R 618

Technical Report

LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS, ~~PART III.~~ pt. 3. GLASS PIPES WITH CONICAL FLANGED ENDS

March 1969

Sponsored by

NAVAL FACILITIES ENGINEERING COMMAND

NAVAL CIVIL ENGINEERING LABORATORY

Port Hueneme, California

This document has been approved for public release and sale; its distribution is unlimited.

LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS— PART III. GLASS PIPES WITH CONICAL FLANGED ENDS

Technical Report R-618

Y-F015-01-07-001

by

K. O. Gray and J. D. Stachiw

ABSTRACT

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Copies available at the Clearinghouse for Federal Scientific & Technical Information (CFSTI), Sills Building, 5285 Port Royal Road, Springfield, Va. 22151

CONTENTS

PROBLEM STATEMENT

The study, exploration, and utilization of hydrospace require the use of instruments and devices which must be "packaged" in sealed housings to protect them from the high-pressure seawater environment. While a few such housings are available as off-the-shelf commercial items, the variety is small, the cost high, and delivery time is frequently long.

Lights for hydrospace illumination are a fundamental requirement for either direct or indirect visual observation of any type of undersea activity. Only in the shallowest portions of the sea, and then only during sunny days, is the natural light adequate for most observational purposes. The majority of lights and lighting systems require a transparent, pressure-resistant housing for the proper functioning of the light-producing element. Certain other devices such as light sensing or measuring instruments also require pressure-resistant transparent housings. These requirements immediately point to the possible application of glass as a housing material since it is transparent, has high compressive strength and in certain formulations is resistant to thermal shock. Glass has an additional attraction in that it is a nonmagnetic material and is transparent to many types of electromagnetic radiation that are attenuated by metallic housing materials. The more obvious advantages of these properties are the applicability of magnetic and photo cell techniques for operating instruments in such packages and for the transmission of data through the instrument housing walls without the requirement for physical penetrations of the housing or end closures.

STUDY OBJECTIVE

This study is a continuation and extension of an investigation into the applicability of commercially available off-the-shelf, dome-shaped glass containers to the design of underwater lights.[1,2] Additional potential applications for pressure-resistant glass housings caused the investigation to be broadened to include larger commercial tubular glass shapes for both lights and instrument housings. After evaluation of selected glass pipes of various diameters and lengths, prototype light and instrument housings were to be designed and fabricated.

PURPOSE OF STUDY

Commercially available off-the-shelf glass pipe used in the chemical industry is available in a variety of lengths and diameters (Figure 1). Due to the wide use of this material in the industry, it is mass produced and is much less expensive than either custom-fabricated or limited-production housings of similar dimensions fabricated from either metal or glass. In addition, the widespread and continuing use of standard glass piping assures a continuing supply of standardized sizes and formulations. Thus, if successful designs of underwater light and instrument housings based on these types of glassware can be developed, low acquisition and replacement costs can be guaranteed for the future.

The purpose of this report is to document the experimental study of readily available glass pipe for use in underwater housings, to present data on its performance, and to offer a limited number of designs for instrument and light housings utilizing glass pipe as the main component of the housing.

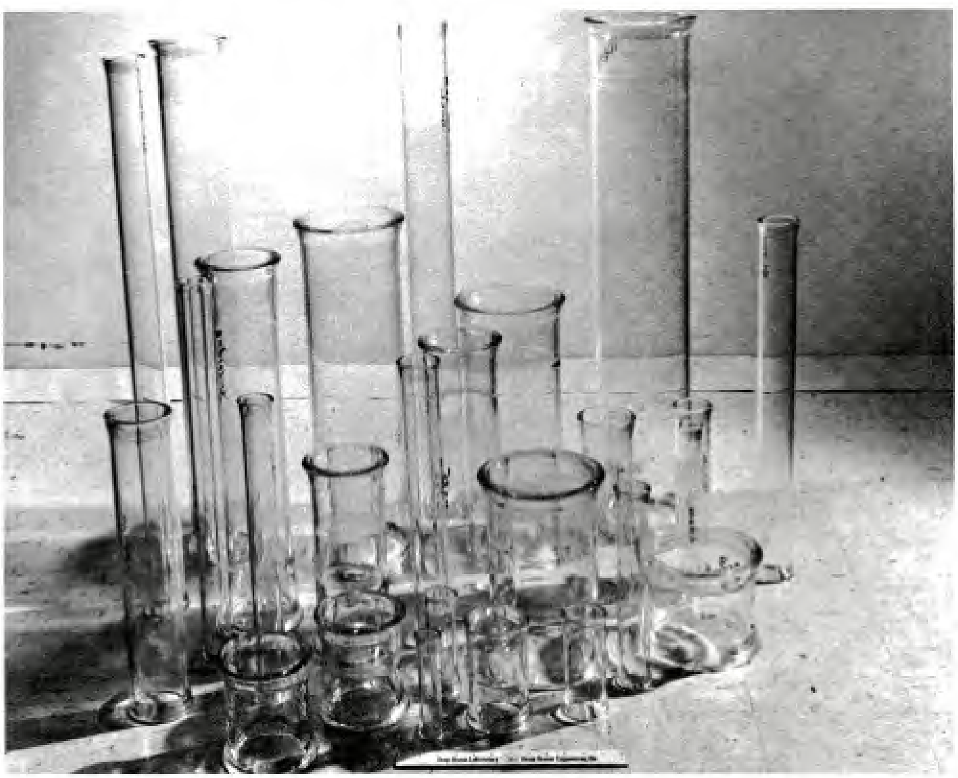


Figure 1. A sampling of the variety of glass pipe diameters and lengths tested in this program.

EXPERIMENTAL PROCEDURE

In order to satisfy the objectives of the overall study, it was necessary to embark on several phases or substudies. These were (Phase I) determination of the relative critical pressure of various diameters and lengths of glass pipe, (Phase II) determination of the best end-closure-to-glass bearing and sealing system, and (Phase III) the testing of prototype light and instrument housings.

Phase I: Short-Term Critical Pressure of Glassware

Relative critical pressure was determined by preparing test assemblies of four specimens of each selected diameter and length combination. Table 1 shows the dimensions and quantity of specimens evaluated in this first phase.

Table 1. Dimensions and Number of Specimens Tested in Phase I of Study

Inside Diameter (in.)	Pipe Length* (in.)				
	6	12	18	24	36
	Number Tested				
1	4	4	0	0	0
1-1/2	4	4	4	4	0
2	4	4	4	4	4
3	4	4	4	4	4
4	4	4	4	4	4
6	4	4	4	4	4

* Lengths of up to 120 inches are available.

Figure 2 shows the component parts of a test assembly. Figure 3 shows one specimen of each length of 4-inch-ID pipe made up into a test assembly and Figure 4 shows 6-inch lengths of five different diameters of pipes made up into test assemblies. Due to the complete destruction of the metal flange assemblies, which generally took place when the specimens imploded, a simplified, less costly hold-down system was devised. This system consisted of three small pieces of Plexiglas which clamped the glass to the end plates. This simplified system (Figure 5) or a tie-rod system was used after the first few tests.

The test assembly end plates, fabricated from 6061-T6 aluminum, were designed to be of sufficient thickness (a) to provide a rigid end closure capable of withstanding pressures up to 20,000 psi, (b) to prevent end-plate thinning (resulting from the metal removal in refinishing the seating surfaces after each use) from weakening them excessively before the experiment was completed. Remachining was necessary after each use because of the damage done to the sealing surface when the specimens either imploded (Figure 6) or failed by cracking without implosion.

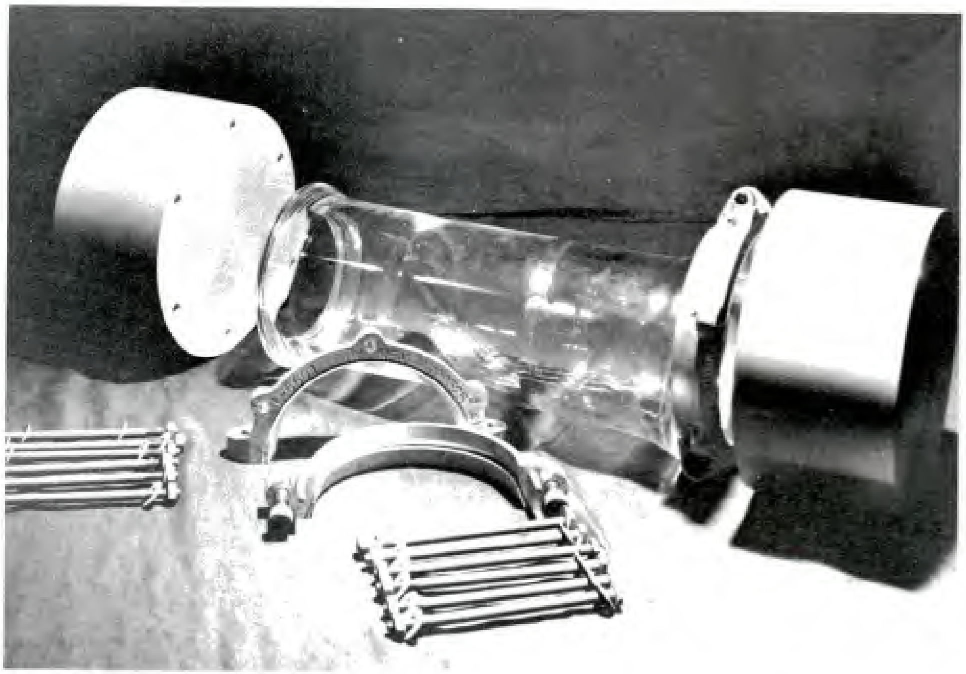


Figure 2. Component parts for a short-term critical pressure test assembly.

The initial thickness of the aluminum end plates for each diameter of pipe was as follows:

Inside Pipe Diameter (in.)	Plate Thickness (in.)
1	0.910
1-1/2	1.365
2	1.820
3	2.730
4	3.63
6	5.45

Figure 3. Four-inch-ID glass pipe test assemblies made up of 6-, 12-, 18-, 24-, and 36-inch lengths.

Figure 4. Six-inch-long glass pipe test assemblies made up with 1-, 2-, 3-, 4-, and 6-inch-ID pipes.

The assembly process was accomplished by first applying room temperature vulcanizing (RTV) silicone rubber paste to the glass pipe flange (Figure 7) then inverting it and placing it on the end closure (Figure 8) using the bolts as alignment pins to center the pipe on the end closure. Figure 9 shows the finished seal just before the bolts were torqued. The bolts were torqued to a uniform load as shown in Figure 10. After about 24 hours, the glass pipe was partially (85% to 90%) filled with water (to reduce the force of the implosion) and the top end closure was assembled in a similar manner. The assembly was then allowed to stand for a minimum of 24 hours before testing.

The standard procedure for testing specimens which would fit into the laboratory pressure vessels was to suspend them from the pressure vessel head (Figure 11) and then place them in the vessel. The vessel was filled with water and then pressurized by means of air-operated piston pumps at a rate of 100 psi per minute. Pressurization was continued at a constant rate until the test assembly imploded, leaked, or the pressure reached 20,000 psi. Pressure was then relieved, the specimen removed from the vessel, and a detailed examination made of the remains.

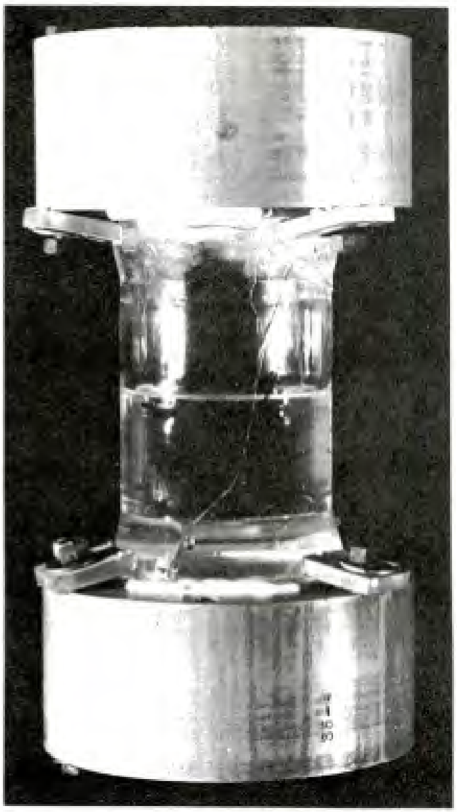


Figure 5. Simplified hold-down system used in majority of short-term critical pressure tests.

Figure 12 shows a 4-inch-ID x 12-inch-long (90% water-filled) specimen assembly and the remains of a similar (except air-filled) assembly after failure by implosion. The violence of the implosion of the air-filled specimen is shown by the granular nature of the remains of the glass pipe (the small pile of white material seen between the end closures).

The 4- and 6-inch-diameter pipes that were too long to fit into the NCEL pressure vessels were assembled with tie-rods (used between the end plates) and were left completely filled with air (no water inside). These pipes were tested by taking them to sea and lowering them into the ocean until they reached implosion depth. The violence of the implosion was sufficient to be heard aboard ship by means of a hydrophone hung in the water below the ship. Thus, the time of the implosion was known, the wire angle and the amount of wire necessary to reach implosion depth was noted. The depth of implosion was then derived. This depth was then converted to critical pressure.

The technique of mitigating equipment damage (from the violent implosion of void specimens) by almost completely filling the test specimen with water brought up the following questions:

1. Was sufficient void space being provided to compensate for the volume reduction of the glass pipe—end closure system at high pressures?

2. Did this technique have any effect on the critical pressure of the specimens tested?

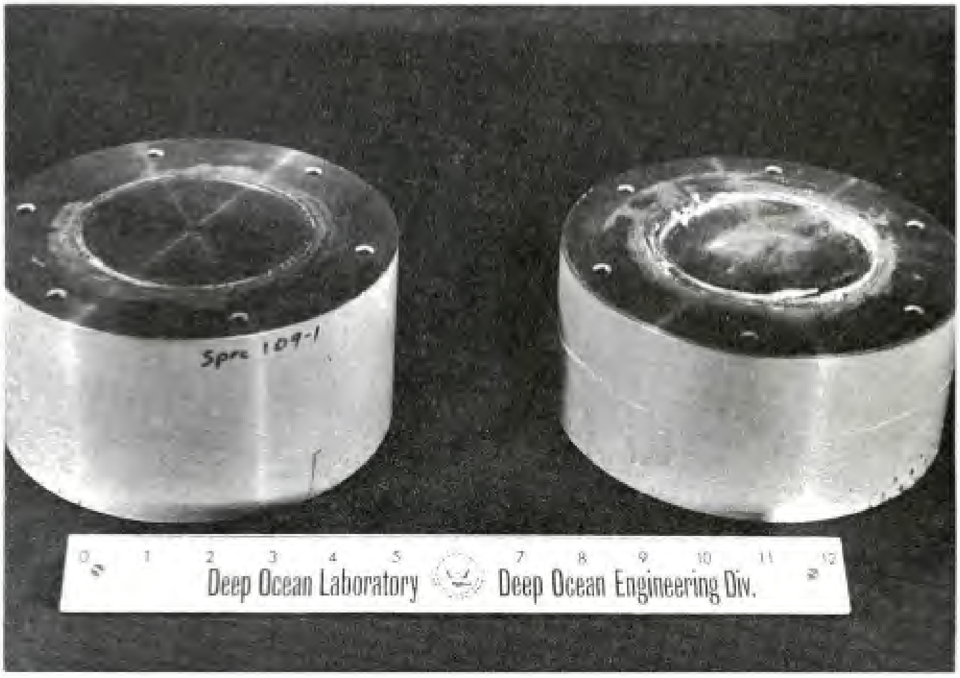


Figure 6. Damage to aluminum end-closure plates resulting from the implosion of a 4-inch-ID, 6-inch-long glass pipe at 16,900 psi.

In order to answer these questions, specimens of different sizes were provided with end-closure plates, which were drilled and tapped to receive a high-pressure tubing connection. The specimens were then assembled, filled completely with water, and the high-pressure connection brought through the pressure vessel end closure (Figure 13). This line was then connected to a graduated cylinder so that the fluid expelled from the specimen assembly in response to increasing external pressure could be measured. Figure 14 illustrates this setup. A graphic plot of the volume change versus pressure for three 4-inch-ID x 6-inch-long glass specimen assemblies is shown in Figure 15.

Experimental data from testing 4-inch-ID x 6-inch-long glass pipe in the manner just described showed the following:

1. The net reduction of volume of the glass pipe, under an external hydrostatic pressure of 20,000 psi, was approximately 40 ml or 3% of the volume at atmospheric pressure. (The void space normally left in specimens tested for critical pressure amounted to about 10% to 15%.)

2. The critical pressure of glass pipes partially filled with water did not vary significantly from those filled with air at atmospheric pressure.

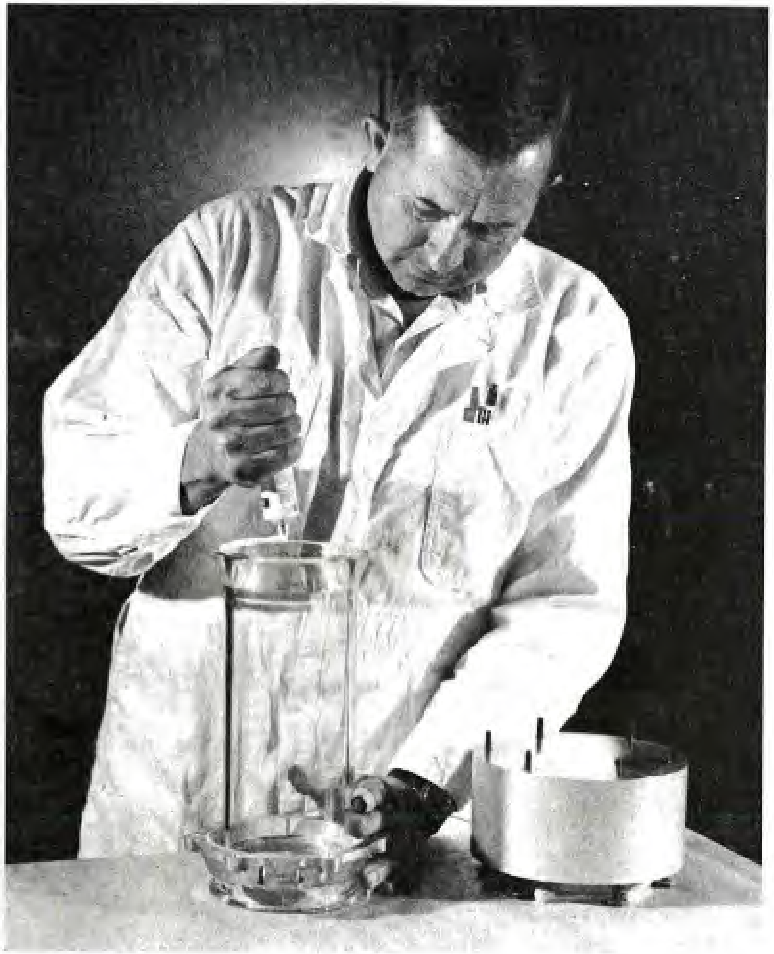


Figure 7. Application of RTV silicone rubber to glass pipe flanges during preparation of specimen assembly.

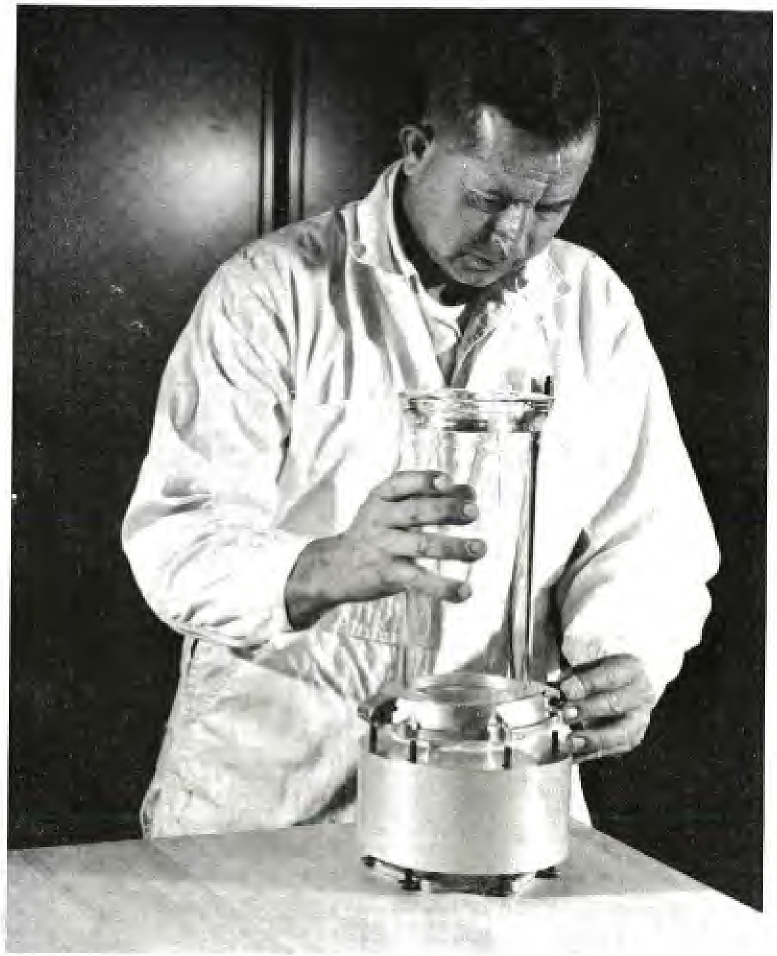


Figure 8. Assembly of silicone-rubber-coated glass pipe to aluminum end-closure plates.

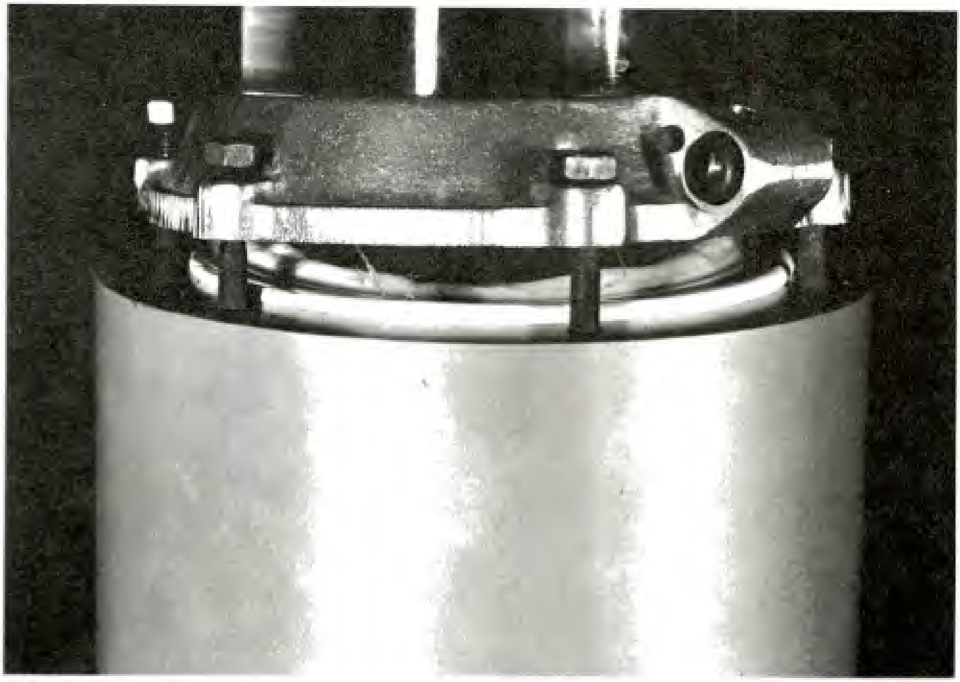


Figure 9. Appearance of glass-to-metal contact area and RTV silicone rubber seal before the tie bolts were torqued.

Based on this information it was concluded that the practice of partially filling the specimens with water did not affect the test results as long as a minimum of 10% to 15% of the total internal volume of the test glass specimen was filled with air at atmospheric pressure when the specimen was assembled.

The test results, presented in Table 2, indicate that "off-the-shelf" glass pipe of the type tested will provide reliable, transparent, nonmagnetic instrument or light housings for a single submersion to any depth to be found in the ocean, provided the appropriate diameter and length are chosen. Relatively large housings (6-inch-ID x 18 inches long) are useful for at least one cycle to depths of 5,000 feet with a safety factor of 2 when used in a system with simple 6060-T6 aluminum end plates.

Phase II: Gasket System Tests

The second phase of the study consisted of testing a number of different glass pipe-to-end closure bearing and sealing systems. The objective was to develop an end-closure system capable of withstanding the maximum critical pressure and satisfactory for repeated exposure to external hydrostatic pressure (pressure cycling service).

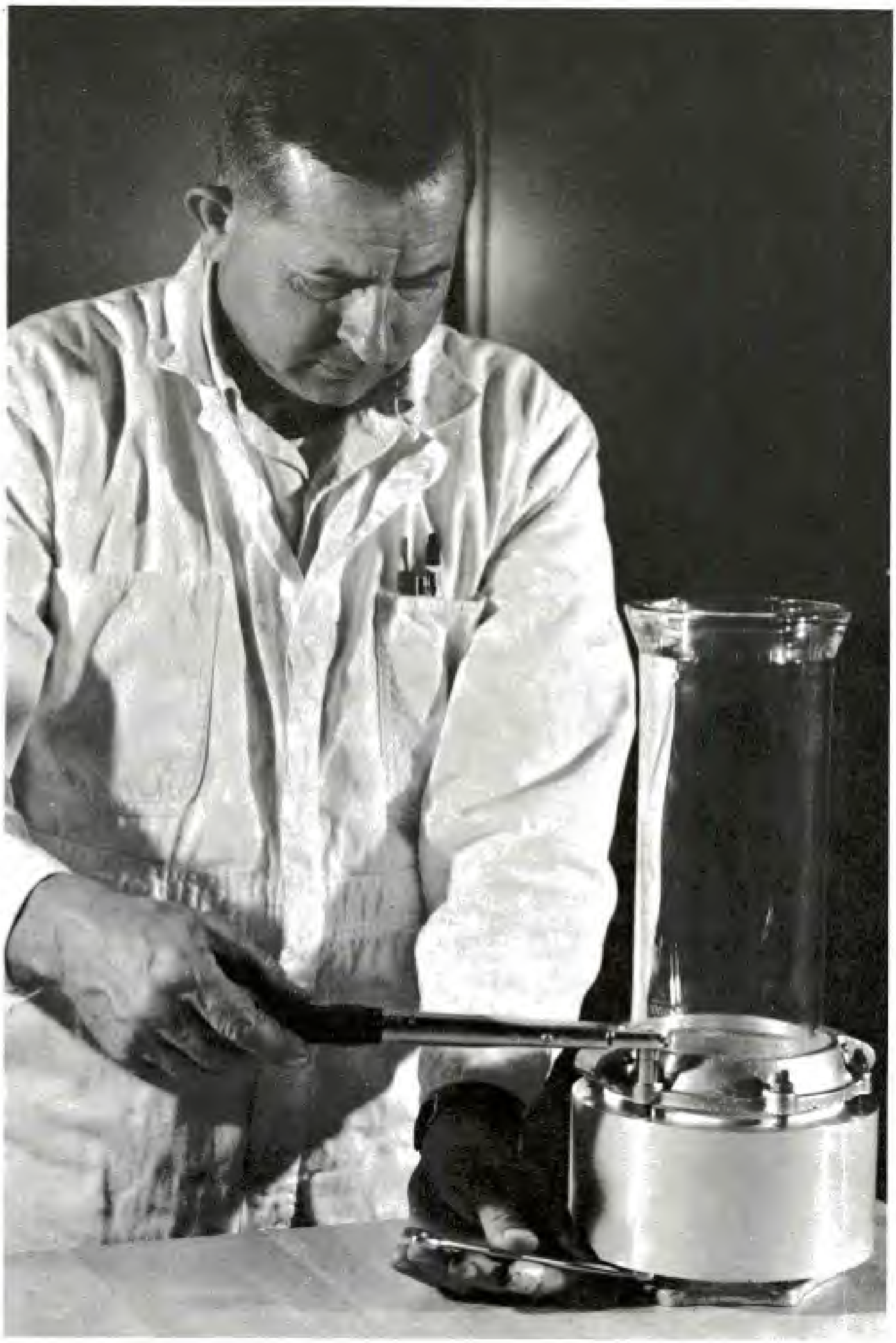


Figure 10. Use of torque wrench to avoid damage to glass pipe through uneven loading of glass flange area during assembly.

Figure 11. Vented glass pipe specimen assembly attached to head of 18-inch pressure vessel (left).

In order to standardize the test procedure and to eliminate as many variables as possible, all tests in this phase were conducted with 4-inch-ID x 6-inch-long Pyrex pipe with flanged ends. The glassware was procured through normal commercial glassware supply channels and was used in the "as received" condition.

Five specimens each of the following simple glass pipe-to-end closure plate combinations were tested to determine critical pressure using the same procedures as described under the Phase I tests.

Aluminum 6061-T6	3.63 inches thick
Stainless steel type 316	2.58 inches thick
Brass (naval)	2.58 inches thick
Titanium (Ti-6Al-4V)	1.82 inches thick
Phenolic resin—glass fiber laminate	4.40 inches thick

Figure 16 shows typical samples of the end closures used.

The variation in end-closure plate thickness from material to material is a result of the relative strengths of the materials and the calculated thickness required to resist failure at an operating pressure of 20,000 psi.

Table 3 shows the results from short-term critical pressure tests with the various end-closure materials.

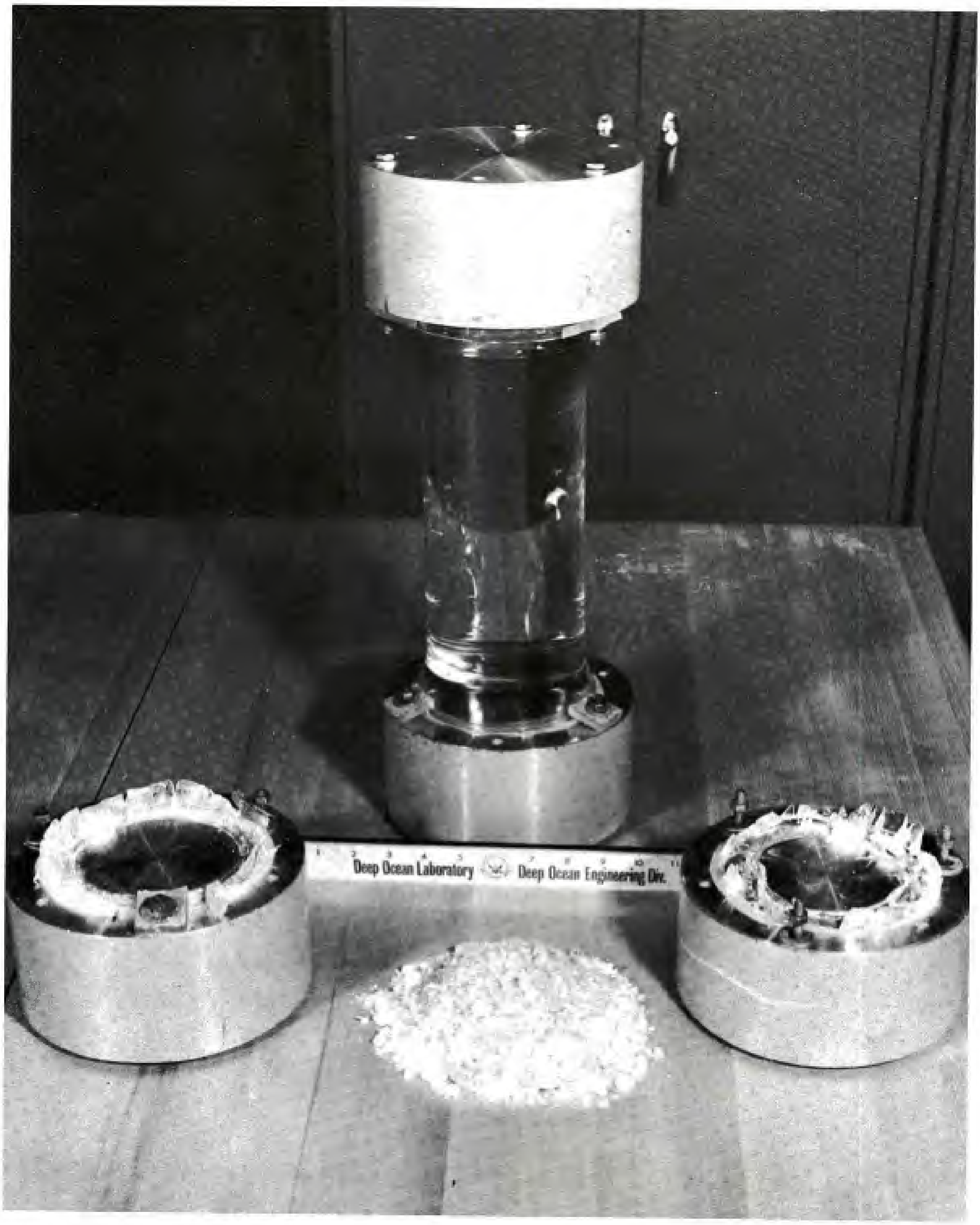


Figure 12. Air-filled 4-inch-ID x 12-inch-long glass pipe assembly before and after implosion at 10,400 psi. The white, granular material is all that remains of the glass pipe following implosion.

Figure 13. Typical water-filled specimen assembly showing connection through the end closure to the pressure vessel head and the atmosphere.

Figure 14. Typical setup used to measure the water displaced from and detect leaks in a test assembly during pressurization. The displaced water is collected and measured in the graduated cylinder at intervals of 1,000-psi pressure increase.

Table 2. Short-Term Critical Pressure of Flanged Glass Pipe With 6061-T6 Aluminum End Plates

(Bracketed numbers—for example, [1] —represent number of specimens tested.)

Inside Diameter and Type of Pipe	Critical Pressure (psi) for Length of—														
	6 Inches			12 Inches			18 Inches			24 Inches			36 Inches		
	Min	Max	Avg	Min	Max	Avg	Min	Max	Avg	Min	Max	Avg	Min	Max	Avg
1-Inch Pyrex**		20,000* [4]			20,000† [4]			‡			‡			‡	
1-1/2-Inch Pyrex		20,000§ [4]		18,400	19,010 [3]	18,636	17,400	17,900 [4]	17,535	19,400	19,700 [3]	19,500		‡	
2-Inch Pyrex	14,600	14,750 [3]	14,650	8,900	9,800 [4]	9,275	9,000	9,500 [4]	9,175	8,400	8,650 [4]	8,560	8,150	8,400 [4]	8,260
3-Inch Pyrex	16,400	17,200 [4]	16,720	5,900	7,100 [4]	6,590	5,300	5,500 [4]	5,380	3,800	5,000 [4]	4,560	3,600	5,000 [4]	4,207
4-Inch Pyrex		20,000§ [4]		8,050	11,700 [3]	10,050	6,850	6,900 [2]	6,875	4,700	5,700 [4]	5,187	3,900	4,000 [4]	3,950
6-Inch Pyrex	2,500	9,700 [4]	6,460	9,250	9,900 [4]	9,540	3,800	7,700 [4]	4,768	3,000	3,400 [4]	3,240	2,250	2,600 [4]	2,350
4-Inch Kimax††	16,900	20,000§ [4]	18,650	7,200	8,600 [4]	7,830	4,800	5,100 [3]	4,933	3,450	4,550 [4]	4,010	3,250	3,550 [4]	3,363

* None failed.

† None imploded; one cracked slightly.

‡ Not tested.

§ None imploded; all were severely cracked.

** Corning Glass Catalogue PE-200

†† Owens-Illinois Catalogue TR-99

Table 3. Short-Term Critical Pressure of 4-Inch-ID x 6-Inch-Long Glass Pipe Specimens When Tested With End Closures of Various Materials

End-Plate Material	Critical Pressure (psi)			Notes on Structural Performance
	Min	Max	Avg	
6061-T6 aluminum	20,000+	20,000+	20,000+	All specimens cracked during pressurization but did not implode or leak until pressure was relieved. Figure 5 shows a typical specimen after testing.
Stainless steel type 316	12,900	20,000+	(?)	Two specimens imploded at 12,900 and 14,350, respectively, during pressurization. Two specimens failed to implode but were severely cracked and fell apart when the pressure was released.
Titanium type Ti-6Al-4V	18,600	20,000+	(?)	One specimen imploded. Two specimens were severely cracked and came apart when removed from the vessel.
Naval brass	9,400	20,000+	(?)	One specimen imploded. Three specimens were severely cracked and came apart when removed from the vessel.
Phenolic resin-impregnated glass fiber laminate	20,000+	20,000+	20,000+	For one specimen no cracking except a large chip spalled off the inside rim. Two specimens were severely cracked but did not fall apart. One specimen cracked and fell apart on the release of pressure.

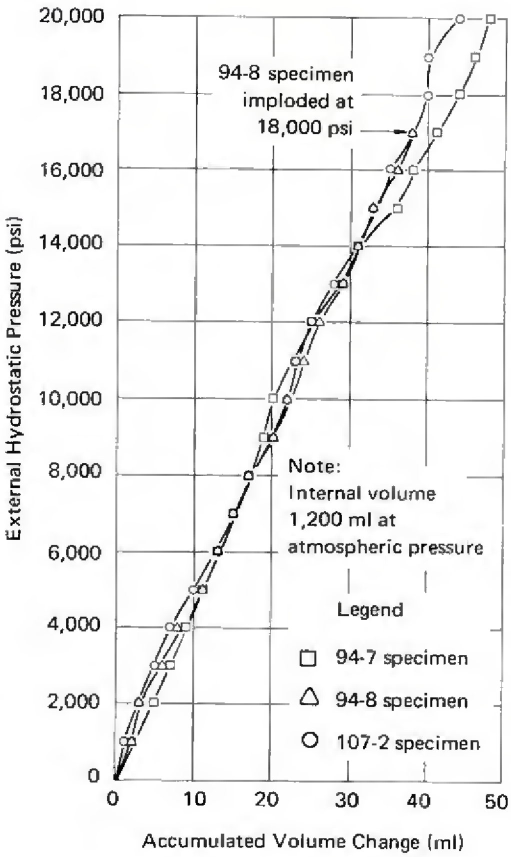


Figure 15. Graphic plot of the volume of water displaced from three water-filled, 4-inch-ID x 6-inch-long glass pipe assemblies subjected to increasing hydrostatic pressure.

The 6061-T6 aluminum and the phenolic resin—impregnated glass fiber laminate materials performed the best under a single pressurization to implosion. However, the cost of both the basic material and the machining of the glass laminate end closures is much greater than that of the aluminum end closures. For this reason the 6061-T6 aluminum is considered the best choice based on these tests.

The second series of five specimens of each glass pipe—end closure combination was tested for cyclical performance. Cyclical tests started with 10 cycles from 0 psi to 5,000 psi. Specimen assemblies were then disassembled and examined, and the next cycling pressure was selected on the basis of the just completed test. Subsequent cycling tests at successively lower pressures were conducted as indicated in Table 4. The pressurization rate was held at 1,000 psi per minute in all tests.

Based on these tests, naval brass appears to be the best choice for limited cycling service under this set of conditions.

A third series of experiments was designed to evaluate the effectiveness of various "gasket" (sealing and bearing) systems designed to interpose a "soft" bearing and sealing element between the end of the glass pipe and the type 316 stainless steel end-closure plate selected for these tests; effectiveness was gaged by the increase in the cyclical depth range of the glass pipe under test.

The gasket systems were designed to serve three functions. (1) to provide a watertight seal between the glass pipe and end-closure disc at both low and high pressures; (2) to eliminate stress concentrations due to point contact between the somewhat uneven glass bearing surface and the steel end-closure material; (3) to provide a "mobile" surface on which the glass pipe end flange could move as the pipe diameter decreased in response to external pressure.

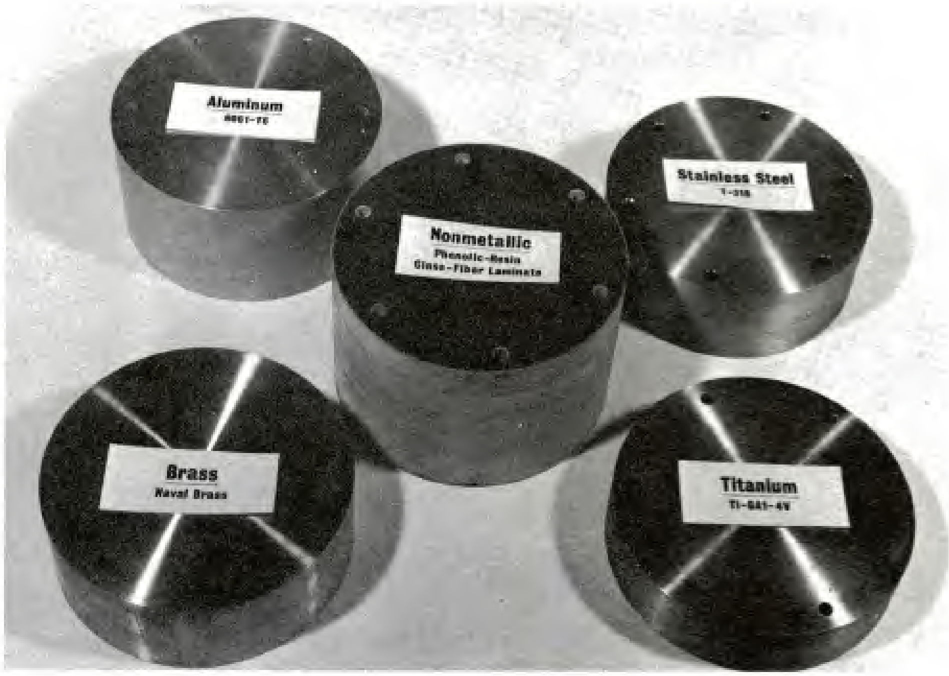


Figure 16. Typical end closures used to evaluate the effect of end-closure material on short-term critical pressure of 4-inch-ID x 6-inch-long glass pipe with flanged ends.

Since the purpose of this series of experiments was to enhance the usefulness of the glass pipe in terms of numbers of use cycles and depth limits, an arbitrary minimum acceptable short-term collapse pressure of 3,000 psi was adopted. Systems which failed at pressures less than this were not further tested to determine what their maximum depth rating would be for cyclical service.

The gasket systems tested were as follows:

1. Soft (60 durometer) 3/32-inch cross section O-rings were placed in the grooves in the glass pipe flanges, lubricated with silicone grease and then compressed between the glass pipe flanges and the steel end plates by tensioning the tie rods holding the assembly together. (See Figure 17.)

In each test assembly the O-ring acted as a wedge and sheared off the external flange lip from the pipe (Figures 18 and 19). This occurred during the first cycle to 3,000 psi for each pipe. Thus the maximum useful depth rating for 3/32-inch O-rings is not known. Substitution of thinner O-rings may eliminate the wedge action, as was shown subsequently in the successful light housing design (Figures 33 and 34).

Table 4. Performance of 4-Inch-ID x 6-Inch-Long Glass Pipe Specimens With End Closures of Various Materials Under External Cyclical Hydrostatic Pressure Loading

(Bracketed numbers—for example, [5] —represent number of test cycles completed.)

End-Plate Material	Test Plan and Results				
6061-T6 aluminum	(Plan: 5 cycles at 5,000 psi) One 2-inch-long chip broke off of inside rim; no other damage. [5]	(Plan: 5 cycles at 4,500 psi) Imploded at 4,000 psi under increasing pressure during second cycle. [1]	(Plan: 5 cycles at 4,500 psi) One crack on inside near rim; no other damage. [5]	(Plan: 5 cycles at 3,500 psi) Leaked under increasing pressure during third cycle; glass cracked through all the way around. [2]	–
Stainless steel type 316	(Plan: 5 cycles at 5,000 psi) Cracked and leaked after second cycle; crack propagated from end-plate area and cracked through all the way around. [1]	(Plan: 5 cycles at 4,500 psi) Very fine crack in one end between inside rim and O-ring groove; no other damage. [5]	(Plan: 5 cycles at 4,000 psi) Very fine cracks in one end between inside rim and O-ring groove; no other damage. [5]	(Plan: 5 cycles at 3,500 psi) Very fine cracks in both ends between inside rim and O-ring groove; some fine spalling inside; no other damage. [5]	–
Titanium type Ti-6Al-4V	(Plan: 5 cycles at 5,000 psi) Fine cracks in both ends, parallel to O-ring groove; some fine spalling inside; no other damage. [5]	(Plan: 5 cycles at 4,500 psi) Fine cracks in both ends parallel to O-ring groove; some spalling inside; no other damage. [5]	(Plan: 5 cycles at 4,000 psi) Fine cracks in both ends parallel to O-ring groove; no spalling or other damage. [5]	(Plan: 5 cycles at 3,500 psi) Cracked and leaked after second cycle; cracks all the way around in both ends about 1 inch above flange. [1]	–
Naval brass	(Plan: 5 cycles at 5,000 psi) No damage to glass. [5]	(Plan: 5 cycles at 4,500 psi) No damage to glass. [5]	(Plan: 10 cycles at 4,500 psi) Imploded at 4,100 psi under increasing pressure during 10th cycle. [9]	(Plan: 10 cycles at 4,000 psi) Sealant failed during eighth cycle; fine cracks and some spalling in both ends. [8]	–
Phenolic resin-impregnated glass fiber laminate	(Plan: 5 cycles at 5,000 psi) Many cracks; leaked after one cycle. [1]	(Plan: 5 cycles at 4,500 psi) Many cracks; leaked after one cycle. [1]	(Plan: 5 cycles at 4,000 psi) Imploded at 2,900 psi under increasing pressure during second cycle. [1]	(Plan: 5 cycles at 3,000 psi) No damage to glass. [5]	(Plan: 20 cycles at 3,000 psi) No damage to glass; same specimen assembly as used in 3,000-psi cycling test. [20]

Note: A new glass pipe specimen and refinished end plates were used for each series of cyclic pressure tests.

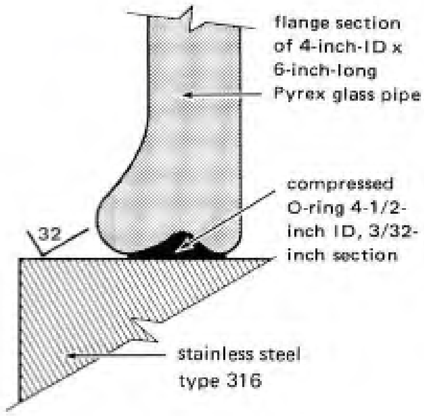


Figure 17. Experimental sealing and bearing system utilizing an O-ring interposed between the glass pipe flange and the metal end-closure plate.

2. Thin (0.023-inch-thick) gaskets made of fiber-reinforced neoprene (Fairprene 5722A), lubricated with silicone grease, were placed between the glass pipe end flanges and the steel end plates and then compressed by tensioning the tie rods holding the assembly together. A watertight seal was provided by applying room temperature vulcanizing (RTV) silicone rubber to the joint between the end closure and pipe flange after compressing the seal. (See Figure 20.)

This bearing—sealing system worked reliably on each of three test assemblies cycled 10 times to 3,000 psi, and on one specimen cycled 10 times to 3,000 psi and then 10 cycles to 4,000 psi. Two others failed on the 15th and 18th cycles to 4,000 psi after being cycled 10 times to 3,000 psi. This system is considered useful for limited cyclical service with 4-inch-diameter flanged glass pipes to depths of 6,000 feet. The same gasket system was found in previous studies[1,2] to perform successfully in cyclic tests on glass domes to depths of 40,000 feet.

3. The gaskets in this case consisted of 1/8-inch-thick polyimide resin (Vespel SP-1) washers fitted into grooves in the steel end plates. These washers were of sufficient width that the glass pipe did not touch the steel end plates. During assembly the watertight seal was provided by applying RTV silicone rubber to the area between the washer and the glass pipe. (See Figure 21.)

Of the two test assemblies cycled, one failed through cracking during the first cycle at 6,000 psi after 10 cycles at 3,000 psi, 10 cycles at 4,000 psi, and 10 cycles at 5,000 psi. The second test assembly completed 10 cycles each at 3,000 and 4,000 psi, cracking on the first cycle at 5,000 psi. This system is considered useful for limited cyclical service with 4-inch flanged glass pipes to 8,000 feet.

4. Composite gaskets were made by soft-soldering 1/8-inch-thick copper washers to 1/8-inch-thick steel washers and then facing off the flat surfaces parallel and providing them with a 32-rms finish. The steel faces were then coated with a 0.0025-inch-thick molybdenum disulfide

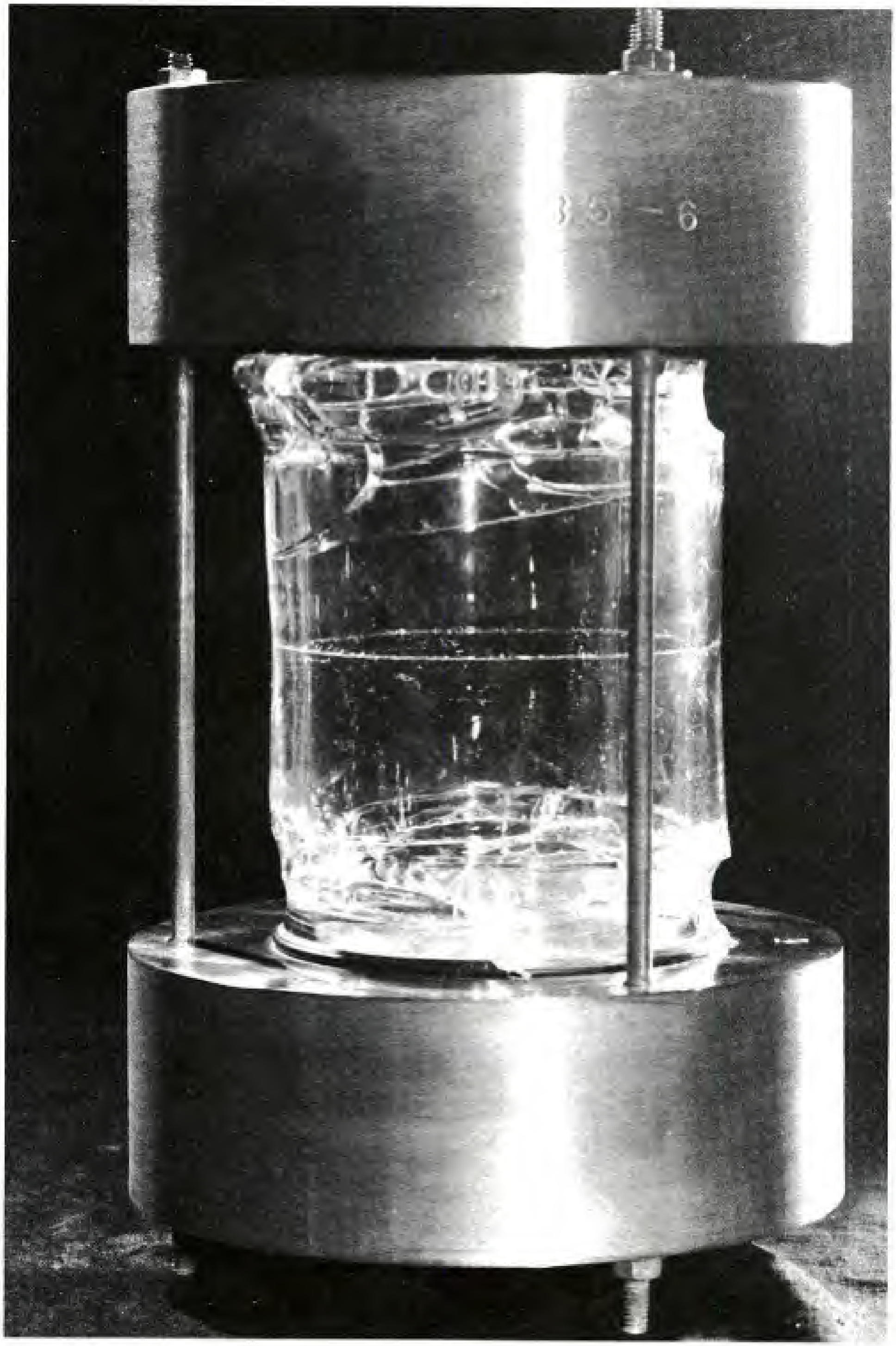


Figure 18. Four-inch-ID x 6-inch-long glass pipe utilizing an O-ring interposed between the glass pipe flange and the metal end-closure plate after exposure to 3,000 psi external hydrostatic pressure for one cycle.

Figure 19. Closeup of failure area of glass pipe tested with O-ring gasket between glass pipe and flange and metal end-closure plate.

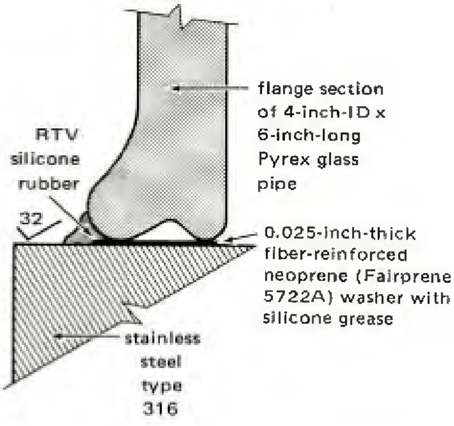


Figure 20. Experimental sealing and bearing system utilizing a fiber-reinforced neoprene washer between the glass pipe end flange and the metal end-closure plate.

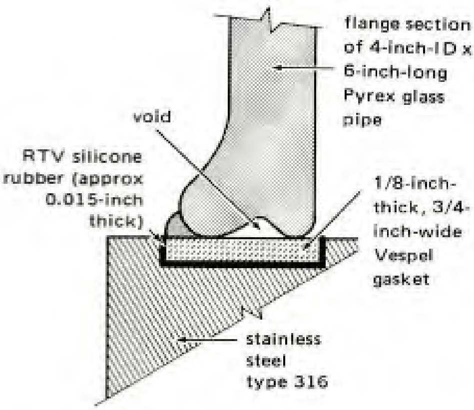


Figure 21. Experimental sealing and bearing system utilizing a Vespel washer between the glass pipe end flange and the metal end-closure plate.

dry lubricant material. Assembly was effected by placing the molybdenum-disulfide-coated face of a washer on the steel end plate, placing the glass pipe flange (coated with RTV silicone rubber) in contact with the copper face, then the second washer on the opposite RTV silicone rubber coated glass pipe flange (copper side down), and finally placing the second steel end plate on top of the washer. The assembly was then compressed by tightening the tie rods. The joints between the washers and end plates were then sealed with RTV silicone rubber. (See Figure 22.)

The purpose behind this system (and those described in paragraphs 5, 6, 7, and 8) was to provide a bearing surface (copper in this case) which would be soft enough to accommodate irregularities in the glass pipe end flanges and thus effectively eliminate stress concentrations due to point loading at those areas. Figure 23 shows the relative indentations in the aluminum (see paragraph 6), copper, and lead (see paragraph 5) bearing surfaces which resulted from one pressure cycle to 3,000 psi. These indentations give an indication of the degree to which these materials "accommodated" the configuration of the glass bearing surface. In addition, it was hoped that the glass and washer assembly would respond to changing external pressure as a unit when changing diameter due to the effects of such external hydrostatic pressure. The molybdenum disulfide coating on the washer—end closure contact area was provided to

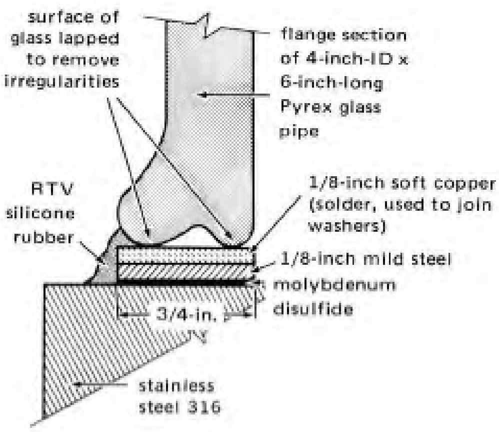


Figure 22. Experimental sealing and bearing system utilizing a composite copper—steel washer between the glass pipe end flange and the metal end-closure plate.

permit the washer to slide on the end closure as the diameters changed with pressure. If this system performed as hoped, it would not only reduce the stress concentrations in the glass—seal bearing area but would also largely eliminate tension cracking of the glass pipe end flanges caused by the glass "dragging" on the end closure as it attempted to change diameter in response to changing external pressure. Neither this system nor the similarly designed lead bearing washer system described below performed satisfactorily at 3,000 psi. Examination of the failed specimens showed many small cracks which originated in the glass—metal bearing area and

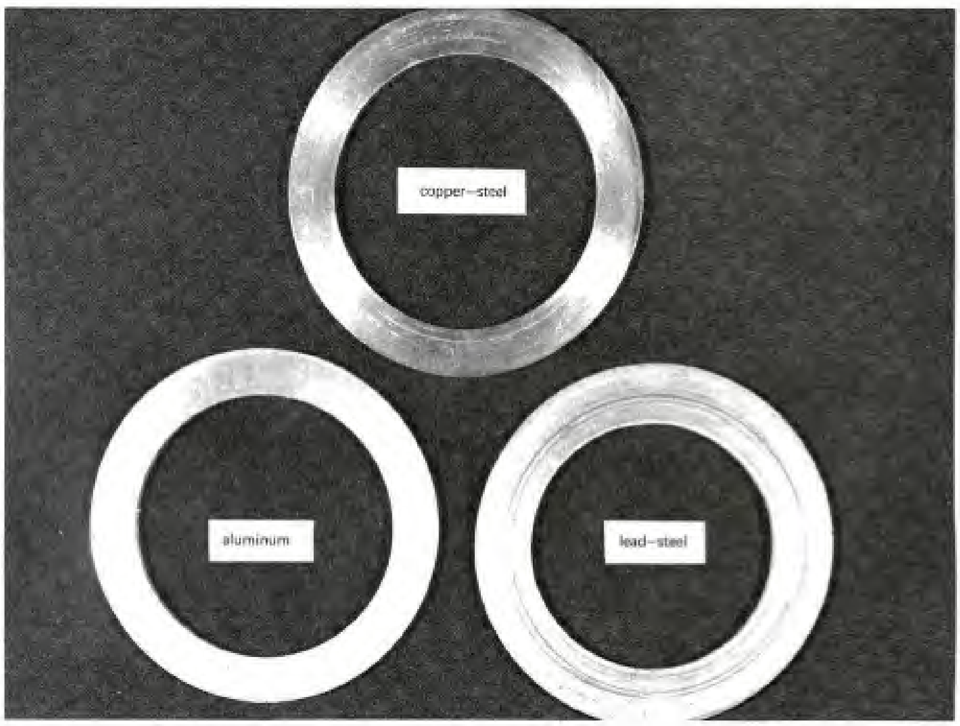


Figure 23. Aluminum, composite copper—steel, and composite lead—steel washers showing the relative depth of indentation resulting from pressurization of experimental sealing and bearing system assemblies to one cycle at 3,000 psi external hydrostatic pressure.

propagated up into the pipe wall. While this system very likely is satisfactory for cyclical service at some pressure less than 3,000 psi, it was not tested at lower pressures and its maximum useful depth rating is not known.

5. Composite gaskets were made by cementing 1/8-inch-thick steel washers to 1/8-inch-thick lead washers with epoxy cement (see Figure 24). The composite gaskets were then treated and assembled to the glass pipe end closures in the same manner as described above for the copper—steel gaskets.

While the lead bearing system provided the most "compliant" of the glass-to-metal bearing systems (see Figure 25), it did not produce the desired critical pressure for the system. All specimens tested failed during the first cycle to 3,000 psi. While this system very likely is satisfactory for cyclical service at some pressure less than 3,000 psi, it was not tested at lower pressure, and its maximum useful depth range is not known.

6. The gaskets in this test assembly consisted of 1/4-inch-thick washers machined from 6061-T6 aluminum and finished to a 32-rms surface finish. A 0.0025-inch-thick molybdenum disulfide coating was then applied to one side of each washer. Figure 26 shows the component parts of the specimen assembly. In the final assembly, the bare sides of the aluminum washers were sealed to the glass pipe end flanges with RTV silicone rubber and the molybdenum-disulfide-coated sides were placed in contact with the steel end-closure discs. The assembly was compressed by tightening the tie bolts, and the joints between the washers and steel end plates were then sealed with RTV silicone rubber (Figure 27).

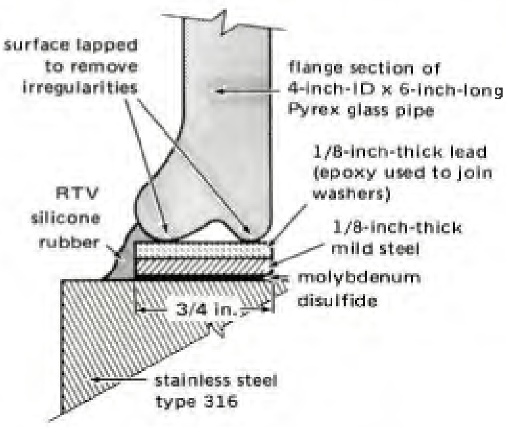


Figure 24. Experimental sealing and bearing system utilizing a composite lead—steel washer between the glass pipe end flange and the metal end-closure plate.

Figure 28 shows the aluminum bearing ring after 10 cycles to 3,000 psi followed by 10 cycles to 4,000 psi. The resulting indentation is very difficult to detect. One specimen assembly survived two 10-cycle tests successively to 3,000 and 4,000 psi. A second specimen failed after one cycle to 3,000 psi and a third specimen survived three 10-cycle tests successively to 3,000, 4,000, and 5,000 psi and six cycles to 6,000 psi.

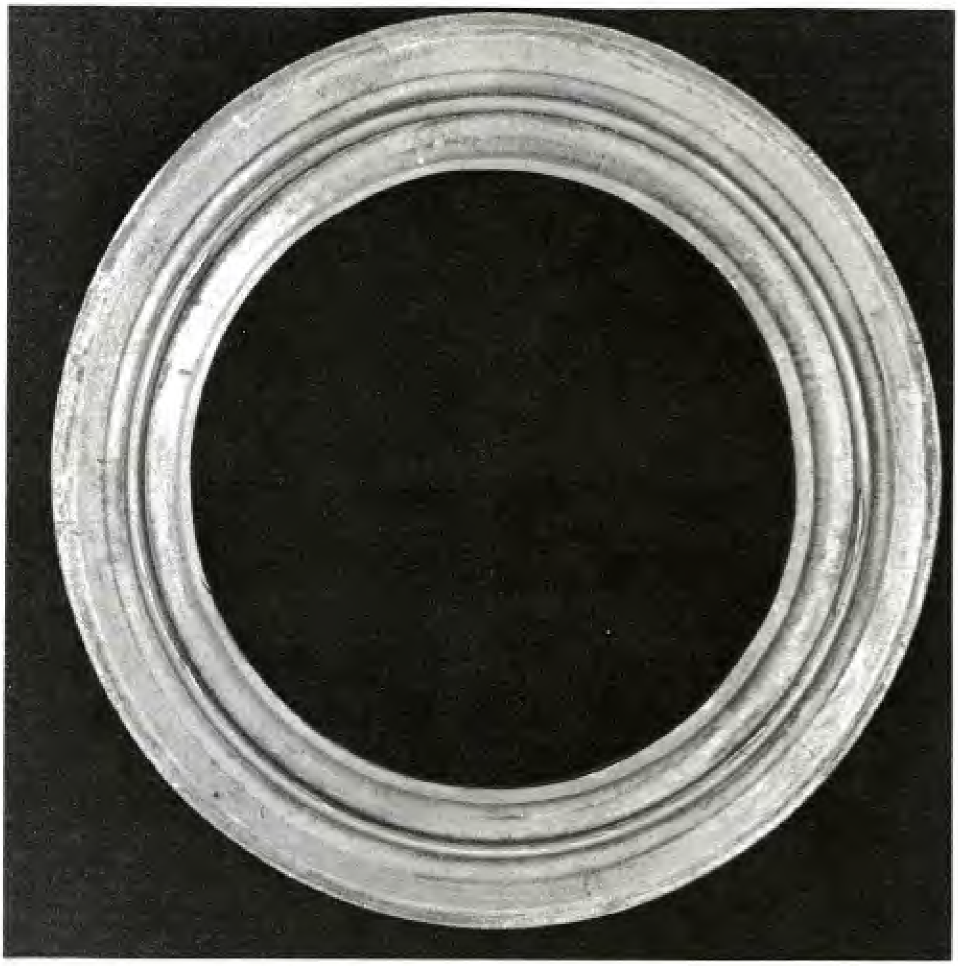


Figure 25. Composite lead–steel washer following external pressurization of test assembly to 3,000 psi. The lead has plastically conformed to the configuration of the glass pipe end flange and its integral O-ring groove.

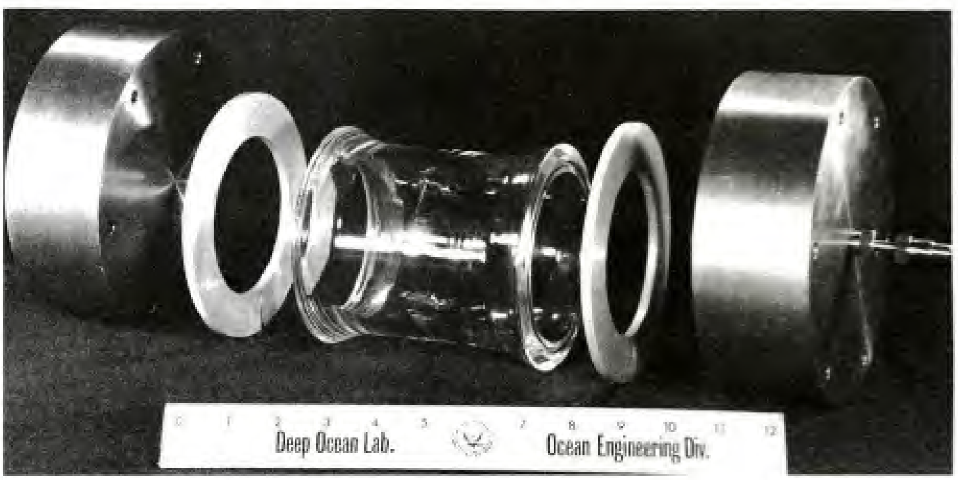


Figure 26. Component parts of an experimental test assembly used to test the seal provided by an aluminum washer interposed between the glass pipe end flanges and the metal end-closure plates.

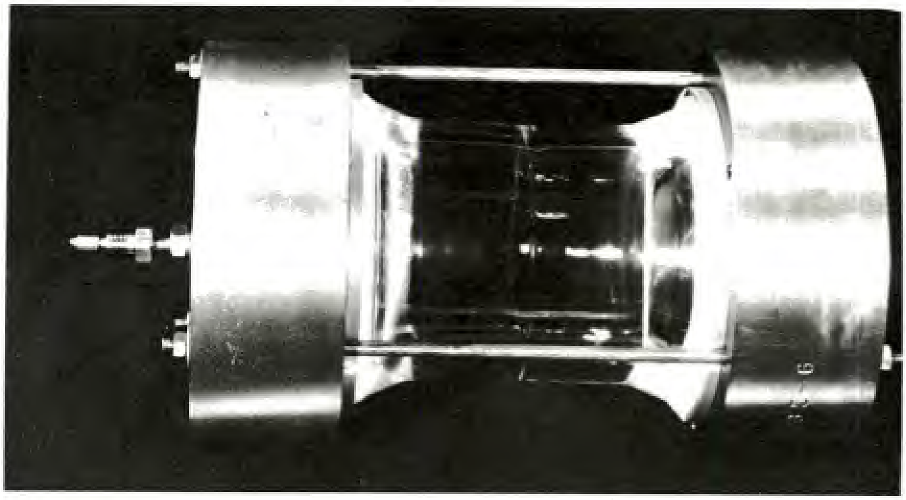


Figure 27. Assembled experimental testing system for aluminum seal ready for insertion in a pressure vessel.

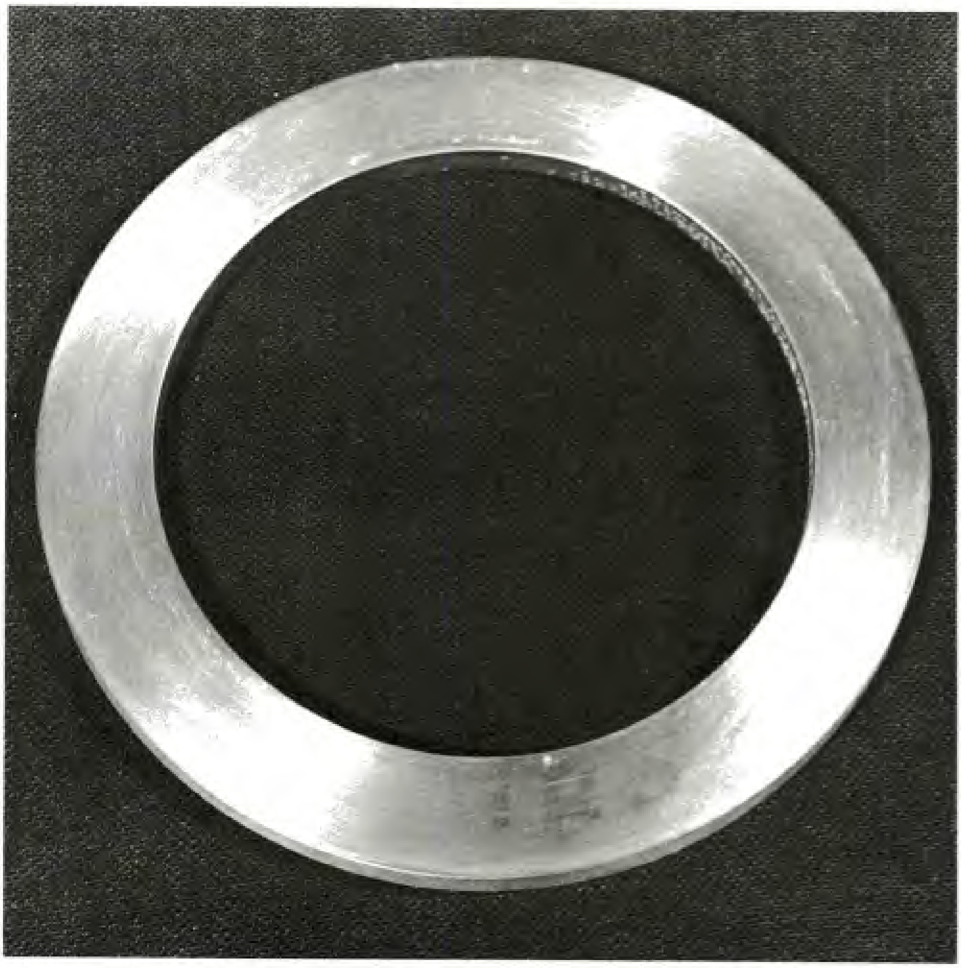


Figure 28. Experimental aluminum washer after being externally pressurized in a test assembly to 3,000 psi. The indentations resulting from the glass pipe pressing on the aluminum are just barely visible.

This system should be suitable for capsules for cyclical service to 6,000 feet. The one failure at 3,000 psi serves to point up the fact that the quality-control criteria applied to this type of glassware do not serve to exclude potential rejects for this type of application.

7. In this bearing—sealing system, two previous concepts were combined. Aluminum washers similar to those described in No. 6 above were fabricated. They differed in fabrication and treatment only in that a recess was machined in one outside edge to provide an O-ring groove when the assembly was made. (See Figure 29.) The O-ring in this groove provided the watertight seal between the washer and the steel end-closure disc. A second O-ring was placed in the groove in the glass pipe end flange. This O-ring made a watertight seal between the washer and the glass pipe. Molybdenum disulfide dry lubricant was applied to the areas of the washers in contact with the steel end plates, and silicone grease was used to lubricate the O-ring between the washer and the end plate, but no RTV silicone rubber was used in this system.

This system did not work satisfactorily at 3,000 psi. Each of the specimen assemblies tested failed from cracks originating in the O-ring groove of the glass pipe end flange—as in the case of the first gasket system tested, which also used an O-ring in the glass pipe flange. While this system very likely is satisfactory for cyclical service at some pressure less than 3,000 psi, it was not tested at lower pressures and its maximum useful depth range is not known.

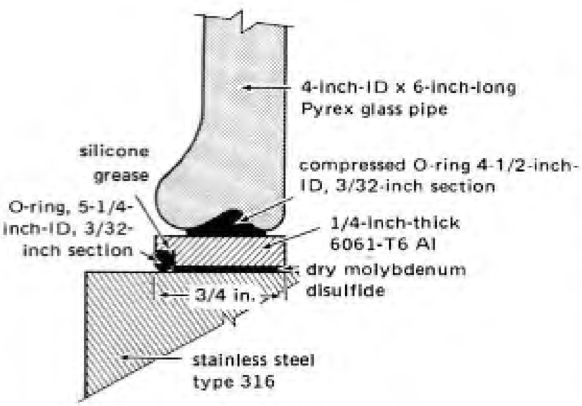


Figure 29. Experimental sealing and bearing system utilizing O-rings and an aluminum washer interposed between the glass pipe end flange and the metal end-closure plate.

8. These gaskets consisted of 1/4-inch-thick 60-shore-hardness neoprene sheet material cut into washers and fitted into grooves in the steel end plates. (See Figures 30 and 31.) The contact area between the glass pipe flanges and the rubber washer was lubricated with silicone grease, and the watertight seal was made after assembly by applying RTV silicone rubber to the joint between the glass and rubber.

This system failed by cracking and leaking in the transition area between the pipe flange and the cylindrical pipe body after seven cycles to 3,000 psi.

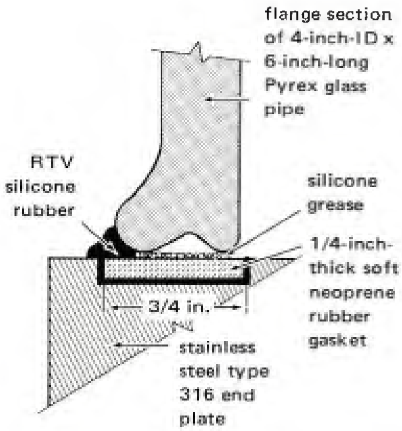


Figure 30. Experimental sealing and bearing system utilizing a 1/4-inch-thick captive neoprene washer between the glass pipe end flange and the metal end-closure plate.

Each specimen assembly of the eight different systems described above was filled with water and the specimen vented through a high-pressure connection (Figures 31 and 32) to a sight glass (outside the vessel) at atmospheric pressure. This enabled the observer to detect any minor leaks in the assembly which might otherwise go undetected until the specimen was removed from the vessel (assuming it was intact) on completion of the series of tests. Previous experience with cycling tests of glass specimens in which the glass cracked but did not fail violently indicated the desirability of a leak detector to determine precisely when a leak occurred. This high-pressure connection from the interior of the

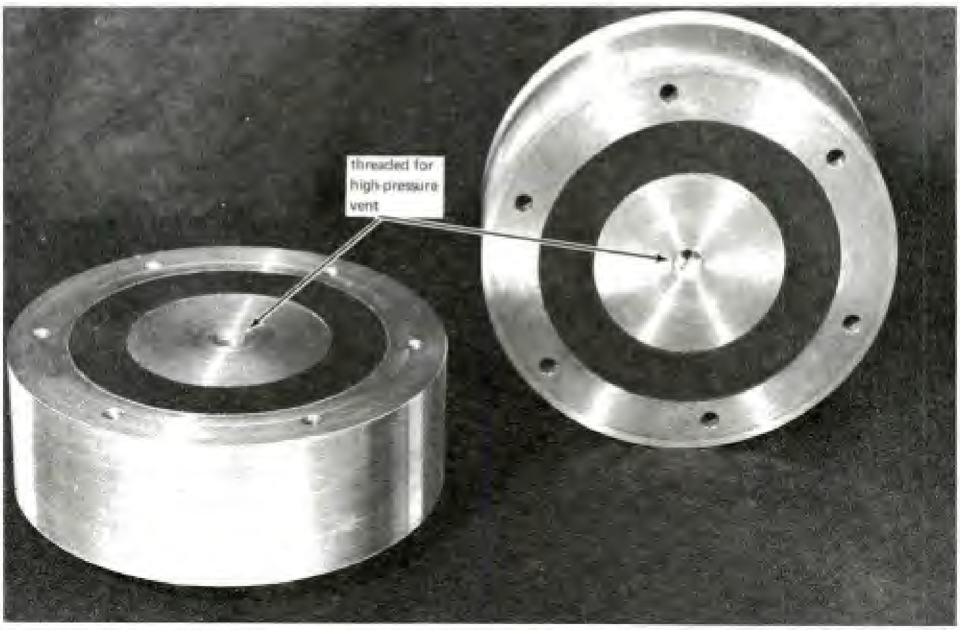


Figure 31. End-closure plates used in testing the Vespel and the neoprene washers (shown here). The high-pressure vent is utilized in the experimental measurement of internal volume change and for attenuation of implosion force.

Figure 32. Typical method of providing a high-pressure vent connection from the interior of a specimen assembly.

water-filled specimen to the atmosphere also served to permit the specimen to fail by cracking or to implode without the potentially damaging shock which results from the implosion of a void or even partially void specimen.

Table 5 summarizes the results of these experiments.

From the viewpoint of simplicity and low cost, the fiber-reinforced neoprene gasket (system 2) is the most satisfactory to depths of 6,000 feet. The systems which promise to give the 8,000-foot depth ratings for cyclical service are 3 and 6, which utilize Vespel and aluminum gaskets, respectively, to eliminate failure from the rubbing of the glass pipe end flanges on the end closure as the glass pipe changes in diameter with changes of external pressure.

Phase III: Prototype Housing Tests

In order to demonstrate typical applications of glass pipe to undersea use, two prototype housing designs were tested, one for a 1,000-watt light and one for an instrument housing.

Light Housing. Figure 33 shows a deep-submergence light specifically developed to utilize a glass pipe housing with conical flanges. This design utilizes a 1-1/2-inch-ID x 6-inch-long, flanged Pyrex pipe as the transparent section. Figure 34 shows the construction.

Operational testing of this assembly indicated a collapse pressure of 9,000 psi. It was subsequently successfully operated and pressure cycled at 2,500 psi. In view of its successful performance at 2,500 psi, it is considered reliable for operation at 5,000 feet.

Instrument Housing. Figures 35 and 36 show a prototype instrument housing developed at NCEL. This housing utilizes two 4-inch-ID, flanged Pyrex glass pipe caps and a section of 4-inch-ID, flanged Pyrex glass pipe

6 inches long. Interposed between the glass elements are two aluminum rings provided with fiber-reinforced neoprene washers, which provide a compliant seat between the glass and metal components. Sealing was effected by the use of a silicone grease, and initial compression was supplied by tensioned tie rods and metallic pipe flanges.

This assembly was pressure tested and found to fail at an average of 2,800 psi. Subsequent pressure cycling at 1,000 psi demonstrated the usefulness and reliability of this system at depths to approximately 2,000 feet in cyclical service.

MODES OF FAILURE

The specimens failed in two ways: cracking followed by implosion and cracking without implosion. Those which cracked without implosion probably would have imploded had pressurization been continued beyond the 20,000-psi limit used in these tests.

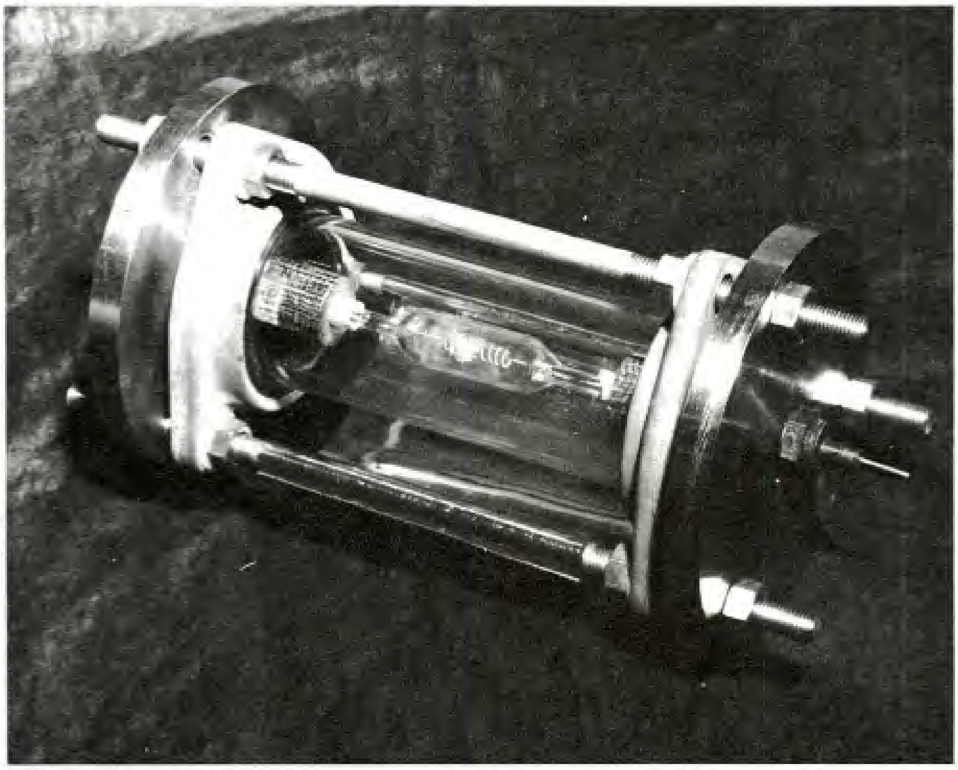


Figure 33. Prototype deep-submergence light for 10,000-foot depth service using 1-1/2-inch-ID x 6-inch-long glass pipe.

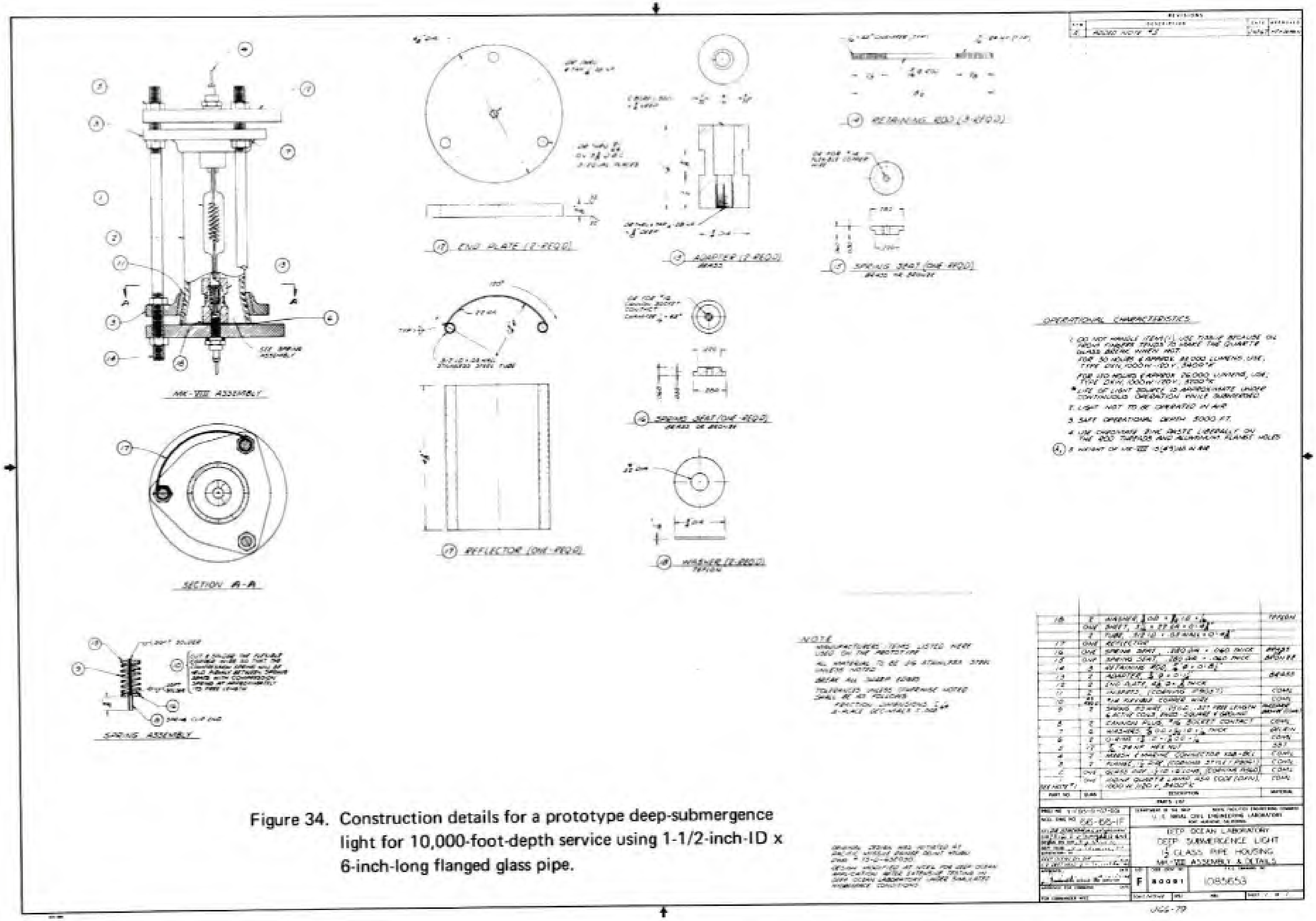


Figure 34. Construction details for a prototype deep-submergence light for 10,000-foot-depth service using 1-1/2-inch-ID x 6-inch-long flanged glass pipe.

Table 5. Results of Pressure Cycling Tests on 4-Inch-ID x 6-Inch-Long Glass Pipe Assemblies Having Various Bearing—Sealing Systems Interposed Between the Glass Pipe Flange and the Metal End Plates

(● = completion of cycle without leak; ■ = failure by cracking and leaking; ▲ = failure by implosion.)

Successive Pressure Cycling Tests on Same Specimen

Bearing System Interposed Between Glass Pipe Flange and Stainless Steel 316 End Plate	0 to 3,000 psi, cycle — 1	2	3	4	5	6	7	8	9	10
1. Neoprene O-ring (60-durometer, 3/32-inch section)	■									
	■									
2. Fiber-reinforced neoprene gasket (0.023-inch thick)	●	●	●	●	●	●	●	●	●	●
	●	●	●	●	●	●	●	●	●	●
	●	●	●	●	●	●	●	●	●	●
3. RTV silicone rubber with Vespel washer	●	●	●	●	●	●	●	●	●	●
	●	●	●	●	●	●	●	●	●	●
4. RTV silicone rubber with copper-steel composite washer plus molybdenum disulfide coating	●	▲								
	●	●	■							
5. RTV silicone rubber with lead-steel composite washer plus molybdenum disulfide coating	●	■								
	●	■								
6. RTV silicone rubber with aluminum washer plus molybdenum disulfide coating	●	●	●	●	●	●	●	●	●	●
	●	■								
	●	●	●	●	●	●	●	●	●	●
7. Neoprene O-ring—aluminum washer plus molybdenum disulfide coating	■									
	■									
	■									
	■									
	■									
8. RTV silicone rubber with thick rubber washer	●	●	●	●	●	●	●	■		

Bearing System Interposed Between Glass Pipe Flange and Stainless Steel 316 End Plate	0 to 4,000 psi, cycle — 1	2	3	4	5	6	7	8	9	10
1. Neoprene O-ring (60-durometer, 3/32-inch section)										
2. Fiber-reinforced neoprene gasket (0.023-inch thick)	●	●	●	●	●	●	●	●	●	●
	●	●	●	●	●	■				
	●	●	●	●	●	●	●	■		
3. RTV silicone rubber with Vespel washer	●	●	●	●	●	●	●	●	●	●
	●	●	●	●	●	●	●	●	●	●
4. RTV silicone rubber with copper-steel composite washer plus molybdenum disulfide coating										
5. RTV silicone rubber with lead-steel composite washer plus molybdenum disulfide coating										
6. RTV silicone rubber with aluminum washer plus molybdenum disulfide coating	●	●	●	●	●	●	●	●	●	●
	●	●	●	●	●	●	●	●	●	●
7. Neoprene O-ring—aluminum washer plus molybdenum disulfide coating										
8. RTV silicone rubber with thick rubber washer										

Bearing System Interposed Between Glass Pipe Flange and Stainless Steel 316 End Plate	0 to 5,000 psi, cycle — 1	2	3	4	5	6	7	8	9	10
1. Neoprene O-ring (60-durometer, 3/32-inch section)										
2. Fiber-reinforced neoprene gasket (0.023-inch thick)	■									
3. RTV silicone rubber with Vespel washer	●	●	●	●	●	●	●	●	●	●
	■									
4. RTV silicone rubber with copper-steel composite washer plus molybdenum disulfide coating										
5. RTV silicone rubber with lead-steel composite washer plus molybdenum disulfide coating										
6. RTV silicone rubber with aluminum washer plus molybdenum disulfide coating	●	■								
	●	●	●	●	●	●	●	●	●	●
7. Neoprene O-ring—aluminum washer plus molybdenum disulfide coating										
8. RTV silicone rubber with thick rubber washer										

Bearing System Interposed Between Glass Pipe Flange and Stainless Steel 316 End Plate	0 to 6,000 psi, cycle — 1	2	3	4	5	6	7	8	9	10
1. Neoprene O-ring (60-durometer, 3/32-inch section)										
2. Fiber-reinforced neoprene gasket (0.023-inch thick)										
3. RTV silicone rubber with Vespel washer	■									
4. RTV silicone rubber with copper-steel composite washer plus molybdenum disulfide coating										
5. RTV silicone rubber with lead-steel composite washer plus molybdenum disulfide coating										
6. RTV silicone rubber with aluminum washer plus molybdenum disulfide coating										
	●	●	●	●	●	●	■			
7. Neoprene O-ring—aluminum washer plus molybdenum disulfide coating										
8. RTV silicone rubber with thick rubber washer										

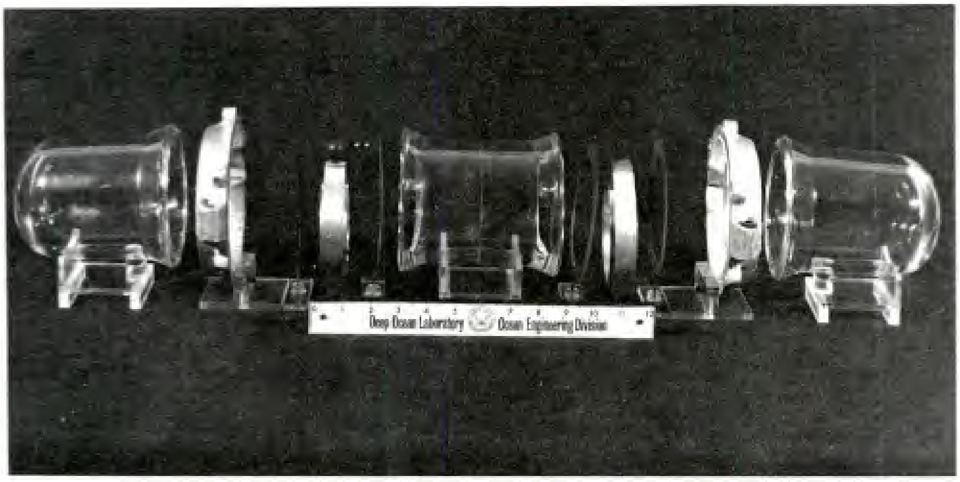


Figure 35. Component parts for a 4-inch-ID prototype glass instrument housing for cyclical service to a depth of 1,000 feet.

Failure by implosion produced a loud, sharp report clearly audible up to 50 feet from the pressure vessel. High-pitched "cracking" noises were generally heard prior to implosion. As implosion pressure approached, these noises increased in amplitude and became clearly-audible to personnel behind a barricade several feet away from the pressure vessel. As the testing program progressed, it became apparent that the inception of the cracking sounds was not being reliably observed due to its initial low amplitude and ambient noise. In order to observe this phenomena and positively determine when it started, a contact microphone and an audioamplifier were procured and the microphone securely attached to the outside of the pressure vessel. Using this technique, it was noted that the first "cracking" frequently took place at relatively low pressures. However, it should be noted that in some cases, even though cracking noises were observed, there was no visible damage to the glass tubes. This cracking noise in many cases may be generated by the glass bearing surfaces dragging and "chattering" on the end closure as the end of the pipe decreased in diameter in response to the external hydrostatic pressure.

The actual cracks observed in specimens appear to be of three general types. ***First*** the cracking and subsequent spalling of very thin shards in the area where the inside rim of the pipe contacts the end plate (Figure 37). ***Second*** are the concentric shear cracks propagating from the pipe surface in contact with the end closure (Figure 38). These cracks, in most cases, result in the cracking off of the portion of the pipe flange which is greater in outside diameter than the main section of the pipe, and thus unsupported. They appear to be a result of a combination of the shearing of the glass pipe flange

Figure 36. Assembled prototype 4-inch-ID glass instrument housing for cyclical service to a depth of 1,000 feet.

from the main body of the pipe and tension resulting from the dragging of the pipe flange across the end plates as the glass pipe diameter is reduced by external pressure. *Third* are the radial cracks extending from the glass-to-metal contact area up into the main body of the pipe. (See Figure 39.) These cracks originate at the ends and propagate radially along the longitudinal axis of the pipe. These cracks often interconnect so that the pipe falls apart when the pressure is relieved, though in many cases the badly cracked pipe may not leak while under increasing or constant pressure. These cracks appear to be the result of stress concentrations resulting from small protruding irregularities on the pipe-flange face. These irregularities on some of the specimens are large enough to cause the pipe to "rock" when placed on a flat surface. These irregularities can be eliminated by grinding and polishing the pipe flanges. This was not done, however, since one objective of this study was to evaluate off-the-shelf glassware, not custom-finished glassware.

PREDICTION OF CRITICAL PRESSURES

Although implosion testing of glass pipes with conical flanges has experimentally determined the critical pressures of various sizes of glass pipes, it is also important to be able to correlate this data with some sort of an analytical expression for the prediction of critical pressures. If the correlation between experimental and calculated critical pressures is good, then such an analytical expression can be used with confidence to predict the critical pressures of glass pipe sizes that have not been tested in this particular experimental program.

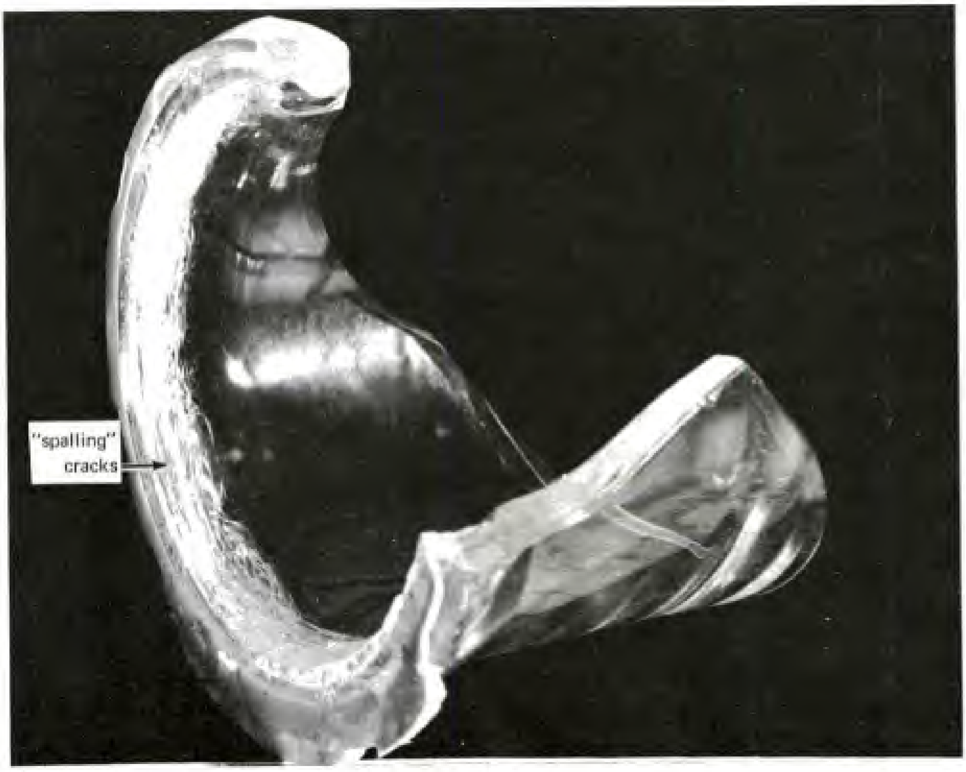


Figure 37. Typical example of the cracking and spalling occurring in a 4-inch-ID x 6-inch-long glass pipe with aluminum end-closure plates externally pressurized to 20,000 psi.

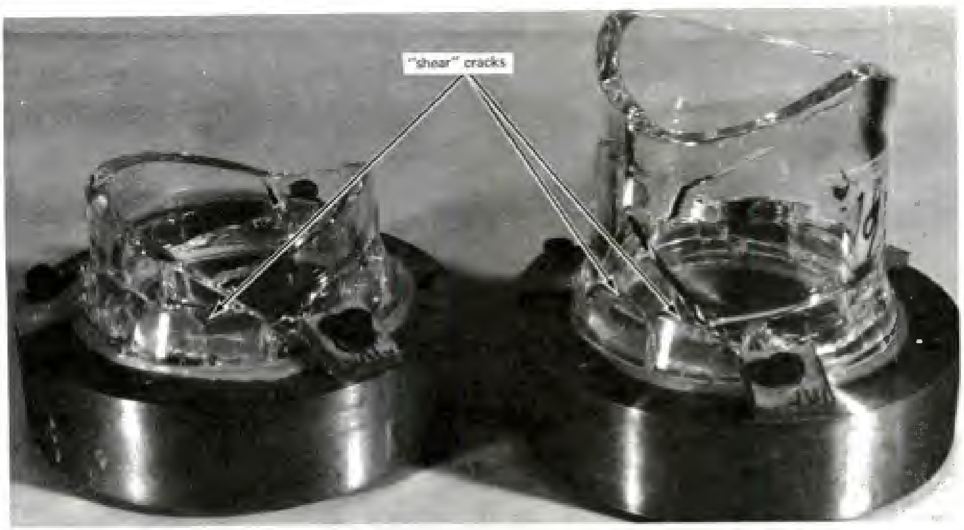


Figure 38. Shear cracks result in the spalling off of the portion of the pipe flange greater in diameter than the outside diameter of the main section of the pipe (4-inch-ID x 6-inch-long pipe with titanium end-closure plates externally pressurized to 20,000 psi).

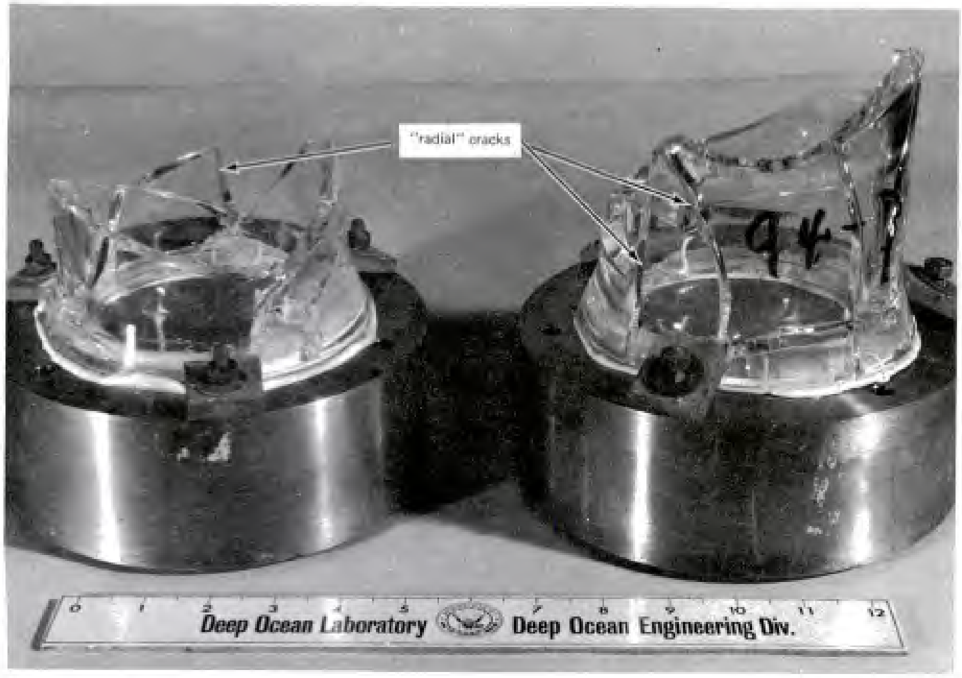


Figure 39. A typical example of radial cracking (4-inch-ID x 6-inch-long pipe with stainless steel 316 end-closure plates externally pressurized to 20,000 psi).

There are many analytical expressions for the calculation of critical pressure in cylinders with or without stiffeners. Generally, the complex analytical expressions are better than the simple expressions for predicting the critical pressure. Thus, at first glance, it would appear that it is much more desirable to use complex expressions than the simple ones since the calculated critical pressures will be much closer to the experimental ones. Unfortunately this is true only so long as detailed measurements and specifications exist for the given test specimen, as when it is an item custom-made to very rigid dimensional and material specifications.

For mass-produced glass pipes with conical flanges detailed specifications to close tolerances are not available because in the mass-production fabrication process large variations in wall thickness, roundness, and quality of glass welds exist. Because of the discrepancy that exists between the nominal and actual dimensions of the pipes, calculated values must differ considerably from experimental values even if the analytical expression used in the calculations is the correct one. Because of this discrepancy, little can be gained by going to elaborate analytical expressions when only the nominal pipe dimensions supplied by the manufacturer are used in the calculations.

In view of these considerations, it was decided for critical pressure calculations to utilize only the nominal pipe dimensions supplied by the manufacturer and an analytical expression which would combine simplicity with fair accuracy. Such an analytical expression is the R. von Mises[3] equation for buckling of monocoque cylinders equipped at their ends with simple supports in the form of ring stiffeners or bulkheads (Equation 1).

$$y = \frac{1-\sigma^2}{n^2+\frac{\alpha^2}{2}-1}\left(\frac{\alpha^2}{\alpha^2+n^2}\right)^2 + \frac{X}{n^2+\frac{\alpha^2}{2}-1}\left[(n^2+\alpha^2)^2 - 2\mu_1 n^2 + \mu_2\right] \ldots \quad (1)$$

where $\mu_1 = 1/2\left[1 + (1+\sigma)\rho\right]\left[2 + (1-\sigma)\rho\right]$

$$\mu_2 = (1-\sigma\rho)\left[1 + (1+2\sigma)\rho - (1-\sigma^2)\left(1+\frac{1+\sigma}{1-\sigma}\rho\right)\rho^2\right]$$

$$X = \frac{t^2}{3D^2}$$

$$y = p\frac{D}{2t}\left(\frac{1-\sigma^2}{E}\right)$$

$$\rho = \frac{\alpha^2}{n^2+\alpha^2}$$

$$\alpha = \frac{\pi D}{2L}$$

To solve Equation 1, one must find that whole number **n** of buckling lobes on the cylinder which would make the collapse pressure p a minimum for a given mean cylinder diameter **D**, wall thickness **t**, and length between supports **L**. Although this analytical expression is rather simple, time consuming and repetitious calculations must be performed before that number **n** of buckling lobes is determined which makes p a minimum for a given cylinder under hydrostatic pressure. For this reason, several approximations of Equation 1 have been developed[3] (Equation 2) which in conjunction with the expression for long unstiffened cylinders[4] (Equation 3) permit rapid calculation of the collapse pressure for any cylinder.

The pair of equations that when substituted for R. von Mises expression permit rapid approximation of the collapse pressure due to buckling are:

$$\rho = \frac{2.42\,E}{(1-\mu^2)^{0.75}} \left[\frac{\left(\frac{t}{D}\right)^{5/2}}{\frac{L}{D} - 0.45\left(\frac{t}{D}\right)^{1/2}} \right] \qquad (2)$$

$$\rho = \frac{2\,E}{(1-\mu^2)} \left(\frac{t}{D}\right)^3 \qquad (3)$$

where L = length of cylinder between stiffeners, inches

D = mean diameter of cylinder, inches

t = wall thickness, inches

E = modulus of elasticity, psi

μ = Poisson's Ratio

ρ = collapse pressure, psi

After solving both equations for the dimensions of a given cylinder, the higher collapse pressure is utilized as the correct one. However, a more desirable solution to Equation 1 would be a set of plotted nondimensional curves that would permit the user in the field to determine immediately the predicted collapse pressure for a given cylinder without involved calculations. Such a plot of Equation 1 has been prepared (Figure 40) in nondimensional form. Because they are nondimensional, the graphs can be used to predict the buckling collapse pressure of monocoque cylinders between simple supports regardless of cylinder composition.

A major problem encountered in the comparison of experimental and analytical data is that test specimens and methods of test do not exactly agree with the basic assumptions of the analytical expression. In the experimental testing program of standard glass pipes with conical flanges, three basic differences (see next paragraph) from the analytical expression exist. The analytical expression is based on the assumptions that: (1) the monocoque cylinder is of uniform wall thickness, (2) the cylindrical radius is uniform throughout the length of the cylinder, (3) the ends of the cylinder are simply supported, (4) the material is perfectly elastic, and (5) the implosion of the cylinder is not initiated by failure of the material, but by elastic instability of the cylinder.

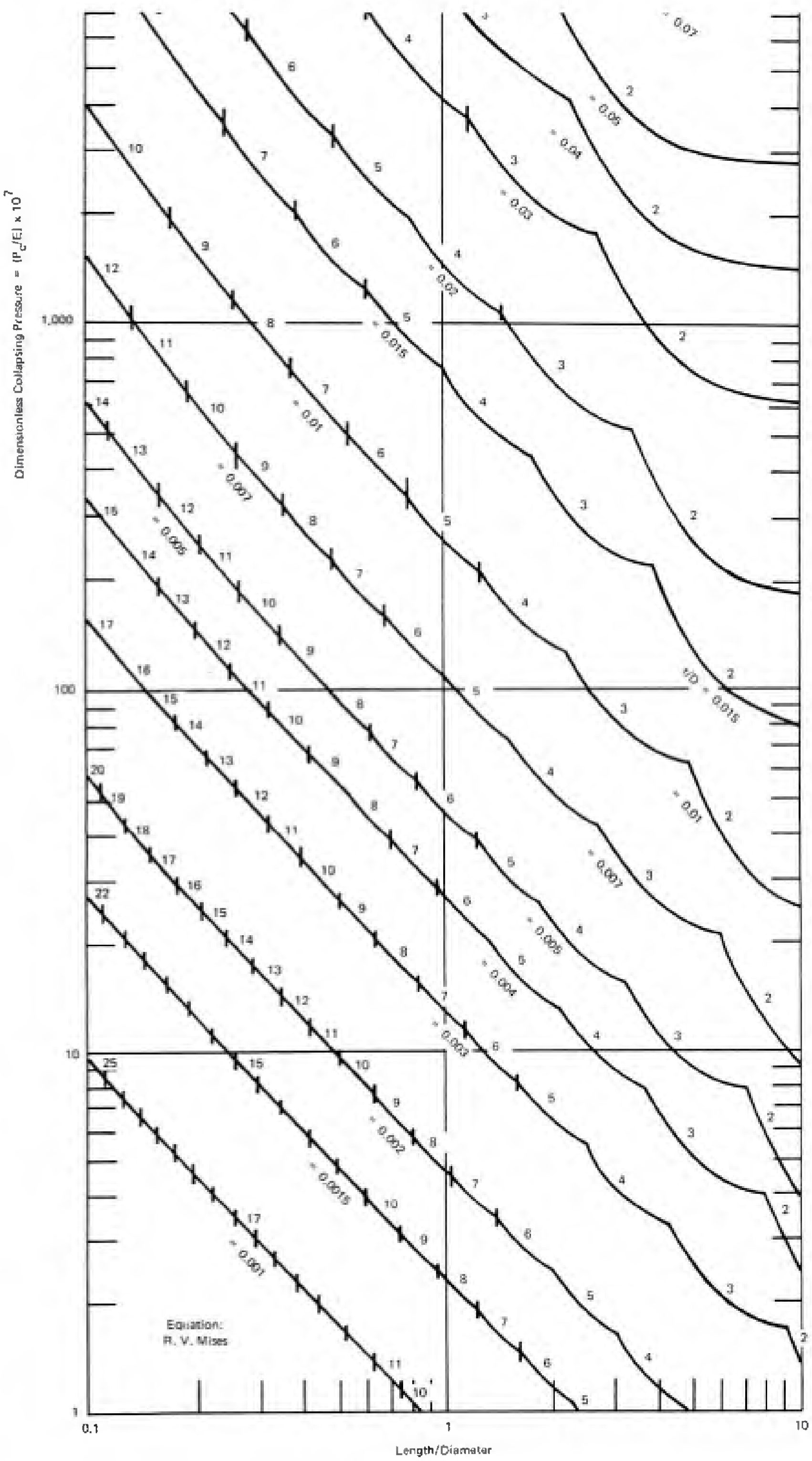
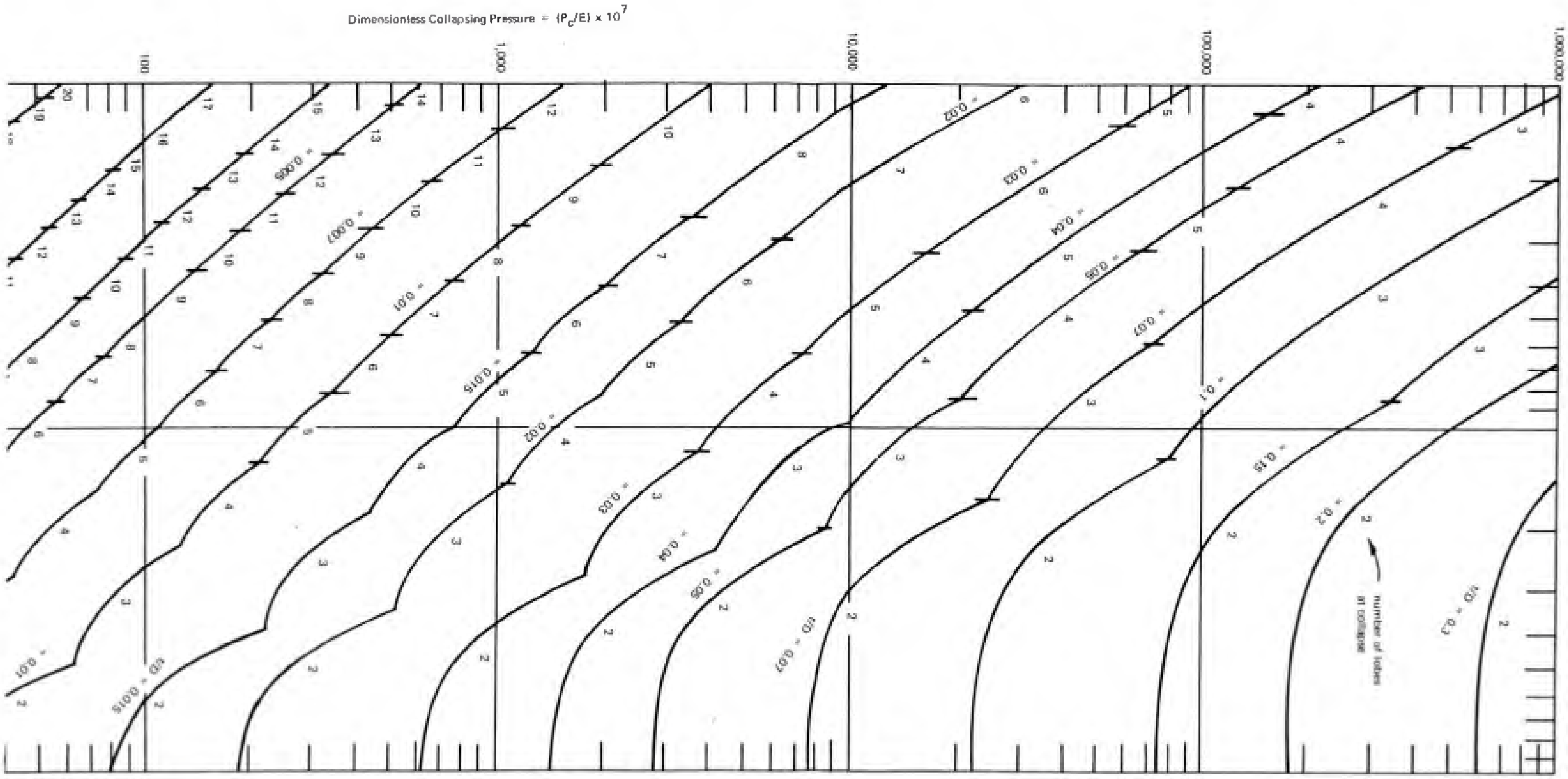


Figure 40. Elastic buckling of cylindrical shells between stiffeners under external hydrostatic pressure.[5]



Dimensionless Collapsing Pressure = $(P_c/E) \times 10^7$
100
1,000
10,000
100,000
1,000,000
t/D = 0.005
= 0.007
= 0.01
t/D = 0.015
= 0.02
= 0.03
= 0.04
= 0.05
t/D = 0.07
= 0.1
= 0.15
= 0.2
t/D = 0.3
number of lobes at collapse

The properties of glass pipes with conical end flanges differ substantially from those assumed in the analytical expression. The pipes, first of all, are not of uniform thickness and do not possess a uniform radius. This is shown by the variation in thickness of plus or minus 20% from nominal value and the plus or minus 4% variation of external radius measured on glass pipes tested. In addition, the conical end flanges do not give the glass cylinder simple end support, but a support which is neither simple nor rigid, but a cross between the two. The taper in the end flanges introduces an additional uncertainty: the cylinder length between end supports. If the overall length of pipe sections (including the tapered flange portion) is used in the equation, different critical pressures will be obtained from the equation than if the length of the cylindrical portion between tapered sections of the pipe is used. In addition to these uncertainties, there is added the presence of a stress raiser in the form of an O-ring groove in the flange that may cause the failure of the glass pipe at lower pressure than if the implosion of the pipe took place due to elastic buckling. Only after the user of the analytical expression understands all of these differences between the assumptions on which the equation is based and the measurements of the actual test specimens is he ready to intelligently apply the equation to the calculation of critical pressure due to buckling for glass pipes with conical flanges.

For the purpose of comparing calculated with experimental implosion pressures, the following dimensions and properties of pipes were used for plotting of experimental results (Figure 41):

D—mean diameter, as determined by subtracting one nominal wall thickness from nominal external diameter specified by manufacturer

t—nominal wall thickness specified by manufacturer of glass pipe

E—nominal modulus of elasticity for borosilicate glass used in the fabrication of pipe, 9.1×10^6 psi

L—nominal length of pipe, the distance between the two flat bearing surfaces on the ends of the pipe

Comparison of experimentally determined implosion values with the calculated curves for the same **t/D** ratio discloses rather good agreement between them. The agreement becomes more remarkable when it is noted that there are so many dimensional variations in the pipes and the flanges at the end of the pipes do not represent the simple supports specified by the analytical expression. Only one pipe size, the 6-inch-ID x 6-inch-long pipe, imploded at pressures significantly lower than the calculated pressure.

Characteristically, this is also the only pipe configuration tested where the distribution of cracks was distinctly different from those in other pipe configurations. In the 6 x 6-inch pipe configuration, all of the cracks were radial oriented along longitudinal axis of the pipe, while in other pipe configurations they were mostly of the circumferential type. It would thus appear that because of the low L/D ratio for the 6 x 6-inch pipe configuration, the stress distribution is such that failure occurs at lower hydrostatic pressure due to failure of material initiated by stress raisers on the bearing surfaces rather than at the higher pressure predicted for it by elastic buckling theory.

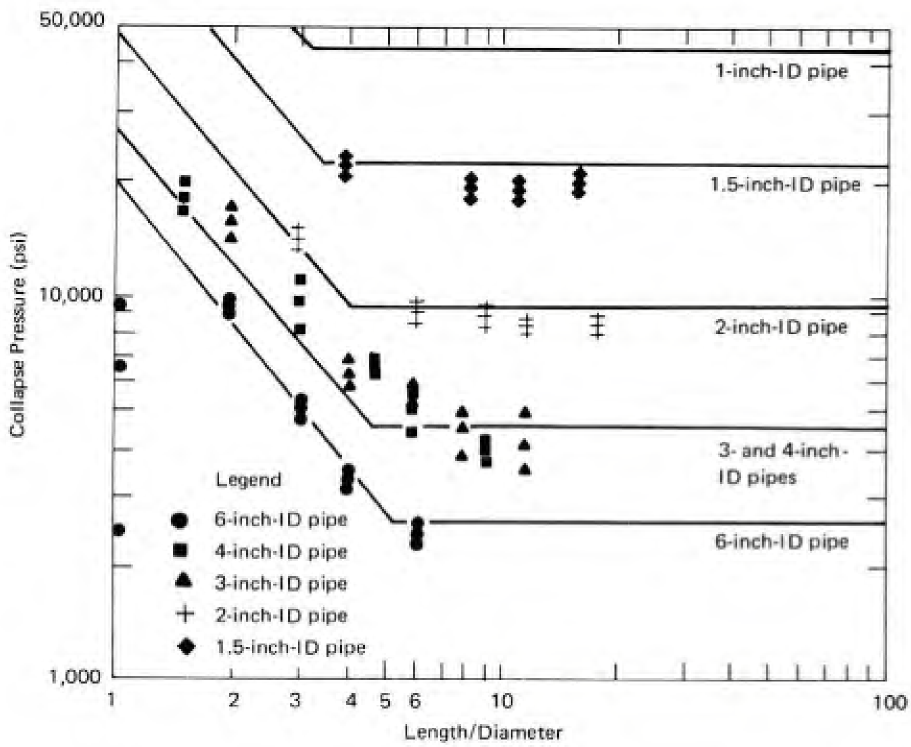


Figure 41. Comparison of actual pipe implosion pressures with pressures calculated on the basis of Equations 2 and 3.

In general, with only one minor exception, the plot of R. von Mises equation has been found helpful in predicting the implosion pressure due to buckling of glass pipes with conical flanges. Needless to say, extensive cracking may, and in most cases does, occur prior to the act of implosion. Because of it, the actual operational depth to which the glass housings may be repeatedly subjected without damage is considerably less. The relationship between the magnitude of the implosion depth and of safe operational depth can be seen from comparison of Tables 2 and 6, which show the implosion pressures and safe operational depths, respectively.

Table 6. Recommended Depth Ratings* and End-Closure Thicknesses for Various Diameters of Flanged Pyrex Glass Pipe

Length of Pipe (in.)	Inside Diameter (in.)	One Way Dive, Depth (ft)	One Round-Trip Dive, Depth (ft)	Multiple Dives, Depth (ft)	Thickness of** End Closures (in.) (6061-T6 Aluminum)
6	1	28,000	19,000	5,000	3/4
	1.5	22,000	15,000	4,000	15/16
	2	16,000	11,000	3,000	1-1/8
	3	16,000	11,000	3,000	1-3/4
	4	16,000	11,000	3,000	2-1/4
	6	3,000	2,000	500	1-1/2
12	1	28,000	19,000	5,000	3/4
	1.5	20,000	13,000	4,000	15/16
	2	10,000	6,000	2,000	7/8
	3	7,000	5,000	1,500	1
	4	7,000	5,000	1,500	1-1/2
	6	7,000	5,000	1,500	2-1/4
18	1	28,000	19,000	5,000	3/4
	1.5	19,000	13,000	3,800	15/16
	2	10,000	6,000	2,000	7/8
	3	6,000	4,000	1,100	1
	4	5,000	3,000	1,500	1-1/4
	6	4,000	2,000	800	1-5/8
24	1	28,000	19,000	5,000	3/4
	1.5	19,000	13,000	4,000	15/16
	2	9,000	6,000	2,000	13/16
	3	4,000	2,500	800	13/16
	4	4,000	2,500	800	1-1/8
	6	3,000	2,000	500	1-1/2
36	1	28,000	19,000	5,000	3/4
	1.5	19,000	13,000	4,000	15/16
	2	9,000	6,000	2,000	13/16
	3	4,000	2,500	800	13/16
	4	4,000	2,500	800	1-1/8
	6	2,000	1,500	500	1-1/8

* These ratings reflect not only the experimental data contained in this report, but also experimental data from NCEL TR-523, TR-559, and past experience with reproducible failure pressures for mass-produced stock glass items.

** For a one-way dive, bare aluminum end closures may be used; for round trip or multiple dive service nylon fiber reinforced neoprene gaskets (Fairprene 5722A) are required.

SUMMARY

Phase I

Off-the-shelf flanged glass pipe can be used to provide transparent, nonmagnetic instrument housings of very simple and inexpensive construction for use in the ocean. Depending on the size requirements, the Pyrex glass pipe tested in this study has a depth capability ranging from 2,000 feet to the greatest depths in the ocean for one-time (no cycling) use.

Phase II

1. Of the readily available materials tested, 6061-T6 aluminum proved the most satisfactory for use in a simple end plate-to-glass (without bearing gaskets) closure system for one-time service.

2. For limited cyclical service, naval brass appears to be the best choice in a simple end plate-to-glass (without bearing gaskets) closure system.

3. Of the various bearing gasket systems tested, fiber-reinforced neoprene washer (Fairprene 5722A), the Vespel washer (6061-T6), and the aluminum washer system showed the most promise.

Phase III

1. A prototype light housing utilizing 1-1/2-inch-ID glass pipe was designed, constructed, and tested; it was found to be useful for cyclical service at depths to 5,000 feet.

2. A prototype instrument housing utilizing 4-inch-ID glass pipe was designed, fabricated, and tested; it was determined to be useful for cyclical service to depths of 2,000 feet.

CONCLUSION

Flanged glass drain pipe provides a useful, inexpensive, transparent capsule material for enclosing lights and instruments for undersea use.

RECOMMENDATIONS

A summary of recommended depth ratings and end-closure thicknesses, based on the available experimental data and experience of the authors is given in Table 6 for various diameters and lengths of Pyrex glass pipe with flanged ends.

REFERENCES

1. Naval Civil Engineering Laboratory. Technical Report R-532: Light housings for deep-submergence applications—Part I. Four-inch-diameter glass flasks with conical pipe flanges, by J. D. Stachiw and K. O. Gray. Port Hueneme, Calif., June 1967. (AD 653293)

2.————. Technical Report R-559: Light housings for deep-submergence applications—Part II. Miniature lights, by J. D. Stachiw and K. O. Gray. Port Hueneme, Calif., Jan. 1968. (AD 663890)

3. David Taylor Model Basin. Report 366: The critical external pressure of cylindrical tubes under uniform radial and axial load, by D. F. Windenburg. Corderock, Md., Aug. 1931. (Translation of: R. von Mises. "Der kritische Aussendruck für allseits belastete zylindrische Rohre," in Stodolas Festschrift, Zurich, 1929, pp. 418-430)

4. "Application of the energy test to the collapse of a long thin pipe under external pressure," Cambridge Philosophical Society, Proceedings, vol. 6, 1888, pp. 287-292.

5. J. D. Stachiw and K. O. Gray. "Instrument capsules for deep submergence undersea use," in Proceedings of the 20th Annual Conference of the Instrument Society of America, Los Angeles, Calif., Oct. 4-7, 1965. (Paper no. 12.1-3-65)

DISTRIBUTION LIST

SNDL Code	No. of Activities	Total Copies	
–	1	20	Defense Documentation Center
FKAIC	1	10	Naval Facilities Engineering Command
FKNI	13	13	NAVFAC Engineering Field Divisions
FKN5	9	9	Public Works Centers
FA25	1	1	Public Works Center
–	15	15	RDT&E Liaison Officers at NAVFAC Engineering Field Divisions and Construction Battalion Centers
–	305	305	NCEL Special Distribution List No. 9 for persons and activities interested in reports on Deep Ocean Studies

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Naval Civil Engineering Laboratory
LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS (Final), by K. O. Gray and J. D. Stachiw
TR-618 48 p. illus Mar 1969 Unclassified

1. Deep-submergence light housings—Commercial flanged glass pipe I. Y-F015-01-07-001

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

Unclassified
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (*Corporate author*) Naval Civil Engineering Laboratory Port Hueneme, California 93041	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP

3. REPORT TITLE

LIGHT HOUSINGS FOR DEEP-SUBMERGENCE APPLICATIONS—PART III. GLASS PIPES WITH CONICAL FLANGED ENDS

4. DESCRIPTIVE NOTES (*Type of report and inclusive dates*)

Final; November 1966—June 1968

5. AUTHOR(S) (*First name, middle initial, last name*)

K. O. Gray and J. D. Stachiw

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
March 1969	48	5

8a. CONTRACT OR GRANT NO. b. PROJECT NO. Y-F015-01-07-001 c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) TR-618 9b. OTHER REPORT NO(S) (*Any other numbers that may be assigned this report*)

10. DISTRIBUTION STATEMENT

This document has been approved for public release and sale; its distribution is unlimited.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	Naval Facilities Engineering Command Washington, D. C.

13. ABSTRACT

The objective of this study was to evaluate commercially available glass pipes with conical flanged ends for application as transparent housings for underwater lights and instruments. Flanged-end glass pipes in diameters ranging from 1 inch to 6 inches and in lengths ranging from 6 inches to 36 inches were imploded under short-term and cyclic external pressure loading. Collapse pressure and recommended operational depth data are presented for one-way trip, round-trip, and cyclical applications. Recommendations for end-closure systems are also presented. One prototype light design and one prototype instrument housing design are described.

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Light housings						
Flanged glass pipes						
End closures						
Sealing and bearing systems						
Implosion						
Cyclical loading						
Short-term loading						