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DEPARTMENT OF THE NAVY

DAVID TAYLOR MODEL BASIN

HYDROMECHANICS

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AERODYNAMICS

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STRUCTURAL MECHANICS

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APPLIED MATHEMATICS

REINFORCED PLASTICS FOR HYDROSPACE VEHICLES

by [redacted]

J. E. Buhl, Jr., J. G. Pulos, DTMB
and
W. R. Graner, BuShips

[redacted]

RESEARCH AND DEVELOPMENT REPORT

August 1961

Report 1524

DEPARTMENT OF THE NAVY
DAVID TAYLOR MODEL BASIN
WASHINGTON 7, D.C.

IN REPLY REFER TO
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Ser 7-393
12 October 1961

From: Commanding Officer and Director, David Taylor Model Basin
To: Chief, Bureau of Ships (634C) (in duplicate)

Subj: Structural model tests of glass-reinforced plastic cylinders; forwarding of report on

Ref: (a) Fonecon between Messrs. W.R. Graner, BUSHIPS (634C), and J.G. Pulos, DATMOBAS (731) of 20 Feb 1961

Encl: (1) DATMOBAS Report 1524 entitled "Reinforced Plastics for Hydrospace Vehicles" 7 copies

1. The Bureau of Ships (Code 634C) had requested the David Taylor Model Basin by reference (a) to participate at the Filament Winding Symposium of the Society of Aerospace Material and Process Engineers held in Pasadena, California, on 28, 29, and 30 March 1961, to the extent of preparing and presenting a paper on the results of research being carried out in the Navy to study the feasibility of new materials such as glass-reinforced plastics for pressure-vessel application.

2. Background information was presented on the use of glass-reinforced plastics (GRP) for the construction of pressure hulls for deep submergence. The characteristics of an ideal material for hulls for deep-depth operation were described, and available information on structural response to static loading of cylindrical hulls made of GRP was summarized to demonstrate their potential advantages over metals. Plans for future research were reviewed.

3. The paper which was presented is being made available in the form of a Model Basin report because the proceedings of the Conference were not published.

E. E. Johnson

E.E. JOHNSON
By direction

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REINFORCED PLASTICS FOR HYDROSPACE VEHICLES

by

J. E. Buhl, Jr., J. G. Pulos, DTMB

and

W. R. Graner, BuShips

August 1961

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FOREWORD

The material in this report was presented, essentially in the same form, at the Filament Winding Symposium of the Society of Aerospace Material and Process Engineers held in Pasadena, California, on 28, 29 and 30 March 1961. It is made available in this form because the proceedings of the Conference will not be published.

Mr. William R. Graner of the Materials Development and Applications Branch, Hull Division, Bureau of Ships, was the Bureau's cognizant engineer for this program and also contributed to the technical content of this paper.

ABSTRACT

This paper presents background information on the use of glass-reinforced plastics (GRP) for the construction of pressure hulls for deep submergence. The characteristics of an ideal material for hulls for deep-depth operation are described, and available information on structural response to static loading of cylindrical hulls made of glass-filament reinforced plastics is summarized to demonstrate their potential advantages over metals. Work to date indicates that experimentally determined pressures obtained from a limited number of tests of structural models fabricated from commercially available GRP materials agree well with pressures computed from formulas based on thin-shell theory for isotropic materials. Plans for future research are reviewed.

BACKGROUND

The great oceans cover 70 percent of the earth's surface and, as shown in Figure 1, 70 percent of these oceans are between 10,000 and 20,000 ft deep. Hence, in man's quest to explore, utilize, and understand the seas around him, he must develop high-strength hull structures which can withstand the pressures at great depths and still be sufficiently light in weight to carry a practical payload of crew and equipment.

The ideal characteristics for metals for pressure-hull application have been summarized and discussed at some length by Owen and Sorkin.[1] These characteristics, which may be extended to apply to any material, are shown in Table 1.

By virtue of their density, strength, and modulus, glass-reinforced plastics (GRP) appear to have great promise as a structural material for deep-diving submersibles. Furthermore, reinforced plastics can be tailor-made to have high strength in preferred directions so that for this material isotropy is not necessarily a desirable characteristic. Other favorable properties of these nonmetallics are their availability, formability, and nonmagnetic characteristics.

Possible creep under long-term hydrostatic pressure, strength reductions due to cyclic fatigue, and aging are potentially serious shortcomings. There is also a problem with regard to achieving high-strength repairs. The magnitude of these shortcomings can only be established after extensive investigations.

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1 References are listed on page 12.

A survey of the literature showed that there was very little information on the design of shell structures of composite materials, such as GRP, to withstand high external hydrostatic pressures. Indeed, most work has involved comparatively thin-walled containers for internal pressure application. In the areas of interest to the pressure-hull designer, information is virtually nonexistent. Barnet and Lloyd[2] have reported on tests of small-diameter GRP cylindrical and hemispherical shells, some made of glass fabric and others of glass-roving type of reinforcement, which were subjected to moderate external pressures. In these tests collapse strengths were significantly reduced by continuous immersion for periods up to one year. However, it appears fair to state that the materials available at this time are structurally more efficient than earlier materials.

The apparent advantage of using glass-reinforced plastics as the material for cylindrical pressure hulls can be seen in Figure 2 for unstiffened cylinders and in Figure 3 for ring-stiffened cylinders. The curves show the predicted strength-weight characteristics for cylindrical shells of various materials loaded by external hydrostatic pressure. In the construction of the figures the same formulas were used for all materials. These formulas have been proven valid for metals but not, as yet, for anisotropic materials such as reinforced plastics. The extensive calculations required were carried out with the aid of the Model Basin's IBM-704 computer.

The physical properties used in the calculations represent the best materials available today as well as those expected to be available in the near future. It was assumed that the properties were the same in both the circumferential and longitudinal directions.

For the glass-reinforced plastics, various combinations of density, strength, and modulus were assumed. The first combination represented materials similar to those available in 1959, i.e., a density of 0.07 lb/cu in., a strength of 70,000 psi, and a Young's modulus of 4.8×10^6 psi. The other combinations of properties for the plastics reflect material properties expected from higher glass content and the use of glass fibers with higher strength and modulus.

The physical properties of the metals used in obtaining the curves of Figures 2 and 3 include HY-100 (100,000-psi yield strength) steel and the higher strength and lighter weight alloys that could be available in the next few years, i.e., HY-150 and HY-200 steels, 60,000-psi aluminum and beryllium, and 150,000-psi titanium.

Where the yield strength of the material shown in Figure 2 is less than that shown in Figure 3, the modulus of elasticity, as reflected in the buckling pressure, is the controlling factor, and no advantage can be gained by increasing the yield strength. The curve for steel in Figure 2 is a good example of this. In order to have an unstiffened steel cylinder fail by elastic shell buckling over the weight-displacement range considered, it is only necessary to have the yield strength of the steel exceed 36,000 psi.

Beryllium is the only material where failure of an unstiffened cylinder occurs by yielding of the shell over a significant portion of the weight range considered in Figure 2.

The material densities used in computing the weights of the metal cylinders were 0.283 lb/cu in. for steel, 0.160 lb/cu in. for titanium, 0.100 lb/cu in. for aluminum, and 0.0658 lb/cu in. for beryllium.

In Figure 2, for unstiffened cylinders, the nonlinear portion of each curve represents the elastic shell buckling mode of failure.[3] The straight-line portion of each curve represents the pressure at which the stresses reach the strength of the material. For the metal cylinders, failure was assumed to occur when the biaxial "effective stress" determined from the Hencky-Huber-Von Mises theory of failure reached the yield strength of the material. The circumferential and longitudinal membrane stresses, $\sigma_c = pR/h$ and $\sigma_L = pR/2h$, respectively, were used in these calculations. For the reinforced plastics, failure was assumed to occur when the circumferential membrane stress σ_c reached the fracture stress of the material. This latter assumption may be conservative, but it was deemed necessary because of the lack of data on the response of GRP materials when subjected to high compressive stresses, particularly biaxial compressive stresses resulting from hydrostatic pressure.

For the curves in Figure 3 it was assumed that the metal cylinders had T-frames and the reinforced plastic cylinders had rectangular frames. For a given material and a given weight-displacement ratio, the most efficient or optimum design was selected. In other words, it was assumed that the frames failed, either by overall instability of the frame-shell combination or by yielding, at the same calculated pressure at which the shell failed by axisymmetric collapse. An accepted amount of out-of-roundness was assumed in calculating the bending stresses in the frames. The overall length of the cylinder was assumed to be two diameters except where noted otherwise.

The various stages of progressive deformation occurring in a ring-stiffened cylinder under hydrostatic pressure are shown in Figure 4. The axisymmetric elastic deformations and stresses occurring in a ring-stiffened cylinder under hydrostatic pressure can be adequately predicted from the theories of Von Sanden and Günther[4] or Salerno and Pulos.[5] In the analysis of the latter investigators, the destabilizing effect of the "beam-column" action arising from the interaction of longitudinal compression and longitudinal bending has been considered. This effect can influence the critical frame and shell stresses by something on the order of 10 percent or more, in the range of applied pressure and material strengths associated with cylindrical pressure hulls designed for deep submergence. Failure of the shell between frames of the metal cylinders was assumed to occur when plastic hinges had formed at midbay and at the frames so that a three-hinge mechanism, as shown in Figure 4d, occurred. Theory for this mode of collapse has been developed by Lunchick.[6] The failure of the shell of the reinforced plastic cylinders was assumed to occur when either the circumferential stress at midbay or the longitudinal stress at a frame reached the strength of the material; these critically stressed regions are shown in Figure 4c. The failure criterion for metals has been reasonably well confirmed by experiment, but the one for plastics has not, although it is probably conservative.

For the reasons indicated in Figures 2 and 3 the Navy, in searching for high-strength materials for hulls for deep submergence, became interested in glass-reinforced plastics.

DAVID TAYLOR MODEL BASIN TESTS AND RESULTS

It is appropriate at this time to trace some of the history pertinent to the development of current interest within the U. S. Navy in the feasibility of composite materials such as GRP for application to naval structures.

In April 1959 the Zenith Plastics Company of Gardena, California, a division of Minnesota Mining and Manufacturing Company, proposed to the Office of Naval Research the possible application of plastic materials developed by the parent company to a variety of plate and shell structures of interest to the U. S. Navy. Subsequent conferences arranged by the Office of Naval Research and attended by cognizant personnel of Zenith, the Office of Naval Research, the Bureau of Ships, and the David Taylor Model Basin led to the establishment of a joint program of exploratory research to study the feasibility of using such materials in cylindrical shell structures.

A relatively large-size unstiffened cylindrical shell was fabricated by Zenith, at their own expense, and forwarded to the Model Basin for instrumentation and testing.[7] The test cylinder, designated Model ZP-3562, had an inside diameter of 31.06 in., a wall thickness of 1.61 in., and an overall length of 59.0 in. The ends of the cylinder were closed by heavy steel plates machined to fit the inside of the cylinder and held together by a tie rod. The end closure plates reduced the effective length of the test section to 55.0 in. A schematic diagram of the cylinder in the test tank is shown in Figure 5.

The cylinder was fabricated by winding a glass-fiber-reinforced epoxy-resin tape on a mandrel. The shell wall consisted of two layers of tape with the glass fibers oriented in the circumferential direction and one layer of sheets with the glass fibers in the longitudinal direction. This buildup was repeated until the required thickness was obtained.

Nominal mechanical properties of the cured material used in the test cylinder were furnished by the manufacturer to be as follows:

Density γ, 0.070 lb/cu in.

Yield strength in the circumferential direction $\sigma_{y\phi}$, 70,000 psi.

Modulus of elasticity in the circumferential direction E_ϕ, 4.8×10^6 psi

Poisson's ratio ν, 0.1

The cylinder was instrumented with two-element strain gage rosettes on the inside and outside surfaces of the shell at midlength and along one generator. The cylinder was subjected to external oil pressure to a maximum of 3000 psi. After completion of tests at the Model Basin, the cylinder was loaded to failure, using water pressure, in the Naval Research Laboratory 4-ft-diameter test tank.

Failure occurred at a pressure of 3735 psi after the cylinder had sustained a pressure of 3750 psi for approximately 2 min. When failure occurred, the cylinder split along one generator with some circumferential tearing at midlength and considerable delamination in the wall thickness as shown in Figure 6.

The strain-sensitivity factors at midlength were used in conjunction with the membrane stresses for an unstiffened tube to compute moduli of elasticity of 4.85×10^6 psi and 3.44×10^6 psi in the circumferential and longitudinal directions, respectively. These moduli were then "weighted" in the same two-to-one ratio as the fiber distribution and an "effective modulus" of 4.38 X 10^6 psi was found. This effective modulus and an effective length of 55.0 in. were then used to compute an elastic lobar buckling pressure of 3674 psi. This value is 1.7 percent less than the experimental collapse pressure of 3735 psi. Additional test details and results can be found in Reference 7.

The excellent agreement of the experimental collapse pressure with the computed buckling pressure and the high strength-to-weight ratio realized from this test were sufficiently encouraging to warrant the establishment of an extensive research program by the Bureau of Ships to investigate the use of GRP materials for naval structures.

The program was established to investigate commercially available materials and winding techniques by testing simple unstiffened cylinders in a manner similar to that used for Model ZP-3562. The testing of a limited number of ring-stiffened cylinders was also included in the initial phases of the program to investigate the applicability of design formulas and theoretical analyses developed for isotropic materials to GRP materials.

To minimize costs and to be able to utilize available test facilities with working pressures in excess of 3000 psi, small-scale structural models with an inside diameter of approximately 6 in. were designed. The unstiffened models contemplated for the initial phase of the investigation are listed in Table 2. The dimensions were selected to cover the range of thicknesses considered to be of interest. The letter prefixed to the model number indicates the manufacturer and the technique employed in winding these cylindrical models.* The B-models are filament-wound in five discrete layers, of which three have circumferential fibers and two have longitudinal fibers, producing a 3C:2L fiber distribution through the shell thickness. The G-models are filament-wound with complete dispersion of fibers to have a 2C:1L buildup. The H-models are also filament-wound but have a buildup of one circumferential layer and two layers helically wound at a 45-deg angle. The Z-models are wound with a glass-fiber-reinforced epoxy-resin tape; each of these cylinders is built up in a sequence of two circumferential plies and one longitudinal ply repeated to produce a 2C:1L distribution of fibers completely dispersed through the thickness.

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*The cylinders were fabricated at the request of the Model Basin. The participating manufacturers were chosen on the basis of available information and inquiries, conducted at the request of the Bureau of Ships, to establish willingness to participate in the program.

Six of the models listed in Table 2 have already been tested to collapse under short-term loading.* The models after failure are shown in Figures 7 through 12. All models were instrumented at midlength on both the inside and outside surfaces with two-element strain gage rosettes. The test procedure used was the same as that for ZP-3562.[7] The measured circumferential and longitudinal strains were used in conjunction with the membrane stresses to compute two values of elastic modulus. An "effective modulus" was then determined and used to predict the elastic buckling pressure. The buckling pressures thus determined and the pertinent test results are given in Table 3. The corresponding information for Model ZP-3562 is included for comparison.

An examination of Table 3 reveals that for the seven models tested under short-term loading, the experimental collapse pressures agree within 5 percent with the calculated pressures except in the case of Model H-1. Model H-1 failed at a pressure 26 percent below that calculated from the elastic buckling equation and appears to have collapsed prematurely either by virtue of its helical construction or some possible flaws in the cylinder. Some additional tests will be carried out to check these possibilities further.

Model Z-2, a duplicate of Z-1, was subjected to long-term loading at various pressure levels to investigate the creep characteristics and their cumulative effect on static collapse strength of GRP cylindrical shells. The highest pressure sustained was 3400 psi for 26 days. At this time the O-ring seal in the test tank failed, causing the pressure to drop to zero. After the O-ring was replaced, the pressure was again raised to 3400 psi. On this last pressure loading it was observed that the strain rate was greater than that on the previous loading to 3400 psi. The model sustained the 3400 psi for 27 hr before collapse occurred. The mode of collapse was similar to that of Model Z-1 except that some circumferential tearing occurred near the ends; see Figure 13.

Strain-time histories recorded from circumferential gages near the region of failure of Model Z-2 are shown in Figure 14. An examination of these curves reveals that the "cycling" resulting from the loading sequence of five pressure runs to collapse resulted in significant changes in the straining. These changes are indicative of a reduction in the material properties, principally modulus, possibly resulting from either creep inherent in this material or mechanical breakdown of the bond between the fibers and the resin. The reduction in moduli resulting from the long-term loading is also reflected in the loss of ultimate load-carrying capacity shown in Table 3.

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*By short-term loading is meant that a total period of about 10 hr was used to reach collapse. This included three pressure runs, each starting from zero load.

In addition to these tests of unstiffened cylinders, two ring-stiffened cylinders, Models ZR-1 and ZR-2, were designed according to the criteria used in developing the curves of Figure 3. These two models had approximately the same weight-displacement characteristics as the unstiffened cylinders ZP-3562 and Z-1. Model ZR-1 had a 2C:1L shell construction, whereas Model ZR-2 had a 1C:1L shell construction. However, both had the same geometry and, in particular, both had end frames which had three times the cross-sectional area of a typical frame. Model ZR-1 failed at a pressure of 5400 psi; ZR-2 failed at 6000 psi. The models after failure are shown in Figures 15 and 16, respectively. It is significant to note that both cylinders failed near the end regions, indicating that the full strength of the typical portions was not realized. The design collapse pressure was on the order of 6500 to 7000 psi. The apparent reduction in strength can be attributed to the rather rigid end frames incorporated in the design. It will be discussed further in the next section.

MODEL TESTS BY ZENITH PLASTICS COMPANY*

The Zenith Plastics Company has fabricated and tested, at their own expense, 38 unstiffened cylinders of approximately 6 in. inside diameter. The cylinders were fabricated from various glass-fiber-reinforced epoxy-resin tapes manufactured by their parent company. The collapse pressures of the cylinders having a 2C:1L buildup and a wall thickness of approximately 0.31 in. are shown in Figure 17. Curves of the elastic buckling pressure computed for assumed Young's moduli of 4×10^6 psi and 5×10^6 psi are shown for comparison. The large spread in experimental collapse pressures can generally be attributed to variations in the modulus of elasticity due to the glass content of the cylinders. The glass content ranged from about 69 percent to 84 percent; the cylinders with the higher glass content had the higher collapse pressures.

The Zenith Plastics Company has also fabricated and tested, at their own expense, nine ring-stiffened cylinders. Six of these models had the same typical geometry as Models ZR-1 and ZR-2. In these six tests the type of material was varied as were the end conditions and the fiber distribution through the shell thickness. Four of the models had shells with a 2C:1L fiber distribution; two with a 1C:1L fiber distribution. The rectangular ring stiffeners were machined from tape-wound tubing which had from 90 to 100 percent circumferential fibers. The highest collapse pressure realized was

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*This information is presented here with the generous permission of the Zenith Plastics Company. The results given were obtained during research conducted by Zenith as part of a no-cost arrangement with the Bureau of Ships and in collaboration with the David Taylor Model Basin.

7300 psi for a cylinder with a 2C:1L shell; the next highest collapse pressure was 6500 psi for a cylinder with a 1C:1L shell. Both of these cylinders had end rings with cross-sectional areas twice the area of a typical stiffener.

The typical center portion of these models was designed by the Model Basin assuming the material had a strength of 70,000 psi and a Young's modulus of 4.8×10^6 psi. It is estimated that the material in the shells of the models that failed at 7300 psi and 6500 psi had a strength of approximately 90,000 psi and a Young's modulus of approximately 6×10^6 psi and that the rings had a strength in excess of 100,000 psi and a modulus of approximately 9×10^6 psi.

DISCUSSION AND CONCLUSIONS

It should be emphasized that the GRP materials presently being investigated as part of the Model Basin program were not developed for the specific purpose of pressure vessels subjected to high external pressure. Thus they are not representative of the ultimate composite material for such application.

The strength-weight curves of Figure 3 for the GRP materials are based essentially on the ultimate strength of commercially available filament-wound GRP materials. This would seem to be reasonable since the curves for the other materials were based on yield strength and on an allowance for the plastic reserve strength, and GRP does not have a yield point. There is concern, however, about the effects of cyclic fatigue and long-term exposure to deep-submergence pressures on the load-carrying capacity of filament-wound structures and consequent limitations in allowable design stresses. Much work must be done to determine the design limitations of filament-wound structures subjected to deep-submergence pressures.

Nevertheless, the general information now available on various GRP materials subjected to long-term and cyclic loading gives ample reason for vigorous prosecution of research and development programs to improve the characteristics of these materials. For example, there has been a considerable amount of information obtained on the effect of cyclic fatigue (tension-compression) on various laminates under different conditions. Such tests indicate that there may be more than a 50-percent reduction in strength (based on short-term compression strength) after 50,000 cycles. With regard to long-term loading, the Forest Products Laboratory[8] has reported failures of unidirectional glass-reinforced epoxy laminates after immersion in water under flexural load equivalent to 56 percent of the short-time flexural strength for 1000 hr. It is obvious that such reductions will greatly reduce the advantages of GRP materials for pressure hulls.

In seeking to obtain improved laminate characteristics, it is important that the interface between the resin and the glass filaments be considered as a system. To date, it appears that the improvement of resins has received too little attention, possibly for lack of new or original approaches to the problem. Although it has not definitely been established which particular characteristics it would be most important to improve for the particular application to deep-submergence hulls, higher compressive strength and modulus, greater affinity of resin to reinforcement, and improved resistance to shear and delamination seem necessary. The resin must provide greater stability to the glass filament and greater resistance to catastrophic failure.

Much work has already been done in the area of improved glasses and finishes, but specific resin-glass systems must be evaluated extensively before the value of these new reinforcements can be appraised. Certainly much more effort should be made to improve reinforcement.

Because the modulus of elasticity of GRP is low compared with that of steel, the possibility of sandwich-type construction to increase the cross-sectional stiffness is also intriguing and deserving of investigation. Here the strength characteristics of available core materials are a limitation. Lightweight, high-compressive-strength core materials resistant to water penetration under high hydrostatic pressures are needed.

Confidence levels must be established to give the designer assurance that the materials in the hull structure will perform in accordance with established design criteria. Quality control practices and inspection test techniques must be devised to give this assurance.

Hull form is another factor which requires investigation. The thought has been widely advanced that shapes other than cylindrical with spherical ends might provide improved resistance to external pressures. Various unconventional, complex configurations have been suggested. Such shapes would be extremely difficult if not impossible to achieve with the high-strength metals but could be fabricated easily using GRP materials.

The Bureau of Ships is supporting material research efforts in some of these areas. The Material Laboratory of the New York Naval Shipyard has been investigating the basic mechanisms of failure for glass-reinforced laminates. It is hoped that from this effort may come better direction and guidance for future materials research programs. The Laboratory has also been working on development of test methods that will provide standards for evaluation of materials and relate these to the type of loading conditions experienced under external hydrostatic pressure. Inspection and nondestructive test techniques are also being investigated.

A program to study resin-glass reinforcement systems under carefully controlled laboratory conditions is being sponsored by the Bureau of Ships to develop materials which have improved compressive strength and modulus.

The David Taylor Model Basin is conducting a broad investigation of filament-wound cylindrical and hemispherical shell structures made of commercially available GRP materials to determine response to both static and dynamic loading and to obtain design information on such details as methods of stiffening, openings, closures, and joints.

It is anticipated that research and development in the areas mentioned will result in significant improvements, not only in filament-wound GRP materials but also in the field of general-purpose laminates as well. At the present time, these materials are competing against the high-strength metal alloys on the basis of superior strength-weight properties. Ultimately, it can be expected that GRP materials will be developed which will demonstrate strength and stiffness characteristics equal or superior to those of the metals.

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TABLE 1

Ideal Characteristics of Materials
for Hydrospace Vehicles

Property	Description
Density	As low as magnesium (0.065 lb/cu in.).
Yield Strength	300,000 psi.
Moduli	Tension and compression 40 X 10^6 psi. Shear 15 X 10^6 psi.
Toughness	Will not fracture in a brittle manner under severe plastic deformation at 0^oF or lower.
Weldability	95-percent joint efficiency or greater in as-welded condition for yield strength, toughness, and fatigue strength.
Formability	Hot-formed or cold-formed to shape without the necessity for subsequent heat treatment.
Repairability	Weld repairable under service conditions.
Fatigue	Notched fatigue strength of welded structure at least 90 percent of unnotched fatigue strength of basic material for low-cycle, high-stress, and plastic-strain conditions wherein number of peak stress cycles is 20,000 or less.
Corrosion	Not susceptible to stress corrosion. Corrosion fatigue strength equal to fatigue strength in air.
Stability	Will not creep or change dimensions significantly under operating stress (75 percent of yield strength).
Isotropy	Mechanical properties identical in any plane.

TABLE 2

Unstiffened Cylinder Models Planned for Glass-Reinforced Plastics Program

Model	Inside Diam. in.	Shell Thick in.	Length in.	Exper. Collapse Pressure psi	Material Density lb/cu in.	Shell Build-Up	Test
B-1	5.987	0.319	11.47	5000	0.078	3C:2L	Short Term
B-2	5.986	0.320	11.40		0.078	3C:2L	
B-3	5.984	0.178	11.43	1225	0.0752	3C:2L	Short Term
B-4	5.983	0.309	11.46		0.0779	3C:2L	
B-5	5.977	0.384	11.46		0.0787	3C:2L	
B-6	5.982	0.457	11.28		0.0780	3C:2L	
G-1	6.	0.180	11.4			2C:1L	
G-2	6.	0.308	11.4			2C:1L	
G-3	6.	0.380	11.4			2C:1L	
G-4	6.	0.455	11.4			2C:1L	
H-1	5.940	0.197	11.40	910	0.0751	1C:2-45^{o}	Short Term
H-2	5.943	0.172	11.41		0.0755	1C:2-45^{o}	
H-3	5.938	0.308	11.41			1C:2-45^{o}	
H-4	5.936	0.305	11.40			1C:2-45^{o}	
H-5	5.938	0.378	11.40		0.0763	1C:2-45^{o}	
H-6	5.937	0.377	11.42		0.0761	1C:2-45^{o}	
H-7	5.937	0.452	11.40			1C:2-45^{o}	
H-8	5.938	0.451	11.40			1C:2-45^{o}	
Z-1	6.012	0.313	11.39	3550	0.068	2C:1L	Short Term
Z-2	6.012	0.313	11.39	3400	0.068	2C:1L	Creep
Z-3	5.994	0.373	11.40	7900	0.0781	2C:1L	Short Term
Z-4	5.997	0.380	11.40		0.0773	2C:1L	
Z-5	6.001	0.450	11.40		0.0754	2C:1L	
Z-6	5.998	0.449	11.40		0.0749	2C:1L	
Z-7	5.995	0.180	11.40	1325	0.0752	2C:1L	Short Term

TABLE 3

Summary of Results of Tests of Unstiffened Cylinders

Model Number	B-1	Z-1	ZP 3562	Z-3	Z-7	H-1	B-3	Z-2 Run 2	Z-2 Run 3
Circumferential Strain μ in/in/psi									
Outside	1.53	2.07	1.69	1.15	2.61	3.12	2.81	1.78	2.01
Inside	1.75	2.40	2.04	1.35	2.79	3.31	3.16	1.96	2.21
Average	1.640	2.235	1.815	1.252	2.701	3.213	2.985	1.870	2.110
Longitudinal Strain μ in/in/psi									
Outside	0.627	1.116	1.14	0.658	1.01	1.24	0.95	0.87	0.93
Inside	0.671	1.182	1.18	0.668	1.03	1.29	0.92	0.84	0.94
Average	0.649	1.149	1.160	0.663	1.024	1.267	0.934	0.855	0.935
Membrane Stresses, psi/psi									
Circumferential	9.884	10.105	10.146	8.535	17.156	15.584	17.309	10.105	10.105
Longitudinal	4.942	5.053	5.073	4.267	8.578	7.792	8.655	5.053	5.053
Modulus of Elasticity psi x 10^6									
Circumferential	5.57	4.10	4.85	6.17	5.87	4.48	5.41	4.94	4.39
Longitudinal	5.40	3.82	3.44	4.90	5.87	4.35	6.12	4.35	3.95
Effective	5.50	3.99	4.38	5.75	5.87	4.44	5.65	4.74	4.24
Collapse Pressure, psi									
Computed Elastic Buckling, p_c	4890	3407	3674	7565	1281	1225	1199	4046	3621
Experimental, p_e	5000	3550	3735	7900	1325	910	1225		3400
Ratio of Collapse Pressures $\frac{p_e}{p_c}$	1.022	1.042	1.017	1.044	1.034	0.743	1.022		

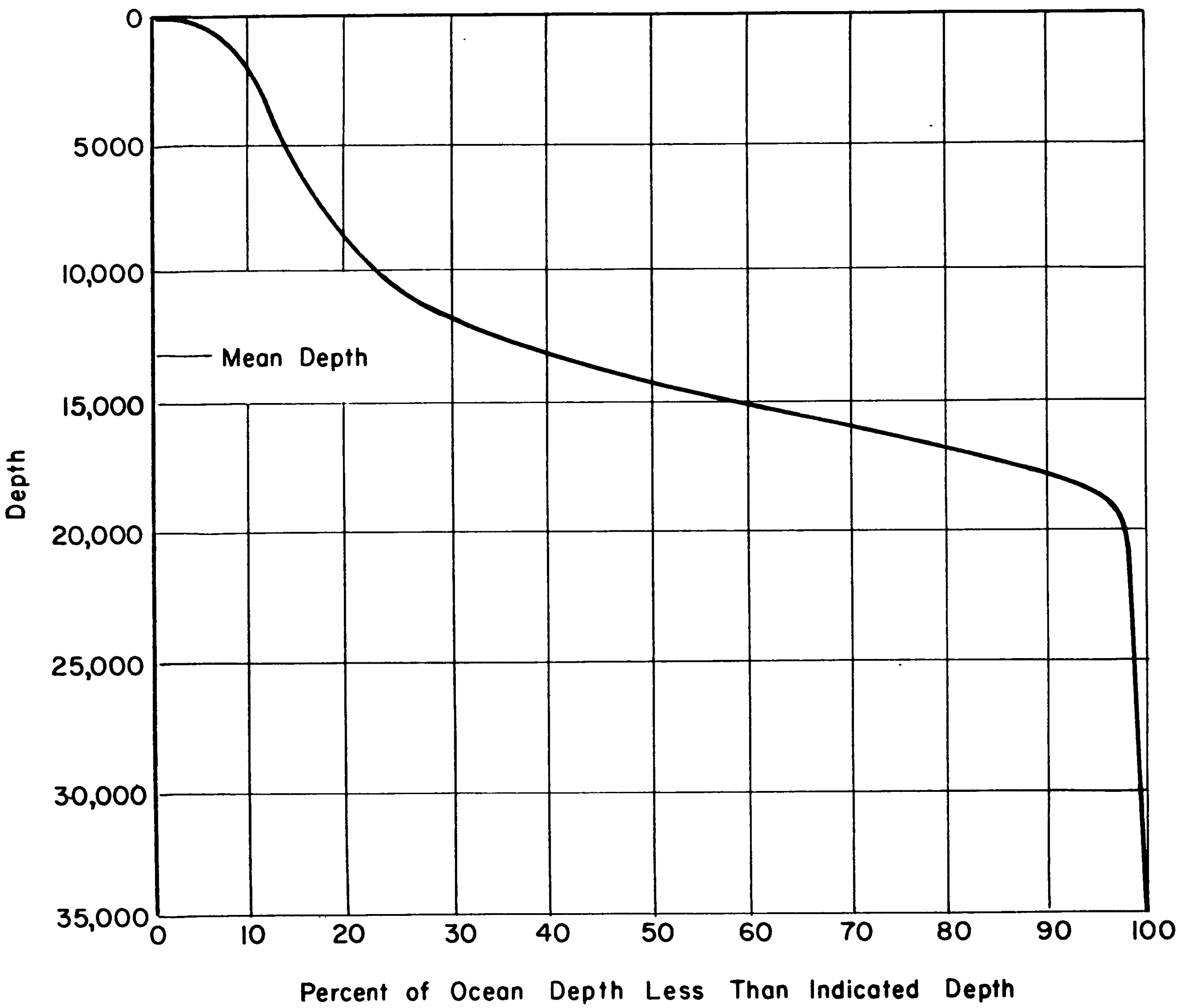


Figure 1 - Percentage Distribution of Depth of Oceans

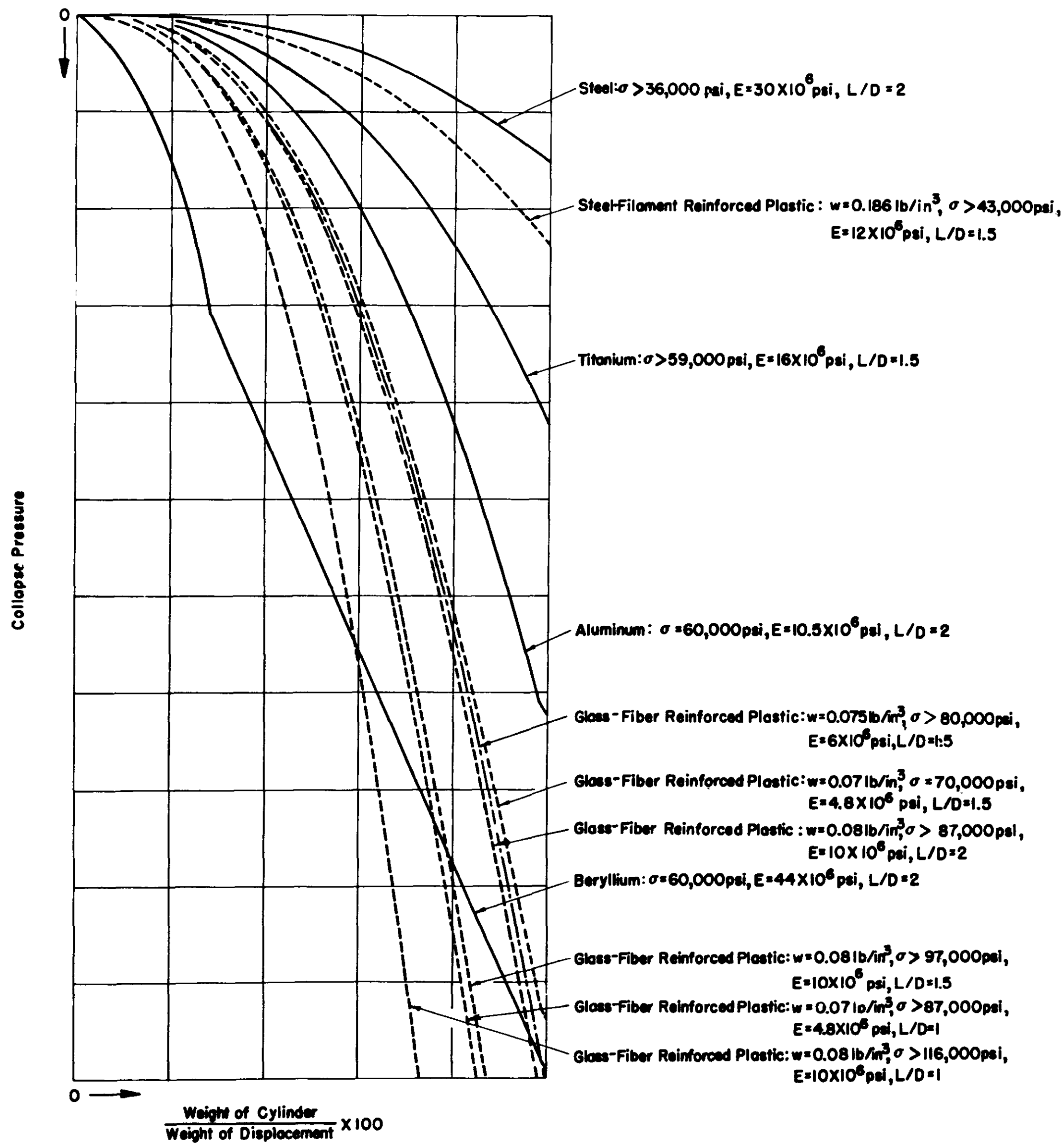
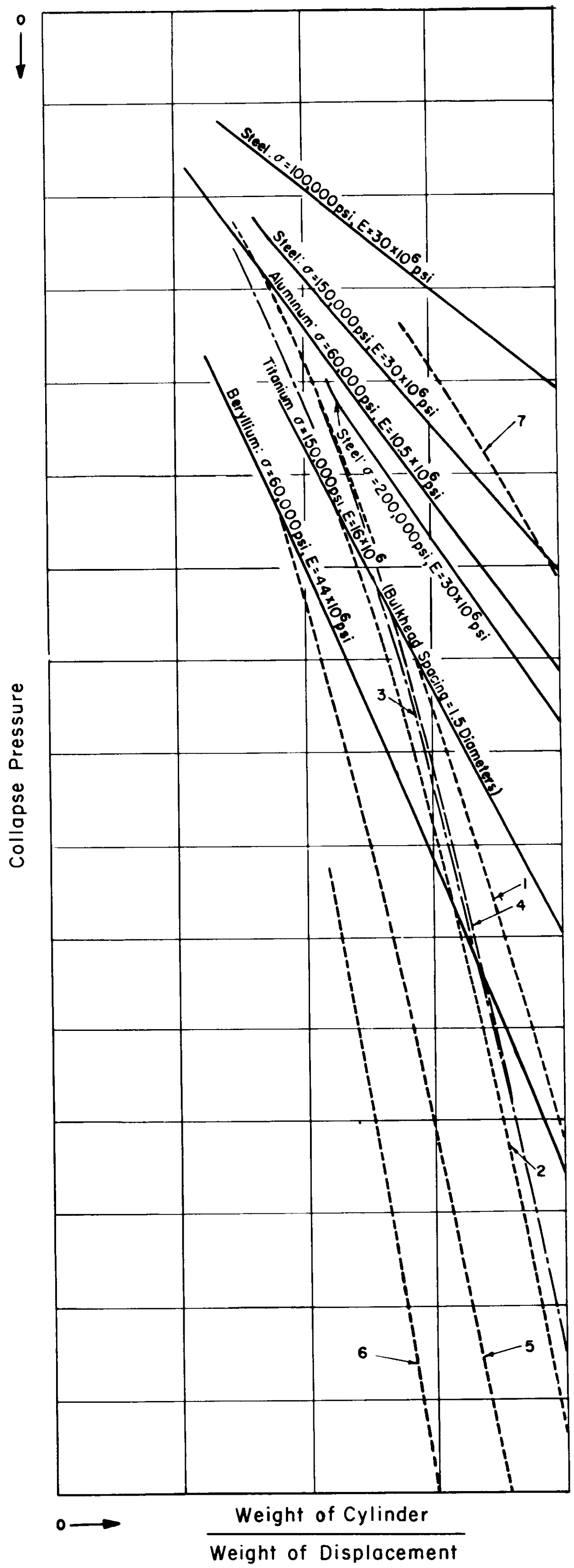


Figure 2 - Strength-Weight Characteristics of Unstiffened Cylindrical Pressure Vessels



Fiberglass Characteristics

Curve	Material	Density, lb/in^3	σ, psi	E, psi	Bulkhead Spacin / Diameter
1	Glass-Fiber Reinforced Plastic	0.07	70,000	4.8×10^6	1.5
2		0.07	100,000	4.8×10^6	1.0
3		0.075	90,000	6.0×10^6	1.5
4		0.08	100,000	10×10^6	2.0
5		0.08	150,000	10×10^6	1.5
6		0.08	200,000	10×10^6	1.0
7	Steel-Filament Reinforced Plastic	0.186	150,000	12×10^6	1.5

Figure 3 - Strength-Weight Characteris of Ring-Stiffened Cylindrical Pressure Vessels

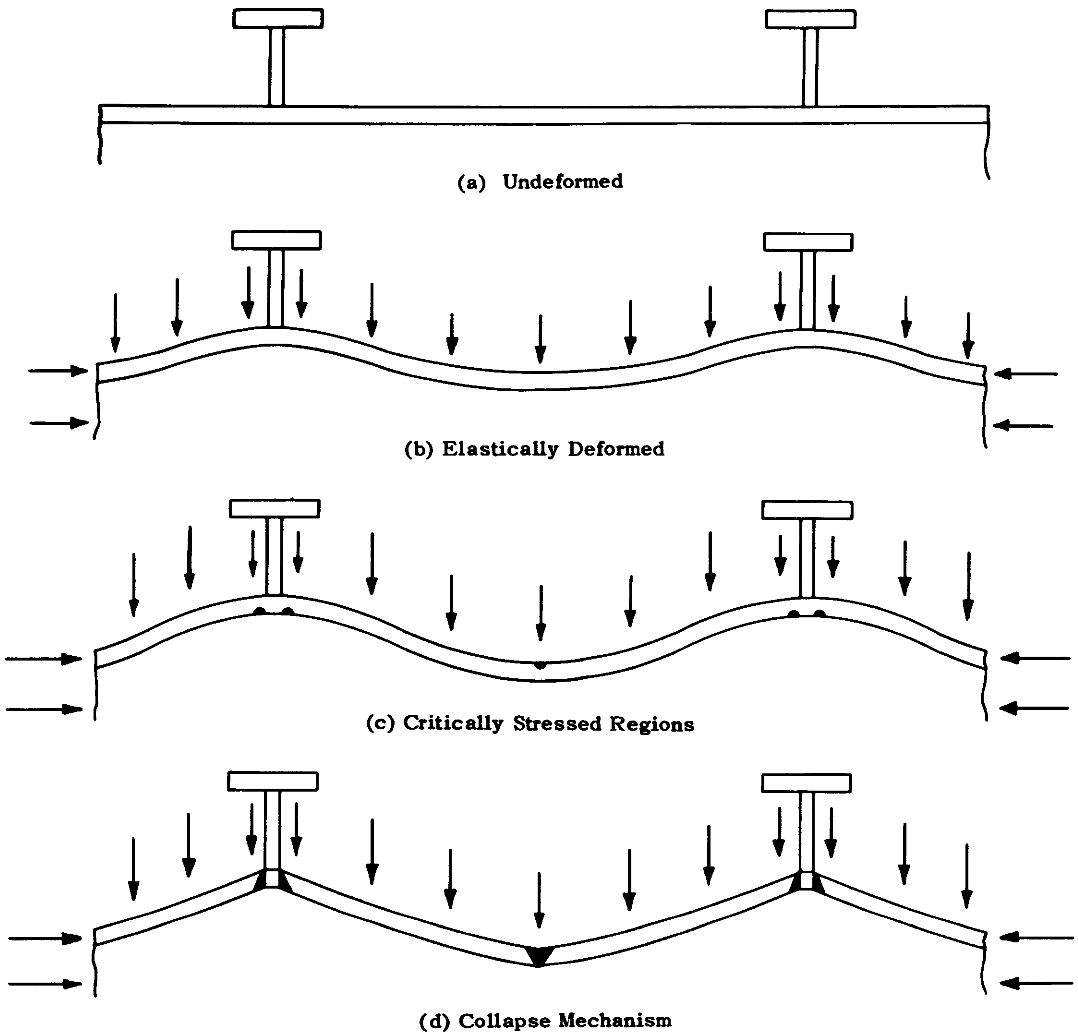


Figure 4 - Mechanism of Failure for Ring-Stiffened Cylindrical Pressure Vessels

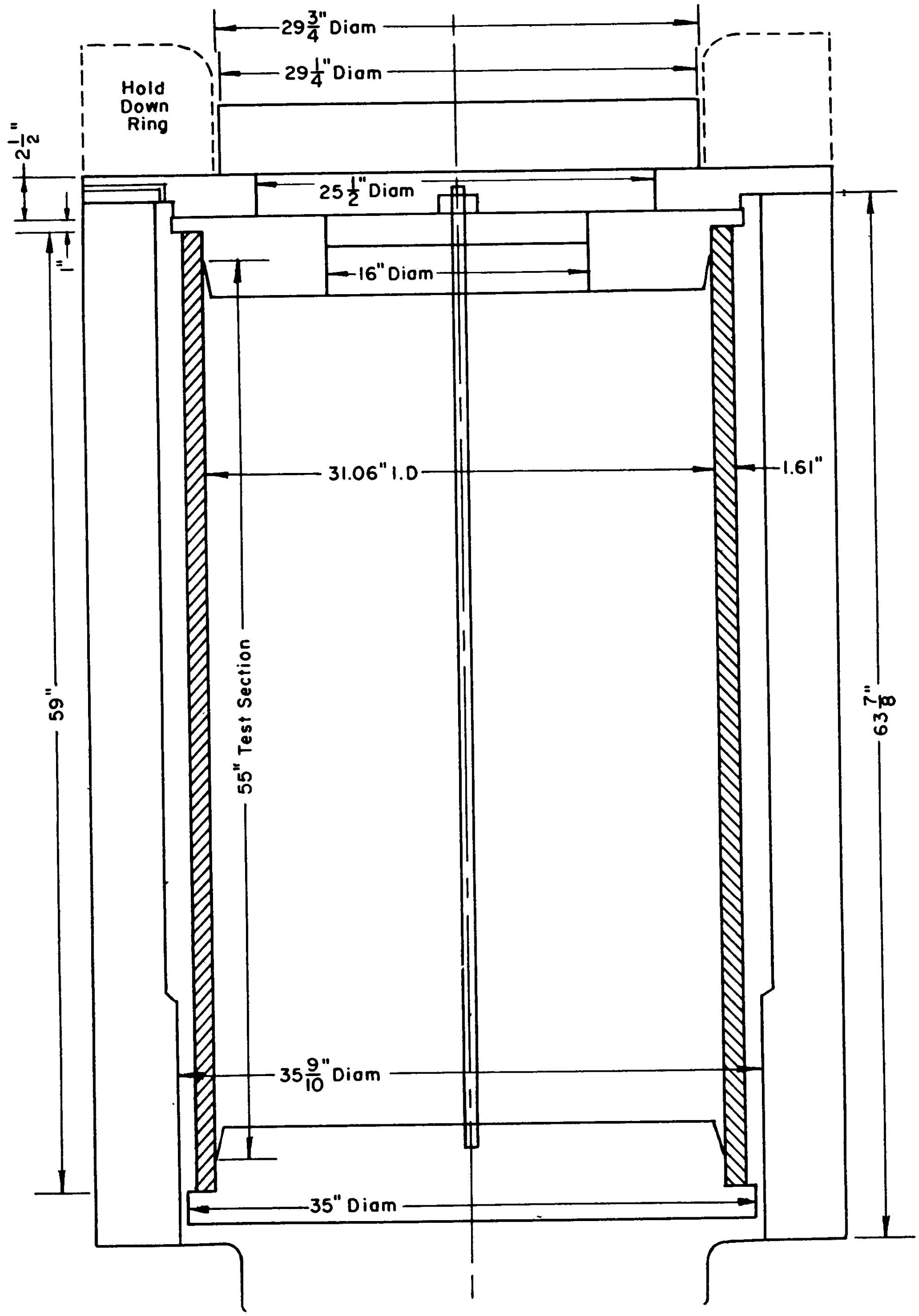


Figure 5 - Schematic Diagram of Model ZP-3562 in Test Tank

Figure 6 - End View of Model ZP-3562 after Failure

Figure 7 - Interior View of Model Z-1 after Failure

Figure 8 - Interior View of Model B-1 after Failure

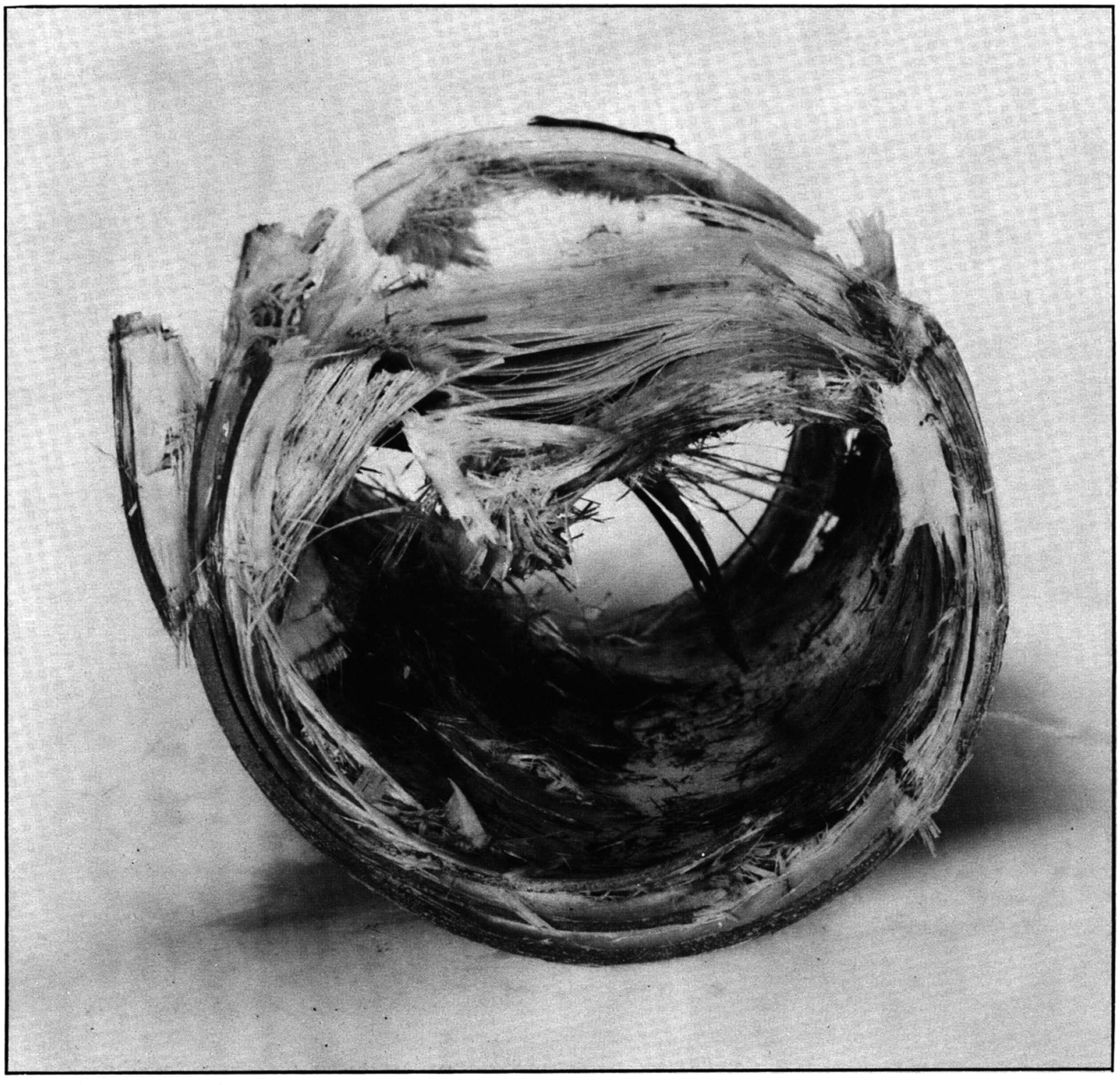


Figure 9 - End View of Model Z-3 after Failure

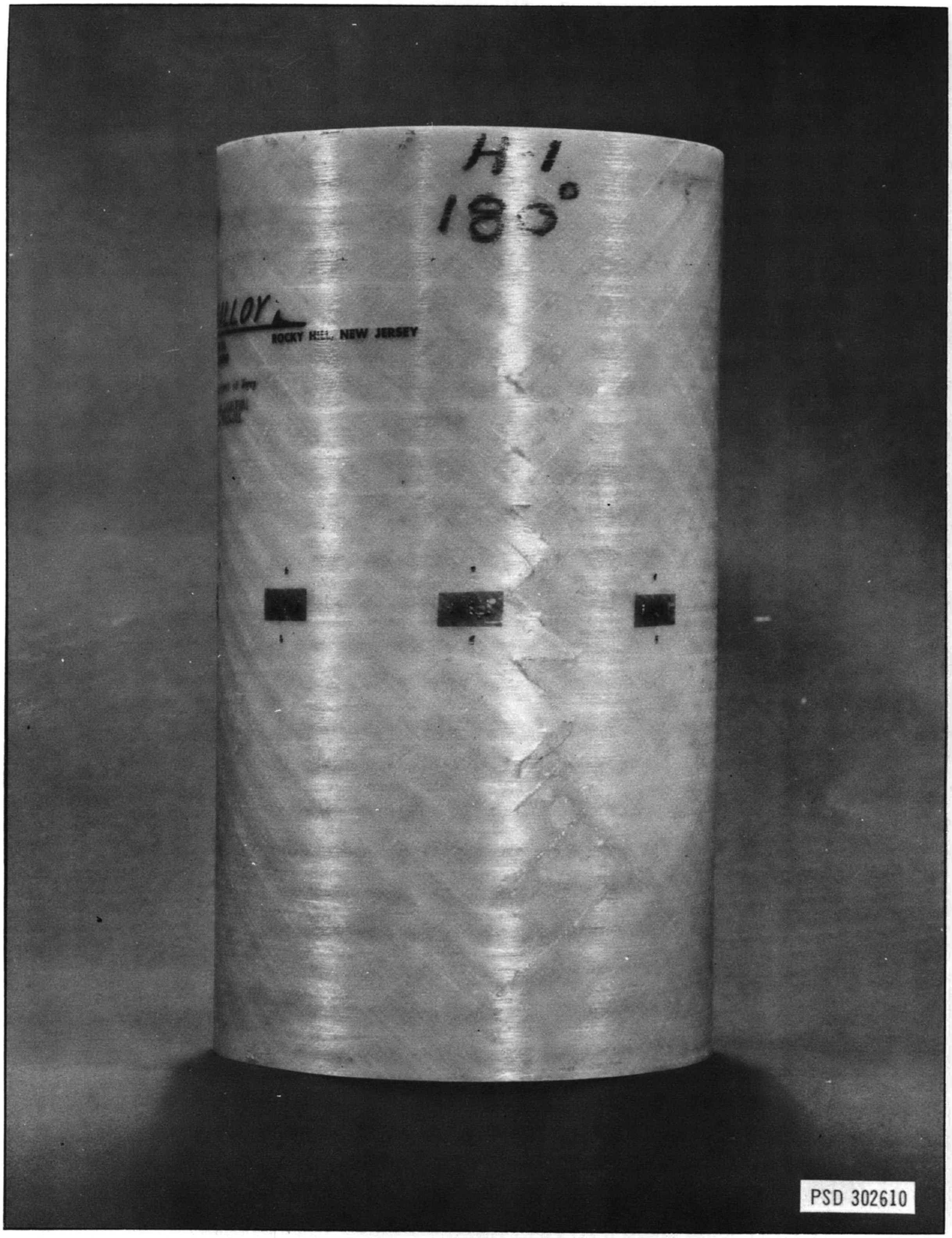


Figure 10 - Exterior View of Model H-1 after Failure

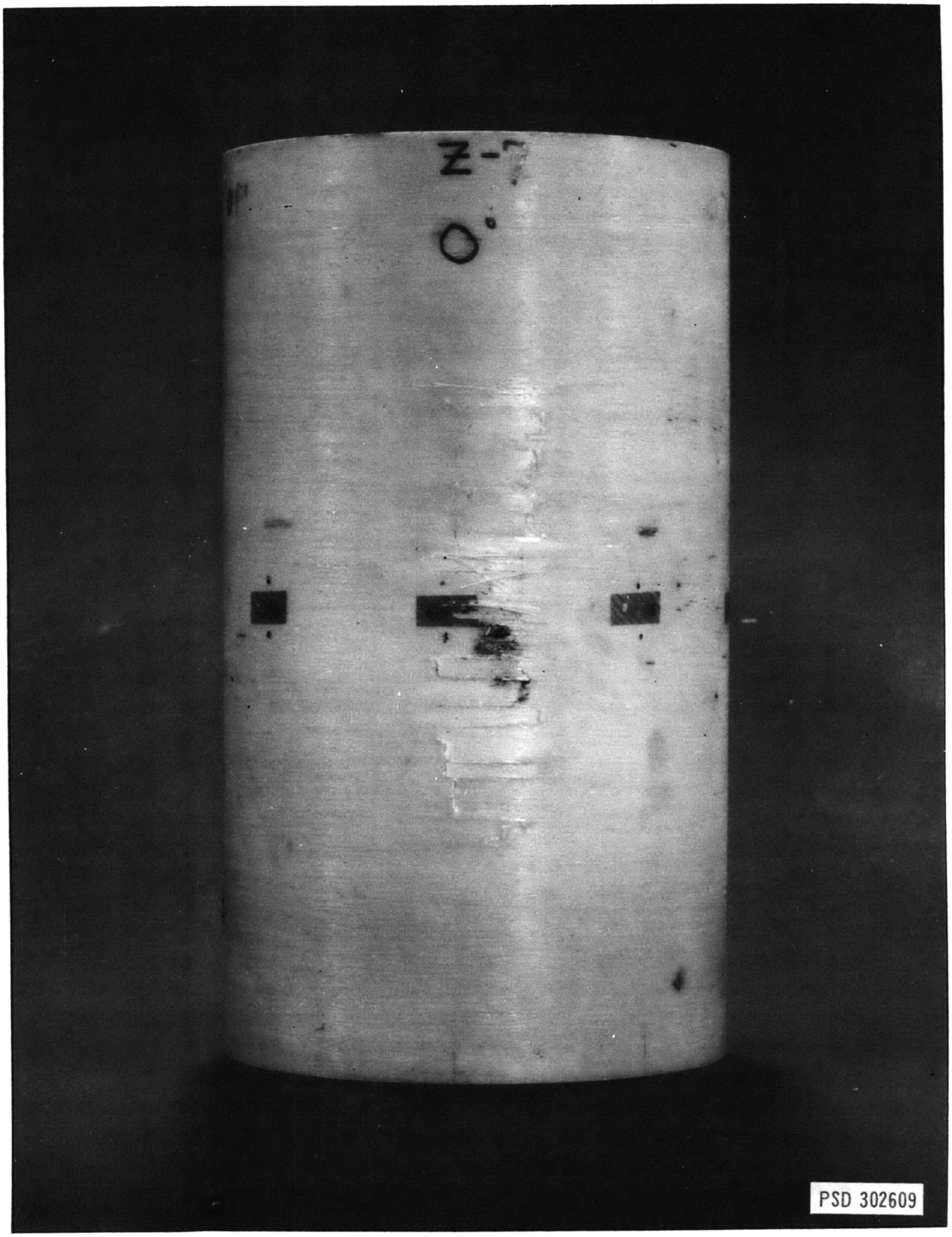


Figure 11 - Exterior View of Model Z-7 after Failure

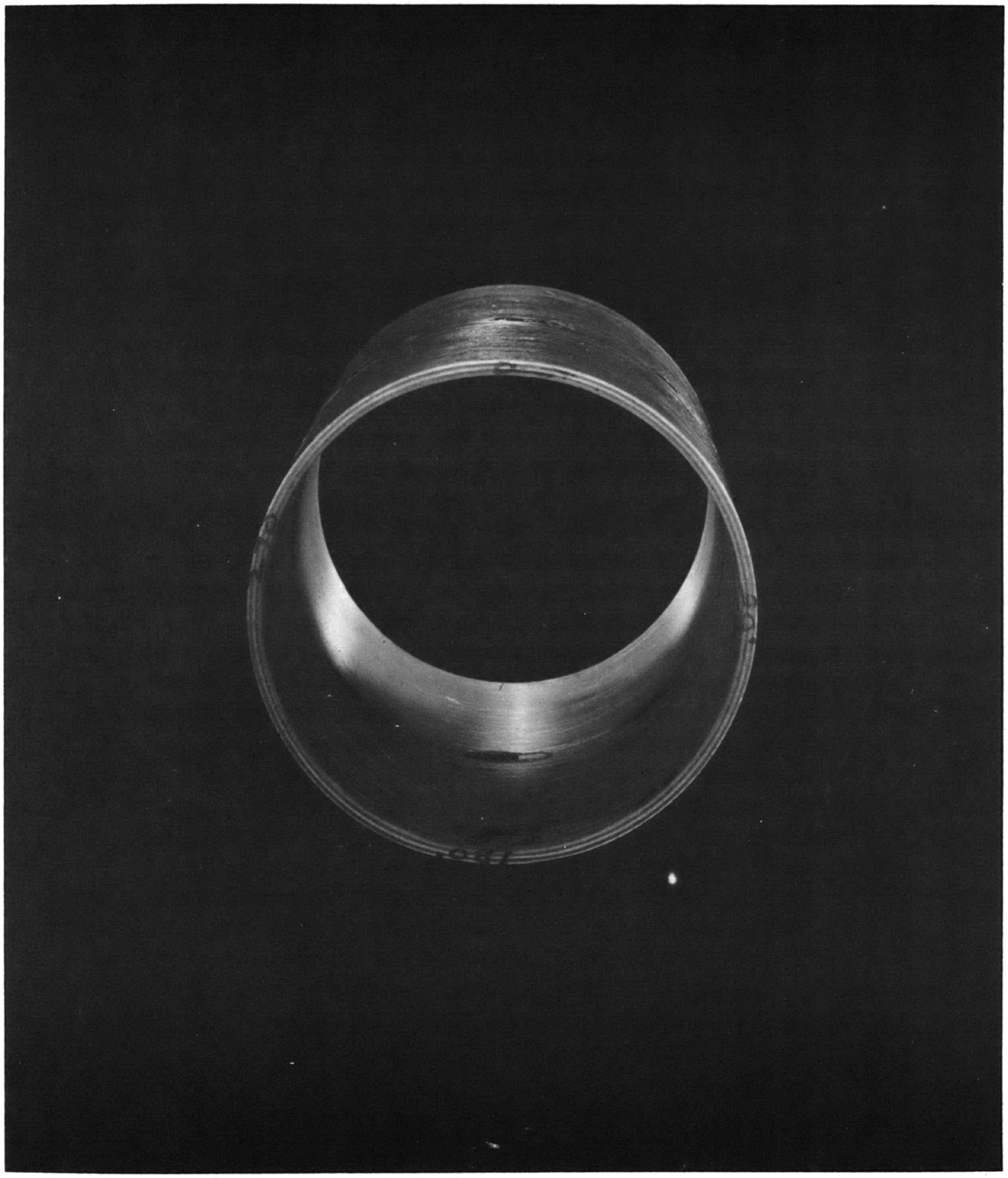


Figure 12 - End View of Model B-3 after Failure

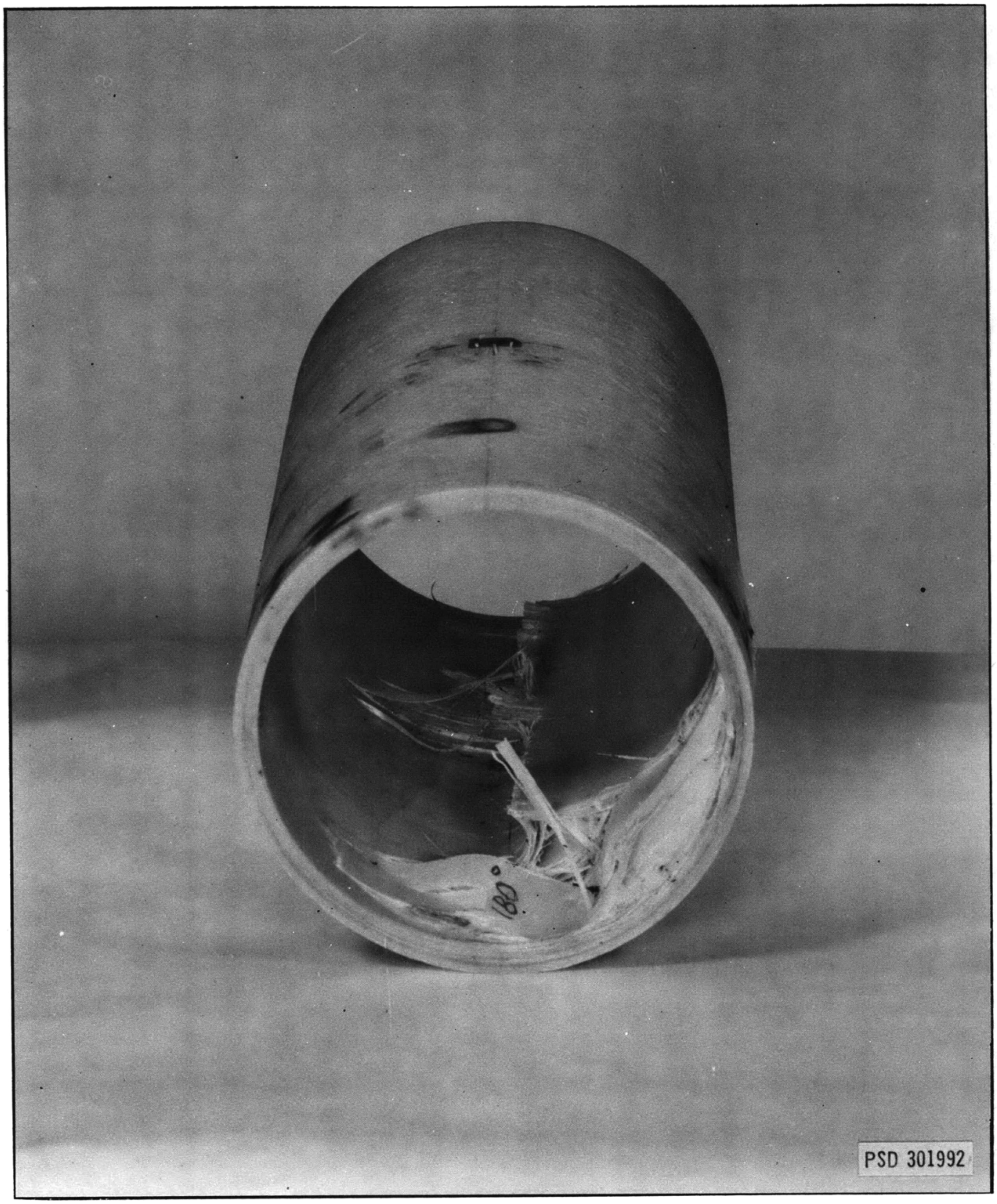


Figure 13 - Interior View of Model Z-2 after Failure

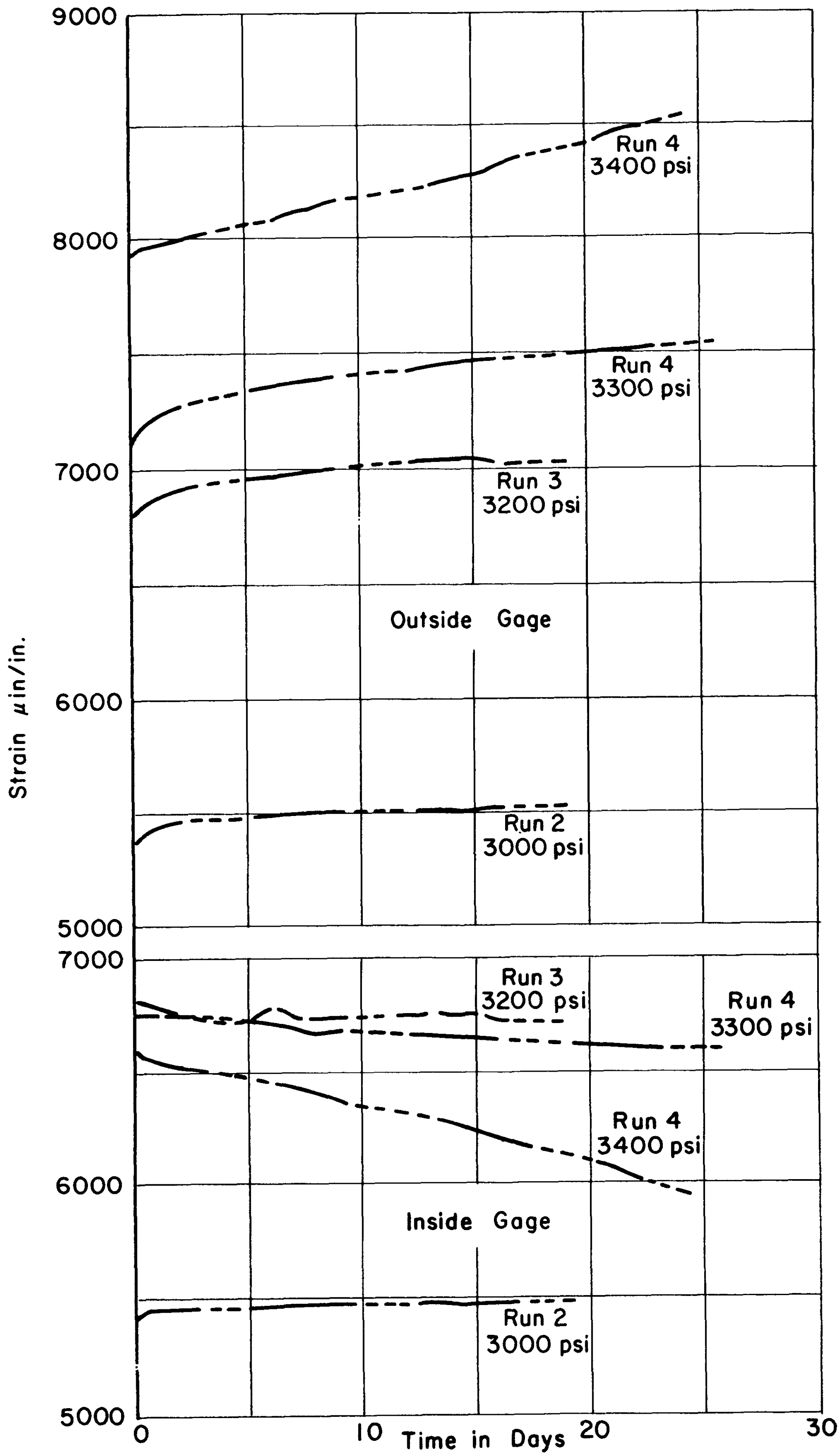


Figure 14 - Strain-Time Plots for Circumferential Gages on Model Z-2

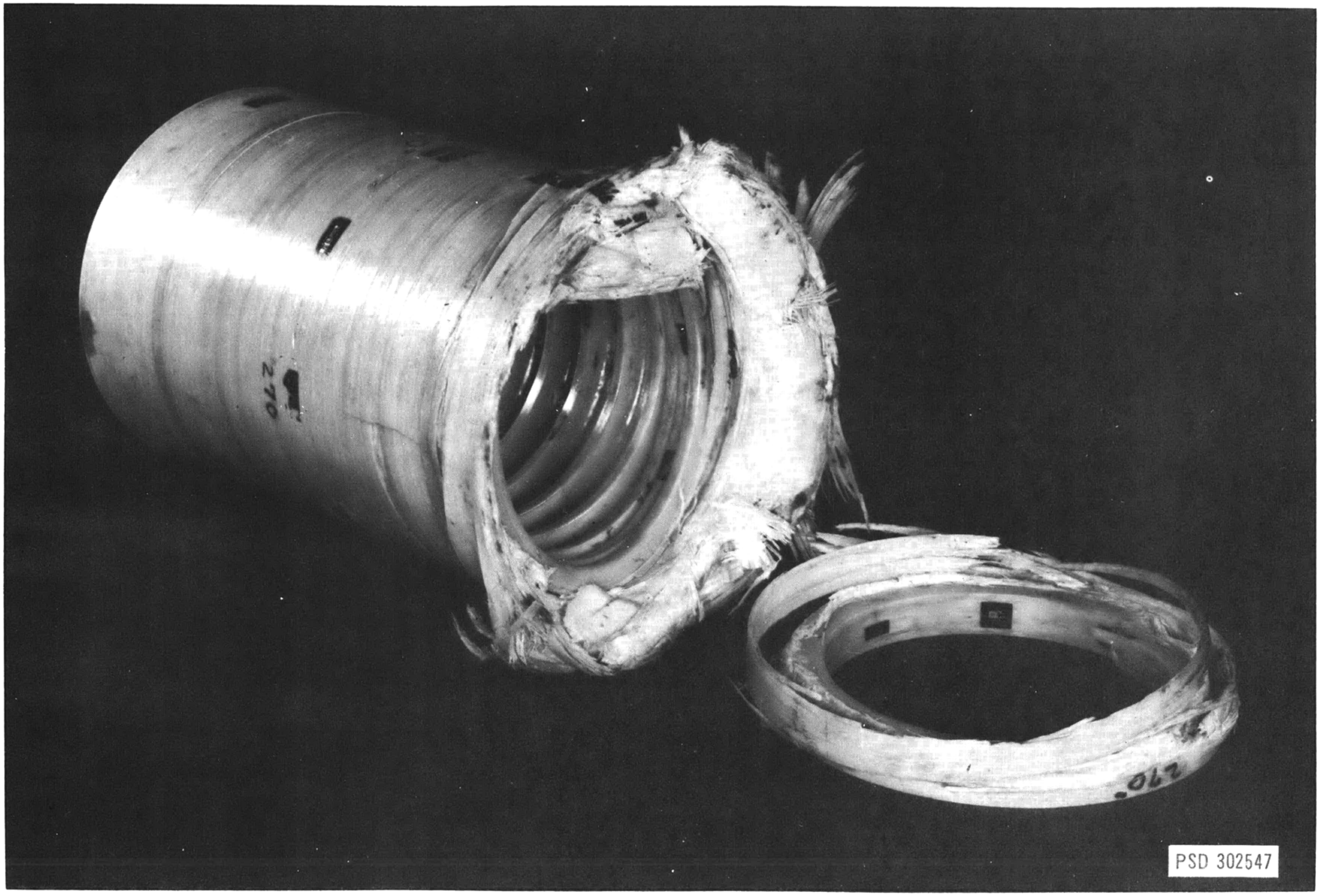


Figure 15 - End View of Model ZR-1 after Failure

Figure 16 - End View of Model ZR-2 after Failure

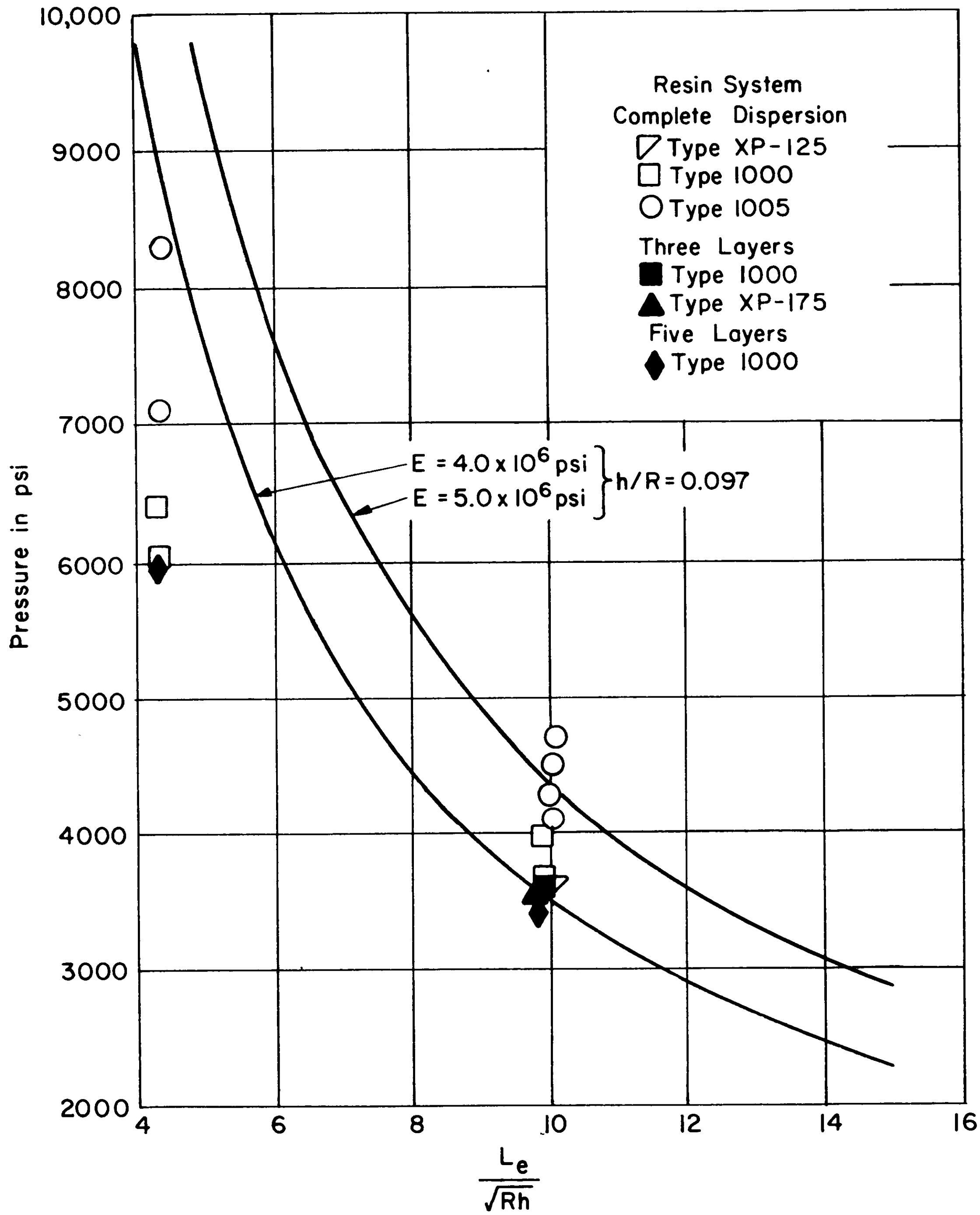


Figure 17 - Buckling Collapse of Unstiffened Zenith Cylinders (2C:1L)

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