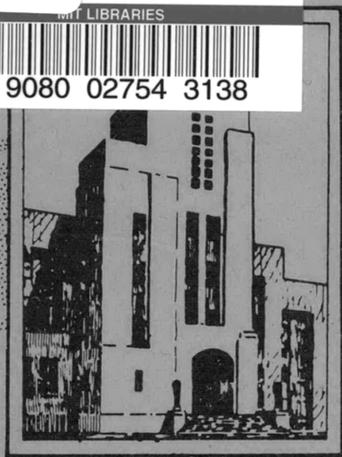
#1



DEPARTMENT OF THE NAVY
DAVID TAYLOR MODEL BASIN

HYDROMECHANICS

○

AERODYNAMICS

○

STRUCTURAL MECHANICS

○

APPLIED MATHEMATICS

YIELD FAILURE OF STIFFENED CYLINDERS UNDER HYDROSTATIC PRESSURE

by

M.E. Lunchick

[redacted]

STRUCTURAL MECHANICS LABORATORY
RESEARCH AND DEVELOPMENT REPORT

January 1959

Report 1291

YIELD FAILURE OF STIFFENED CYLINDERS UNDER HYDROSTATIC PRESSURE

by

M.E. Lunchick

Reprinted from Proceedings of the
Third U.S. National Congress of
Applied Mechanics
held at Brown University
Providence, Rhode Island
June 11-14 1958

January 1959

Report 1291
NS 731-038

YIELD FAILURE OF STIFFENED CYLINDERS UNDER HYDROSTATIC PRESSURE

Experimental results are presented of hydrostatic tests of stiffened cylinders failing by axisymmetric yielding. These results are compared with existing theories predicting the collapse pressure as well as a new method presented in this paper. This method is based on an elastic-plastic analysis of a stiffened cylinder comprised of an ideally plastic material. The experimental collapse pressures agree best with theories using a Hencky-von Mises criterion of yielding and allowing for plastic reserve strength after the initiation of yielding.

M. E. Lunchick
David Taylor Model Basin

Introduction

In recent years there has been an increasing tendency to design stiffened cylinders for carrying external hydrostatic pressure with heavier frames and closer frame spacing than heretofore. As a result these stiffened cylinders have had a greater propensity toward failure by axisymmetric yielding of the shell between frames. To design the structural elements of a stiffened cylinder to withstand shell yielding, the relationship between collapse pressure, geometry, and material properties must first be determined theoretically. If a minimum weight design is desired, the collapse pressure must be predicted accurately. Test results will be presented and compared to various theories to indicate their reliability in predicting failure. In addition, a new procedure for determining collapse will be presented which is considered to improve on current practice.

A Method for Determining the Plastic Reserve Strength of a Stiffened Cylinder

In the Appendix an expression, equation (24), is derived for determining the collapse pressure by axisymmetric yielding of a stiffened cylinder under hydrostatic pressure. Using $k_\phi = \mu k_x$, $\sigma_{b\phi}/\sigma_{m\phi} = 6\beta_\phi$, and $\sigma_{mx}/\sigma_{m\phi} = K_x/K_\phi$, the design curves shown in Fig. 1 are obtained. In these curves the plastic reserve strength expressed as the ratio of collapse pressure to yield pressure P_c/P_y is plotted against the ratio of circumferential bending stress to circumferential membrane stress $\sigma_{b\phi}/\sigma_{m\phi}$ for different values of longitudinal membrane stress to circumferential membrane stress $\sigma_{mx}/\sigma_{m\phi}$.

The collapse pressure of a stiffened cylinder can be determined by applying the curves of Figure 1 to a point on the exterior surface of the shell at midbay. The

(USE)

stress ratios, $\sigma_{b\phi}/\sigma_{m\phi}$ and $\sigma_{mx}/\sigma_{m\phi}$, can be obtained by the theory of Salerno and Pulos [1] or by the theory of von Sanden and Gunther [2], or if the non-linearity between pressure and strain is not appreciable such as occurs for steel cylinders with a yield strength below 60,000 psi. These stress ratios are then used to enter Figure 1 which gives the ratio P_c/P_y. The collapse pressure is obtained when the ratio P_c/P_y is multiplied by the pressure P_y at which yielding initiates at midbay by the Hencky-von Mises criterion.

Comparison of Theory with Experiment

To evaluate the validity and usefulness of any theory, a comparison should be made between predicted values and experimental results. For the geometries of seven stiffened cylinders listed in Table 1, experimental collapse pressures are presented in Table 2, together with

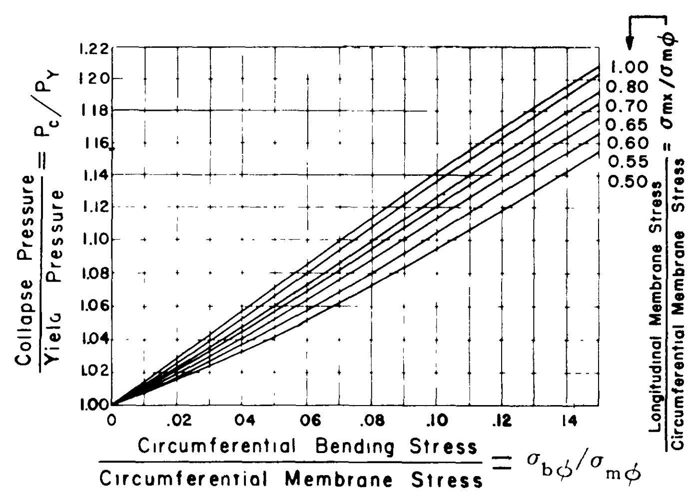


FIGURE 1. DESIGN CURVES FOR DETERMINATION OF PLASTIC RESERVE STRENGTH OF STIFFENED CYLINDERS.

TABLE 1

MATERIAL AND GEOMETRICAL PROPERTIES OF TEST CYLINDERS

Test Cylinder	1	2	3	4	5	6	7
Thinness Ratio*	0.41	0.42	0.46	0.59	0.61	0.67	0.70
Hodge's Parameter, C**	0.932	0.857	0.857	1.388	1.951	1.290	2.077
Ratio of Shell Thickness to Diameter, h/D	0.00880	0.00784	0.00784	0.00599	0.00814	0.00474	0.00581
Ratio of Frame Spacing to Diameter, L/D	0.114	0.095	0.095	0.110	0.202	0.109	0.168
Ratio of Frame Area to Shell Area, A/Lh	0.646	0.745	0.745	0.282	0.288	0.479	0.228
Yield Strength of Shell, σ_y, in psi	46,800	47,500	59,200	44,800	47,100	49,600	47,000

*The thinness ratio is defined as:

$$\lambda = \sqrt[4]{\frac{(L_{eff}/D)^2}{(h/D)^3}} \cdot \sqrt{\frac{\sigma_y}{E}}$$

where Leff is the effective frame spacing or clear length of shell between frames.

**Hodge's parameter C is defined as:

$$C = L_{eff}/\sqrt{Dh}$$

values of pressures predicted by a number of theories as well as the method presented in the Appendix. To facilitate comparison, the ratios of theoretical to experimental collapse pressure are also presented in Table 2.

All cylinders listed were externally framed. Internally framed cylinders were purposely omitted from the tables since they usually have an initial axisymmetric deflection between stiffeners which could lower the collapse pressure. On the other hand, for the externally framed cylinders, the initial axisymmetric deflection increases their strength. In reality, however, this deflection varies over the cylinder so that values close to zero are present; hence, the strength of an externally framed cylinder approaches the strength of a cylinder with no initial deflection. The theories listed in Table 2 neglect the effects of this initial axisymmetric deflection which, however, has been described and its influence analyzed by Lunchick and Short in [3].

Formulas 92 and 92a of von Sanden and Gunther give values on the average 16 percent lower than the experimental collapse pressure. Both formulas result from the solution of the equation of equilibrium for a stiffened cylinder. The solution is derived for the boundary conditions of no rotation of the shell at the frames and for deflections at the frames determined by treating the frame as a ring and satisfying equilibrium of radial forces acting on the frame. Formula 92 gives the pressure at which the longitudinal stress on the *interior* surface of the shell *at the frame* reaches the yield strength of the material; Formula 92a gives the pressure at which the circumferential stress on the *exterior* surface of the shell *at midbay* reaches the yield strength. Both formulas, therefore, determine the *pressures* associated with the onset of yielding using the criterion of maximum normal stress. In addition, both formulas predict a linear relationship between pressure and the deflections and stresses.

The theory of Salerno and Pulos predicts pressures within about one percent of those obtained by Formulas 92 and 92a. This theory differs from that of von Sanden and Gunther in that it accounts for the moments developed in the longitudinal direction due to the pressure on the ends of a closed cylinder. Furthermore, the theory accounts for the influence of the axial stress in the shell tending to expand the frame [4]. The resulting relationship between pressures and stresses is nonlinear by this theory and is considered more exact in describing elastic behavior than that given by von Sanden and Gunther.

When the Hencky-von Mises criterion of yielding is used in conjunction with the von Sanden and Gunther theory at an exterior midbay point, closer agreement occurs with experiment than when the maximum normal stress criterion is used. The values by this method are shown on line 4 of Table 2 and it must be emphasized that the values listed are associated with the onset of yielding. None of the theoretical collapse pressures discussed so far make any allowance for any plastic reserve strength between the initiation of yielding and final collapse.

When a pressure is determined such that the membrane stresses satisfy the Hencky-von Mises criterion, some allowance for the plastic reserve strength is made. These pressures are shown on line 6 of Table 2 and are indicated as being associated with yielding at the *middle surface* of the cylinder at *midbay*. Since the bending stresses above the middle plane are neglected, this pressure is larger than that associated with yielding on the outer surface. The experimental results indicate better agreement when the plastic reserve strength is taken into account in this manner. Agreement between experiment and this method is good (within 9 percent) but the test results indicate that the method tends to predict high.

The agreement between experiment and a theory developed by Hodge [5] is within 7 percent. Hodge's analysis is based on a rigid-plastic material which assumes that the elastic deformations are so small compared to the plastic deformations that they can be neglected. The material is assumed to obey the Tresca or maximum shear stress criterion of yielding. The theory does not account for the elastic deflections of the rings used on the cylinders tested but assumes no deflections at the ends of a cylinder. On the other hand, the theory assumes no rotation (clamped conditions) at the rings as is the case for the actual cylinders.

The method developed in the Appendix predicts collapse pressures within 7 percent for the seven cylinders tested. For four of the seven cylinders agreement by this procedure is better than by the other theories listed. This procedure accounts for the deflection of the stiffening rings and the elastic deformations prior to collapse. In addition, it is based on the Hencky-von Mises criterion of yielding and includes an allowance for the plastic reserve strength.

Wonk?

Conclusions

The experimental data for the range of geometries listed in Table 1 indicate the following:

1. The use of the Hencky-von Mises criterion of yielding rather than the maximum normal stress criterion results in predicted collapse pressures in better agreement with test results.

2. The plastic reserve strength between the initiation of yielding and final collapse can be appreciable and should be taken into account.

3. Despite the discrepancies between the assumptions in Hodge's theory and reality, the agreement between this theory and experiment is good. If the deflection of the reinforcing rings were adequately taken into account in a similar analysis, a more rigorous and reliable analysis than all the other theories presented could result.

4. The theory of satisfying the Hencky-von Mises criterion with the membrane stresses at midbay gives

TABLE 2

COMPARISON OF THEORETICAL AND EXPERIMENTAL COLLAPSE PRESSURES

Line	Test Cylinder		1	2	3	4	5	6	7
1	Experimental Collapse Pressure, psi		1385	1300	1502	700	955	585	610
2	Von Sanden and Gunther Formula 92	Theoretical Pressure, psi	1024	997	1243	655	805	494	584
		Theoretical Pressure / Experimental Pressure	0.739	0.767	0.828	0.936	0.843	0.844	0.957
3	Von Sanden and Gunther Formula 92a	Theoretical Pressure, psi	1179	1054	1314	595	777	514	545
		Theoretical Pressure / Experimental Pressure	0.851	0.811	0.875	0.850	0.814	0.879	0.893
4	Stresses by theory of von Sanden and Gunther with Hencky-von Mises yield criterion at exterior midbay point (rad used from his incl.s. method)	Theoretical Pressure, psi	1220	1114	1388	666	876	560	619
		Theoretical Pressure / Experimental Pressure	0.881	0.857	0.924	0.952	0.917	0.957	1.015
5	Stresses by theory of Salerno and Pulos with Hencky-von Mises yield criterion at exterior midbay point	Theoretical Pressure, psi	1197	1119	1394	660	866	533	605
		Theoretical Pressure / Experimental Pressure	0.864	0.861	0.928	0.943	0.907	0.945	0.992
6	Stresses by theory of von Sanden and Gunther with Hencky-von Mises yield criterion at middle plane and midbay	Theoretical Pressure, psi	1393	1235	1539	721	948	628	662
		Theoretical Pressure / Experimental Pressure	1.006	0.950	1.025	1.030	0.993	1.074	1.085
7	Rigid-plastic analysis by Hodge	Theoretical Pressure, psi	1266	1176	1464	720	927	600	649
		Theoretical Pressure / Experimental Pressure	0.914	0.905	0.975	1.029	0.971	1.026	1.063
8	Theory predicting fully plastic hinge at midbay derived in Appendix	Theoretical Pressure, psi	1351	1210	1510	698	930	600	629
		Theoretical Pressure / Experimental Pressure	0.975	0.931	1.005	0.997	0.974	1.026	1.031

comparison between 6 & 8   -3% -2% -2% -3% -2% -5% -5%

good results. However, the theory tends to predict pressures higher than the experimental pressures.

5. The method developed in the Appendix predicts collapse pressures in good agreement with the test results presented. It is more conservative than the theory based on the yielding of membrane stresses. For this reason it is recommended that the procedure be used in design of stiffened cylinders where geometries are in the range of failure by axisymmetric yielding.

APPENDIX

Derivation of an Expression for the Plastic Reserve Strength

After a review of the literature, it was concluded that any new procedure for predicting the collapse pressure of a stiffened cylinder under hydrostatic pressure should be based on the following concepts. First, the elastic stresses are most accurately described by ~~using the~~ solution ~~of Salerno~~ and Pulos. Second, the criterion of yielding should be that of Hencky-von Mises. Third, any yielding that first begins at the frames is very localized; hence, the shell can be treated as elastic until yielding starts at midbay [6]. In other words, yielding at the frames does not appreciably influence the pressure at which yielding begins at midbay. Finally, the critical condition for collapse exists at midbay [7], and collapse occurs when the load-carrying capacity of the shell ~~is~~ completely exhausted at that location. This last concept is, in reality, an assumption that, ~~by the time a~~ completely plastic hinge develops at midbay, the plastic zones at the frames have markedly progressed from the surface into the thickness of the shell to form plastic hinges. In effect, a three-hinge mechanism of failure is formed [6]. Furthermore, since the stress gradients at midbay are usually low, once plasticity has started there, it spreads rapidly over and through the middle third of the shell between frames. When the plastic zone has spread over a considerable portion of the shell, the load-carrying capacity of the cylinder vanishes.

In order to determine the collapse pressure of a stiffened cylinder, the load-carrying capacity of an element situated at midbay subject to biaxial loading will be derived. An element one unit square will be analyzed to simplify the derivation. It will be loaded by compressive membrane forces N_x and N_ϕ and edge moments M_x and M_ϕ, the subscripts x and ϕ referring to the longitudinal and circumferential directions, respectively.

The forces F and moments M acting on the element will be assumed linear with pressure and can be expressed as follows:

$$N_x = \sigma_{mx} h = K_x ph \qquad (1)$$

$$N_\phi = \sigma_{m\phi} h = K_\phi ph \qquad (2)$$

$$M_\phi = k_\phi p \qquad (3)$$

$$M_x = k_x p \qquad (4)$$

where σ_m denotes membrane stress and K, k are constants of proportionality. In effect, the assumption is that the relationship between the pressure and forces or moments remains the same for the plastic as well as the elastic range as the relationship is very nearly linear before yielding occurs. For the parameter K_x this is known to be true, for its value is $\frac{R}{2h}$ from $N_x = \frac{pR}{2}$ whether yielding proceeds or not. If the pressures in the plastic range are not too much greater (15 percent or less) than the pressure associated with the initiation of yielding, the assumed relationships for K_ϕ, k_x, and k_ϕ are probably good first approximations.

In this analysis the material will be assumed to be an ideally plastic material with a stress-strain relationship in compression consisting of a straight line through the origin for the elastic range and a horizontal line for the plastic range. For an ideally plastic material, the changes in shape of stress distribution with increases in load are shown in Figure 2. Admittedly, the stress distributions in Figure 2 are conjectured, but they are considered reasonable. The stress distribution for complete penetration of plasticity through the thickness of the material is assumed to be that indicated in Figure 2e. Specifically, the tensile stresses in any direction are constant through the thickness, and the compressive stresses, although of different magnitude, are constant as well.

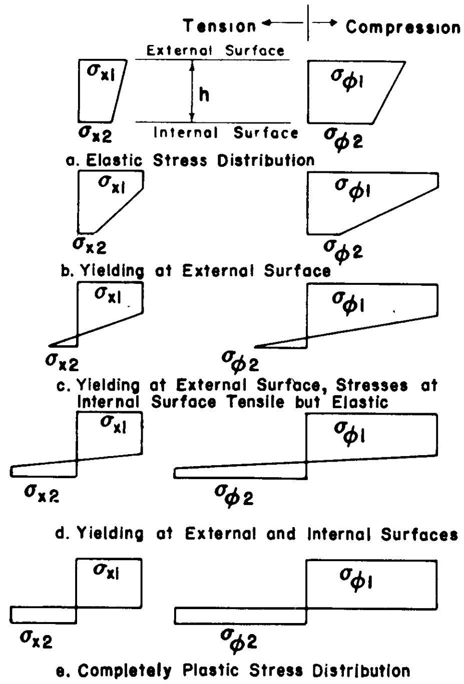


FIGURE 2. PROGRESSIVE CHANGES IN STRESS DISTRIBUTIONS THROUGH THICKNESS OF CYLINDRICAL SHELL.

The pressure associated with the "double-rectangle" stress distributions is the pressure at which a fully plastic hinge develops. This pressure is obtained by satisfying equilibrium of forces in the x and ϕ directions, equilibrium of moments about corresponding axes, and the yield criterion of Hencky-von Mises for the compressive and tensile stresses. The equilibrium expressions on the forces are as follows:

$$\Sigma F_x = 0; \quad \sigma_{x1}\, d - \sigma_{x2}\,(h - d) - K_x ph = 0 \tag{5}$$

$$\Sigma F_\phi = 0; \quad \sigma_{\phi 1}\, d - \sigma_{\phi 2}\,(h - d) - K_\phi ph = 0 \tag{6}$$

where the subscripts 1 and 2 refer to compressive and tensile stresses, respectively. The equilibrium expressions for the moments can be written as:

$$\Sigma M_x = k_x p = \sigma_{x1}\, d\left(h - \frac{d}{2}\right) - K_x p\frac{h^2}{2} - \sigma_{x2}\frac{(h-d)^2}{2} \tag{7}$$

$$\Sigma M_\phi = k_\phi p = \sigma_{\phi 1}\, d\left(h - \frac{d}{2}\right) - K_\phi p\frac{h^2}{2} - \sigma_{\phi 2}\frac{(h-d)^2}{2} \tag{8}$$

The Hencky-von Mises yield conditions on the stresses are:

$$\sigma_y^2 = \sigma_{x1}^2 + \sigma_{\phi 1}^2 - \sigma_{x1}\sigma_{\phi 1} \tag{9}$$

$$\sigma_y^2 = \sigma_{x2}^2 + \sigma_{\phi 2}^2 - \sigma_{x2}\sigma_{\phi 2} \tag{10}$$

where σ_y is the compressive yield strength of the material. When (5) through (10) are solved simultaneously for the pressure, the following expression results:

$$P_c = \frac{\sigma_y}{K_\phi}\frac{\gamma}{\sqrt{4\theta_1 + 4\theta_2\gamma + \theta_4\gamma^2}} \tag{11}$$

where P_c is the ultimate or collapse pressure,

$$\theta_1 = B_\phi^2 - B_x B_\phi \frac{K_x}{K_\phi} + B_x^2\left(\frac{K_x}{K_\phi}\right)^2 \tag{12}$$

$$\theta_2 = B_\phi - \frac{1}{2}(B_\phi + B_x)\frac{K_x}{K_\phi} + B_x\left(\frac{K_x}{K_\phi}\right)^2 \tag{13}$$

$$\theta_3 = \theta_1/\theta_2 \tag{14}$$

$$\theta_4 = 1 - \frac{K_x}{K_\phi} + \left(\frac{K_x}{K_\phi}\right)^2 \tag{15}$$

$$\gamma = \frac{1 - 2\theta_3 + \sqrt{1 + 4\theta_3^2}}{2} \tag{16}$$

$$B_\phi = \frac{k_\phi}{h^2 K_\phi} \tag{17}$$

and

$$B_x = \frac{k_x}{h^2 K_x} \tag{18}$$

Equation (11) expresses the pressure at which a fully plastic hinge develops as a function of the applied loads and the yield strength of the material.

Interest in the collapse pressure P_c centers around the reserve strength it represents above the pressure P_y at which yielding first occurs. The yield pressure P_y can be accurately determined by the theory of Salerno and Pulos. Nevertheless, if the yield pressure P_y is expressed in terms of the same parameters as that for P_c (see (11)), the ratio P_c/P_y of collapse pressure to yield pressure could be established in terms of a given set of loads. This ratio, which will always be greater than one, would give the plastic reserve strength mathematically.

An expression for the yield pressure P_y will now be derived in terms of the loads on the unit square element but with a stress distribution of the form shown in Figure 2a. As indicated in Figure 2a the stress distributions through the thickness of the element are linear and trapezoidal in shape. The yield pressure is that minimum pressure at which the stresses, σ_{x1} and $\sigma_{\phi 1}$, on the compressive fiber satisfy the Hencky-von Mises yield criterion together with the expressions of equilibrium for the forces and moments. The equilibrium expressions of the forces are:

$$\Sigma F_x = 0; \quad \frac{(\sigma_{x1} - \sigma_{x2})}{2}h - K_x ph = 0 \tag{19}$$

$$\Sigma F_\phi = 0; \quad \frac{(\sigma_{\phi 1} - \sigma_{\phi 2})}{2}h - K_\phi ph = 0 \tag{20}$$

For the moments the equilibrium expressions are:

$$\Sigma M_x = k_x p; \quad \sigma_{x2}\frac{h^2}{2} + (\sigma_{x1} - \sigma_{x2})\frac{h}{2}\cdot\frac{2h}{3} - K_x p\frac{h^2}{2} = k_x p \tag{21}$$

$$\Sigma M_\phi = k_\phi p; \quad \sigma_{\phi 2}\frac{h^2}{2} + (\sigma_{\phi 1} - \sigma_{\phi 2})\frac{h}{2}\cdot\frac{2}{3}h - K_\phi p\frac{h^2}{2} = k_\phi p \tag{22}$$

The Hencky-von Mises criterion on the compressive fiber stresses is

$$\sigma_y^2 = \sigma_{x1}^2 + \sigma_{\phi 1}^2 - \sigma_{x1}\sigma_{\phi 1} \tag{9}$$

Solving (19) to (22) and (9) simultaneously for the pressure yields

$$P_y = \sigma_y/\{K_\phi\sqrt{(K_x/K_\phi)^2(1+6B_x)^2 - (K_x/K_\phi)(1+6B_x)(1+6B_\phi) + (1+6B_\phi)^2}\} \tag{23}$$

When (11) is divided by (23), the expression for the ratio of collapse pressure to yield pressure results as follows:

$$\frac{P_c}{P_y} = \left[\frac{1 + 36\phi_1 + 12\phi_2}{1 + 8\phi_1 + 4\sqrt{4\phi_1^2 + \phi_2^2}}\right]^{1/2} \tag{24}$$

where

$$\phi_1 = \theta_1/\theta_4 \tag{25}$$

and

$$\phi_2 = \theta_2/\theta_4 \tag{26}$$

When $k_x = K_x = 0$ and B_ϕ goes to infinity, the case of a beam in pure bending is obtained. For this case, (24) results in $P_c/P_y = 1.5$, or, since moment $= K_\phi p$, then the ratio of moment at collapse to yield moment is 1.5. This value is the theoretically correct one for the case of a beam in pure bending composed of an ideally plastic material [8].

Also, when $k_x = k_\phi = 0$, a state of stress without bending exists. For this case $P_c/P_y = 1.0$, which is to be expected for an ideally plastic material. Thus (24) degenerates to known solutions.

Bibliography

1. Salerno, V. L. and Pulos, J. G., "Stress Distribution in a Circular Cylindrical Shell Under Hydrostatic Pressure Supported by Equally Spaced Circular Ring Frames," Polytechnic Institute of Brooklyn Aeronautical Laboratory Report No. 210, December, 1952.
2. von Sanden, K. and Gunther, K., "Über das Festigkeitsproblem querversteifter Hohlzylinder unter allseitig gleichmässigem Aussendruck," ("The Strength of Cylindrical Shells Stiffened by Frames and Bulkheads under Uniform Pressure on All Sides"), *Werft and Reederei*, Vol. 1, Nos. 8, 9, and 10 (1920), and Vol. 2, No. 17 (1921); also Taylor Model Basin Translation 38 (March 1952).
3. Lunchick, M. E. and Short, R. D., "Behavior of Cylinders with Initial Shell Deflection," *Jour. Applied Mechanics, Trans. ASME*, Vol. 24, 1957, pp. 559–564.
4. Viterbo, F., "Sul problema della robustezza di cilindri cavi rinforzati traversalmente sottoposti da ogni parte a pressione esterna," Vol. IV, July, 1930, pp. 146-456; August, 1930, pp. 531–540.
5. Hodge, P. G., "The Rigid-Plastic Analysis of Symmetrically Loaded Cylindrical Shells," *Jour. Applied Mechanics, Trans. ASME*, Vol. 21, 1954, pp. 336–342.
6. Kempner, J. and Salerno, V. L., "Analysis of the Inelastic Behavior of Transversely Reinforced Cylindrical Shells under Hydrostatic Pressure," Polytechnic Institute of Brooklyn Aeronautical Laboratory Report No. 172, August, 1950.
7. Trilling, C., "The Influence of Stiffening Rings on the Strength of Thin Cylindrical Shells under External Pressure," Taylor Model Basin Report No. 396, February, 1935.
8. Seely, F. S. and Smith, J. O., "Advanced Mechanics of Materials," *John Wiley and Sons*, 2nd Edition, 1952, p. 534.

INITIAL DISTRIBUTION

Copies

15 CHBUSHIPS, Library (Code 312)
- 5 Tech Library
- 1 Tech Asst to Chief (Code 106)
- 1 Prelim Des Br (Code 420)
- 1 Prelim Des Sec (Code 421)
- 1 Ship Protection Sec (Code 423)
- 1 Hull Des Br (Code 440)
- 2 Sci & Res Sec (Code 442
- 1 Structure (Code 443)
- 1 Submarines (Code 525)
- 1 Hull Arrang, Struc, & Preser Br (Code 633)

2 CHONR, Mech Br (Code 438)

1 OPNAV (Op 373)

1 CDR, USNOL

1 DIR, USNRL, Attn: TID

2 NAVSHIPYD PTSMH

2 NAVSHIPYD MARE

1 NAVSHIPYD NORVA
Attn: UERD (Code 270)

2 SUPSHIPINSORD, Groton, Conn.

1 Elec Boat Div, General Dynamics Corp, Groton, Conn.

2 SUPSHIPINSORD, Newport News, Va.

1 NNSB & DD Co, Newport News, Va.

2 SUPSHIPINSORD, Pascagoula, Fla.

1 Ingalls Shipbuilding Corp, Pascagoula, Fla.

1 ASST SECDEF (R & D), Attn: Tech Library

1 CO, USN Admin Unit, MIT, Cambridge, Mass.

1 O in C, Postgrad School, Webb Inst of Tech, Glen Cove, L.I., N.Y.