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EXPLORATORY INVESTIGATION OF NONWELDED PRESSURE HULLS FOR HYDROSPACE VEHICLES

by

[redacted]

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STRUCTURAL MECHANICS LABORATORY

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ABSTRACT

The feasibility of fabricating a deep-submergence pressure hull composed of rings joined by means other than welding was explored in tests of three structural models (two of aluminum and one of titanium) designed for a collapse strength of 10,000 psi. These tests demonstrated that hulls can be built without welding and that these hulls can have collapse strengths comparable to monolithic hulls. Longitudinal strength, watertight integrity, and corrosion protection, which were not explored by these tests, can be provided by any of several mechanical techniques without compromising collapse strength.

INTRODUCTION

Growing interest in the ocean depths both in scientific and military circles has emphasized the need for vehicles capable of operating at great depths. It is recognized that the materials and the fabrication techniques now in use place a severe limitation on the size and maximum operating depth of positively buoyant vehicles. Much interest has been shown in using light, high-strength materials such as aluminum, titanium, glass-reinforced plastics, and high-strength steels. Solid glass is also being investigated.

The most difficult problem in utilizing these new materials is that very few of them can be welded and still retain their strength characteristics. In addition, the thick sections required for larger diameter hulls make welding extremely difficult and expensive even where it is possible.

Techniques have been proposed for eliminating or reducing the welding problem where large rings are used to form the strength elements of the cylindrical pressure hull. Reynolds proposed a hull of aluminum rings held together with tension rods and covered with a thin coating of pure aluminum for corrosion protection.[1] The latest design of the ALUMINAUT pressure hull cylinder is composed of rings held together by bolts.[2] The David Taylor Model Basin is currently investigating composite-type construction in which the strength rings are secured and protected by a thin outer jacket of weldable or otherwise easily fabricated material.[3]

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[1]References are listed on page 10.

For this study three structural models, two of aluminum and one of titanium were assembled from machined rings and were tested to determine whether hulls built from separate rings could have collapse strengths comparable to monolithic hulls. This type of hull, consisting of separate rings, would be adaptable to a variety of techniques for attaining longitudinal strength and watertight integrity. This report summarizes the results of the tests of these three models.

DESCRIPTION OF MODELS

The three models were designated PJ-1S, PJ-1L, and PJ-2. Models PJ-1S and PJ-1L were made from 7079-T6 aluminum alloy and Model PJ-2 from 6Al-4Va titanium alloy. Yield strengths of 62,000 and 150,000 psi were used in the design calculations for aluminum and titanium, respectively. Based on these strengths, the models were designed for a collapse strength of 10,000 psi, A value of 10.5×10^6 psi was assumed for the Young's modulus of the aluminum, and a value of 16.0×10^6 psi for the titanium. A Poisson's ratio of 0.3 was assumed for all material. The models are shown in Figure 1, and relevant dimensions are given in Figure 2.

The aluminum models (PJ-1S and PJ-1L) had rectangular inside frames and heavy shell segments of uniform thickness. The typical bay weighed 67.9 percent of its displacement weight in sea water. The area per unit length of the hull section was obtained by allowing the average two-dimensional Hencky-Von Mises stress to reach about 62,000 psi at a pressure of 10,000 psi. A highly stable shell structure is required to permit utilization of the full-yield strength of the material. To this end, the frames were designed for a general-instability pressure[4] of about 30,000 psi. A short frame spacing was selected to minimize bending in the shell due to hydrostatic loads and to provide a high elastic-shell buckling strength. Model PJ-1S was 1.7 diameters long, representing a finite compartment length, and was closed at the ends with flat aluminum plates; Model PJ-1L was four diameters long, approximating a semi-infinite cylinder, and had hemispherical end closures. Models PJ-1S and PJ-1L were designed to evaluate the effect of bulkhead spacing. Both models had identical typical bays and had grooves at each frame. These grooves were filled with an aluminum-impregnated epoxy, which sealed the joint and provided limited longitudinal strength.

The titanium model (Model PJ-2) had a double shell separated by thin webs. The typical bay weighed 53.8 percent of its displacement weight in sea water. In this model, area per unit length of the hull section was obtained by allowing the average circumferential stress over the section to reach 150,000 psi at a pressure of 10,000 psi. The depth of the web and the average thickness of the shells were chosen to obtain an elastic general-instability pressure for a semi-infinite cylinder of about 25,000 psi. The shell segments were designed to eliminate bending due to the hydrostatic load.[5] This was done by varying the thickness of the shell in the longitudinal direction. Also, as in the first two models, the end bays of Model PJ-2 were shorter than the typical bay.

Model PJ-2 was 1.6 diameters long and was closed at the ends with flat steel plates. Grooves on the inside and outside at each web were filled with an artificial rubber compound. The artificial rubber was chosen for ease of application in the laboratory, rather than as a suggested prototype material. It was also used to seal the closure plates to the model.

TEST PROCEDURES AND RESULTS

The models were instrumented with foil-type resistance strain gages. The arrangement of the gages is shown in Figure 3. Model PJ-1S was tested to collapse in the 17-inch diameter, high-pressure test tank at the Model Basin. Model PJ-1L was tested almost to collapse in the 17-inch pressure tank and collapsed in the 9-inch-diameter, high-pressure test tank. Model PJ-2 was tested to 7000 psi in the 13-inch diameter, high-pressure test tank and collapsed in the 9-inch pressure tank. Strain data were obtained only during tests in the 17 and 13-inch tanks. At least three runs were made for each model to obtain strain data. The loads were applied to Model PJ-2 at the same rate as the compression specimens.

Model PJ-1S collapsed at 12,500 psi, Model PJ-1L at 11,900 psi, and Model PJ-2 at 10,100 psi. Strain sensitivities devised from the initial slopes of the pressure-strain plots are given in Figure 3. Figure 4 shows Models PJ-1L and PJ-2 after test.

DISCUSSION

The measured strains, presented in Figure 3, are compared with theoretical strains in Table 1. The agreement with theory is very good, indicating that the elastic strains were not affected by the structural discontinuities at the frames. The calculations are from the theory of Salerno and Pulos[6] as presented by Lunchick and Short.[7]

All of the models failed by inelastic general instability in the n=2 mode. This is indicated by the appearance of the models after test (Figure 4) as well as by the theoretical calculations. The computed collapse pressures for Models PJ-1S and PJ-1L indicate a very high degree of stability in the elastic shell buckling modes and a margin of at least 20 percent in the inelastic shell buckling modes; see Table 2.

Because of the highly stable shell design, the theoretical inelastic shell buckling pressures[8,9,10] correspond to strain levels in excess of those measured in tests of material compression specimens. The ratios reported in Table 2 correspond to average strain levels of 1.5 percent. No theory is available to compute the shell-buckle pressures for nonuniform shells such as those of Model PJ-2. All of the theories presented in Table 2 assume the models to be of monolithic construction.

The experimental collapse pressures and the scaled collapse strengths are given in Table 3. To compare the ring models with monolithic hulls, data from two other models are also included in Table 3. Model DSRV-P,[3] a small machined-aluminum model, similar to Model PJ-1S was 1.4 diameters long and had a modified-Bryant critical buckling pressure of 3.55 times its collapse pressure. Since it is impossible to machine a monolithic sandwich hull such as Model PJ-2, a similar two-piece hull is included in this discussion for comparison. Model OV-4 was made by inserting a cylinder with outside rectangular frames into a closely fitted jacket, which formed the outer shell. Model OV-4[11] had nearly the same semi-infinite, elastic, general-instability collapse pressure as Model PJ-2 and was 4 diameters long.

Generally, the collapse strength of models that fail inelastically is proportional to the yield strength of the material and to the weight-to-displacement ratios. This permits comparison of collapse strengths among similar models. The scaled collapse pressures for Models PJ-1S and PJ-1L were obtained by scaling the model yield strengths to 62,000 psi. The scaled collapse pressure for Model PJ-2 represents a semi-infinite hull of the same typical bay geometry and with a yield strength of 150,000 psi. The collapse strength of Model DSRV-P was scaled to 62,000 psi yield strength and a weight-to-displacement ratio of 67.9 percent. The collapse strength of Model OV-4 was scaled to a yield strength of 150,000 psi and a weight-to-displacement ratio of 53.8 percent.

The comparisons shown in Table 3 illustrate that ring construction need not result in any sacrifice in collapse strength compared to monolithic hulls. Since the distortion and weakening effect of welding stresses are not present, ring construction may permit some increase in collapse strength relative to welded hulls.

A submarine pressure hull is designed principally to resist external hydrostatic pressure. However, in addition to the hydrostatic loads, a submarine or other structure is subjected to overall bending moments. To resist these bending moments, the structure must possess longitudinal tensile strength in addition to its hydrostatic collapse strength. This tensile strength is required only when operating on or near the surface. At deeper depths, the longitudinal, hydrostatic compressive load exceeds the tensile load of the bending moments and less longitudinal tensile strength is required. For example, an oceanographic research station and bottom-based vehicles could be assembled on site with a minimum of tensile bending.

For vehicles which must operate on the surface or at high speed, some form of mechanical joining is required. The fundamental considerations in designing a joining device are weight, volume, and stress concentration in the pressure hull. For a given level of longitudinal strength, it is desirable that the least excess weight be added to the structure by the joints. In addition, it is desirable to consume as little as possible of the valuable interior space. Any joining procedure which induces stress concentration in the pressure hull may lower the collapse strength or induce a fatigue problem. Many mechanical joining techniques have been proposed for this application. A few are considered here. The following discussion presents several typical ideas and some of their strengths and weaknesses.

The rings could be held in place by tension wires or rods secured at the bulkheads (Figure 5a). This method has the advantage of low weight since very high-tensile-strength steel may be used. One problem involves the location of the rods. They cannot pierce the frames or webs since, this would introduce severe stress concentrations particularly for nonductile materials. If they are inboard of the frames, they are less efficient structurally and consume valuable interior space. Ideally, they would be placed outside the hull, but there they are subject to mechanical damage and to corrosion.

Bolts might be used to secure the rings (Figure 5b). This technique is being considered for ALUMINAUT. An important problem here is the stress concentration in the bolt holes. Large, round taper pins might overcome the stress-concentration problem of the holes (Figure 5c), but they would be loaded in shear and would have a stress-concentration problem under longitudinal loading.

Clamps and similar devices carry no hydrostatic load and may be prohibitively heavy for joining individual shell and frame rings. However, sections consisting of several frame and shell sections could be joined by a system as shown in Figure 5d. The sections could be assembled by one of the other techniques considered here or, for small diameter hulls, could be machined. This procedure is being considered for deep-running torpedoes. The clamping ring may be formed in several segments which are joined by bolts.

In the technique of composite construction, the rings are inserted into a jacket of weldable or otherwise easily fabricated material (Figure 5e). The jacket yields during the initial submergence, holds the rings in place, and provides watertight integrity and corrosion protection. The main difficulty with this technique is in the repair of the jacket. Some materials, particularly aluminum and glass-reinforced plastics, are heat sensitive, and heat applied in welding the jacket might seriously weaken the inner rings. This problem could be overcome by using an epoxy or glass-reinforced plastic jacket or, possibly, an intermediate heat shield.

The rings could also be designed to be self-locking. This could be done by allowing for positive interference and shrink-fitting the rings together (Figure 5f). Aside from the difficulties in fabrication, this technique precludes later separation of the rings and may be a source of stress concentration if close tolerances are not maintained. The rings could also be made with threads and screwed together (Figure 5g); here the main problem is the stress concentration at the root of the threads.

One of the most interesting joining and sealing techniques available is the use of adhesives. Adhesives have the significant advantages of low weight and economy of application. They can be used either alone or in combination with other mechanical bonds.

The availability of epoxies or similar materials with the necessary bonding properties and tensile strengths has not been established. It is felt that the rapidly expanding adhesive technology will be able to provide a suitable material if the need is established. In addition, adhesives have two critical defects which impair their usefulness in deep-submergence applications; one is a problem in joint design and the other a problem in materials.

An epoxy adhesive was used on the deep-submergence Krupp sphere of the bathyscaphe TRIESTE to hold the three segments of the sphere together. The joint was a simple glue-line with a layer of epoxy between the two metal surfaces. When the Krupp sphere surfaced after the first deep dive (Dive 61), the two joints were parted. Fortunately, there was no immediate danger to the occupants of the bathyscaphe since the segments were held together by the pressure of the water; for further dives, the sphere was held together by a system of rings and bands.[12]

The failure of the TRIESTE bond was probably caused by the deterioration of the epoxy under high compressive loads. Epoxy resins and other adhesives have relatively low Young's moduli and compressive yield strengths. This means, of course, that they may experience excessive deformation when subjected to the same stress as a metal. Conversely, if the adhesive is subjected to the same deformation as a metal, it will carry a very much smaller stress. The models in this report illustrate one type of joint where the adhesive may act as a bond and seal, without being subjected to high compressive stresses. Much work is required to evaluate this, and other, improved designs for adhesive joints. It is important that the presence of the groove for the adhesive joint did not seem to impair the collapse strength of the models.

In addition to deterioration due to loading, adhesive materials are subject to eventual deterioration due to exposure to salt water and to biological fouling. These material problems must be solved by developing new adhesives and material-protection techniques. Until they are solved, adhesives will be limited to short-term applications or will require periodic replacement.

The primary advantage of ring construction is that it permits the use of many light, high-strength nonweldable materials for deep-submergence pressure hulls; but there are a number of other features which may produce considerable savings. The combining of rings into a pressure hull by mechanical means is essentially a faster and less expensive operation than welding and requires less time spent in the shipway. This potential saving is balanced by the greater machining costs associated with ring construction. In addition, it is often necessary to open a hull to make repairs or to replace machinery. It is not possible to open a hull by cutting or burning a ring since it cannot ordinarily be rewelded; but it may be practical to separate two rings at a joint, and then reassemble the section after the repair work is completed.

Present forging capacity limits the maximum size of nonweldable metals to a diameter of about 12 to 15 feet. Aluminum must be forged to obtain its highest yield strength, but titanium and steel may be welded and then heat treated to higher strengths. It is felt that the primary applications of ring construction will be in structures of smaller diameter. The possibilities of ring construction in onsite assembly of undersea laboratories and in oceanographic research vehicles have already been mentioned. In addition, the technique can find military applications for deep-running-torpedo housings or for submerged missile-silos.

SUMMARY

1. A submarine pressure hull built of separate rings may be so designed as to have a hydrostatic collapse strength comparable to that of a monolithic hull.

2. The technique of ring construction permits an increase in static collapse strength compared to current steel hulls of the same weight, through the use of light, high-strength, nonweldable materials. This technique may also result in savings in fabrication and assembly costs, which would make it practical for use with weldable materials.

3. A wide variety of mechanical techniques is available to provide the necessary longitudinal strength, watertight integrity, and corrosion protection for hulls of ring construction. The expanding technology of epoxy plastics may provide materials suitable for these applications.

RECOMMENDATIONS

Future research should include studies of the effects of hull bending moments and longitudinal tensile loads on various types of mechanical joints.

ACKNOWLEDGMENTS

The author wishes to acknowledge the assistance of Mr. Martin A. Krenzke who initiated the project and designed Models PJ-1S and PJ-1L.

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Figure 1 - Models before Test

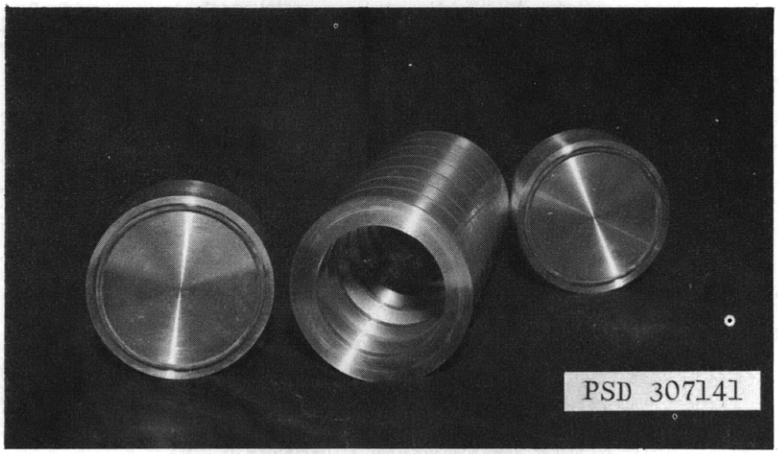


Figure 1a - Model PJ-1S

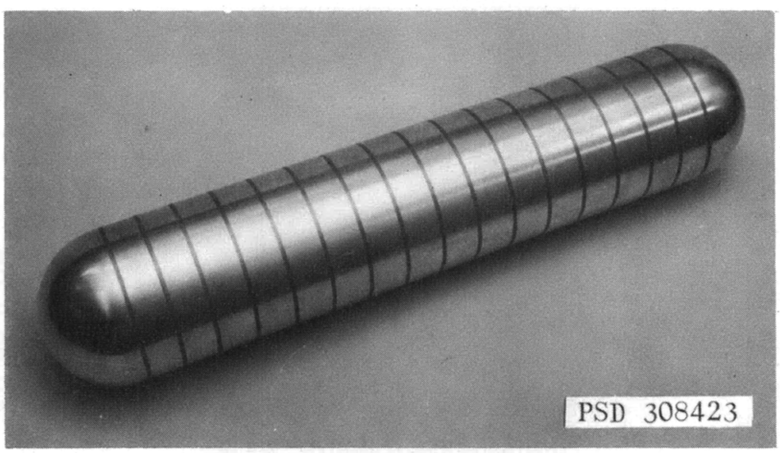


Figure 1b - Model PJ-1L

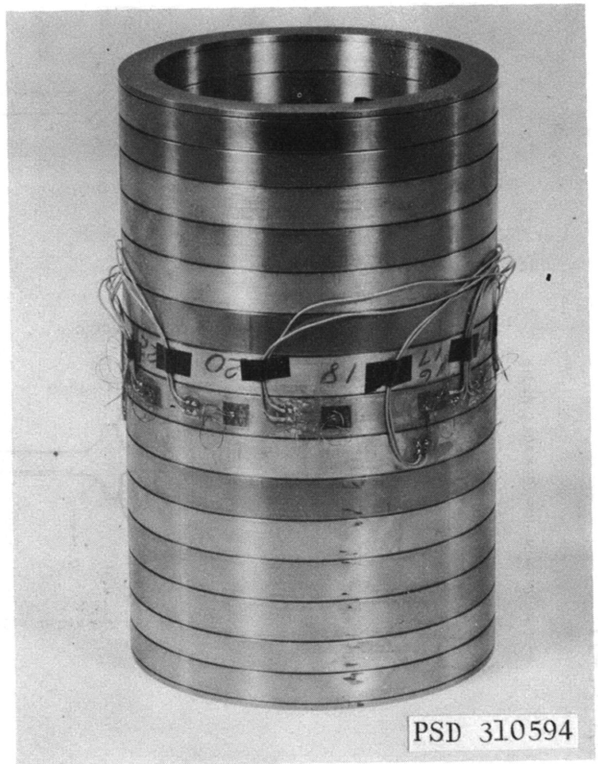


Figure 1c - Model PJ-2

Figure 1d - Model PJ-2 (Showing Rings)

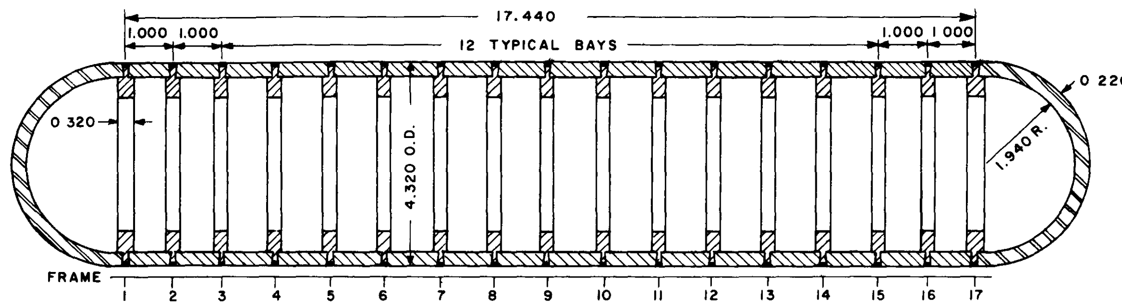


Figure 2a - Model PJ-1L

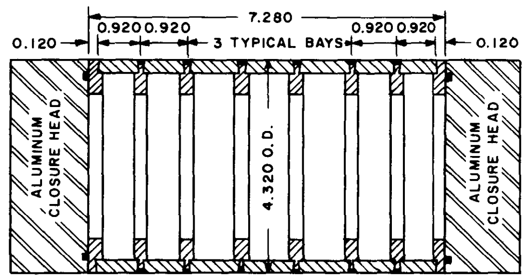


Figure 2b - Model PJ-1S

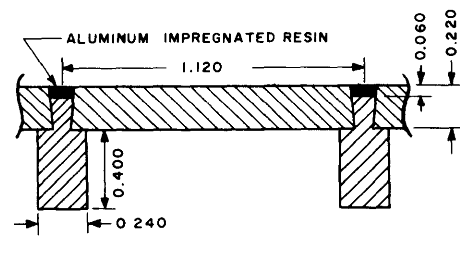


Figure 2c - Typical Bay

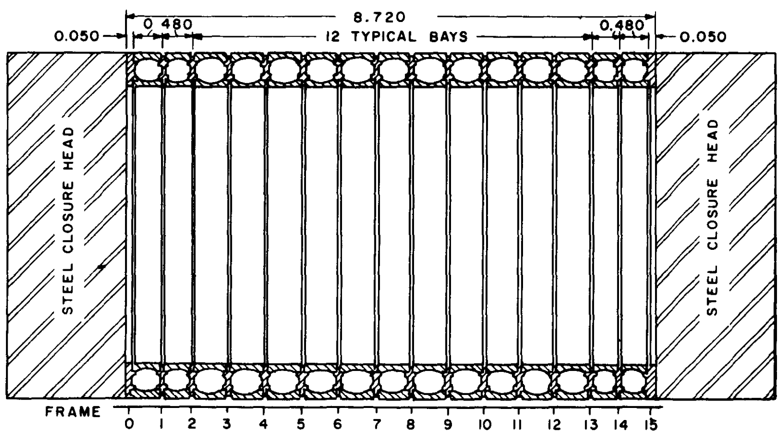


Figure 2d - Model PJ-2

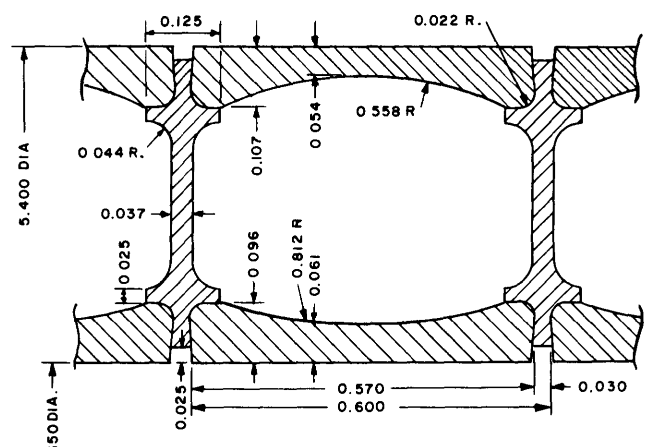
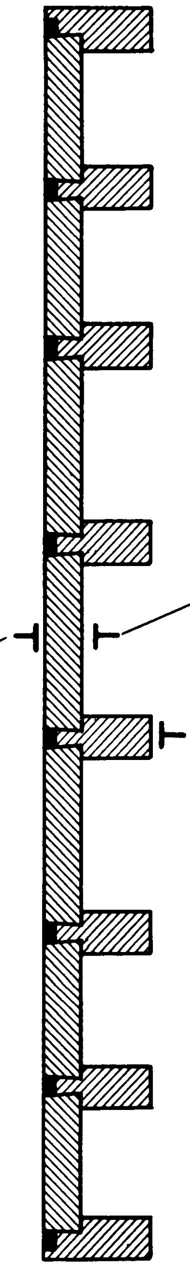


Figure 2e - Typical Bay PJ-2

Figure 2 - Sketches of Models

Figure 3 - Measured Strain Sensitivities

Outside		
θ	ϵ_ϕ	ϵ_χ
0	-0.49	-0.32
90	-0.49	-0.33
180	-0.49	-0.34
270	-0.50	-0.34



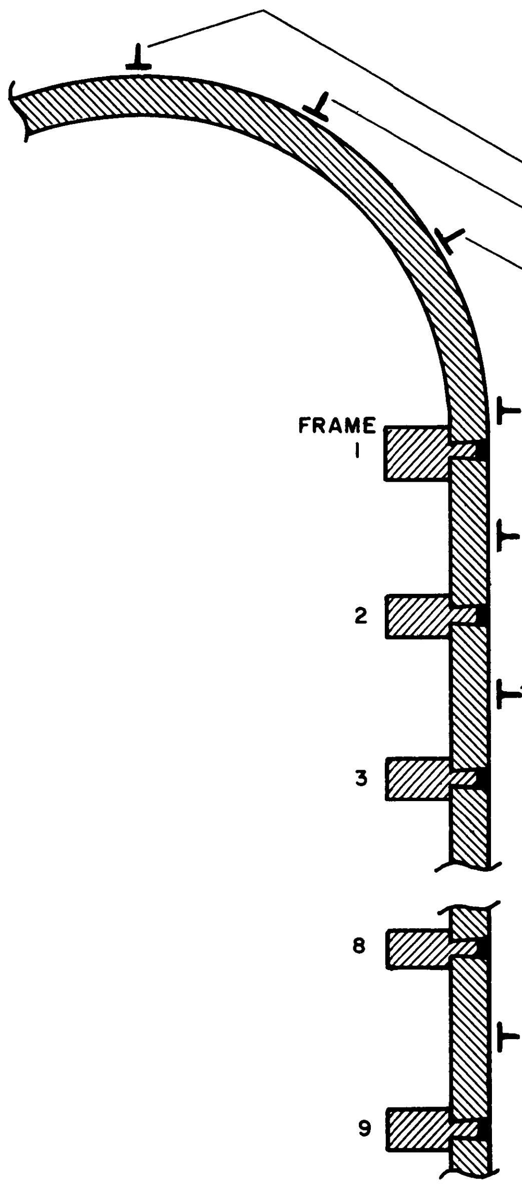
Inside		
θ	ϵ_ϕ	ϵ_χ
0	-0.57	-0.15
90	--	--
180	-0.58	-0.16
270	-0.58	--
0	-0.71	
30	-0.70	
60	-0.68	
90	-0.72	
120	-0.71	
150	-0.72	
180	-0.71	

Figure 3a - Model PJ-1S

θ **Circumferential location of gages**

ϵ_ϕ **Circumferential strain sensitivity in μin/in per psi**

ϵ_χ **Longitudinal strain sensitivity in μin/in per psi**



0 Degrees		90 Degrees	
ϵ_ϕ	ϵ_χ	ϵ_ϕ	ϵ_χ
	-0.28		-0.28
-0.29	-0.29	-0.29	-0.28
-0.28	-0.28	-0.29	--
-0.25	-0.12	-0.26	--
-0.45	-0.47	-0.47	-0.37
-0.50	-0.37	-0.49	-0.38

θ	ϵ_ϕ
0	-0.51
30	-0.52
60	-0.50
90	-0.50
120	-0.49
150	-0.51
180	--

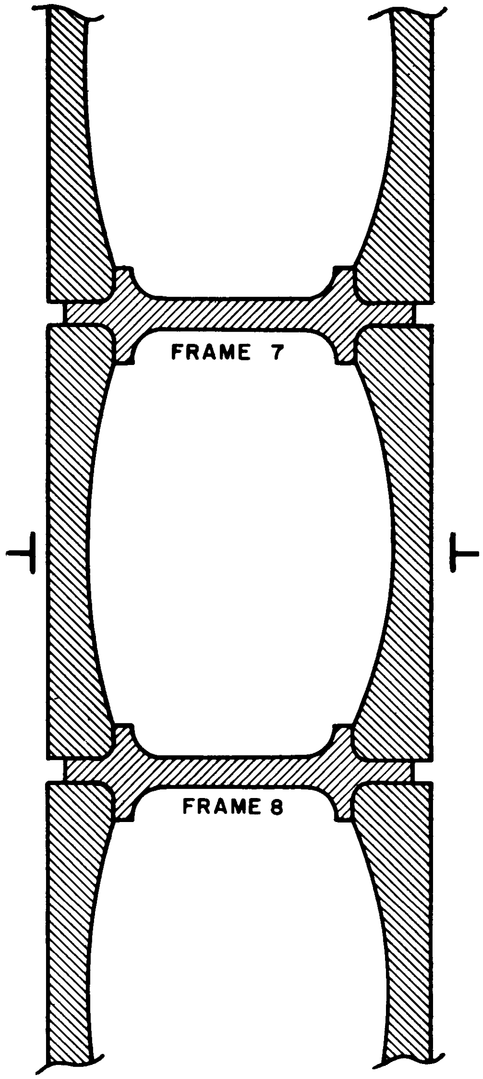
Figure 3b - Model PJ-1L

θ **Circumferential location of gages**

ϵ_ϕ **Circumferential strain sensitivity in μin/in per psi**

ϵ_χ **Longitudinal strain sensitivity in μin/in per psi**

Inside		
θ	ϵ_ϕ	ϵ_χ
0	-0.62	-0.76
30	-0.63	
60	-0.63	
90	-0.60	
120	-0.60	-0.58
150	-0.63	
180	--	
210	-0.56	
240	--	-0.56
270	-0.62	
300	-0.57	
330	-0.64	



Outside		
θ	ϵ_ϕ	ϵ_χ
0	-0.68	-0.62
30	-0.66	
60	-0.60	
90	-0.62	
120	-0.62	-0.54
150	-0.60	
180	-0.65	
210	-0.65	
240	-0.70	-0.64
270	-0.61	
300	-0.61	
330	-0.65	

Figure 3c - Model PJ-2

θ **Circumferential location of gages**

ϵ_ϕ **Circumferential strain sensitivity in μin/in per psi**

ϵ_χ **Longitudinal strain sensitivity in μin/in per psi**

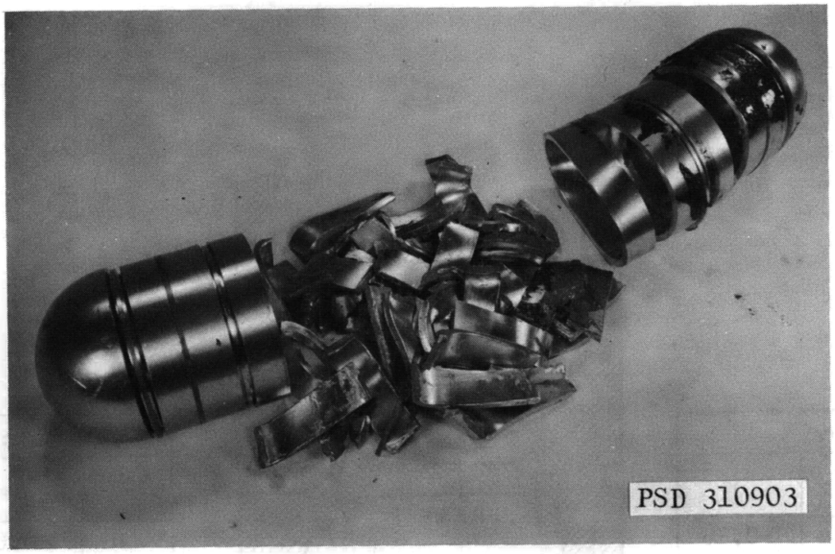


Figure 4a - Model PJ-1L

Figure 4b - Model PJ-2

Figure 4 - Models After Test

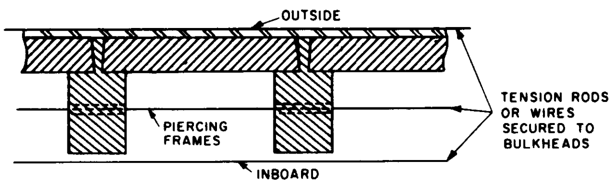


Figure 5a - Tension Wires or Rods

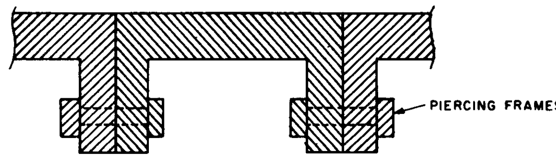


Figure 5b - Bolts

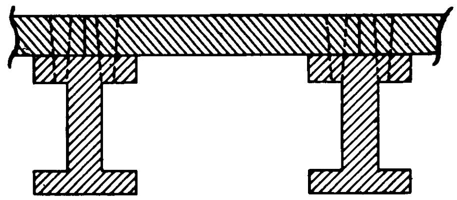


Figure 5c - Taper Pins

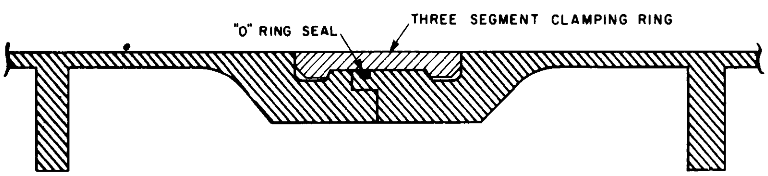


Figure 5d - Clamping Ring Joining Sections of Several Frames

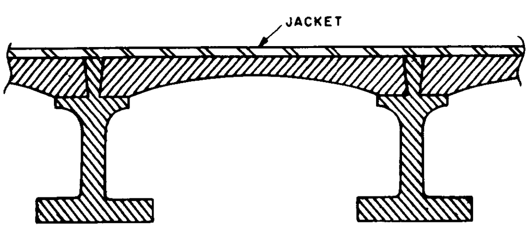


Figure 5e - Composite Construction

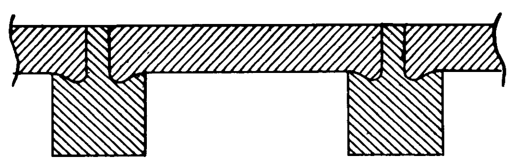


Figure 5f - Self-Locking

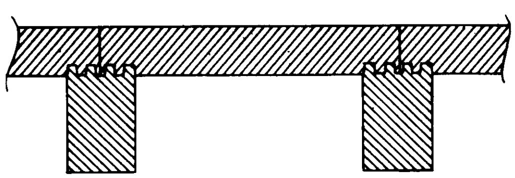


Figure 5g - Threads

Figure 5 - Mechanical Joining Techniques

TABLE 1

Comparison of Experimental
and Theoretical Strain Sensitivities* at Midbay

Location	Theoretical**	Average PJ-1S	Average PJ-1L
Circumferential			
Inside	0.56	0.58	
Outside	0.50	0.49	0.50
Longitudinal			
Inside	0.15	0.16	
Outside	0.34	0.34	0.38†

*Strain sensitivities in μin/in/psi.

**Theory of Salerno and Pulos.

†Bay 2 1/2 only.

TABLE 2

Ratios of Theoretical Collapse Pressures to Experimental Collapse Pressures

Theory of Collapse	Ratio of Theoretical to Experimental Collapse Pressure		
	Model PJ-1S	Model PJ-1L	Model PJ-2
Modified Bryant[4] (n = 2)	3.13	2.54	3.37
Inelastic general instability[3] (n = 2)	0.93	0.98	1.02
Reynolds' asymmetric elastic[8] shell buckling	11.7	12.3	--
Lunchick's axisymmetric[9] elastic shell buckling	13.5	14.1	--
Reynolds' asymmetric[8, 10] inelastic shell buckling	>1.2	>1.2	--
Lunchick's axisymmetric[9, 10] inelastic shell buckling	>1.2	>1.2	

TABLE 3

Experimental and Prototype Collapse Pressures

Parameter		Model PJ-1S	Model PJ-1L	Model DSRV-P[3]	Model PJ-2	Model OV-4[11]
Experimental collapse pressure in psi		12,500	11,900	12,400	10,100	11,100
Yield strength (0.2 percent offset) in psi	Actual model	75,900	77,000	81,000	144,500	138,000
	Scaled	62,000	62,000	62,000	150,000	150,000
Weight-to-displacement ratio in percent	Actual model	67.9	67.9	66.0	53.8	63.1
	Scaled	67.9	67.9	67.9	53.8	53.8
Scaled collapse pressure in psi		10,200	9,600	10,300	10,200*	9,800

*Scaled pressure for Model PJ-2 computed for a corresponding semi-infinite cylinder.

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