4 electrical-resistance strain gages to study their elastic and inelastic behavior during pressure loading and to facilitate interpretation of the results. The shells were instrumented in order to obtain data for comparison with the elastic strains computed from the Von Sanden and Günther theory,[6] which theory constitutes the basis for designing ring-stiffened cylindrical shells subjected to hydrostatic pressure. The rectangular frames were not instrumented since the width of the frames was too narrow for strain gages.

The models were tested to failure in two or more pressure runs. Preliminary runs were made to obtain sufficient data on the elastic behavior of each model for comparison with the elastic strains computed from the Von Sanden and Günther theory.[6] A period of 5 min. was allowed to elapse before strains were recorded to permit stabilization of stresses in the models. All models were tested to collapse in the Model Basin 10-in. diameter hydrostatic pressure tank.

TEST RESULTS

Six of the ten models (Models 22, 23, 55, 56, 62, and 73) failed in the shell by lobar buckling. The other four models (Models 12, 13, 54, and 65) failed in a combined mode of lobar buckling and axisymmetric collapse.

TABLE 2

Experimental Collapse Pressure, Mode of Failure, and Thinness
Ratio for Each Model

Model	Experimental Collapse Pressure psi	Mode of Failure	Thinness Ratio λ
12	975	BY[*]	0.855
13	1160	BY	0.986
22	735	B	1.035
23	705	B	1.354
54	695	BY	1.340
55	730	B	1.358
56	725	B	1.443
62	1335	B	1.111
65	1695	BY	0.893
73	475	B	1.671

[*]B denotes lobar buckling; BY denotes combination of lobar buckling and axisymmetric collapse.
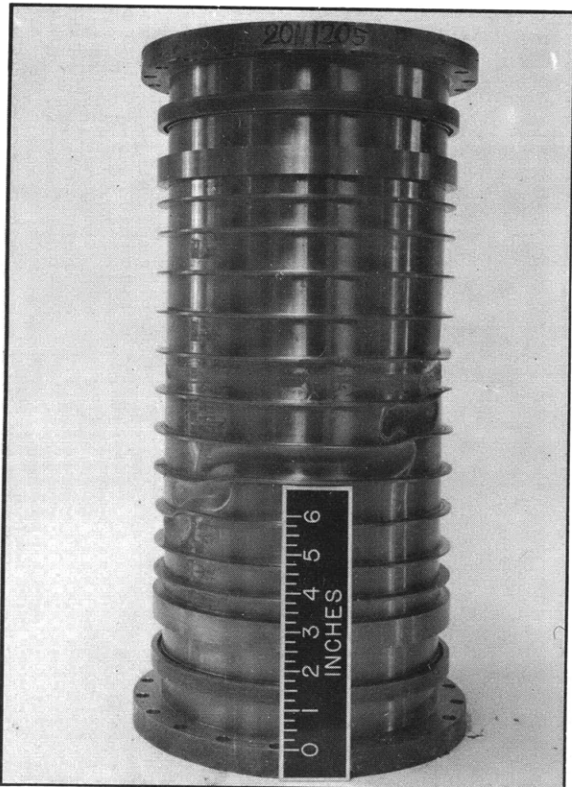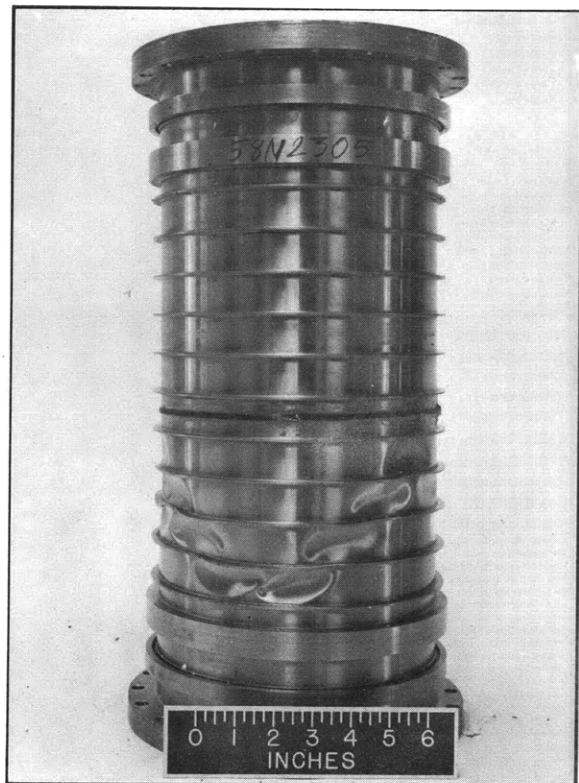
PSD-68939
Figure 2a - Model 12 ($\lambda = 0.855$)
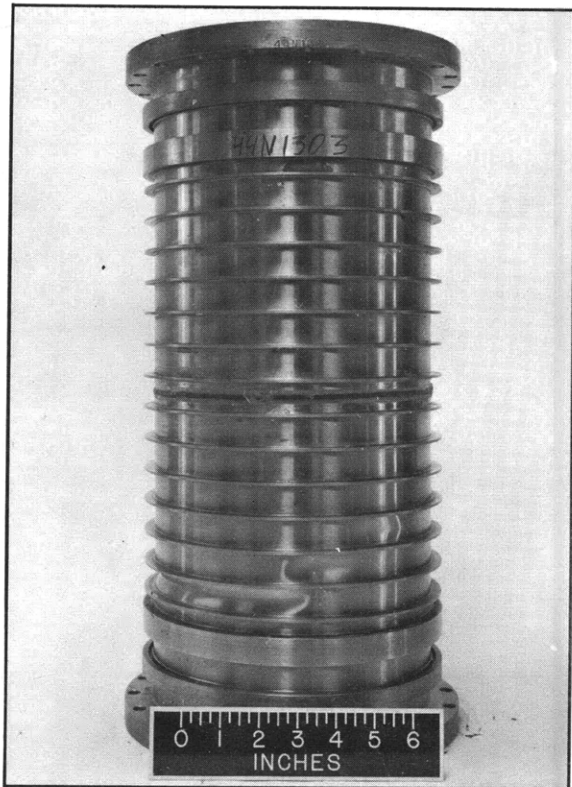
PSD-68937
Figure 2b - Model 13 ($\lambda = 0.986$)
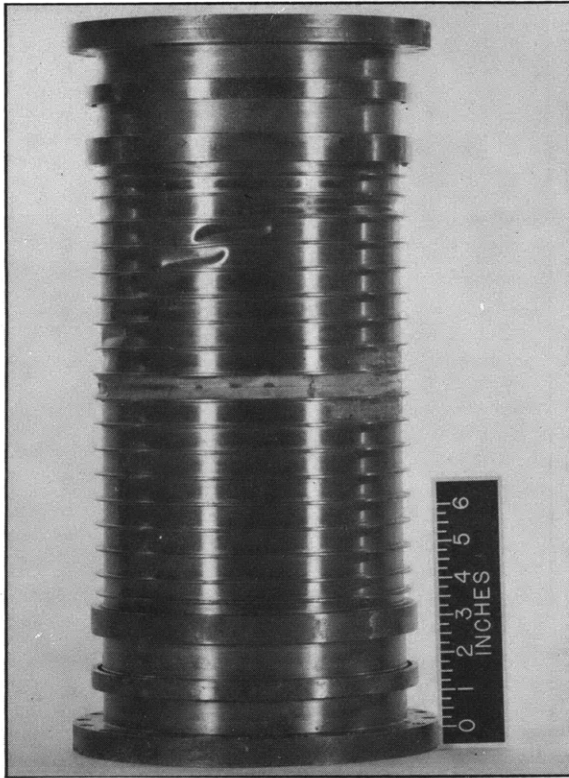
PSD-68936
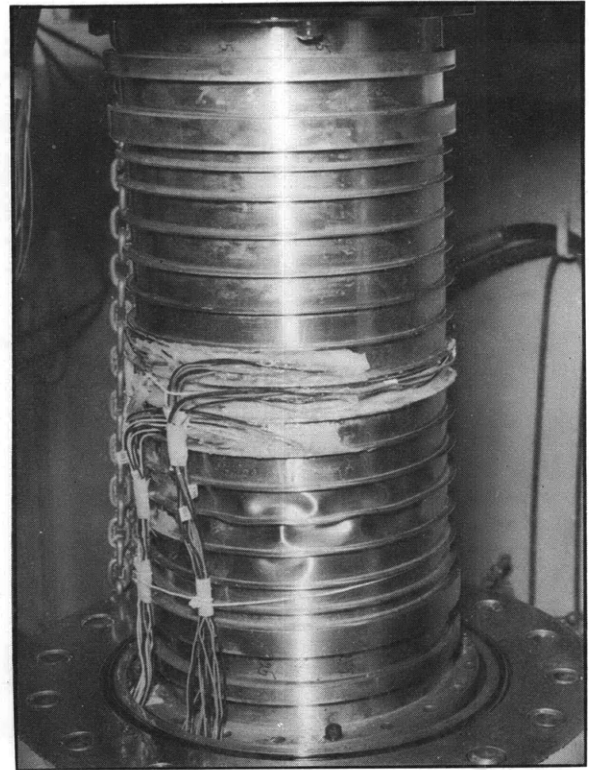Figure 2c - Model 22 ($\lambda = 1.035$)

PSD-68943
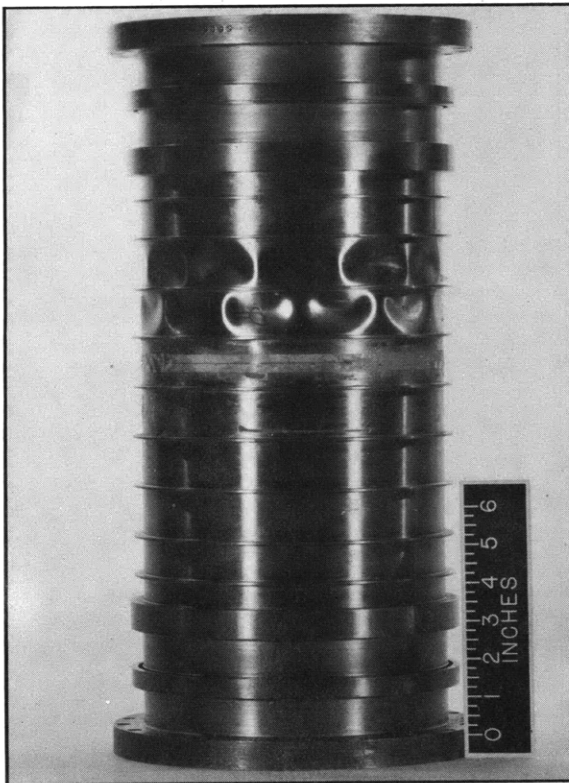Figure 2d - Model 23 ($\lambda = 1.354$)

Figure 2 - Models after Collapse Showing Modes of Failure
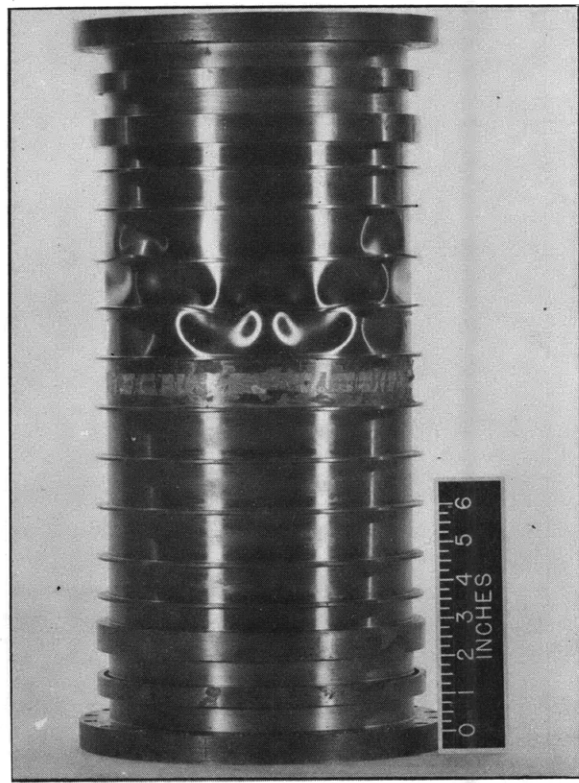
PSD-301949

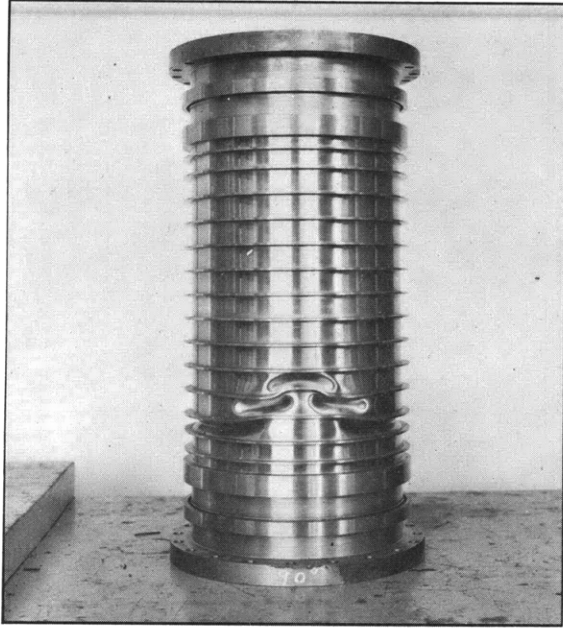Figure 2e - Model 54 ($\lambda = 1.340$)

PSD-91586

Figure 2f - Model 55 ($\lambda = 1.358$)

PSD-301951

Figure 2g - Model 56 ($\lambda = 1.443$)

PSD-301952

Figure 2h - Model 62 ($\lambda = 1.111$)

PSD-94515

Figure 2i - Model 65 ($\lambda = 0.893$)
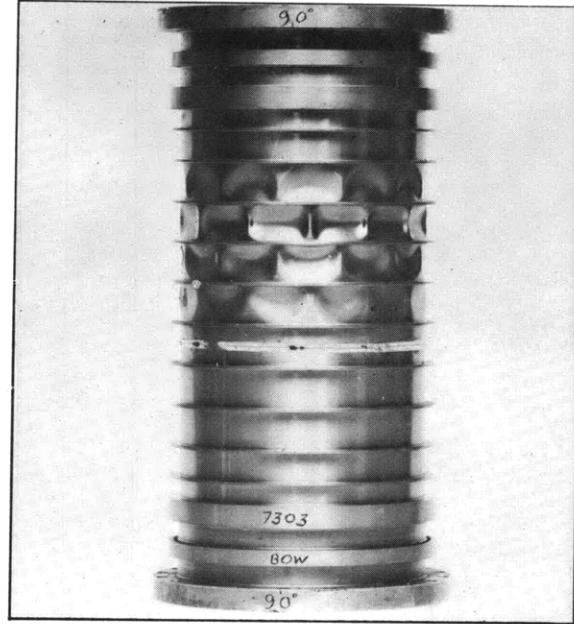
PSD-99758

Figure 2j - Model 73 ($\lambda = 1.671$)

The experimental collapse pressures, modes of failure, and corresponding values of "thinness ratio" λ^{*} for each model are summarized in Table 2.

It is interesting to note that three of the four models which failed in the combined lobar buckling and axisymmetric mode had the lowest values of the thinness ratio λ. The mode of collapse of each model is shown in Figure 2.

A strain-sensitivity factor was determined for each strain gage. This factor is the slope of the linear portion of the pressure-strain curve and is measured in microinches per inch per psi of pressure. Pertinent strain data from gages located in the center region of each model are summarized in Table 3 together with strain-sensitivity factors computed from the theory of Von Sanden and Günther.[6] The corresponding experimental and theoretical strains compare favorably.

$* \quad \lambda = \sqrt[4]{\dfrac{(\ L/2R\ )^2}{(\ h/2R\ )^3}} \cdot \sqrt{\dfrac{\sigma_y}{E}}$

where $L = L_f - b$

9

TABLE 3

Comparison of Measured and Theoretical Strain-Sensitivity Factors
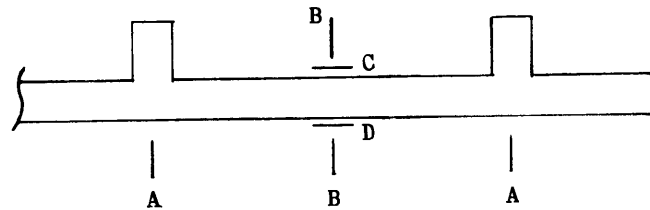
| Model | Factors in μ in/in/psi | | | | | | | |
| | A | | B | | C | | D | |
	Experimental	Theoretical[6]	Experimental	Theoretical[6]	Experimental	Theoretical[6]	Experimental	Theoretical[6]
12	- 1.47(5)*	- 1.56	- 2.18(12)	- 2.17	- 1.57(4)	- 1.48	+ 0.26(4)	0
13	- 1.53(3)	- 1.63	- 1.99(16)	- 2.11	- 1.77(6)	- 1.80	+ 0.16(7)	- 0.06
22	**	- 1.90		- 2.90		- 1.78		+ 0.05
23	- 2.16(5)	- 2.18	- 3.36(18)	- 3.46	- 2.05(6)	- 1.93	+ 0.32(6)	+ 0.02
54	- 2.29(6)	- 2.55	- 3.23(21)	- 3.45	- 3.61(8)	- 3.33	+ 0.15(9)	+ 0.01
55	- 2.10(6)	- 2.36	- 3.28(18)	- 3.51	- 2.76(8)	- 2.53	+ 0.32(9)	+ 0.12
56	- 1.78(5)	- 2.03	- 3.20(20)	- 3.38	- 1.59(8)	- 1.51	+ 0.02(9)	- 0.13
62	- 1.36(6)	- 1.52	- 2.17(19)	- 2.24	- 1.26(6)	- 1.23	+ 0.06(9)	- 0.03
65	- 1.40(3)	- 1.50	- 1.75(9)	- 1.77	- 1.42(2)	- 1.36	0(3)	- 0.19
73	- 1.62(9)	- 1.75	- 4.10(21)	- 4.28	- 1.86(7)	- 1.94	+ 0.24(9)	0

*Numbers in parentheses denote number of strain gages used in averaging process to determine experimental value.

**Strain data not available.

TABLE 4

Theoretical and Experimental Collapse Pressures

Model	Thinness Ratio λ	Experimental Collapse Pressure	Axisymmetric Collapse Pressures				Inelastic Lobar Buckling Pressures[2]	Elastic Lobar Buckling Pressures		
			Von Sanden and Günther Formula [92a][6]	Hencky-Von Mises at Outer Fiber[10]	Plastic Hinge[8]	Inelastic Axisymmetric Buckling[1]		Reynolds[7]	Von Sanden and Tölke[5]	EMB Formula [10][9]
12	0.855	975(BY)*	788	859	946	978	938	2253	2014	1786
65	0.893	1695(BY)	1451	1543	1703	1723	1674	3810	3544	2969
13	0.986	1160(BY)	1104	1149	1301	1325	1203	2598	2190	1807
22	1.035	735(B)	622	686	747	763	734	1203	1054	963
62	1.111	1335(B)	1208	1348	1446	1516	1300	1879	1702	1628
54	1.340	695(BY)	712	719	834	786	[711]**	1009	742	616
23	1.354	705(B)	732	816	878	893	626	811	693	665
55	1.358	730(B)	732	787	877	871	[703]	892	680	651
56	1.443	725(B)	820	931	977	992	[652]	744	665	654
73	1.671	475(B)	644	731	770	780	[381]	460	386	387

*(B) denotes lobar buckling failure; (BY) denotes combination of lobar buckling and axisymmetric mode of failure.

**The theory of Reference 2 indicates that the values in brackets are elastic buckling pressures. In the elastic range, the theory of Reference 2 degenerates to a solution similar to the Von Sanden and Tölke theory.[5]

DISCUSSION AND INTERPRETATION OF RESULTS

Theoretical collapse pressures for various modes of failure for each of the ten models are given in Table 4 in order of increasing thinness ratio λ together with the corresponding experimental collapse pressures and observed modes of failure. The parameters given in Table 1 were used in calculating the collapse pressures in Table 4. The secant and tangent moduli determined from uniaxial stress-strain curves of the material for each of the models were used to calculate the inelastic axisymmetric buckling and asymmetric (lobar) buckling pressures from the theories of References 1 and 2, respectively.

In Table 4, the first three theoretical pressures are for axisymmetric collapse of stiffened cylinders fabricated from an ideally plastic material. Since the geometry and material properties of the models tested were not in this range, these theoretical pressures are given in Table 4 merely for comparison with the theoretical pressures that are associated with the lobar buckling mode of failure. The pressures computed from Formula [92a] are those at which the outer-fiber circumferential stress midway between adjacent frames reaches the strength of the shell material. The Hencky-Von Mises pressure is the pressure at which yielding begins on the outside surface of the shell. It is the pressure at which the biaxial stress, computed by applying the Hencky-Von Mises criterion[10] to the Von Sanden-Günther stresses on the outer fiber of the shell at midbay, reaches the yield strength of the shell material. The "plastic hinge" pressures were computed from a theory,[8] based on the Hencky-Von Mises criterion, which allows for the plastic reserve strength after yielding begins. This plastic hinge analysis assumes that the shell material possesses a stress-strain relationship in compression which consists of a straight line through the origin for the elastic range and a horizontal line for the plastic range, that is, the material is elastic, perfectly plastic. The yield strength necessary for computing these three pressures can only be arbitrarily determined for materials exhibiting a continuous stress-strain curve. Standard procedure at the Model Basin has been to use the 0.2-percent-offset method for determining the yield strength of such materials.

The pressures mentioned so far apply to cylinders which fail due to axisymmetric yielding. It has been found from measurements prior to collapse that this type of failure is usually accompained by large nonlinear strains in the shell of the cylinder. In contrast, for these ten models the non-linear strains measured prior to collapse were relatively small, and the failures occurred rather abruptly. This is indicative of a buckling phenomenon. Therefore, the three theoretical collapse pressures discussed so far do not apply as such to the models presented in this report.

The axisymmetric buckling[1] pressures and asymmetric (lobar) buckling[2] pressures given in Table 4 were computed from analyses which are representative of inelastic modes of failure encountered by ring-stiffened cylinders fabricated from strain-hardening materials. It can be seen that the pressures computed from Reference 1 are higher than those computed from Reference 2. This is not at all suprising since the geometries of the models were in the lobar buckling range ($\lambda > 0.8$). However, excellent agreement was observed between the experimental collapse pressures of Models 12 and 65, the models with the lowest λ values, and the corresponding theoretical collapse pressures computed from Reference 1.

The elastic lobar buckling pressures given in Table 4 as computed from the theories of Von Sanden and Tölke[5] and Von Mises[9] are lower than the corresponding pressures computed from the theory of Reynolds,[7] which is based on more realistic assumptions as mentioned earlier. Both the Von Mises[4] and Von Sanden and Tölke[5] elastic buckling theories are based on the assumption of simple support at the shell edges whereas Reynolds' theory[7] considers the torsional restraint afforded the shell by the reinforcing rings. Further, Reynolds assumed a more representative prebuckling state of stress in the shell, reflecting the reinforcing action of the rings, which leads to a higher critical buckling pressure.

Therefore, to compare the experimental collapse pressures with theory in a logical way, attention must be focused on the inelastic and elastic lobar buckling pressures computed from References 2 and 7, respectively. Table 4 shows that, for stiffened cylinders with λ values from 0.85 to 1.10, excellent agreement was realized between the experimental collapse pressures and the corresponding theoretical pressures computed from Reynolds' inelastic lobar buckling theory.[2] Likewise, for stiffened cylinders with

λ values greater than 1.40, excellent agreement was realized between the experimental collapse pressures and the corresponding theoretical pressures computed from Reynolds' elastic lobar buckling theory.[7] However, only fair agreement with the inelastic theory of Reference 2 was observed for those cases with λ values between 1.10 and 1.40, which are still in the realm of inelastic buckling but which tend toward the elastic range. The fact that this agreement is not as good as for cases where $\lambda < 1.10$ is due primarily to the simplified form of the elastic buckling terms used in developing the inelastic buckling theory of Reference 2. In the limit, the inelastic theory of Reference 2 predicts an elastic buckling pressure which represents an upper bound of the inelastic influences on the buckling process. However, this upper bound or elastic buckling pressure of Reference 2 is much lower than the corresponding elastic pressure computed from Reference 7. Therefore, if more rigorous elastic buckling terms, such as those used in Reference 7, could be incorporated in the inelastic theory of Reference 2, better agreement could be expected between this latter theory and stiffened cylinders with λ values between 1.10 and 1.40; however, this would lead to a very cumbersome and complicated buckling theory.

CONCLUSIONS

1. The prebuckling strains measured on the stiffened cylinder models agreed very well with strains computed from the axisymmetric elastic theory of Von Sanden and Günther.[6]

2. For the stiffened cylinders with λ values ranging from 0.85 to 1.10, excellent agreement was realized between the experimental collapse pressures and the corresponding pressures computed from Reynolds' inelastic lobar buckling theory.[2]

3. For the stiffened cylinders with λ values greater than 1.40, excellent agreement was realized between the experimental collapse pressures and the corresponding pressures computed from Reynolds' elastic lobar buckling theory.

4. For the stiffened cylinders with λ values between 1.10 and 1.40, only fair agreement was realized between the experimental collapse pressures and those calculated from the theory of Reference 2. The disagreement appears to be due to the simplified form of the elastic buckling influences used in developing the theory of Reference 2.

14

ACKNOWLEDGMENT

REFERENCES

1. Lunchick, M.E., "Plastic Axisymmetric Buckling of Cylindrical Shells of Strain-Hardening Materials Subjected to External Hydrostatic Pressure," David Taylor Model Basin Report 1393 (Jan 1961).

2. Reynolds, T.E., "Inelastic Lobar Buckling of Cylindrical Shells under External Hydrostatic Pressure," David Taylor Model Basin Report 1392 (Aug 1960).

3. Yoshiki, M., Yamamoto, Y. and Suzuki, T., "Axisymmetric Collapse of Circular Cylindrical Shells," David Taylor Model Basin Translation T-298 (in preparation).

4. Von Mises, R., "The Critical External Pressure of Cylindrical Tubes under Uniform Radial and Axial Load," Experimental Model Basin Report 366 (Aug 1933).

5. Von Sanden, K. and Tölke, F., "On Stability Problems in Thin Cylindrical Shells," David Taylor Model Basin Translation 33 (Dec 1949).

6. Von Sanden, K. and Günther, K., "The Strength of Cylindrical Shells Stiffened by Frames and Bulkheads, under Uniform External Pressure on All Sides," David Taylor Model Basin Translation 38 (Mar 1952).

7. Reynolds, T.E., "Elastic Lobar Buckling of Ring-Supported Cylindrical Shells under Hydrostatic Pressure," David Taylor Model Basin Report (in preparation).

8. Lunchick, M.E., "Yield Failure of Stiffened Cylinders under Hydrostatic Pressure," David Taylor Model Basin Report 1291 (Jan 1959).

9. Windenburg, D.F. and Trilling, C., "Collapse by Instability of Thin Cylindrical Shells under External Pressure," Transactions, American Society of Mechanical Engineers, Vol. 56, No. 11 (Nov 1934); also Experimental Model Basin Report 385 (Jul 1934).

10. Murphy, G., "Advanced Mechanics of Material," McGraw-Hill Book Co., Inc., New York (1946), pp. 76-84.

INITIAL DISTRIBUTION

Copies

9 CHBUSHIPS
   3 Tech Info Br (Code 335)
   1 Lab Mgt (Code 320)
   1 Prelim Des Br (Code 420)
   1 Prelim Des Sec (Code 421)
   2 Sci & Res Sec (Code 442)
   1 Struc Sec (Code 443)

1 CHONR
   1 Struc Mech Br (Code 439)

1 CDR, USNOL

1 DIR, USNRL (Code 2027)

2 NAVSHIPYD PTSMH

2 NAVSHIPYD MARE

1 NAVSHIPYD CHASN

1 DTMB, UERD (Code 780)

1 SUPSHIP, GROTON

1 Electric Boat Div, General Dynamics Corp

1 SUPSHIP, Newport News

1 NNSB & DD Co.

1 SUPSHIP, Pascagoula

1 Ingalls Shipbuilding Corp

1 CO, USNROTC & NAVADMINU, MIT

1 DIR DEF R & E, Attn: Tech Library

1 O in C, PGSCOL, Webb

1 Prof. N.J. Hoff, Stanford Univ

1 Prof. J. Kempner, Brooklyn Polytechnic Inst

1 Dr. E. Wenk, Jr., White House

1 Dr. R.C. DeHart, Southwest Res Inst

1 Dr. V.L. Salerno, Fairleigh Dickinson Univ

1 Prof. G. Gerard, College of Eng, New York Univ

1 Prof. H. Becker, College of Eng, New York Univ