R666

Technical Report

PRESSURE VESSEL CONCEPTS

Exploratory Evaluation of Stacked-Ring and Segmented-Wall Designs With Tie-Rod End-Closure Restraints

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by

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ABSTRACT

An exploratory experimental study was conducted to evaluate the stacked-ring and segmented-wall pressure vessel concepts. The evaluation consisted of (1) testing to destruction stacked-ring and segmented-wall pressure vessel models with tie-rod end-closure restraints and (2) evaluating a series of seal designs utilized in the sealing of the joints between the pressure vessel end closures and the cylindrical pressure vessel body. The test results indicate that the stacked-ring pressure vessel design is approximately 50% heavier than a multilayered pressure vessel of same internal diameter, length, material, and pressure capability. The segmented-wall pressure vessel design is approximately 8 to 9 times heavier than a multilayered pressure vessel of same diameter, length, material, and pressure capability. The free-floating, self-energizing radial seal system provided the most reliable and extrusion-proof sealing for vessels with considerable radial dilation and axial end-closure movement.

CONTENTS

INTRODUCTION

Statement of the Problem

The Navy deep-submergence effort requires well-equipped deep-ocean simulation test facilities with pressure vessels of sufficient size to accommodate at least the largest single component of any deep-submergence vehicle or habitat. This represents only the minimum requirement, while a more desirable requirement would be to be able to test any full-size deep-submergence vehicle or habitat to its design depth in a pressure vessel, so that the whole system receives a thorough proof test.

The current Navy's research and development program requires pressure vessels with an operational capability of 13,500 psi and at least 120 inches in inside diameter and 360 inches in internal length. Such a pressure vessel could test to collapse the structure of an average-sized construction vehicle, or scale model of a habitat to the limit of its 1.5 safety factor for a 20,000-foot depth. But the size of the above-mentioned pressure vessel is not the ultimate in projected pressure vessel requirements. Larger pressure vessels will be required as the size of the deep-submergence manned vehicle and habitat hulls increases. In addition, more emphasis will be placed on proof-testing complete deep-submergence vehicle and habitat systems in controlled laboratory environment rather than in the ocean environment, where even the slightest malfunctioning of a system component spells irretrievable loss of the vehicle or habitat and of its crew.

To meet these future pressure vessel requirements, the exploratory hydrostatic pressure vessel study was conducted by NCEL under NAVFAC sponsorship. Its results are presented in this report.

Background Information

As indicated in Appendix A, traditional construction techniques are hard put to satisfy the operational requirements of the new generation of pressure vessels that are not only much larger in diameter, but also operate at higher pressures than earlier pressure vessels. For many years prior to the invention and successful use of the multilayer construction technique the single-wall welded or forged monolithic construction of pressure vessels was the only technique available for their fabrication. When single-wall thicknesses of more than 2 inches were required for vessels fabricated from welded plate, a point of diminishing returns was reached, as the tensile strength properties of rolled alloy steel plate began to decrease with further increase in plate thickness.

The introduction of the multilayer pressure vessel construction technique overcame this wall thickness limitation. This technique permitted thick pressure vessel walls to be built up from thin sheets or plates, thus obtaining thick walls with material properties equal to those found in thin sheets or plates. Because of this, the multilayer construction technique has been widely accepted and remains today the most reliable and proven technique for fabricating large-diameter, high-pressure vessels.

There are, however, two shortcomings inherent in the multilayer construction technique that become more and more pronounced as the sizes and operational pressures of the pressure vessels increase. The *first* shortcoming is the reliance on longitudinal and circumferential welds for joining the many layers in the wall. Since reliable welding methods as a rule lag behind the development of new steel alloys, reliance on welding forces the multilayer fabricators to use generally only lower strength alloys for which reliable welding techniques have been already developed. At a first glance this does not seem to be much of a disadvantage, as instead of the thin vessel walls of new, higher strength alloys, the old, lower strength alloys could be utilized in thicker vessel walls. Such a substitution would be quite acceptable if the distribution of stresses in the vessel wall remained the same regardless of wall thickness. Unfortunately, this is not the case; the distribution of stresses becomes less and less uniform as the wall-thickness-to-vessel-diameter ratio increases (Figure 1), making thick-wall vessels uneconomical in terms of their internal pressure capability (Figure 2).

The *second* shortcoming of the layered pressure vessel construction is its monolithic mass that makes it impossible to transport such a vessel by land if its dimensions are large and its pressure capability is high. There is a limited solution to this problem whereby the individual vessel layers are welded on site, and the finished assembly is never moved again from its foundations. This solution is acceptable, but the welding and stress relieving is done under conditions less than ideal and future removal of the vessel for repair or maintenance is extremely difficult and expensive.

Although the two above-mentioned shortcomings of layered vessel construction are not serious enough to preclude its use for pressure vessels of any size or pressure capability, they are grave enough to warrant investigation of other types of vessel construction. In the previous survey of pressure vessel construction conducted at NCEL (Appendix A), all the available and proposed methods of vessel construction were reviewed. Only two were found to merit

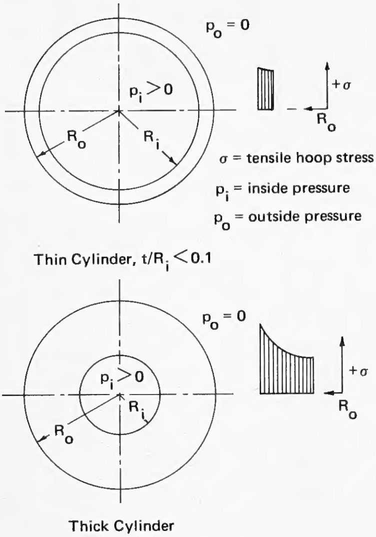


Figure 1. Distribution of stresses in thick-walled and thin-walled cylinders under internal hydrostatic pressure.

further study for application to steel vessels of more than 10-foot diameter and with operational pressure in excess of 10,000 psi. The two pressure vessel construction techniques which are considered to be at least on par with multi-layered construction so far as their applicability to large high-pressure steel vessels is concerned are the stacked-ring and segmented-wall module construction techniques. This report deals with the exploratory evaluation of these concepts from economical, engineering, design, construction, and operational viewpoints.

Objective

The objective of the study was to experimentally investigate the stacked-ring and segmented-wall modular concepts for internal pressure vessels. In addition, seal systems required for such pressure vessel designs were to be explored and evaluated.

This study is an exploratory evaluation of pressure-containing capability of stacked-ring and segmented-wall designs for pressure vessels of equal interior dimensions. The experimental evaluation of the two pressure vessel concept designs, together with the discussion of economical and operational considerations, will be useful in determining the desirability of these concepts when selection of a design for large pressure vessels required in future hydrospace simulation facilities is made. The experimental evaluation of the many available seal designs for large pressure vessels provides a brief overview of available seal systems for high-pressure vessels and their sealing capability.

Scope of Investigation

The study was limited both in *scope* and *depth*. In *scope* the study was limited to only two types of pressure vessel design concepts—the stacked-ring and the segmented-wall modular concepts with tie-rod end-closure restraints. This scope was set by a preceding study (Appendix A) which briefly reviewed

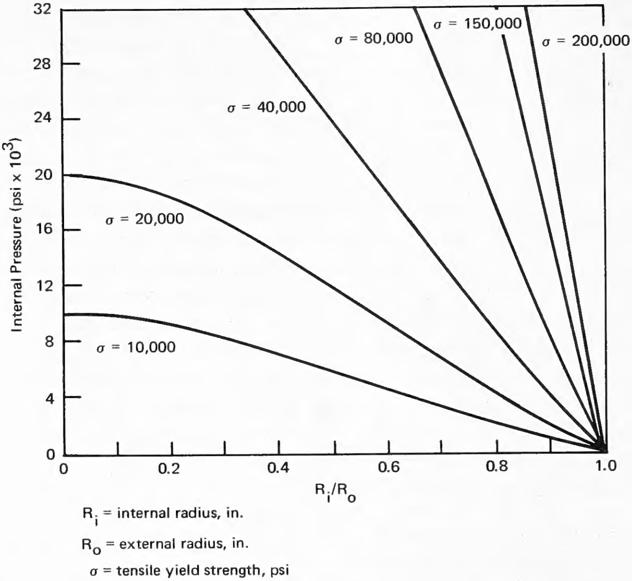


Figure 2. Internal pressure capability of a cylindrical pressure vessel as a function of its thickness-to-diameter ratio.

the existing design concepts for large pressure vessels and selected the stacked-ring and the segmented-wall module concepts with tie-rod end-closure restraints as the most promising candidates for future large-diameter, high-pressure vessels and recommended their further study, preferably by experimental means. In *depth* the study was limited only to a single conceptual design of each concept under consideration. The two design concept models tested were to be made only from one material, acrylic plastic.

Besides the experimental evaluation of the two modular vessel design concepts, one of four seal systems was to be experimentally evaluated for use with large-diameter, high-pressure vessels of modular construction (Appendix B).

DISCUSSION OF CONCEPTS

Stacked-Ring Pressure Vessel

The stacked-ring pressure vessel is a very simple concept[1] which relies for its strength on two separate sets of structural members—one set for carrying the axial stresses, the other for carrying the circumferential stresses.

Radial Restraint. The set of structural members for giving the vessel strength to resist radial forces generated by hydrostatic pressure consists of a series of rings stacked upon each other and a liner inserted inside these rings for sealing the joints between individual rings. Since the rings are only required to carry circumferential stresses, no welding or mechanical bolting is required between individual rings to hold them together. The minimum dimensions of a ring for a given vessel diameter are determined by two parameters: (1) the hoop and radial stresses inside the ring and (2) the twisting moment imposed on the ring by the radial hydrostatic pressure. The *maximum dimensions* of a ring are on the other hand determined by the forging capability of U. S. industry and the weight handling capability of the crane at the pressure vessel assembly place.

Axial Restraint. The set of structural members for giving the vessel strength to resist axial forces generated by hydrostatic pressure consists of the two end closures and the end-closure-retaining tie rods, or a yoke. The end closures and their tie rods (or a yoke) constitute a separate structural assembly in no way interconnected with the stacked rings that resist the radial forces on the vessel. The end closures are of the free-floating type, that is, they displace relative to the stacked-ring assembly when internal hydrostatic pressure is applied. The tie rods (or a yoke) holding the end closures together are of the nonprestressed design, so that upon locking in place there is no axial tensile stress in them prior to pressurization of the pressure vessel's interior. Upon pressurization, the stress in the tie rods (or yoke) is proportional to the hydrostatic pressure inside the vessel, and the resulting elongation of the tie rods (or yoke) permits the end closures to float freely inside the pressure vessel liner enclosed by stacked rings.

Although both the tie rods and the yoke provide axial restraint on the end closures, there is a considerable difference in their effect on the design of end closures because of the manner in which the restraint is imposed upon the end closures under an axial thrust generated by the hydrostatic pressure inside the vessel. The *yoke* type of restraint girds the vessel along its longitudinal axis, thus retaining both pressure vessel end closures at the same time. Since the yoke passes directly over the vessel end closures, and since during the

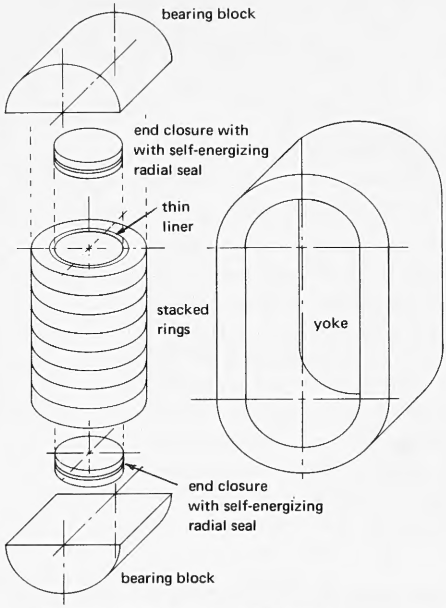


Figure 3. Engineering concept of a stacked-ring cylindrical pressure vessel with continuous-yoke end-closure restraint.

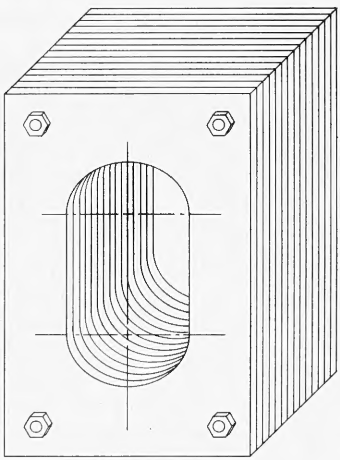


Figure 4. Typical laminated yoke.

hydrostatic pressurization of the vessel the end closures bear directly against the yoke, the end-closure assemblies can be designed to utilize this bearing stress to their advantage.

A typical design that utilizes the bearing stress of the end closure against the yoke is shown in Figure 3. Here the yoke acts upon a bearing block that distributes the bearing stresses evenly over the area of the flat end-closure disc. Because of the even bearing pressure, equal in magnitude to the internal hydrostatic pressure, the end closures can be thin, as they are not required to withstand any bending moments or shear loads. Its sole function is simply to act as a free-floating seal piston, within the cylindrical vessel, while the bearing block functions only as a load distributor and spacer.

Since the yoke can be, and generally is made quite massive to lower the tensile stresses in it, low-carbon hot-rolled steel suffices for this application. To lower the cost of fabrication, such a yoke is generally assembled from many thin plates (Figure 4) in which the proper opening has been cut, or it is built up by winding steel bands (Figure 5) around a yoke frame. In either case, the nominal tensile stresses are very low, and the high ductility of the low-carbon steel tends to prevent stress concentrations from generating fractures. Thus from the engineering research viewpoint, the yokes are not worth an exploratory investigation as their design,

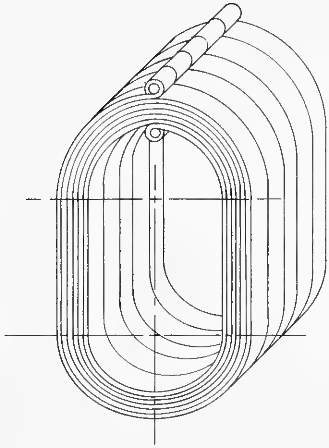


Figure 5. Typical steel band yoke.

fabrication, and operation are quite well understood and within the scope of routine engineering design.

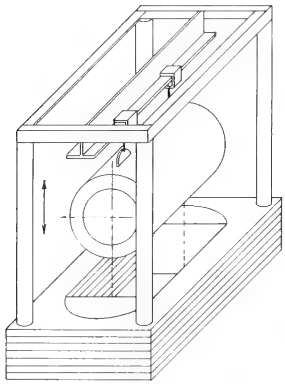
Although from the design and fabrication viewpoint the use of a yoke for retention of end closures is a desirable design feature, from the operational viewpoint it leaves a lot to be desired. Regardless of whether the vessel is placed horizontally or vertically, cumbersome and complicated mechanisms must be employed to gain access to the interior of the vessel for test specimen placement and removal. The opening and closing of the vessel is a time-consuming operation, primarily because of the weights involved—regardless of whether the yoke is stationary and the vessel movable, or vice versa (Figure 6).

The tie-rod system of restraining end closures is quite different from the yoke type of restraint system. The tie-rod restraint system is first of all not a continuous band that girds the vessel about the end closures (Figure 7). It relies rather on a series of tie rods to act upon retaining flanges, that in turn restrain the end closures. Because this restraint system is an assembly of several structural components, it can be taken apart piecemeal for access to the vessel's interior, rather than moving the whole restraint system assembly, or pressure vessel, as is the case with the yoke restraint system. This possible operational advantage, however, is coupled with serious structural disadvantages. These include severe stress concentrations at load transfer points from one restraint component to another, and the need for high-strength materials. The low-grade structural steel generally employed in the yoke-restraint system is inadequate to carry the axial loading distributed among a few tie rods whose number is limited by the circumference of the vessel. Since in the tie-rod restraint system the hydrostatic pressure on the end closures cannot be counteracted by the bearing stresses on the end closures provided by the yoke system girdle, the design must be quite different from yoke restraint system design. This difference not only extends to the shape of the end closure, which in this case cannot be flat, but rather must be hemispherical, but also to the magnitude of, and complexity of stresses in it. With yoke restraint, the design and calculation of stresses in the flat end closure and bearing pad are rather routine; in the case of tie-rod restraint the calculations are difficult because there are stress concentrations whose magnitude must be both analytically and experimentally determined during the design phase.

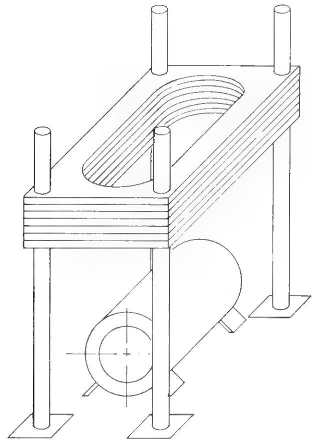
Because (1) very little was known about design and operation of vessels with tie-rod end restraints at the beginning of this exploratory study while the design of yoke restraints and associated end closures was quite well understood, and (2) the application of yoke restraint severely handicaps the access to vessels' interiors and slows down the use of such vessels for hydrostatic tests, it was decided to explore experimentally only the tie-rod restraint system. (The tie-rod restraint system promises to alleviate those difficulties.) It was felt that by exploring it experimentally (1) some design and stress distribution data would be generated where none was available before, and (2) some experience would be gained in the operation of the tie-rod restraint system that would permit rational comparison between the operational desirabilities of tie-rod- and yoke-restrained systems. After selection of the tie-rod restraint for investigation within the objective and scope of this exploratory study, no further discussion of the yoke-restraint subsystem will be made until the section on conclusions and recommendations.

Construction. The fact that the stacked-ring pressure vessel relies for its strength not on any welds, but on isotropic homogeneous forgings permits the use of high-strength steel alloys for which the welding techniques have not yet been developed, or are only in the development stage. Specifically speaking, it permits the construction of a pressure vessel from structural components forged from maraging steels with yield points of up to 250,000 psi.

Although the stacked-ring pressure vessel design and fabrication technique permits the assembly of rather large high-pressure-capacity pressure vessels from smaller structural components, there is a limit to how large a pressure vessel can be assembled in such a manner. This limit on the size of a stacked-ring pressure vessel is determined by the forging capability of the steel industry. The largest structural components in a stacked-ring pressure vessel are the retaining rings and the end closures; therefore the maximum size of these components that can be forged by the steel industry will determine the maximum diameter and pressure capability of a stacked-ring pressure vessel. To determine the largest retaining ring or end closure the industry can forge at any given date is almost impossible without a detailed survey of each forging press in the world. A limited inquiry has shown, however, that the steel industry can easily forge structural components of such size as to permit the assembly of pressure vessels with an operational pressure of about 13,500 psi, a 10-foot internal diameter, and a 30-to-40-foot length.

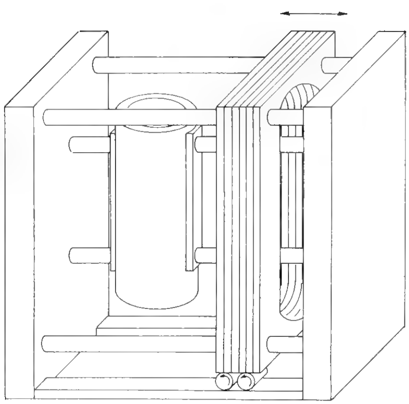


(a) Yoke is stationary.

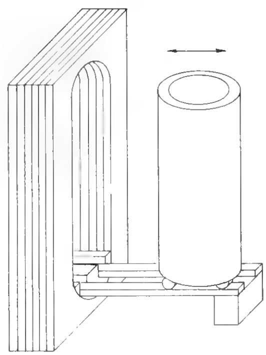


(b) Vessel is stationary.

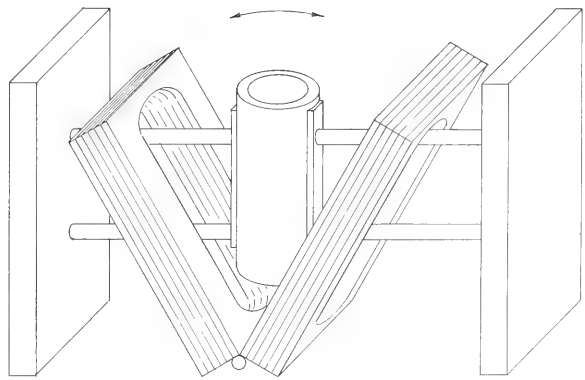
Vessel in Horizontal Position



(c) Vessel is stationary.



(d) Yoke is stationary.



(e) Vessel is stationary.

Vessel in Vertical Position

Figure 6. Mechanisms for operating vessels equipped with yokes.

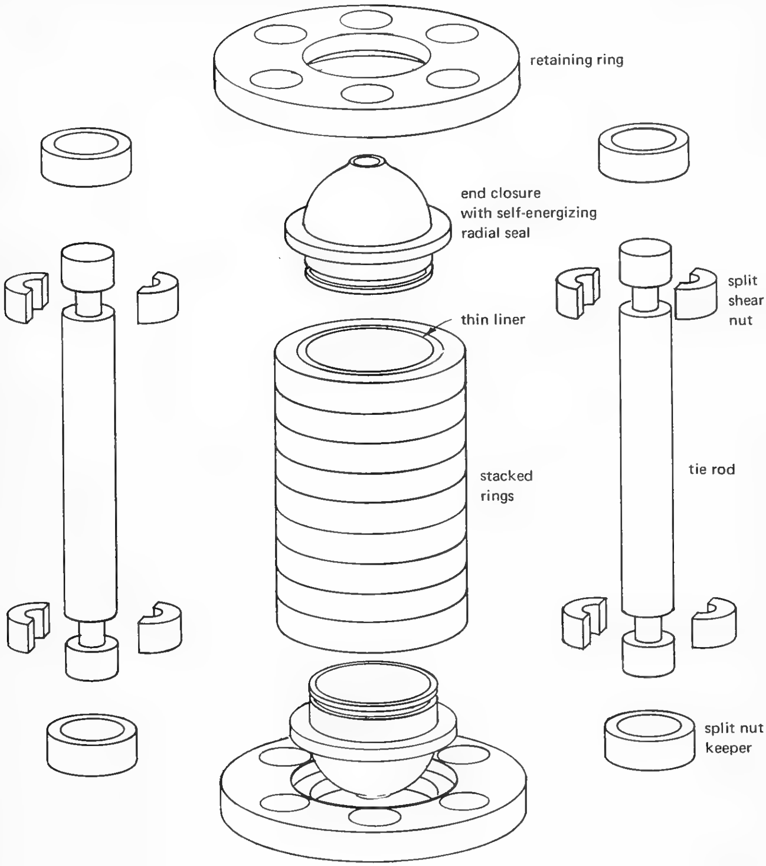


Figure 7. Typical tie-rod end-closure restraint system.

If industry can forge a flat retaining ring larger than the hemispherical end closure, the latter can be modularized so that the factor limiting pressure vessel size is the retaining ring forging, and not the end-closure forging.

One modular design breaks the monolithic end closure down into many spherical polygons permitting the assembly of the end closure from many small, easily forgeable structural modules. Since welding or bolting those end-closure modules would considerably reduce the capacity of such a modular end-closure

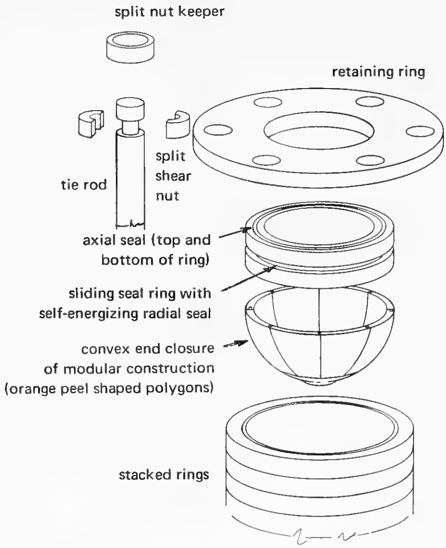


Figure 8. Typical segmented-dome end closure.

assembly to carry tensile stresses, a concept is required that would permit the hemispherical end closure to carry only compressive stresses. If the end-closure modular assemblies do not have to carry any tensile stresses, the capability of the closure to retain internal hydrostatic pressure is not diminished in any way by the presence of the joints between the individual spherical polygon modules, particularly since a thin liner makes the joints watertight. In such a case, the bolted joints between the individual modules serve only to hold them together for handling of the end closure by a hoist during the opening and closing of the pressure vessel. To achieve this, a concept has been proposed (Figure 8) which transforms tensile stresses of the end closure to compressive stresses by substituting a convex hemispherical end closure (as seen from the inside) for a concave one. In this position, the end closure *acts like a dome under external hydrostatic pressure* and the axial force acting on it is absorbed by the retaining ring pressing against its base.

Besides the above-mentioned advantages accruing from the use of convex (as the pressurizing medium "sees" it) end closures, there are also some disadvantages. The major disadvantage is the decrease of internal usable space in the pressure vessel, as the convex hemispherical end closures take up one diameter of internal length. This constitutes a severe weight, and cost penalty, if the pressure vessel must be lengthened to compensate for the loss of the internal space. But for pressure vessels of such large diameter that fabrication of monolithic hemispherical end closures is impossible, the space taken up by the convex end closures is more than compensated for by the fact that without the use of a modular end closure, a vessel of such a large diameter would not be feasible at all.

Assembly. The assembly of the stacked-ring pressure vessel from many structural components without recourse to welding permits the vessel to be transported to its installation site in the disassembled state, and to be assembled on site with the hoists or cranes used for the removal of end closures or placement of test objects during the regular operation of the pressure vessel after assembly. Because of this, even the heaviest stacked-ring pressure vessel component weighs less than 20% of the total pressure vessel weight. The economies accruing from transporting and placing such a pressure vessel are considerable. Instead of having to transport the complete pressure vessel by barge or ship, when its assembled weight is over 250 tons, the vessel components can be shipped to its permanent location by rail or truck. At the permanent location, the many vessel components are then easily placed sequentially into the vessel pit without recourse to special hoisting equipment. For the assembly of a stacked-ring pressure vessel, only an overhead crane is required that later, after the assembly is completed, becomes part of the pressure test facility. Pressure vessels that must be lowered fully assembled into a pit require a group of specialized hoists and cranes. This requirement becomes more stringent when the weight of the assembly exceeds 250 tons. This weight is generally exceeded by pressure vessels 10 feet in diameter or larger, with an operational pressure of 13,500 psi.

Inspection and Safety. The additional desirable features of a stacked-ring pressure vessel design during its operational life are the ease of inspection of the load-carrying structural members, and the ease with which they can be individually replaced in case of actual or incipient failure. In the stacked-ring pressure vessel, every component, except the liner, is removable and replaceable without cutting or welding. This ease of maintenance is bound to save many dollars over the life of the vessel, which because of this component replaceability feature, is much longer than for monolithic vessels. The inspection of individual structural components for incipient cracks is relatively easy, as the individual tie rods, end closures, and retaining rings are easily accessible for inspection on all of their surfaces. The stacked rings are accessible only from the external surface, but because of their homogeneity and isotropic character, accessibility from one surface is sufficient for ultrasonic or radiographic investigation to locate incipient cracks.

There is one further facet of vessel operation that is not often discussed, but merits further investigation: the stacked-ring design is safer than multilayer or unilayer design. Although pressure vessels are designed with safety factors to prevent failure in service under load, they nevertheless do fail once in a while; when failure occurs, damage to equipment and injury to personnel is extensive. The safety feature of stacked-ring pressure vessel design lies in the separateness of each load-carrying structural member. Since it is quite unlikely that an incipient crack would become self-propagating in more than one structural member at the same time, the internal hydrostatic pressure will be relieved by failure of only one ring member. Thus the failure of any of the individual stacked rings is a local failure, and not a general catastrophic failure. The same applies in a limited measure to the end-closure tie rods. If failure in one of the rods occurs, then only one or two more rods will fail with it before the pressure inside the pressure vessel is relieved. Because of this, the damage to the vessel, as well as to the facility, will be slight and the vessel can be easily repaired. The failure of the end closure, or of the end-closure retaining ring, needless to say, will be just as disastrous as in a multilayer or unilayer vessel, but much easier to repair than in such vessels. The top closure is replaceable in all types of pressure vessels, but the retaining flanges and the bottom closure are not. The stacked-ring pressure vessel does permit, however, the replacement of these structural components also.

Segmented-Wall Pressure Vessel

Although the stacked-ring pressure vessel concept alleviates most of the fabrication and handling problems associated with large monolithic or layered high internal pressure vessels, it does not eliminate them completely. The limitation on the diameter of the vessel for stacked-ring vessel design still remains the forging capability of the steel industry. To be sure, this limitation is less severe for forging rings than for forging monolithic cylinders, but it is nevertheless severe enough to make the stacked-ring construction somewhat less than an optimum solution to the problem of large pressure vessel construction. The sizes of forged rings that industry will produce in the near future will, of course, increase from year to year, but even so it is doubtful whether thick-walled rings of larger than 20-foot diameter and 1-foot thickness will be feasible to fabricate. Consequently the segmented-wall module design has been proposed for the fabrication of high-pressure vessels with diameters beyond the capability of the stacked-ring fabrication technique.[2]

The basic attractiveness of the segmented-wall module design lies in its reliance on small segmentlike modules for the construction of the cylindrical vessel wall. The segments, held together by shear pins extending the length of the cylinder, permit the assembly of very large diameter thick-walled pressure vessels from relatively small interchangeable structural modules (Figure 9) that are easy to fabricate, transport, and assemble at the pressure vessel installation site. In this type of design as with the stacked-ring design, the axial loads on the end closure are carried by a series of tie rods or by an external yoke. One further advantage of the segmented-wall design is that a modular design can also be applied to the end-closure retaining rings, if a tie-rod end-closure restraint system is used, eliminating size and weight of the end closure as the limitation on the maximum diameter of vessel that could

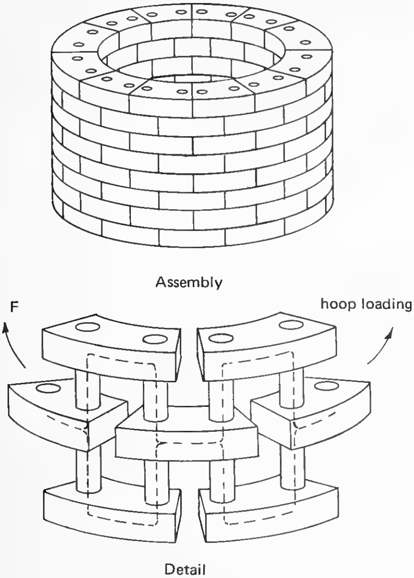


Figure 9. Engineering concept of a segmented wall for cylindrical pressure vessels.

be fabricated by the steel industry. The end closures, again as in the stacked-ring pressure vessel design, can be made as a single forging, if such is feasible in view of its size or as a convex closure assembled from spherical polygons (pentagon shape, orange peel, etc.) (Figure 10). If a yoke end-closure restraint is used, the yoke and bearing block are already of laminated construction, making the vessel completely modular (Figure 11).

Together with the advantages enumerated in the preceding paragraph, there are also disadvantages. The major disadvantages of the segmented-wall module design are the increased weight of the structure over a typical stacked-ring vessel design of same interior dimensions and materials, and considerably greater machining costs of the vessel's component parts. The increased weight of the cylindrical wall structure is primarily a function of a factor not encountered in the monolithic, layered, or stacked-ring vessel designs. This factor, inherent in segmented-wall module design, is the shear-pin linkage of individual wall modules. The shear-pin linkage weakens a vessel wall by introducing shear-pin holes. These shear-pin holes (1) decrease the wall's load-carrying cross section at their location and (2) create stress concentrations, or stress raisers, whose magnitude decreases the effective pressure capability of the vessel. Besides this, the shear-pin linkage in effect reduces the pressure-carrying ability of a cylinder of a given length by one-half because actually only alternate layers of segment modules form a load-carrying hoop around the vessel.

Thus, when one takes into account (1) the approximately 50% decrease in pressure resistance resulting from load bearing by only alternate layers of segments, (2) the presence of tensile stress raisers[3] around shear-pin holes of approximately 3.5 magnitude (as compared to average stress level in segment), and (3) that the shear-pin holes decrease the effective wall thickness by approximately 25%, it would appear that the pressure-containing capability of a

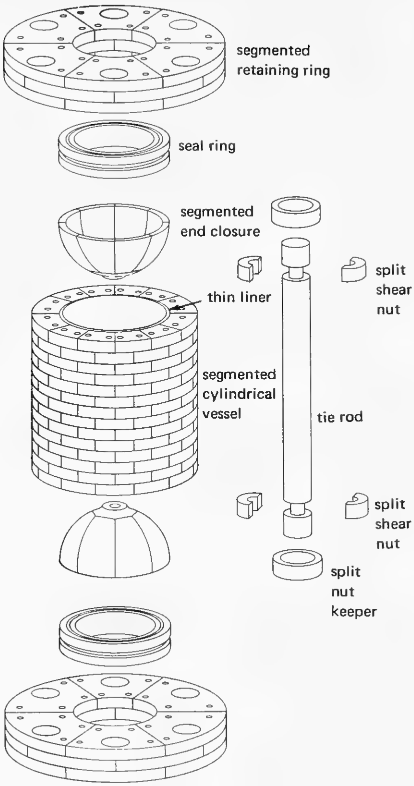


Figure 10. Concept of a pressure vessel composed of a segmented-wall cylinder, segmented end-closure retaining flange, modular end closure and tie-rod end-closure restraint.

segmented-wall module design is only one-eighth to one-ninth of a stacked-ring wall of equal internal dimensions and overall weight. However, because of the many unknowns present, it is impossible to postulate with reasonable accuracy what the internal pressure capability of such a vessel would be without constructing a model of it and pressurizing it to failure.

One further disadvantage of a segmented-wall vessel is that all the areas on modules and shear pins where stress raisers may initiate fracture cannot be inspected satisfactorily without disassembly. Of course, if the fracture does take place, the failure of the vessel will be local, similar to that of a stacked-ring vessel and easily repaired. The number of shrapnel fragments will be somewhat larger than in the stacked-ring vessel because each module is a potential projectile, but with proper precautions (for example, placing the vessel in a pit) this hazard can be virtually eliminated.

The construction of the cylindrical portion of the pressure vessel from modules permits the assembly of the cylinder from easily manufactured, transported, and assembled segment modules. This, however, does not make the size of the segment module forging the sole factor limiting the cylinder's diameter, for the end-closure retaining flanges and the end closure itself are generally substantial forgings, similar to those found in the stacked-ring assembly vessel and much larger than the individual segment module. Clearly, to eliminate the forging of the end-closure retaining flange or of the end closure itself as the factors limiting the vessel's size, it is necessary to make them modular also, or to gird the whole vessel with a yoke restraint of laminar construction. The end-closure flanges could be modularized by assembling them from segments similar to those found in the cylindrical section of the vessel.

The end closure also could be assembled from some smaller modules as it was already proposed for the end closure on the stacked-ring pressure vessel.

The assembly of end closures from modules makes the modularization of a pressure vessel complete, since the external tie rods that hold the end flanges together can be considered modules. If the vessel is completely modularized, the internal diameter of the vessel can be increased by a factor of 2 to 5 over that of a stacked-ring pressure vessel, and 5 to 10 times over a monolithic pressure vessel of 13,500-psi pressure capability. It would thus appear that if the pressure vessel designs are ranked according to their adaptability for constructing pressure vessels over 10 feet in diameter, the segmented design with modularized retaining flanges and end closures (or laminated yoke and bearing blocks) is the more adaptable. When ranked in terms of overall weight and cost for a vessel diameter size that can be built either by the segmented or stacked-ring method, the stacked-ring structure weighs and costs considerably less. The real advantage of the segmented vessel design lies simply in the fact that by using that particular design approach, pressure vessels of much larger diameter can be built for the same pressure than by using the stacked-ring design.

EXPERIMENTAL STUDY DESIGN

General

Since the experimental study on the evaluation of stacked-ring and segmented-wall pressure vessel designs was only exploratory, most of the effort was devoted to evaluating a selected vessel design rather than studying structural parameters that control the structural integrity of such vessels. In other words, the approach was to (1) design and fabricate a stacked-ring and a segmented-wall pressure vessel of comparable size without taking the stress raisers into consideration and (2) pressurize the vessels to failure to determine deviation from the predicted failure pressure, which was selected to be the same for both. The difference between the predicted and experimental performance of the vessels would serve as a good indicator of the magnitude of stress raisers in the structure, while the comparison of experimental failure pressures from the stacked-ring and segmented-wall vessels would show which is more economical on the basis of psi/lb of structure weight. Also, if time permitted, some exploratory investigations could be undertaken into structural details that could have contributed to the early failure of the model vessels.

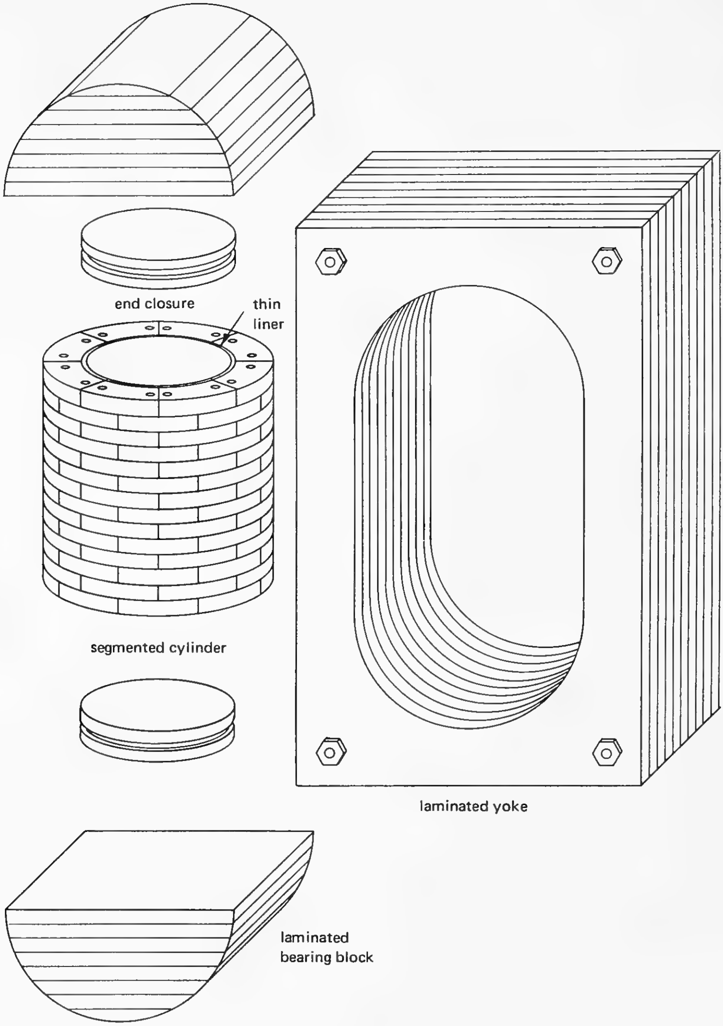


Figure 11. Concept of a pressure vessel composed of a segmented-wall cylinder, laminated bearing block, flat end closure, and laminated yoke end-closure restraint.

Design

The stacked-ring and the segmented-wall pressure vessel models (Figures 12 and 13) were designed to represent in 1:10 scale the full scale 10-foot-diameter, 10,000-psi vessels (operating pressure) made from maraging steel. Since little was known on the magnitude of stress concentrations in such vessels, they were designed on the basis of ordinary engineering calculations. It was calculated that the failure of a given structural member was initiated when the maximum tensile stress in the member became equal to the ultimate tensile stress of the material under uniaxial tension, without taking the stress raisers into consideration. Since the distribution of forces acting on individual members of the vessel was not completely understood in many cases, engineering assumptions were made in their place.

The two vessels were designed to fail at 40,000 psi if they were constructed from maraging steel with 300,000-psi ultimate tensile strength. A design failure pressure of 40,000 psi would give the vessels an apparent safety factor of 4 based on an operating pressure of 10,000 psi while the use of 300,000-psi steel would give the vessel the lightest structure made possible by existing steel alloys applicable to construction of 10-foot-diameter pressure vessels.

Fabrication

Although the actual dimensioning of vessels was based on the 300,000-psi steel, the material selected for actual fabrication of models was not maraging steel, but acrylic plastic. The reasons for using plastic material were twofold. *First*, small forgings of 18% nickel maraging steel were not available at reasonable cost; and *second*, fabrication of the models from a material that had half the ductility of maraging steel would make the model much more sensitive to stress concentrations, causing it to fail when the stresses in the material at the stress raiser reached its ultimate strength. If the model was made from steel, it would probably only yield locally at the stress raiser without any external indications of yielding. Yet, in full-scale vessels, local yielding would in many cases cause the vessel to fail at lower cyclic pressure than predicted on the basis of static failure pressure.

Since acrylic plastic has a tensile strength of about 9,000 psi, while that of 18% nickel maraging steel is about 300,000 psi, the failure pressure of the model is scaled down to 1,200 psi in direct proportion to the lower tensile strength. The operational pressure of the acrylic vessel would be 300 psi instead of the 10,000-psi value for a steel vessel.

The structural components of the stacked-ring model vessels were machined from commercially available acrylic stock (Figures 14 and 15). The rings, and the end-closure retaining flanges, were turned from flat acrylic plates of 1 and 4 inches thickness, respectively.

The tie rods were turned from 2-1/4-inch-diameter acrylic rods, while the hemispherical end closures were contour-machined from 14-inch-diameter by 12-inch-long custom acrylic castings. For the fabrication of modules for the segmented-wall vessel, 1/4-inch-diameter rods were used for shear pins and 1/16-inch and 1/2-inch sheets for wall and retaining flange segments (Figures 16 through 20). Test specimens were taken from the commercial acrylic stock to check on its conformance to the required 9,000-psi tensile strength. Without exception, they have met this requirement by failing in the 9,200-to-9,500-psi tensile-stress range.

Instrumentation

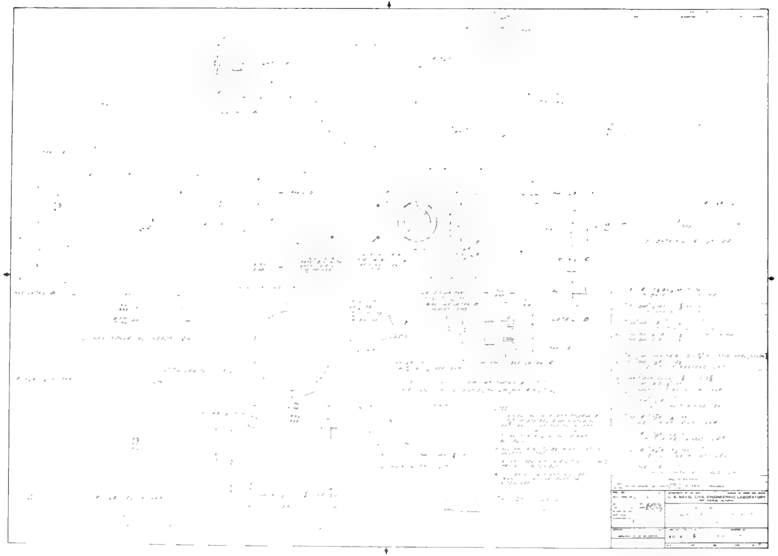
Instrumentation of the models tested to destruction under internal hydrostatic pressure consisted of pressure gages and electrical-resistance strain gages. The pressure gages were used with all of the vessels, while the electrical-resistance strain gages were only used on the stacked-ring pressure vessel.

The reasons for limiting the strain-gage instrumentation to the stacked-ring pressure vessel were as follows:

(1) Since both the stacked-ring and the segmented-wall vessel models utilized the same tie-rod system and hemispherical end closures, there was no need to instrument them twice, as the strains measured during pressurization of the stacked-ring model would be the same as during pressurization of the segmented-wall model.

(2) Only in the stacked-ring model was it possible to measure the actual strains on the end-closure retaining flange and on the rings. In the segmented-wall vessel, the failure of the end-closure retaining flange was predicted to be due to rupturing of pins in that flange, and the failure of the wall segments by shearing of pins and rupturing of segments. In neither case would it be possible to attach strain gages to those structural members at the points of high stress concentration and measure the actual strains.

The actual strain gage installation on the stacked-ring pressure vessel consisted of 15 rosettes placed on major structural components of the pressure vessel model (Figure 21). Six of the rosettes were placed on the end closure, five on the end-closure retaining ring, two on the tie rods, and two on the rings. Only two rosettes, those at a penetration in the end closure, were sufficiently close enough to a stress raiser to measure maximum stresses at a stress concentration. The other rosettes simply measured the general stress level in the structural part of which they were located.



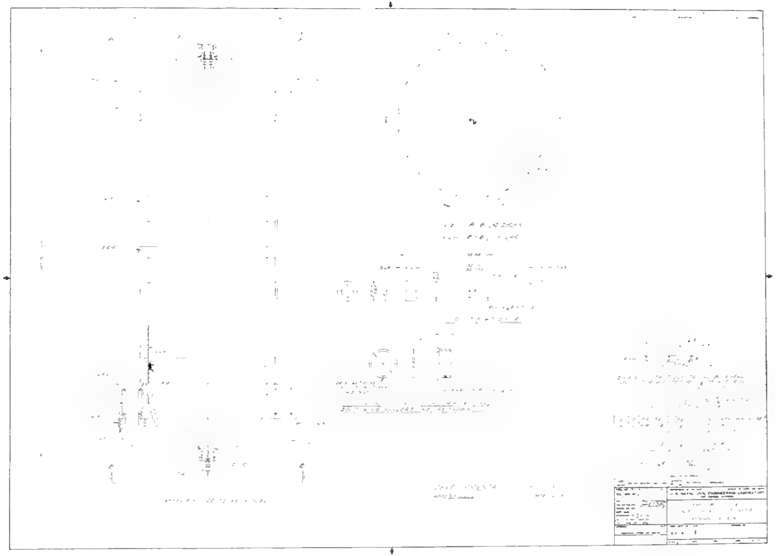
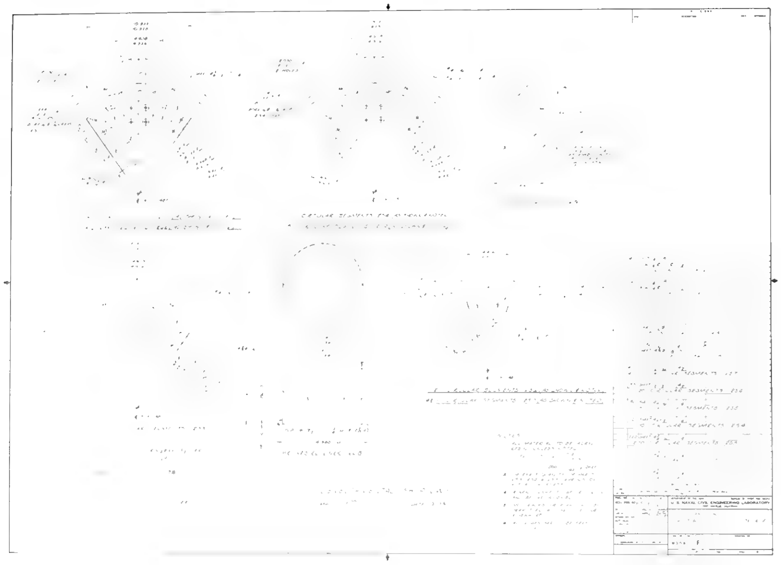


Figure 12 Acrylic model of a maraging steel stacked-ring pressure vessel with 120-inch internal diameter for 10,000-psi pressure service, (a) disassembled, (b) assembled



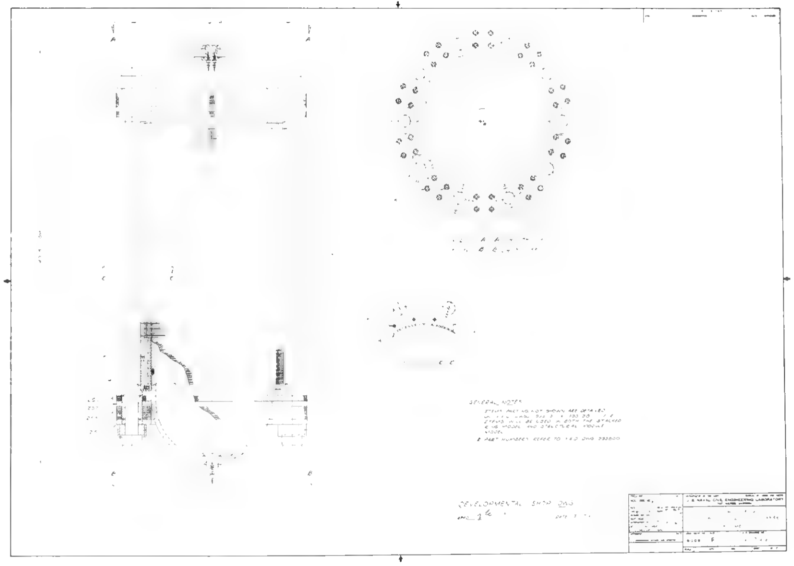


Figure 13. Acrylic model of a maraging steel segmented-wall pressure vessel with 120-inch internal diameter for 10,000-psi pressure service; (a) disassembled, (b) assembled.

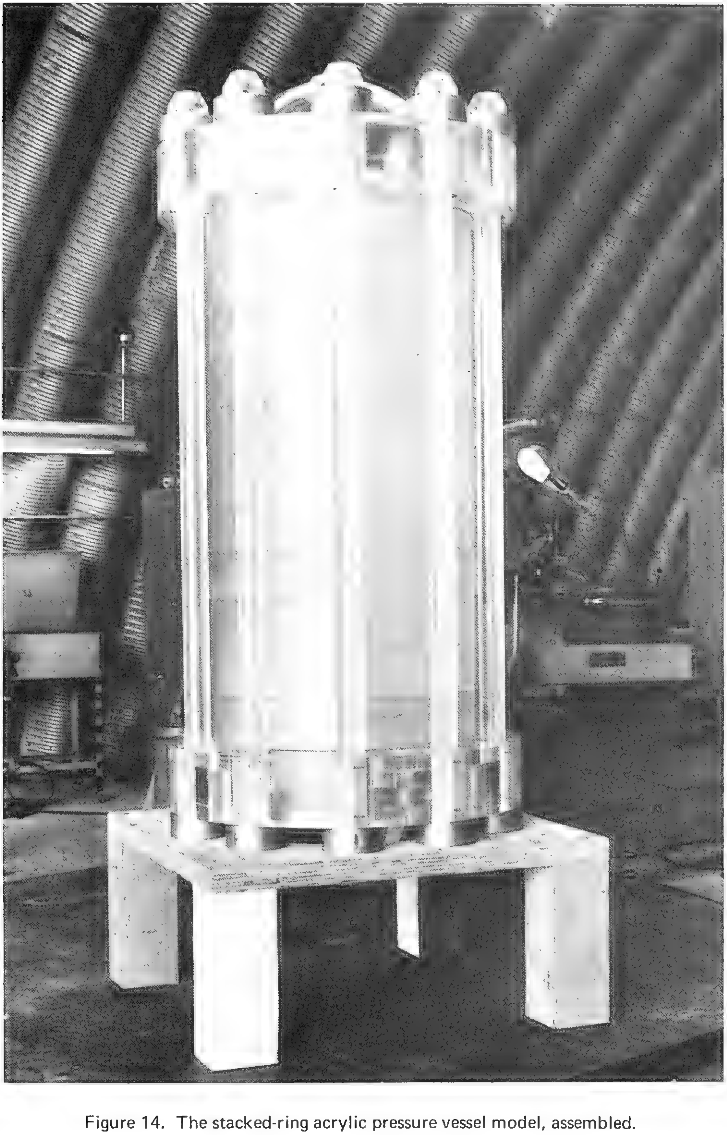


Figure 14. The stacked-ring acrylic pressure vessel model, assembled.

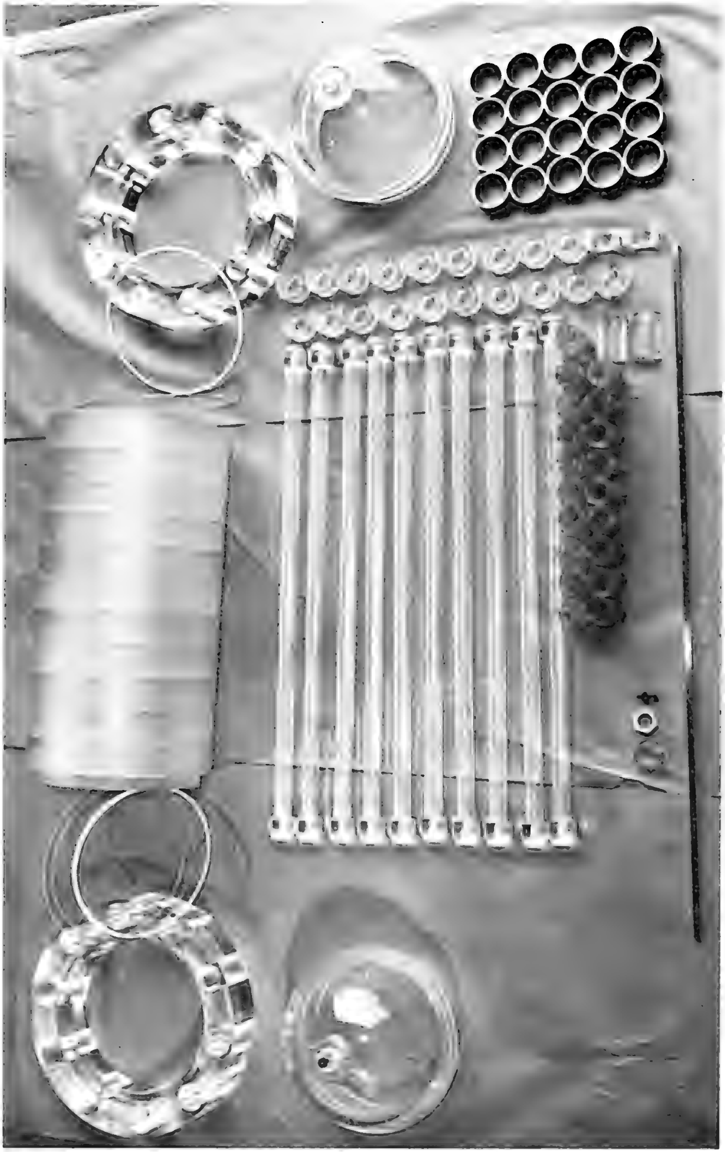


Figure 15. The stacked-ring acrylic pressure vessel model, disassembled.

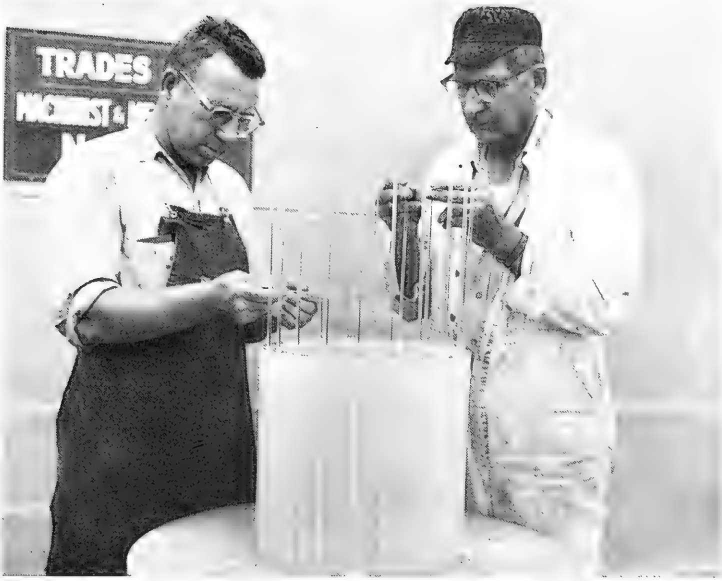


Figure 16. Assembling the segmented-wall pressure vessel from mass-produced 0.067-inch-thick acrylic segments.

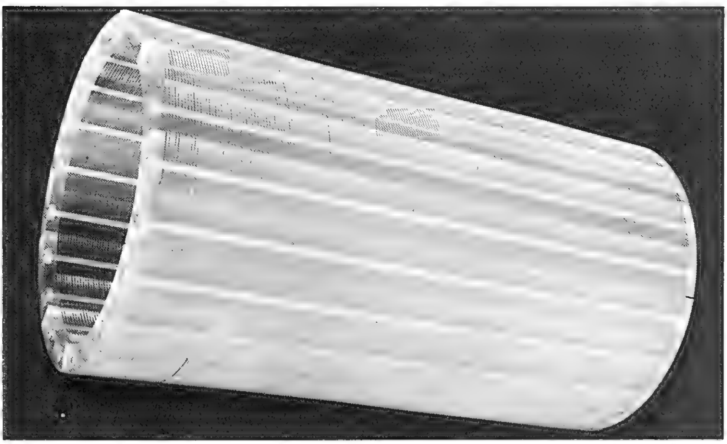


Figure 17. Assembled segmented-wall cylinder.

Figure 18. Segmented-wall cylinder with internal liner in place

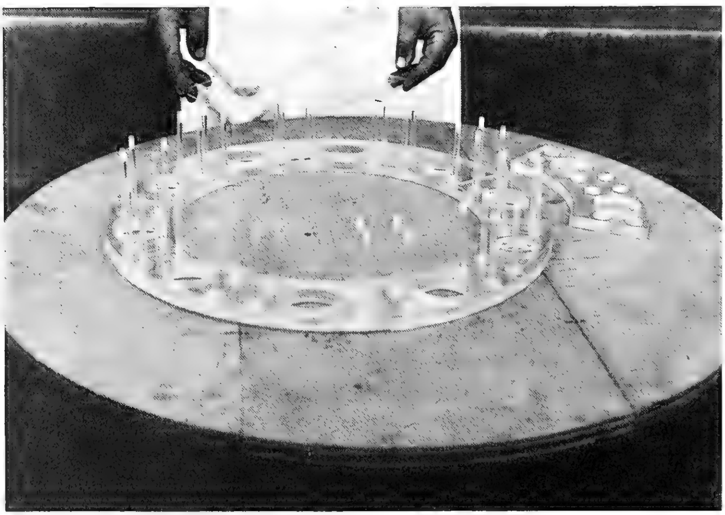


Figure 19. Assembling the segmented end-closure retaining flanges from 0.067-inch-thick acrylic segments.

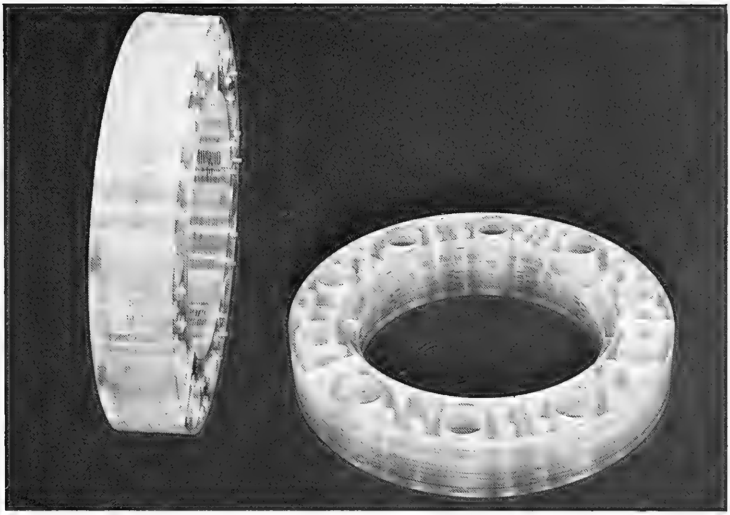


Figure 20. Assembled segmented end-closure retaining flanges.

Testing

Stacked Ring. The stacked-ring pressure vessel was tested with internally applied hydrostatic pressure generated by positive-displacement air-operated pumps. The pressurizing medium was tap water at 65^{O}F. The testing of the stacked-ring pressure vessel was conducted in three distinct steps that were dictated by failure of various structural components at different pressure levels. The *first* test consisted of pressurizing the vessel at a rate of 100 psi/minute (Figure 22) until at 380 psi the test was terminated by fragmentation of the hemispherical end closure. During the test, strain readings were taken at 100-psi intervals (Figure 23).

The *second* test consisted of pressurizing the vessel until the tie rods failed in tension at the base of their heads at 400 psi. For this test the hemispherical end closures were replaced with 2-inch-thick flat aluminum discs that fitted the interior dimensions of the acrylic end-closure retaining ring. Because of this, no change in strain distribution took place in the end-closure retaining ring or the rods (Figure 24).

The *third* test consisted of placing the stack of rings between flat steel end closures held together by steel tie rods. When the interior of this vessel was pressurized, the failure of one of the rings took place at 1,200 psi (Figure 25).

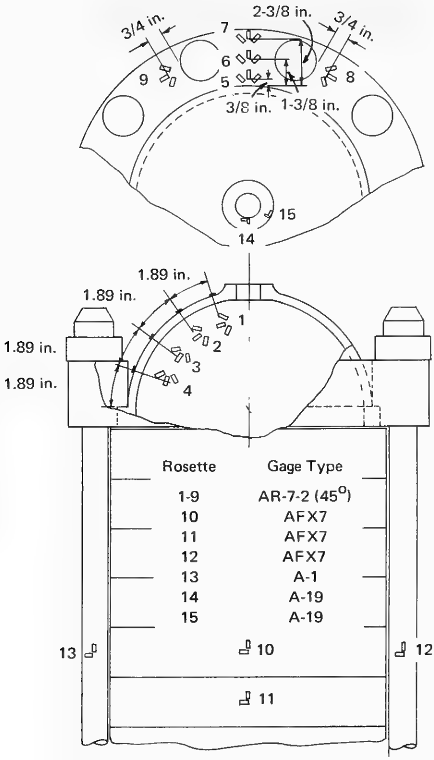


Figure 21. Location of electrical resistance strain gages on the stacked-ring pressure vessel model.

Segmented Wall. The segmented-wall vessel was tested with internally applied hydrostatic pressure in the same manner and at the same temperature as the stacked-ring pressure vessel (Figure 26). The testing of this vessel took place in two steps that were also dictated by the failure of structural components at different pressure levels.

The first test consisted of pressurizing the segmented vessel (Figure 27) until the test was terminated at 140 psi by the tensile failure of the shear pins holding the laminated end-closure retaining ring together.

The second test consisted of pressurizing the segmented wall to destruction at 180 psi of internal hydrostatic pressure (Figure 28). For this test, the segmented-wall cylinder was positioned between two flat steel end plates held together by steel tie rods. The setup was identical to the one used for testing a stacked-ring cylinder to destruction.

FINDINGS

Stacked-Ring Vessel

1. The highest *principal stresses* were measured on the surface of the hemispherical end closures. Since the highest principal stress was recorded in meridional direction at rosette 4, while the hoop stress at rosette 4 was no larger than at rosettes 3, 2 and 1, it appears that considerable flexural stress exists at the base of the hemispherical dome in meridional plane (Figures 23 through 34).

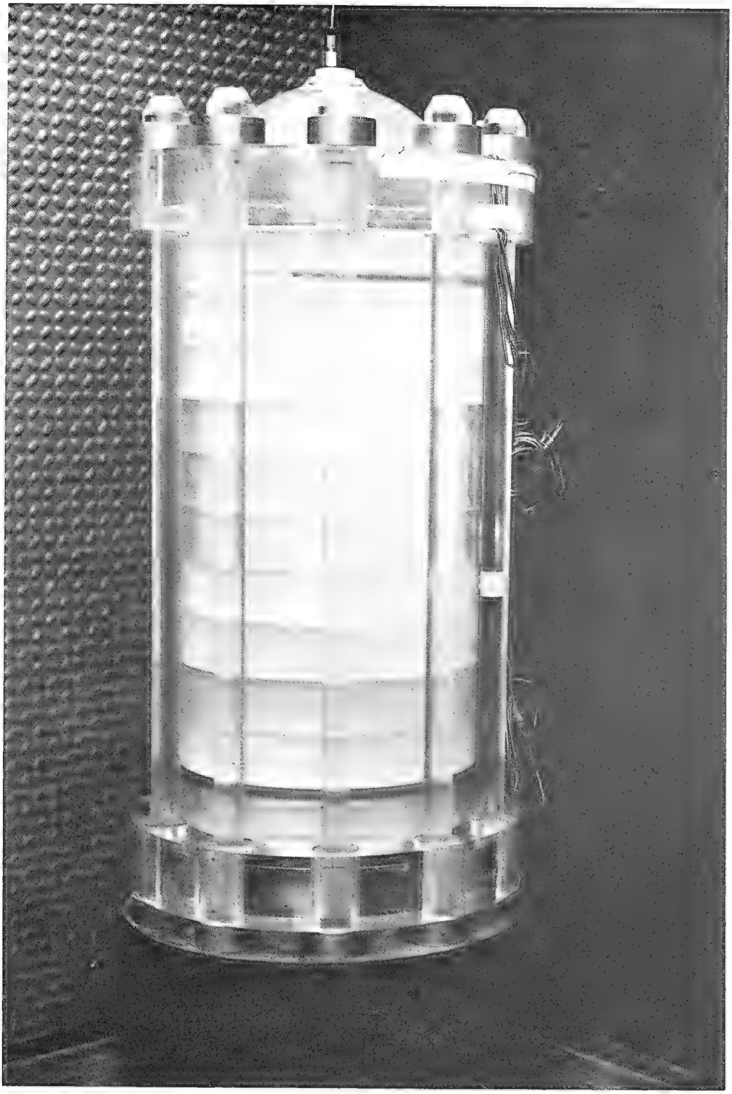


Figure 22. Stacked-ring pressure vessel under internal pressure testing at the Deep Ocean Simulation Facility at NCEL.

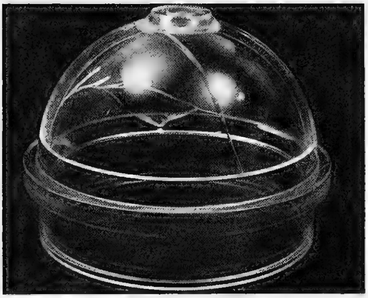
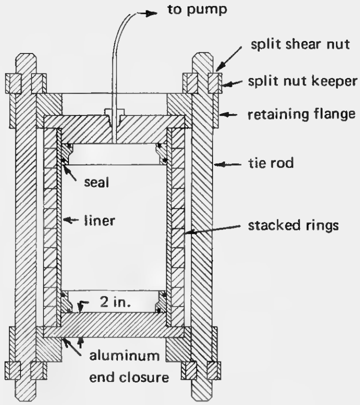


Figure 23. Failed end closure from the stacked-ring pressure vessel. Note the circumferential fracture which caused the end closure to separate from its flange.



Note: All structural components are of acrylic plastic except the aluminum end closure and split nut keepers.

Figure 24. Test arrangement for hydrostatically testing tie rods to failure.

2. Principal stresses of almost the same magnitude as on the hemispherical end closures were measured on the tie rods in axial directions. Since rosettes 12 and 13 were located away from the rod heads, the stresses indicated by them represent the average stress in the tie rods (Figures 35 and 36).

3. The principal stresses on stacked rings in the hoop direction were next in magnitude. The absence of tensile stress in the axial direction indicated that the stacked ring, as postulated, did resist only radial forces exerted by the internal hydrostatic pressure in the vessel (Figures 37 and 38).

4. The principal stresses on the monolithic end-closure retaining ring were the least in magnitude, indicating that unless the magnitude of stress raisers at the root of the flange instep was high, the failure of the vessel would probably not be initiated in this structural component of the vessel (Figures 39 through 43).

5. Fracture of the structural components generally took place either in locations where stress raisers were either known or surmised to exist (Appendix C). Thus, it was surmised prior to the destructive testing that the failure of the hemisphere would take place in equatorial plane somewhat above the flange on the hemispherical end closure. The failure that did take place there was

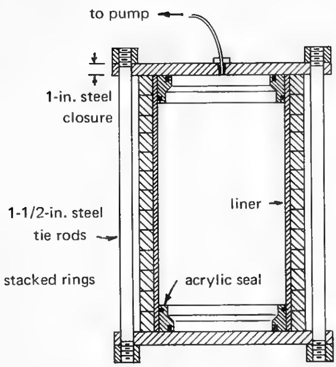


Figure 25. Test arrangement for hydrostatically testing stacked-ring cylinder to failure.

forecast by rosette 4 located approximately 1 inch above fracture plane. This rosette had shown that the maximum principal stress was oriented along the hemisphere's meridian and that it was approximately 50% higher than the hoop membrane stresses measured at other locations on the end closure. Since rosette 4 was 1 inch away from the fracture plane, it did not show the actual stress at the fracture plane that caused the failure. Some exploratory investigations of this tensile stress concentration conducted subsequently have shown that its magnitude in the meridional plane is approximately 3.3 (Appendix C).

6. The failure of the rods was not forecast by the strain gages as they were not located in areas of the highest stress on the rods. The tie rods failed in tension at the very base of their heads where the abrupt change in cross section acted as a stress raiser of unknown magnitude. The failure that took place at approximately 1/3 of calculated failure pressure in the vessel indicated that there must exist in the tie rod at the base of the rod head a stress approximately 3 times higher than the average tensile stress in the middle of the rod's length. Some exploratory investigations of this stress concentration conducted subsequently have to a large measure confirmed this (Appendix C).

7. The fracture of the stacked rings at 1,200 psi showed that the rings were free of stress raisers as they were the only structural components of the vessel to fail at design failure pressure based on the approximately 9,000-psi tensile strength of acrylic. Thus, it appears that the ring is the only structural component of the stacked-ring pressure vessel whose failure can be truly determined on the basis of engineering calculations that do not take stress raisers into consideration. For the other structural components, combinations of stresses and stress raisers must be taken into consideration at otherwise the actual strength of the structural members will be considerably below the calculated one.

Figure 26. Segmented-wall acrylic pressure vessel model undergoing internal pressure test.

Segmented-Wall Vessel

1. The weakest components of the segmented-wall vessel were found to be the tie-bolts holding the individual laminations of the end-closure retaining ring, as they were the first to fail. An increase in their number or diameter would have probably sufficiently raised the strength of the end-closure retaining ring that it would not fail at lower pressure than the segmented wall of the vessel.

2. The segmented wall of the vessel failed at a pressure that is only approximately two-thirteenths of the stacked rings. This indicates that the segmented-wall construction is approximately one-ninth as strong on weight basis as the stacked-ring wall, since the stacked-ring wall is 24% lighter than the segmented wall per unit length of the vessel.

3. The failure of the segmented wall appeared to have been triggered at several locations by tensile failure of the individual laminae at the shear pins followed by shearing of the shear pins themselves.

4. Since the cross section of the individual wall-segment laminae at the shear-pin hole carrying the hoop stresses is identical to the cross section of the stacked ring, and since it takes two layers of segment laminae to provide a complete path for hoop stresses, the difference between bursting pressures of the segmented and stacked-ring walls indicates that the tensile stress concentration around shear pins in the individual segments is probably on the order of 3.3. Subsequent investigation of this stress concentration has, in a large measure, confirmed this (Appendix C).

Figure 27. Failed segmented end-closure retaining ring.

DISCUSSION OF FINDINGS

Although the structural components of the acrylic stacked-ring and segmented-wall pressure vessels failed at different pressures, and in many cases below their expected load-carrying capacity, several generalizations can be made about the behavior of these two different vessel designs.

First, it appears that the stacked-ring modules are the only structural components in the two vessel designs that: (1) possess no stress raisers, (2) can be stress-analyzed reliably, (3) have a failure stress level independent of their fit with other structural components, or machining tolerances, and (4) have the optimized shape for carrying the loading imposed on them. Therefore, they should be utilized in the construction of ocean-environment simulators as large in diameter and high in pressure capability as the fabrication capability of the steel industry permits. In cases where the material properties of thick high-strength forgings are well known, forgings are to be preferred over laminated rings, as both the stress analysis and quality control are well understood. Where a sufficiently thick ring forging cannot be made, or the properties of thick forgings are uncertain, welded concentric laminations can be used for individual stacked-ring fabrication.

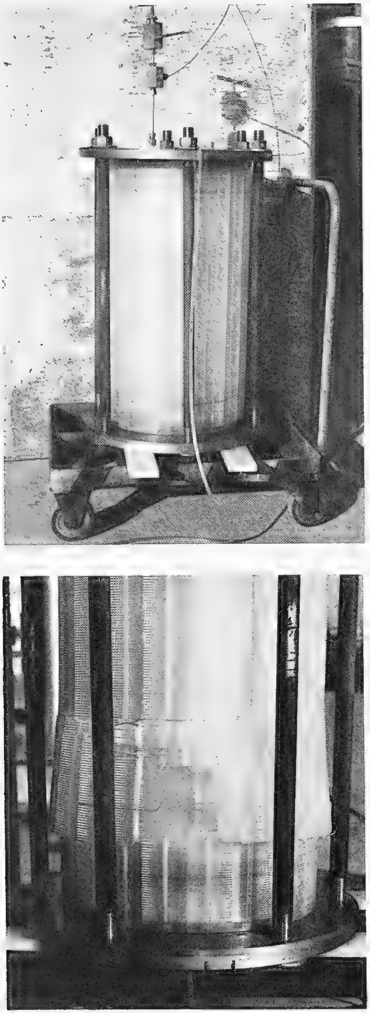


Figure 28. Testing the segmented-wall cylinder to failure; (a) test arrangement, (b) cylinder after failure.

Second, the segmented-wall construction, consisting of small segment modules held together with shear pins, is a feasible method of assembling cylindrical pressure vessels where the axial forces on the end closures are not resisted by the cylinder but by other structural members. This construction method appears to be desirable, however, only for those applications where stacked-ring construction is not feasible because the dimensions of the ring exceed the fabrication capability of the industry. The major drawback of this cylinder construction technique is that it requires approximately 9 to 10 times as much steel as the stacked-ring construction method. In addition, there is considerably more machining required on individual segments than on stacked rings, but the increased amount of machining is probably offset by the mass-production techniques that can be applied to their fabrication. From the stress analysis viewpoint, the segmented-wall construction presents also a real problem not only because of the magnitude of stress concentrations at the shear-pin holes, but also because this magnitude depends to a large degree on the clearance between the pin and the opening, and on the alignment of the shear-pin holes in successive segment layers. Misalignment of holes between segment layers also can induce bending strains in the shear pins causing them to fail at lower internal pressure loading than expected.

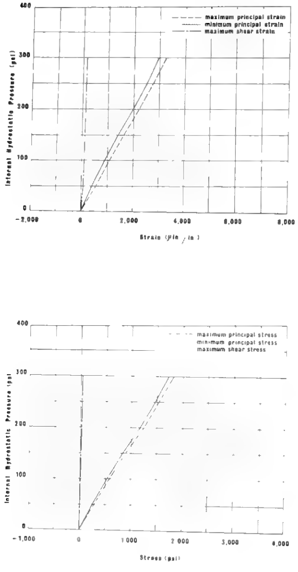


Figure 29 Principal strains and stresses on the hemispherical end closure, location 1.

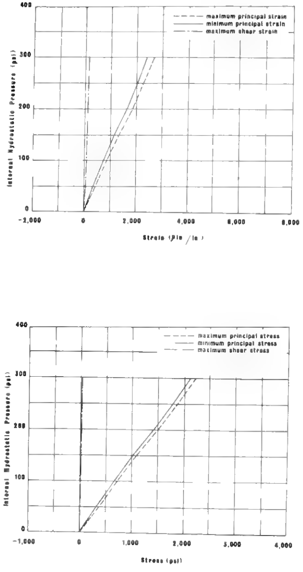


Figure 30 Principal strains and stresses on the hemispherical end closure; location 2.

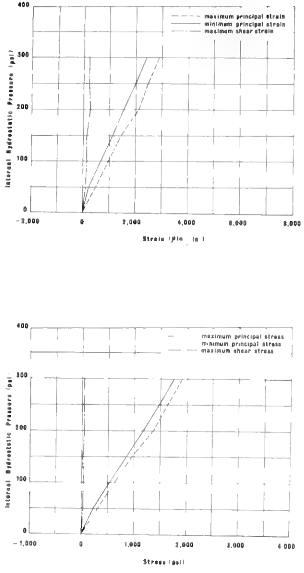


Figure 31 Principal strains and stresses on the hemispherical end closure; location 3.

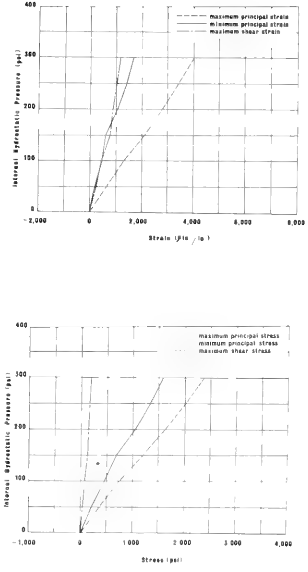


Figure 32 Principal strains and stresses on the hemispherical end closure; location 4

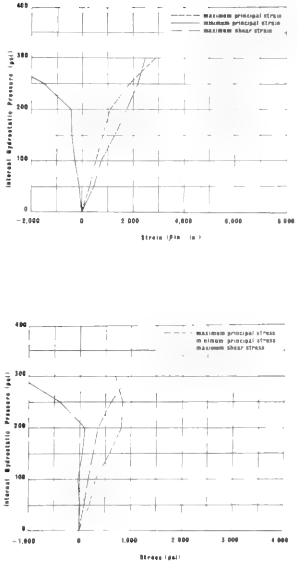


Figure 33 Principal strains and stresses on the hemispherical end closure; location 14.

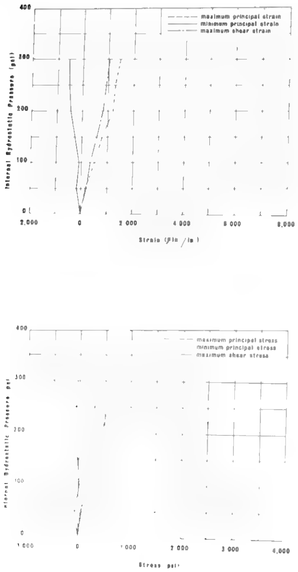


Figure 34 Principal strains and stresses on the hemispherical end closure, location 15

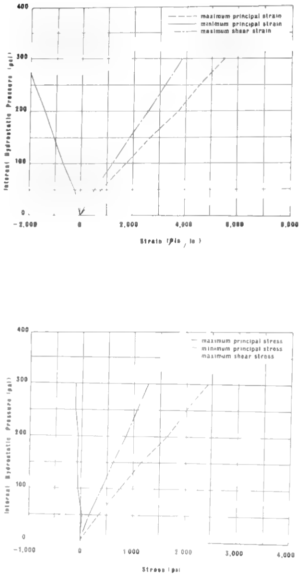


Figure 35 Principal strains and stresses on the tie rods, location 12

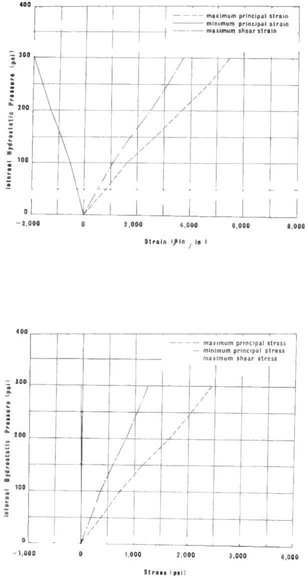


Figure 36. Principal strains and stresses on the tie rods, location 13.

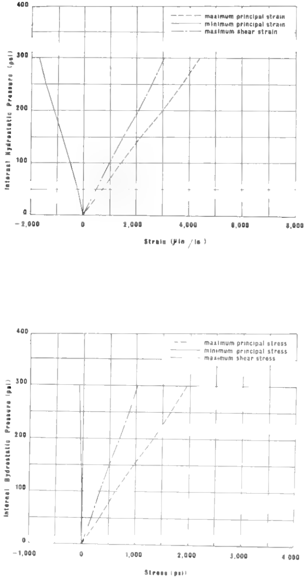


Figure 37. Principal strains and stresses in the stacked rings, location 10

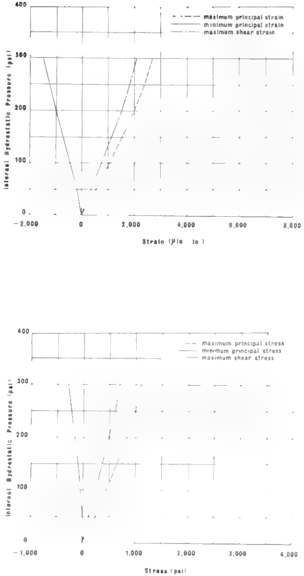


Figure 38. Principal strains and stresses in the stacked rings, location 11.

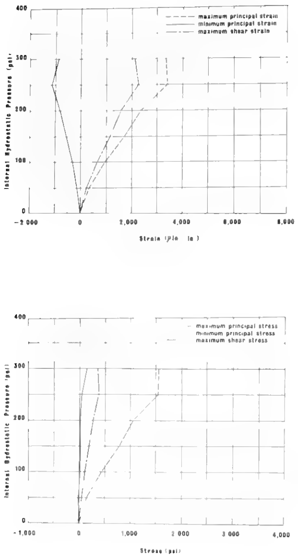


Figure 39. Principal strains and stresses in the retaining ring, location 5

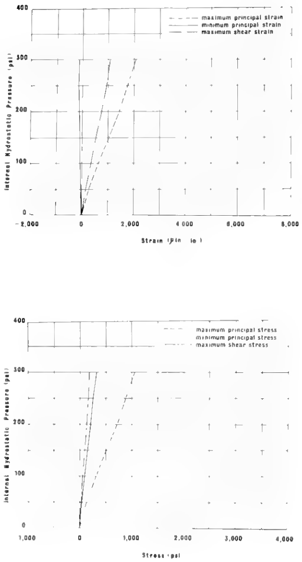


Figure 40. Principal strains and stresses in the retaining ring, location 6

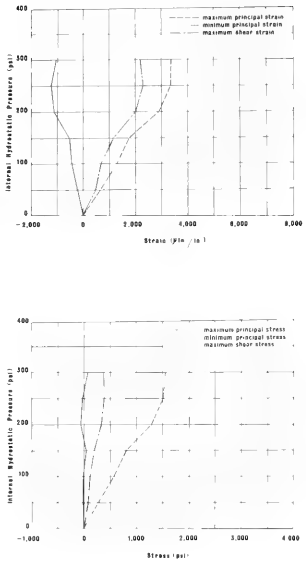


Figure 41. Principal strains and stresses in the retaining ring, location 7.

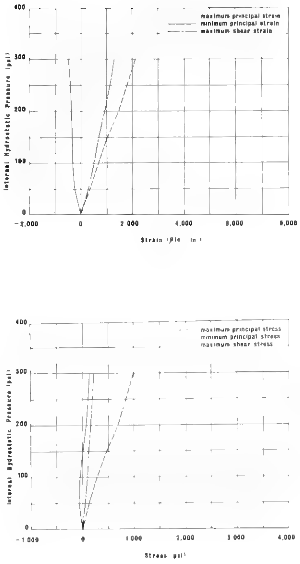


Figure 42. Principal strains and stresses in the retaining ring, location 8.

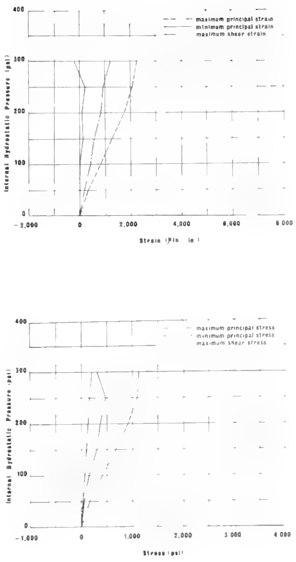


Figure 43. Principal strains and stresses in the retaining ring, location 9

Third, the use of tie rods and retaining flanges for restraining the hemispherical end closure proved to be feasible. As anticipated, this restraint was easy to operate in opening and closing the vessel. However, from the structural viewpoint, this restraint left much to be desired as the stress level in the structural components was higher than calculated. The high level resulted from stress concentrations introduced by the geometry as well as by the machining tolerances. This was shown quite clearly by the failure of tie rods and segmented retaining flanges at hydrostatic pressures considerably lower than those for the stacked rings. The stress concentration in the tie rods appeared to have a magnitude of 3 based on the comparison of hydrostatic pressure, at which tie rods and stacked rings failed. In view of this, it would appear that in order for tie-rod restraint to operate properly, the nominal stress level in the tie rods would have to be decreased by a factor of 3 through enlargement of the tie-rod diameter, or the tie-rod head would have to be redesigned so that the stress raiser effect is considerably decreased.

The same applies to the hemispherical end closure that failed at approximately one-third of its predicted failure pressure. There the problem can also be resolved either by lowering the average stress level in the end closure by a factor of 3 through increase in thickness of the hemisphere, or the transition zone between the end-closure flange and the hemisphere would have to be redesigned. In either case, it appears that the design of the hemispherical end closure with the tie-rod restraint system requires more than nominal engineering stress calculations, and that the weight of this system would have to be increased considerably.

Fourth, in view of the previous discussion, it appears that the tie-rod restraint system with hemispherical end closures, even though proven to be successful operationally, leaves a lot to be desired from the structural viewpoint. It appears, therefore, that the tie-rod restraint system with which the stacked-ring and segmented-wall vessel designs were equipped is less desirable and structurally safe than the continuous-yoke system with bearing blocks and flat end closures discussed earlier in this report.

Fifth, the radial seals utilized on the end closures of the stacked-ring and segmented-wall vessel designs performed satisfactorily without any leakage during all of the hydrostatic tests to which the acrylic models were subjected. For higher pressures, such as those that would be encountered in the steel vessels, the self-energizing radial seals experimented with in this study should be utilized (Appendix C). Thus, it appears that radial seal designs experimented with in this study adequately meet the operational needs of large vessels with 10,000-psi or higher operational pressure.

CONCLUSIONS

1. Both the stacked-ring and the segmented-wall cylindrical pressure vessel concepts are technologically and operationally feasible for construction of large-diameter, high-pressure cylinders without recourse to welding. The stacked ring is more economical and structurally sound than the segmented wall, in which stress concentrations dictate the use of thicker walls and also serve as potential sources of fracture. However, when interior size and pressure capabilities are the only considerations, the segmented-wall concept permits construction of considerably larger cylindrical pressure vessels than the stacked-ring concept.

2. The tie-rod end-closure restraint system is technologically and operationally feasible and can be used with stacked-ring or segmented-wall pressure vessels, but it is structurally less sound than the continuous laminated-yoke system because of the many stress concentrations inherent in this concept.

3. When a laminated-yoke end-closure restraint system is mated with a stacked-ring cylinder, it results in an economical and structurally sound pressure vessel for diameters and pressures in excess of 10 feet and 10,000 psi, respectively.

ACKNOWLEDGMENT

The pressure vessel models tested in this study were designed by Mr. R. O. Doty and Mr. B. M. Merrill of NCEL's Design Division, and the photoelastic analysis of structural components was performed by Mr. J. R. Keeton of the Material Sciences Division.

Appendix A

SUMMARY OF NCEL STUDY GROUP REPORT* ON PRESSURE VESSEL CONCEPTS AND IMPLOSION EFFECT

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* The original letter report was prepared on 24 August 1964 and submitted to NAVFAC for their consideration in response to their request for methods of constructing a 10-foot-diameter vessel for 10,000-psi internal pressure operation.

PRESSURE VESSEL CONCEPTS

Stacked-Ring Concept

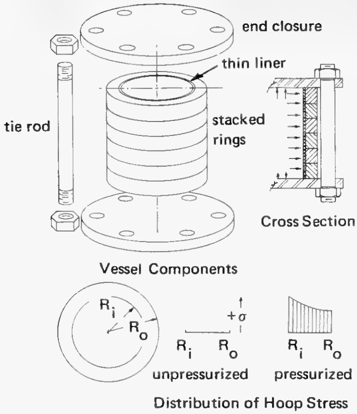


Figure A-1. Stacked-ring concept of pressure vessel construction.

Discussion. The stacked-ring concept (Figure A-1) consists of an inner liner surrounded by reinforcing rings. Since the rings are stacked upon each other along the axis of revolution of the vessel, they give continuous radial support to the liner. In this manner, the liner serves primarily as a pressure seal while the rings take the radial and circumferential stresses. It is felt that this system is much preferable to one in which a space is left between the reinforcing rings, since the latter method would necessitate a heavier liner in order to withstand bending induced by the nonuniform support. Longitudinal anchor bolts of nominal cross-sectional area would be used to hold the rings together. This type of vessel requires tie rods of sufficient size to carry the axial load.

The stresses carried by the rings may be computed in the same way as those in a monobloc forging, or other continuous shell. It is possible to reduce the external radius by the shrink-fitting method, or the autofrettage procedure. For example, calculations indicate that a suitable "ring" could be fabricated by shrink-fitting a large ring onto a smaller ring. However, the increased fabrication costs probably outweigh any saving realized by reduction of size.

This method of design has been used successfully in a small pressure chamber.[4] However, extensive changes in the tie-rod and end-closure design must be made in order to (1) permit rapid access to the vessel's interior, and (2) decrease the weight of the end closure, which because of its flat design would result in such a thick forging for the 10-foot-diameter vessel that it could not be manufactured.

Desirable Features

1. The individual rings are within the size and weight capabilities of fabrication and transportation facilities. Final assembly would be done at the site.

2. Since a ring would be required for the upper flange of all vessels under serious consideration, the additional rings required for the body of a stacked-ring vessel can be obtained without additional tooling-up costs.

3. If desirable, an extra ring could be fabricated for use in metallurgical tests and test of fabrication suitability.

4. No welding would be required on the shell body. Thus, fabrication costs are reduced and reliability is enhanced.

5. Failure of the liner would result in loss of water from the tank, but would not cause failure of the rings. Even if one ring were to fail, the cost of repair would be much less than the cost of replacement of the entire tank. The facilities available for handling the closure would be adequate to disassemble the vessel ring by ring and replace the damaged ring.

6. Analysis of the ring behavior is fairly straightforward since the end closures are not attached to the shell body and each ring behaves in approximately the same manner.

Undesirable Features

1. The total weight of steel used in the construction will be at least 50% greater than in a multilayer construction because a separate system of structural members must be employed to restrain the end closure.

2. Design of the end closures and of the discontinuous tie-rod restraint systems will be difficult as little is known about them.

Conclusions. From the standpoint of feasibility of fabrication, cost of fabrication, reliability (including inherent safety, ease of inspection, etc.), ease of operation, and maintenance, the stacked-ring concept rates very high. An independent device (yoke or tie rods) is required for taking the axial load, but such a device appears to be desirable regardless of the type of tank employed.

Recommendations. It is recommended that an exploratory design be made according to this concept in order to obtain firm cost estimates for fabrication of a stacked-ring pressure vessel.

Multilayer* Concept

Discussion. A multilayer pressure vessel is made up of a number of concentric cylindrical shells. Construction of a multilayer vessel begins with rolling and welding of the vessel's inner cylindrical shells, which may be made

* A. O. Smith trademark.

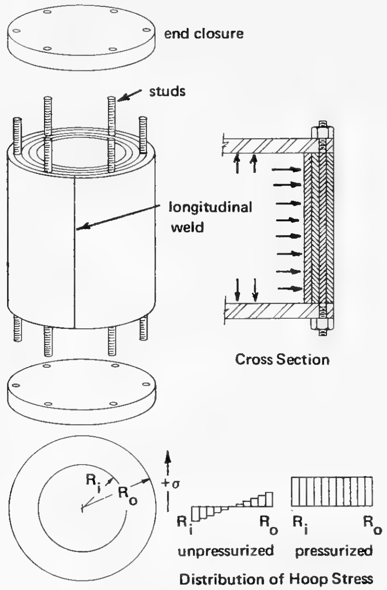


Figure A-2. Multilayer concept of pressure vessel construction.

of corrosion-resistant steel. Successive layers are wrapped (Figure A-2) around and the longitudinal welds join the longitudinal edges of the rolled plates to form concentric cylindrical shells. Shrinkage of the welds is controlled so that the interior layers of the shell attain a desired compressive prestress.

Desirable Features

1. The individual layers are constructed from relatively thin plates which are readily available and whose quality is controllable.

2. Heavy welds are not required, and the welds can be inspected as each layer is added.

3. Only a relatively thin inner shell of corrosion-resistant steel is required. The other layers may be of another grade.

4. Failure occurring in one layer of the vessel would not necessarily propagate to other layers unless the test pressure were sufficient to cause burst of all the layers.

5. Only the inner shell is pressure-tight. The remaining layers are vented to the outside so that overpressure causing rupture of the inner shell would not rupture the entire tank.

6. The fabrication experience and safety record associated with this proprietary construction technique render the behavior more predictable than the behavior of vessels constructed according to the separated layer concept.

Undesirable Features

1. Shipment of a completed 10-foot-diameter multilayer cylinder would involve a 350-ton object whose external diameter of about 13 feet is close to railroad size limits.

2. Replacement of any portion of the vessel would require costly repair procedures. The installed laboratory lifting facilities would not be sufficient to assist in any disassembly.

3. Welds, although made on relatively thin individual layers, except for the end-closure flanges, would nevertheless be an added source of uncertainty with regard to behavior under impact loading, cyclic stressing, etc.

4. The fabrication is restricted to basically one company due to the proprietary nature of this concept.

Conclusions. The multilayer method has been sucessfully used in some previous applications with operational pressures of 10,000 psi and could be extended, with reasonable surety, to the 10-foot size required for the present application.

Recommendations. A complete design and cost estimate should be obtained from the fabricators.

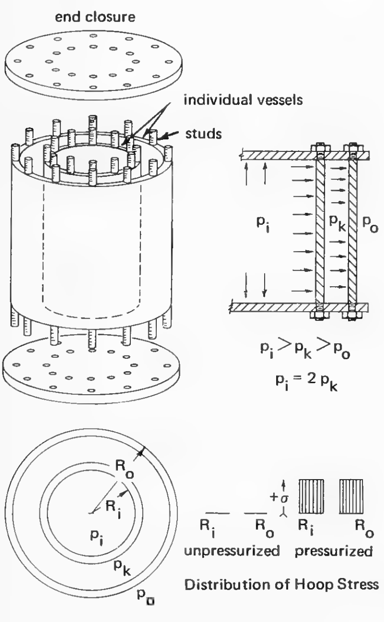


Figure A-3. Separated layer concept of pressure vessel construction.

Separated Layer Concept

The separated layer concept consists of fabricating a vessel from a series of individual shells (Figure A-3) separated by annular fluid spaces. Two systems have been briefly considered, one allowing for continuous control of the annular space pressures and the other providing the initial pressurization to some prescribed values with the subsequent magnitude of the annular space pressures being determined by the deformation of the vessels and the compressibility of the fluid.

A separated layer vessel theory has been developed which assumes that the maximum shear stress, T_{max}, at the interior of each layer, has the same value.

In order to keep the time required to open and close the vessel within practical limits, independent sealing of each tank is precluded. Closure would have to be provided by a common end closure, or closures.

Desirable Features

1. By using several vessels separated by a small, fluid-filled annular space, the wall thickness of the individual shells is reduced. Fabrication operations including forging, rolling, welding, etc., are less expensive for the thinner vessels.

2. Individual vessels could be fabricated elsewhere and assembled at the site. However, for such on-site assembly, it is desirable to reduce welding operations to a minimum.

3. The inner vessel could be constructed of a corrosion-resistant steel while other vessels could be of different material.

4. Compared with the multilayer construction, the separated layer concept provides more flexibility in controlling the stresses in the vessel. If the annular space pressurization proceeds simultaneously with the test chamber pressurization, it becomes unnecessary to obtain large compressive hoop stresses near the interior by prestressing.

Undesirable Features

1. Complicated systems for initial or continuous pressurization of the annular fluid spaces are required.

2. Differences in strains between individual vessels (for example, unequal axial shortening) could lead to difficulties in sealing.

3. Dynamic behavior of a separated layer vessel resulting from implosion of a test object or other causes would require careful analysis.

4. Whereas in a monobloc or multilayer shell the plastic flow of the interior portion of the shell is restrained by the elastic outer portions until yield has proceeded through the shell wall, the behavior of a separated layer vessel at pressures above that required to cause yielding of the interior tank has not been established. For instance, the compressibility of the fluid between tanks might allow the inner vessel to burst with little restraint being offered by the outer layers. Hence, the factor of safety against burst would not be significantly larger than the factor of safety against initial yielding.

5. Sudden depressurization of the test volume could lead to buckling of the vessel, so that this factor would require consideration in design of the inner vessel. This would require thorough study of the annular space pressures resulting from depressurization of the test chamber.

Conclusions. From an operational standpoint, the separated layer vessel is more complex than other concepts studied, because of the annular space pressurization required. Fabrication, which requires fairly extensive welding, would be more costly and less reliable than fabrication of vessels requiring less welding. The stresses are controlled by the annular space pressures as well as the test chamber pressure so that the stressing of the vessels may be made to suit the individual test pressure. This concept merits further study for use in larger pressure vessels.

Recommendations. It is recommended that further studies be made of the problems associated with the separated layer pressure vessels, namely, deformation of the vessels under pressure, effect of implosions or other dynamic disturbances including the possibility of buckling of the inner vessel, stresses in the vessels near the end flanges, burst strength, etc.

In view of the above-mentioned uncertainties, the separated layer vessel concept is not recommended for the 10-foot-diameter, 10,000-psi pressure vessel under immediate consideration.

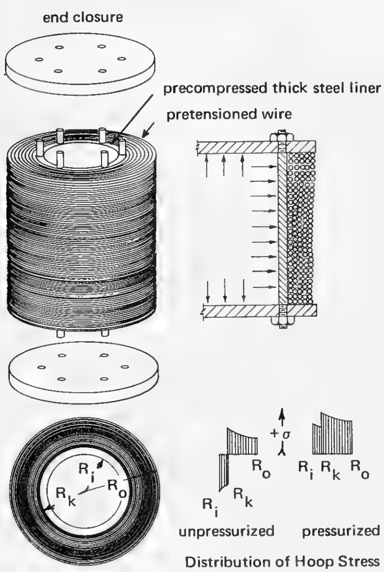


Figure A-4. Pretensioned-wrapped-wire concept of pressure vessel construction.

Wire-Wound, Cylindrical Steel-Core Vessel

The concept of wire-wound cylindrical steel-core vessels subjected to high internal pressures has been used for reinforcing gun barrels (Figure A-4), in which the wire windings are used only for absorbing hoop and radial stresses. The windings offer no resistance to axial loads and an inner monobloc or multilayer steel core must be used to absorb the axial internal-pressure load, or an outer yoke must be employed for the same purpose.

In the absence of internal pressure, the windings exert an external pressure on the core which results in compressive stresses in the core. Internal pressures then act to induce hoop-tension stresses in both the inner monobloc and its wire windings. Thus, under operating conditions the inner core may be considered to have both internal and external pressures acting upon it and the windings to have induced stresses resulting from the winding tension and the internal pressure.

Acceptable design procedures are available for wire-wound cylindrical shells, which are based on the allowable stress in the inner core. Further investigation is required to determine the benefits of applying the windings at a variable tension to produce a constant tension under operating conditions.

It has not been possible to obtain any information on companies which currently undertake wire winding of cylinders of the size contemplated. It is not likely that such companies exist within the United States. It is felt that should this concept be accepted, a considerable amount of time, and therefore expense, will be involved in setting up a facility whereby the fabrication could be accomplished, particularly for the preferable on-site fabrication.

Desirable Features

1. Imposition of prestress on the inner vessel shell by tensioned wires makes thinner vessel walls feasible than in stacked-ring or monobloc vessels.

2. Wire utilizes steel with strength in excess of 250,000 psi that is not available for multilayer or monobloc vessels. This permits further reduction in vessel thickness as compared to multilayer vessels.

3. Fracture crack propagation will be arrested at the inner vessel—wire layer interface.

Undesirable Features

1. Retaining the wire windings at each end of the vessel may be difficult.

2. Yielding the wire in one or more places during winding could occur without the fabricator's knowledge.

3. Abrasion and friction would occur between the wires in loading and unloading cycles.

4. Redistribution of tensions within the winding due to creep may occur.

5. Early fatigue failure of wires in cyclic loading may result from stress raisers in the form of localized abrasion and corrosion.

6. The expense involved in setting up an on-site winding facility will far exceed the transportation costs of a large vessel based on alternative concepts.

Recommendations. It is recommended that in view of the lack of fabrication facilities and the several factors which seriously influence the reliability of such a vessel, the concept of a wire-wound pressure vessel not be considered for immediate application to construction of 10-foot-diameter vessels, and that further investigation into the design of such a vessel should not be undertaken at this time.

Segmented Modular Vessel

One of the major problems that confronts all large pressure vessels during their fabrication is the unavailability of large enough fabrication facilities, and the limitation imposed on their size by the railroad bridges and tunnels. Transportation by ship may obviate some of the latter problems but then, all fabrication facilities and vessel location sites are not always located at harbors capable of unloading such large structures.

This problem would be eliminated if the pressure vessel could be built upon location from readily transportable small construction modules. Such modules could possibly have the shape of long mechanically interlocking cylindrical staves, or of small curved interlocking segment modules (Figure A-5). Inside the cylinder made up of these cylinder construction modules would be a thin steel liner of highly ductile steel which would make the assembled cylinder watertight. To retain the end closures, a flange ring would have to be mechanically attached to the ends of the staves, while in the segmented modular construction; the closure would be kept in place by a yoke girdling the whole cylinder around its axis, or a series of circumferentially spaced tie rods.

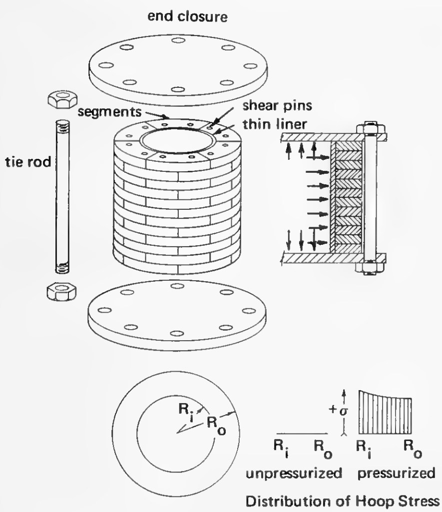


Figure A-5. Segmented-wall concept of pressure vessel construction.

Desirable Features

1. Cylindrical high-pressure vessels of diameter in excess of 20 feet can be built utilizing this concept, as even for the very large vessels the size of individual segments would be under 20 tons.

2. High-strength nonweldable materials can be utilized, as no welding is required in this vessel. To a large extent, the use of high-strength materials can compensate for the need for additional wall thickness to accommodate stress concentrations around shear pins.

3. Individual segments can be transported by common commercial carriers without any trouble.

4. At the erection site only the hoists associated with the pressure vessel test facility need be employed in the assembly of the vessel.

5. The assembly of the vessel can take place after the test facility building has been completed, since individual segments can easily pass through the doors of the facility. Because of this, the overall construction time of the facility may be reduced, as the vessel does not have to be fabricated and installed before the building can be built.

Undesirable Features

1. This construction concept is very new and a very extensive R&D program will have to be conducted to develop safe design and fabrication techniques.

2. This vessel will probably require 5 to 10 times as much steel as a multilayer vessel of same material because of the stress concentrations in modules and due to extra material needed for a yoke or tie-rod end-closure restraint system.

3. Machining of modules for a segmented vessel will require at least 100 times more machine shop time than for a multilayer vessel.

4. The assembly time of such a vessel in situ is longer than for welding a multilayer vessel in the shop, or in situ.

Conclusion. The construction of pressure vessels by the segmented modular method is a new concept that has not been extensively applied. If practive proves it successful, it will mean a breakthrough in the technology of fabricating large, high-pressure metallic vessels.

Recommendation. The segmented vessel construction concept is not recommended for immediate consideration in the construction of large pressure vessels because of complete absence of design or experimental data. However, a study should be immediately initiated to explore this vessel concept.

Prestressed Concrete

The possibility of using prestressed concrete as material for constructing a deep-sea-pressure simulation vessel appears attractive and competitive with other fabrication methods. Prestressed concrete is quite commonly used for liquid containers such as storage tanks and elevated tanks, and for much larger structures—powerhouses, penstocks, pressure pipelines, etc. In the case under consideration, there are no restrictions on size and weight if the vessel is built on site; whereas, size and weight considerations become restrictive for a shop-fabricated vessel transported by rail or water to the site. There are other advantages in the prestressed-concrete concept, the most important being that the vessel and building foundations may be integrally designed for more useful load bearing and distribution capacity. The handling equipment for installing and removing a prefabricated vessel (500 tons plus), unless a modular steel construction is used, is dispensed with.

Work on prestressed-concrete design aspects and dynamic action of reinforced concrete structures to shock loads has been underway at NCEL for the past 14 years. With the concentration of talent in this field it appears likely that a design could be evolved. This method has been applied by NAVFAC as far back as 1941 for a water storage tank, and the first prestressed-concrete barge manufactured in the United States for the Navy is still in service.

The principle of prestressed-concrete pressure vessels (Figure A-6) is that the hoop stresses are assisted by high-strength steel wires under full load and under no load the tensile stress in steel places the concrete in compression. Longitudinal tension to retain the end closure is resisted by means of high-strength tensile bars or studs going the full height of the vessel and which are anchored in the bottom slab. An inner liner of steel or some resilient material is recommended.

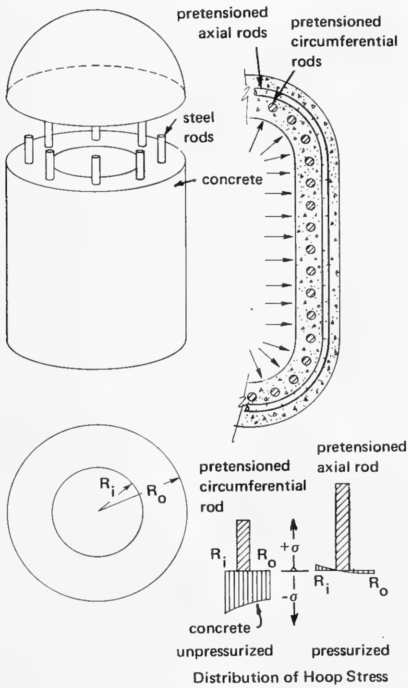


Figure A-6. Prestressed-concrete concept of pressure vessel construction.

Desirable Features

1. Internal pressure vessels of almost any size can be erected in place using this concept.

2. Several vessels of this concept have been built with diameters in excess of 10 feet, and have been found to perform successfully.

3. Cost of building the vessel in situ is less than the cost of any other vessel concept of similar pressure capability, diameter, and length.

4. This vessel is safe in operation as the propagation of a fracture crack in the wall is not accompanied by fragmentation. As soon as the overpressure relieves itself through the crack in the concrete, the pretensioned wires and rods in the vessel close the crack.

Undesirable Features

1. Internal pressure rating of the vessel depends on the compressive strength of the concrete. Since currently the strength of concrete is less than 10,000 psi, the highest internal pressure that such a vessel can contain is also less than 10,000 psi.

2. No design data is available on the incorporation of rapid opening end-closure mechanism into a concrete pressure vessel.

3. The information on behavior of concrete under cyclic loading in triaxial stress field is at best fragmentary and inadequate.

4. Inspection of the vessel during service for incipient failure is very difficult.

Conclusions. The prestressed-concrete pressure vessel concept will permit with reasonable confidence the construction of pressure vessels with pressures less than 5,000 psi and large enough for testing assembled fleet submarines. This pressure vessel concept is at the present time not applicable with currently commercially available Portland cements to the construction of the 10-foot-diameter, 10,000-psi pressure vessel. If cements with compressive strength in excess of 15,000 psi become commercially available prestressed-concrete pressure vessels should be considered for such an application.

Recommendations. The prestressed-concrete pressure vessel should not be considered for the immediate construction of the 10,000-psi, 10-foot (internal diameter) pressure vessel. If requirements arise for construction of very large (10 feet < diameter < 100 feet) vessels with less than 5,000-psi pressure requirements, the prestressed-concrete pressure vessel concept should receive first consideration. In the meantime, experimental studies are recommended for development of concrete pressure vessel technology to meet such requirements.

Glass-Fiber—Epoxy Laminate

Although glass-fiber—epoxy laminated internal pressure vessels have been produced by industry for many years, the proposed NCEL 10-foot-diameter pressure vessel presents severe structural demands that have not been imposed on glass-fiber—epoxy lamination technology. The fact that the proposed pressure vessel must safely contain 10,000 psi of hydrostatic pressure for long periods of time, must be able to withstand full-range pressure cycling for at least 20,000 cycles, and must permit the utilization of the whole internal volume of the vessel, puts the NCEL vessel design in a completely different class from that for missile air bottles or hydraulic accumulators.

The containment of hydrostatic pressure for long periods of time necessitates derating the high short-term tensile strength glass fibers to such an extent that their original advantage of possessing high tensile strength is largely lost. The effect of cycling on the strength of the fibers makes it further mandatory to derate the short-term tensile strength of the fibers. When both of these effects are taken into account, it can be postulated that the original +100 kpsi short-term tensile strength of the glass-fiber—epoxy laminate has been derated to 30 kpsi. At this low tensile strength, the laminate is not competitive with steels available on the market for pressure vessel construction, whose tensile strength under identical load conditions is at least 2 or possibly 3 times as high.

The utilization of the whole internal volume of the pressure vessel requires that one end of the vessel be removable for insertion of specimens to be tested. It does not suffice for this application to have a manhole with a diameter less than that of the vessel itself. Because of this, it is impossible to rely on glass-fiber—epoxy laminate alone to keep a metallic flange attached to the body of the vessel, as otherwise one would have to depend on shear forces between the windings and the flange skirt. To circumvent this difficulty, either an external yoke, or an inner steel liner, would have to be used to which the closure mounting flange would be welded. This liner would carry all the axial thrust on the contained hydrostatic pressure.

From the fabrication viewpoint, such a vessel presents quite a few problems. The thick inner liner cannot be made from one thickness of steel plate, but instead must be made up of many layers, further complicating the fabrication process. Winding glass-fiber preimpregnated tape does not present any special problems for the 10-foot-diameter vessel, but its curing in all probability will because of the unusually thick wall.

For reliability, this method of constructing pressure vessels leaves a lot to be desired. Since the strength of the vessel is derived primarily from a close interaction between the stresses in the liner and those in the overwrap, any discrepancy between the design values of strain in one or the other drastically decreases the pressure-containing capability of such a vessel. When one considers that in a multilayer lining (1) some layers are already in compression while others are in tension, and (2) that the amount of prestress to be expected from very heavy overwrap is not a precisely predictable quantity, it must be concluded that the interaction between the strains in the multilayer liner and the overwrap will be unpredictable.

The cost of a steel—fiber glass laminate vessel has been estimated to be in the $5 to $10 per pound range. The rather high cost of such a construction can be traced to the fact that there are two different fabrication processes involved, each one of them requiring a different fabricator. Each fabricator's profits, overhead, and transportation charges will make such a tank more expensive than it would be if only one fabricator was involved. Furthermore, quality glass-fiber—epoxy laminate is an expensive material, justifiable only where rigidity or weight reduction is desirable. When to the already high cost is added the premium that the fabricator of the overwrap will demand to cover uncertainties of the process when applied to a large vessel, the price of a pressure vessel constructed in this manner probably becomes uncompetitive with other fabrication processes.

The composite vessel consisting of a compressed steel liner with a pretensioned glass-fiber—epoxy laminate overwrap (Figure A-7) can be fabricated today if modifications are made to existing glass-fiber wrapping and curing facilities. The 10-foot internal diameter is already pushing existing facilities to the limit, and if there were a requirement for a 40-foot-diameter vessel, it would necessitate the erection of new fabrication facilities located in a place from where the vessel could be transported by ship to its location in some seashore installation.

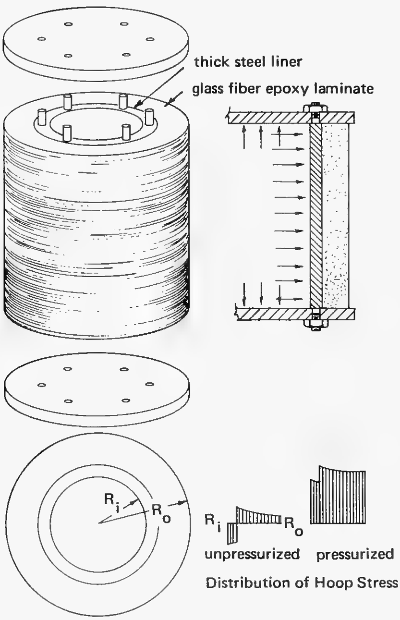
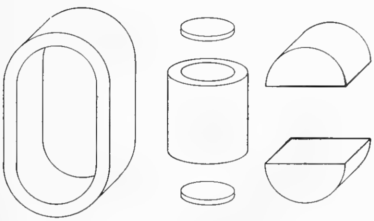


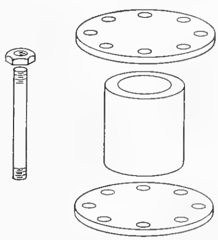
Figure A-7. Pretensioned-glass-fiber—epoxy-laminate concept of pressure vessel construction.

Conclusions. The fabrication technique employing a compressed steel liner and a pretensioned glass-fiber—epoxy laminate overwrap can produce a 10,000-psi internal working pressure vessel of 10-foot internal diameter and 20-foot length. Its low reliability and high cost place it at a disadvantage in comparison to a pressure vessel of equal internal dimensions and pressure capability fabricated by the multilayer or stacked-ring process. The cost of the composite pressure vessel is estimated to be 3 to 5 times higher than for a multilayer vessel.

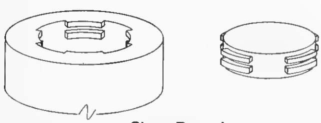
Recommendations. It is not recommended that this type of fabrication be considered at the present time for the proposed NCEL vessel of 10-foot internal diameter and 10,000-psi operating pressure.



Yoke Restraint



Tie-Rod Restraint



Shear Restraint

Figure A-8. Typical end-closure restraints.

END-CLOSURE RESTRAINT SYSTEMS

Restraints

The following criteria apply to the design of end-closure restraint systems:

1. The closure must accommodate the forces exerted by the end caps of a cylindrical vessel.

2. A pressure-tight seal must be incorporated.

3. Comparatively simple and rapid closure or opening of the vessel must be possible.

4. Penetrations through the closures must be provided for transmission of electric cables and hydraulic lines to the vessel's interior during the tests.

Three different end-closure restraint systems are currently considered applicable to the deep-ocean simulation vessels. The three different systems are (Figure A-8):

1. Continuous- or interrupted-thread and shear-block systems
2. Continuous external-yoke system
3. Tie-rod system

Of these three end-closure restraint systems, the threaded and shear-block restraint systems are the most limited in terms of internal pressure and size because of the small shear surface engagement in the end flange. The continuous yoke will operate at the highest pressure limitation, while the tie-rod system occupies a middle position with respect to pressure limits.

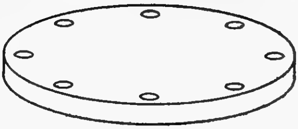
The three different end-closure restraint systems provide different degrees of accessibility to the vessel interior via feedthroughs in the end closures. The threaded and shear-block restraint systems provide maximum accessibility to the end closure for installation of feedthrough, while the continuous-yoke closure provides minimum or complete absence of accessibility. Here again the tie-rod restraint system is midway between the two others. It provides less accessibility to the end closures than the threaded and shear-block system, but more than the continuous-yoke system.

The end-closure restraint systems also vary in the ease of opening and closing the vessel at the beginning and end of each test program. The continuous-yoke system is here the most cumbersome and requires a very expensive and elaborate opening and closing mechanism to perform a reasonably speedy opening or closing operation. Threaded and shear-block restraint systems can be easily mechanized, resulting in very fast opening and closing operations. The tie-rod system is less cumbersome than the continuous yoke, but still more so than the threaded and shear-block systems. It has the potential, however, of resulting in an efficient system if an R&D effort is devoted to it.

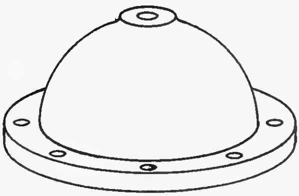
Conclusions. Tie-rod and continuous-yoke restraint systems are superior to interrupted-threaded and shear-block systems for 10-foot-diameter pressure vessels of 10,000-psi pressure service because the small shear surfaces of the latter make them inadequate for high pressure.

Recommendations. It is recommended that the tie-rod end-closure restraint system be investigated further as there is less known about it than the continuous-yoke system. It promises to be more efficient in operation than the continuous-yoke system, if a successful design is found for it.

Closure Shapes



Flat End Closure



Hemispherical End Closure

Figure A-9. Typical end closures.

End closures may be flat or hemispherical (Figure A-9). Flat end closures are more economical to fabricate than the spherical closures. However, because of the severe bending moments that are generated in flat closures by hydrostatic pressure when they are restrained by threaded, shear-block, or tie-rod restraints, flat closures are limited to diameters of less than 3 feet in the pressure range above 5,000 psi. For higher pressures and larger diameters, they become rapidly unwieldly and uneconomical, as forging thicknesses in excess of several feet become necessary to withstand the high bending moments.

The hemispherical end closures require much less steel than the flat closures because of more favorable stress distribution in them, but the saving in steel is offset here by the cost of forging and machining an intricate shape. There are indications, however, that a technique for fabricating layered hemispherical end closures may be developed that instead of expensive forgings utilizes formed plate segments welded into a continuous structure. Because of this new development, the current pressure and diameter limitation on hemispherical end closures may be eliminated.

Large, flat end closures are feasible for high internal hydrostatic pressures only if a continuous-yoke end-closure restraint system is used on the vessel. In such a case, a bearing block under the continuous yoke at the end of the vessel restrains the flat closure from flexing, and only a nominal thickness is required for the closure to retain the necessary seals around its circumference.

Conclusion. It appears that the hemispherical end closures are more desirable for large diameters and internal pressures than flat ones unless the continuous-yoke end-closure restraint system is used on the vessel.

Recommendation. There is no requirement for thick, flat end closures for large vessels, since with the continuous-yoke restraint system, a thin end closure suffices. Investigations into economical end closures for large-diameter vessels need to be concentrated on hemispherical shapes, particularly of layered, welded construction.

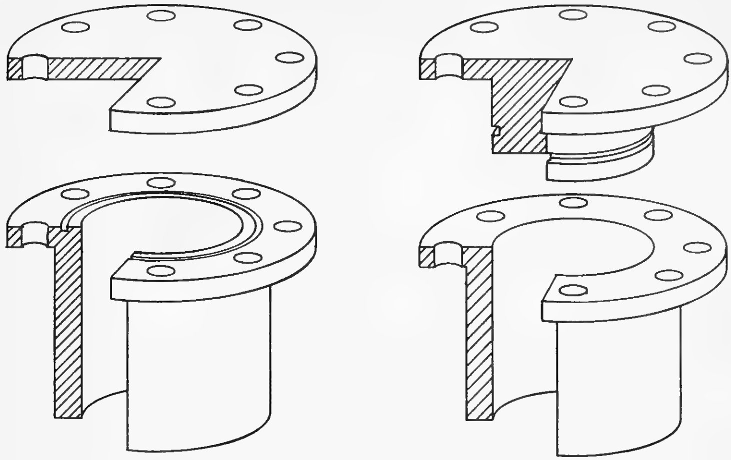
Seals

High-pressure seals should be:

1. Simple to assemble
2. Self-energizing (sealing ability increases with pressure)
3. Unlikely to jam
4. Easy to install

Although a host of proven seal designs is commercially available, none of them are ideally suited to large-diameter vessels for high internal pressure. Their shortcomings lie principally in their requirement for either a high precompression or fine dimensional tolerances between seal surfaces for proper sealing. Those seals that can tolerate rough sealing surfaces and loose dimensional tolerances on the vessel flange require such a high precompression to seal effectively at 10,000-psi hydrostatic pressure that they are inapplicable to high pressure vessels of 10-foot diameter. Almost all the axial compression seals (Figure A-10) fall in this category. Those seals, on the other hand, that do not require axial precompression to seal properly at 10,000-psi hydrostatic pressure require such fine finish and dimensional tolerances on the internal diameter of the vessel that it cannot be satisfied with ordinary machining tolerances for cylinder openings of 10 feet. Only by premium surface finishing techniques and meticulous attention to diameter tolerances on the internal surface of the vessel can those seals be made to work successfully at 10,000 psi. Most radial compression seals fall into this category.

Conclusions. It appears that no currently available sealing system is ideally suited for 10-foot-diameter vessels with 10,000-psi hydrostatic pressure where repeated removal of end closure is required. However, of the two classes of seals available, the radial compression seals are more applicable. It is not feasible to mechanically apply sufficient pretensioning to the end-closure restraint system to insure sufficient compression of axial seals to seal at 10,000-psi operational pressure unless the very cumbersome thermal shrink technique is applied.



Axial Seal

Radial Seal

Figure A-10. Typical end-closure seals.

Recommendation. It is recommended that experimental investigations be initiated for development of an improved self-energizing radial seal suited for 10-foot-diameter vessels and 10,000-psi operational pressure.

IMPLOSION LOADING OF PRESSURE VESSELS

Past experience at laboratories equipped with internal pressure test vessels* has shown that when implosion of models occurs, a severe shock wave is generated which causes the test vessel to be moved laterally or vertically, damaging in the process auxiliary equipment attached rigidly to the pressure chamber. Although there is no record of a pressure vessel rupturing because of an implosion inside of it, this can be attributed in a large extent to the high safety factor of 4 used under the ASME code, the very ductile materials employed, and the low hydrostatic pressures involved in the testing. With the present trend in test vessel design aimed at larger vessels, higher working pressures, materials with higher yield-points but lower ductility, and reduced safety factors, it is only a matter of time before a catastrophic failure of a vessel will occur because of an imploding test object.

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* For example, the Southwest Research Institute, San Antonio, Texas; the Ordnance Research Laboratory, State College, Pennsylvania; the Navy Ordnance Laboratory, White Oaks, Maryland; the Navy Underwater Test Station, Newport, Rhode Island; the David Taylor Model Basin, Carderock, Maryland.

Conclusion

To forestall this type of failure, information must be made available to the vessel designers, and safe vessel operation techniques must be taught to the pressure vessel operators. Such information to be of real value as a design guide must constitute a theoretically postulated and experimentally verified series of equations.

Recommendation

In order to obtain the needed information to design vessels resistant to implosion damage, and to insure the safe operation of vessels already in existence, a program must be initiated to investigate the effect of implosions on pressure vessel life. Such a program should consist of experimental and analytical studies running concurrently. Only from the continual cross-referencing of experimental and analytical work will biable design criteria emerge from such a program.

SELECTION OF SAFETY FACTOR

The safety factor for pressure-vessel operation generally is based on four considerations. These are:

1. Foreseeable inaccuracies in the stress analysis during design on the vessel

2. Predictable discrepancies between the properties of the material samples, and the actual properties of the material in the vessel

3. Unforeseen loads that will act on the vessel while under maximum working pressure

4. Number of pressure cycles to which vessel will be subjected during its life

In the proposed pressure test facility, only items 2, 3, and 4 are decisive, if a vessel construction concept with known design criteria is chosen. The discrepancy between the properties of the specified material and those actually found in the vessel structure will be very large since the construction of the proposed vessel requires that very thick forgings be employed for the closures and flanges. The actual magnitude of discrepancy is not known since very little is known about this subject for very heavy forgings. The same may be said of our knowledge in the generation of shock loads in pressure vessels by implosion of test models. That large shock waves are triggered by implosion is well known, but how high the dynamic stresses in the vessel actually are is only a calculated guess. The fact that those dynamic stresses also fatigue the vessel material *only* further complicates the matter. This fatigue effect, when added to the fatigue caused by static pressure cycling, makes it necessary to reduce considerably the safe stress level that can be tolerated by the vessel material during a projected 20-year lifetime.

Conclusion

A safety factor of 2 based on yield of the material is considered inadequate. A safety factor of at least 3, and preferably 4, should be used. The safety factor should be based on yield of the vessel's material under static pressure loading to insure not only a statically safe vessel but also a long cyclic life at pressures equal to static pressure.

Recommendation

A minimum safety factor of 3, and preferably 4, based on the yield strength of the material, should be applied in the design of the proposed pressure vessel.

OVERALL CONCLUSIONS AND RECOMMENDATIONS

Conclusions

1. The group concurs that at the present time the stacked-ring or multilayer construction concepts are the most feasible concepts for the construction of a 10-foot-diameter deep-ocean simulation vessel with a 10,000-psi operating pressure. Of the two, the stacked-ring concept possesses the added advantages of in-situ assembly, interchangeability and replaceability of individual construction modules, and absence of welds.

2. The most promising closure system for the stacked-ring concept from the viewpoint of accessibility to penetrations, speed of operation, ease of manufacture, and cost, appears to be composed of tie rods and hemispherical end closures. Although it is a promising system, very little design experience is available for its design.

3. The projected types of tests that will take place in the vessel and the impact on the national deep-submergence effort that the loss of such a vessel would create, make a safety factor of 2 inadequate. A minimum safety factor of 3, or preferably 4, based on yield of material, should be utilized.

4. There are no quantitative or analytical data that could be applied to the design of the pressure vessel facilities to eliminate the possibility of vessel failure because of internal implosion. Qualitative observations of implosions, however, have shown that the shock waves unleashed by implosions are of such magnitude that they must be considered in safe vessel design.

Recommendations

1. Conceptual designs of the stacked-ring and multilayer vessels should be prepared and quotations on their fabrication should be solicited. The stacked-ring and multilayer vessel concepts are in the opinion of the study group the leading candidates at the present time for the construction of a 10-foot-diameter, 10,000-psi vessel.

2. The segmented and the stacked-ring vessel concepts should be further explored and refined, as they have great potential for construction of pressure vessels with diameters and pressures in excess of 10 feet and 10,000 psi, respectively.

3. An exploratory study of the implosion effects inside pressure vessels should be immediately initiated. The analytical and experimental data gathered by such study will be of importance in the design of future pressure vessels.

Appendix B

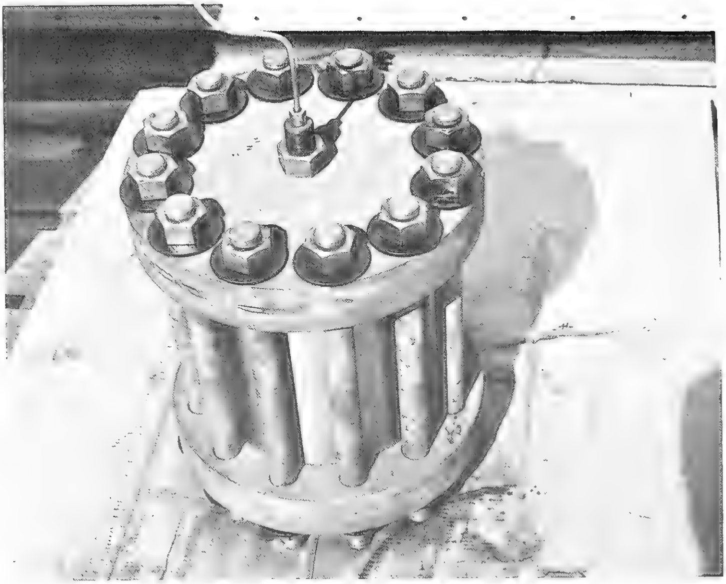
EXPERIMENTAL EVALUATION OF RADIAL END-CLOSURE SEALS

BACKGROUND

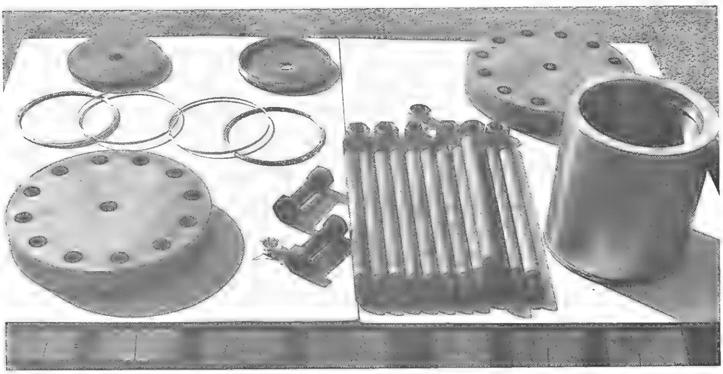
For successful operation each pressure vessel requires seals at joints between removable pressure vessel components. Since seals at best are potential sources of leakage, a concentrated effort is generally made to minimize their number. Such a minimum is represented by a single O-ring in the upper removable closure. No way has been found to eliminate it from a pressure vessel because access to the interior is mandatory for the insertion of test specimens. In the case of stacked-ring or segmented modular design, in which both the upper and lower head closures are removable, the irreducible minimum of seals is two O-rings, one in the top and one in the bottom closure, sealing the joint between the closures and the walls of the vessel. Naturally, more than two O-rings may be and generally are used even with such a design. The additional O-ring seals however, are only a convenient substitute for some other type of seal, for example, threaded pipe fittings.

EXPERIMENTAL DESIGN

To evaluate some of the large variety of existing, or feasible joint seals for high pressure vessels, a small pressure vessel was designed in which seals of varying design could be tested between the closures and vessel body (Figures B-1a and B-1b). In order to simulate the problems that will be encountered in the operation of the full-sized stacked-ring or segmented pressure vessel, the seal test vessel was also designed with free-floating end closures. In this design, the end closures were permitted some vertical motion when internal hydrostatic pressure was applied. In the seal test vessel, the end closures were affixed to the pressure vessel by means of tie rods, which extended only a known and limited amount when the interior of the vessel was pressurized. Although this vertical movement of the end closures was very small (on the order of a 1/32 of an inch at pressures of 10,000 psi), it was sufficient for the end closures to be free floating. The fact that the end closure was free floating made it impossible to utilize with it any of the seals associated with nonfloating end closures. Such seals generally rely on the wedging action between the end closure and the vessel body to squeeze the seal so that it forms a watertight barrier. With free-floating closures, seals must be employed that do not lose their sealing action because of the upward movement of the end closure under load.



(a) Assembled.



(b) Disassembled.

Figure B-1. Pressure vessel for evaluation of different radial seals at 10,000 psi of internal hydrostatic pressure.

The seals associated with the free-floating end closures generally rely for their sealing action on radial compression of the seal body between the end closure and the interior of the vessel body. The design ingenuity of such seals lies primarily in the provision for sealing the increasingly wider gap between the end closure and the interior of the vessel as the vessel expands radially under the internal hydrostatic pressure. Without provision for this gap during the pressurization of the vessel, the seal will extrude into the gap and out of the vessel, losing all of its sealing ability. For this discussion, it is obvious that an ordinary O-ring under radial compression would retain its sealing ability under very low hydrostatic pressure only, as the presence of a gap of several thousands of an inch would make it impossible to retain pressures of even 2,000-to-3,000-psi magnitude. Obviously, other approaches to the seal design besides an ordinary O-ring in radial compression had to be sought and evaluated.

The seal designs that were evaluated in the seal-test vessel (Figure B-2) were the wedge ring seal, O-ring with continuous antiextrusion wedge ring, O-ring with a split antiextrusion wedge ring, and twin O-ring seal in a self-energized elastic follower ring (Figure B-3). Each of these seal designs was thought to be promising and worthy of investigation; the most desirable one was to be selected on the basis of its performance under hydrostatic pressure in the vessel.

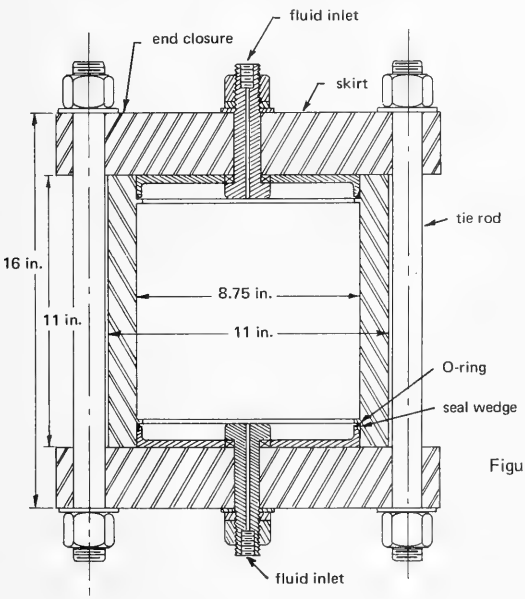
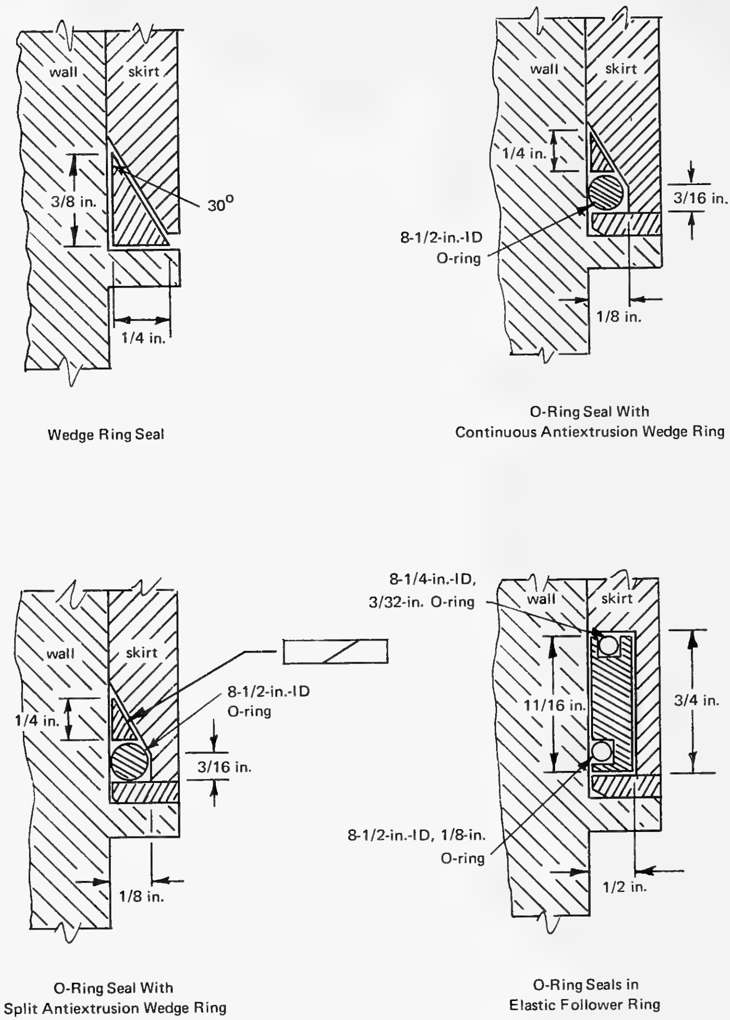


Figure B-2. Location of seals in the pressure vessel during their evaluation under hydrostatic pressure.



Wedge Ring Seal

O-Ring Seal With
Continuous Antiextrusion Wedge Ring

O-Ring Seal With
Split Antiextrusion Wedge Ring

O-Ring Seals in
Elastic Follower Ring

Figure B-3. Seals selected for evaluation in the 10,000-psi pressure vessel.

EXPERIMENTAL EVALUATION

Wedge Ring Seal

The seal was fabricated for the experimental evaluation program from nylon and from brass. Its initial sealing depends on the wedging of the seal between the end closure and the interior of the pressure vessel. This wedging is accomplished by the weight of the end closure pressing upon the wedge, which is restrained from moving by a lip protruding from the interior wall of the vessel. Once the initial sealing is accomplished, hydrostatic pressure within the vessel will tend to wedge the seal in further by pushing axially and radially upon it. To make sure that the hydrostatic pressure acts on the wedge along the vertical axis of the vessel, small serrations were provided on the base of the wedge resting on the lip protruding from the wall of the pressure vessel.

The experimental evaluation of the wedge ring seal has shown that it is not very desirable for end closures that must be closed and opened often. Its shortcomings are serious. First of all, it often fails to seal at low pressures before hydrostatic pressure wedges it between the end-closure skirt and the internal surface of the pressure vessel wall. Thus, to make the seal perform at zero pressure, some force other than hydrostatic must wedge it between the end-closure skirt and the vessel's interior surface. In the experimental evaluation, this force was provided by the weight of the whole end closure pressing against the wedge that rests on the circumferential ledge around the vessel's circumference. In addition to the problems associated with sealing at low pressures, the seal does not perform well at pressures above 5,000 psi. At about that pressure, the plastic seal becomes forced completely into the clearance between the end-closure skirt and the vessel wall; when the internal pressure approaches 10,000 psi, it is forced completely through with an explosive release of pressure. The high-pressure capability of the wedge ring can be increased by substituting metal for plastic. With the metal seal, there is almost no low-pressure sealing capability, as it is very difficult to apply enough force to the metallic wedge at zero pressure to make it seal.

O-Ring Seal With Continuous Antiextrusion Wedge Ring

A marked improvement over the simple plastic wedge seal is a wedge seal combined with an O-ring (Figure B-4). The O-ring acts as a seal at low pressures (0 to 1,000 psi) since it is radially compressed even at zero hydrostatic pressure by the end-closure skirt and the vessel's interior wall. As the pressure rises inside the vessel, the O-ring causes the wedge to seat itself tight and to keep the O-ring from extruding into the radial clearance between the end-closure skirt and the vessel wall. However, when the internal pressure approaches 10,000 psi, this plastic wedge, like the preceding seal type, plastically extrudes and releases compressed water (Figure B-5).

This seal represents a marked improvement over the preceding seal type, as with this type no sealing difficulties are encountered at low pressures, and it is only in the 5,000-to-10,000-psi range that this seal fails by extruding. Both these seal arrangements have an unlimited capability to follow axial displacement of the end closure, but only very limited capability to follow the vessel's radial dilation. Both seal arrangements should have a plastic rather than a metallic continuous wedge ring as otherwise the seals will not follow the radially dilating vessel wall with sufficient compliance to assure a continuous seal.

O-Ring Seal With Split Antiextrusion Wedge Ring

This seal arrangement is basically the same as that of the preceding seal except that a split metallic ring has been substituted for the plastic continuous ring. With this arrangement, the O-ring seals well at zero and low pressures, while at high pressures the metallic wedge ring is much more difficult to extrude than the plastic wedge ring described above. However, if the clearance between the end-closure skirt and the vessel wall became of the same magnitude as the width of the wedge, it would be forced into that space by the hydrostatic forces acting on the O-ring. Once the metallic wedge was lost into the space between the vessel wall and the end-closure skirt, it would cause the end closure to jam and might prevent the removal of the end closure.

This seal arrangement, like the preceding seal arrangements, can follow any axial displacement of the end closure. It has only limited ability to follow the radial dilation of the vessel, and the magnitude of radial dilation of the vessel that this seal can compensate for is determined by the width of the split metallic antiextrusion ring.

Although this seal arrangement has overcome the shortcoming of the first seal of not sealing properly at zero internal pressure, and the shortcoming of the second seal of not sealing at pressures in the 5,000-to-10,000-psi range, it had not overcome the single shortcoming common to all: incapability to compensate for large radial dilation of the vessel wall. Thus another seal arrangement was conceived with the objective to seal well at zero pressure, at high pressure of any magnitude, and to follow axial displacement of the end closure and any magnitude of radial dilation of the vessel wall. Furthermore, to make the seal installation simple and inexpensive, it was to utilize only commercially readily available O-rings and a minimum of custom machined parts.

Figure B-4. Radial O-ring seal with a plastic antiextrusion backup.

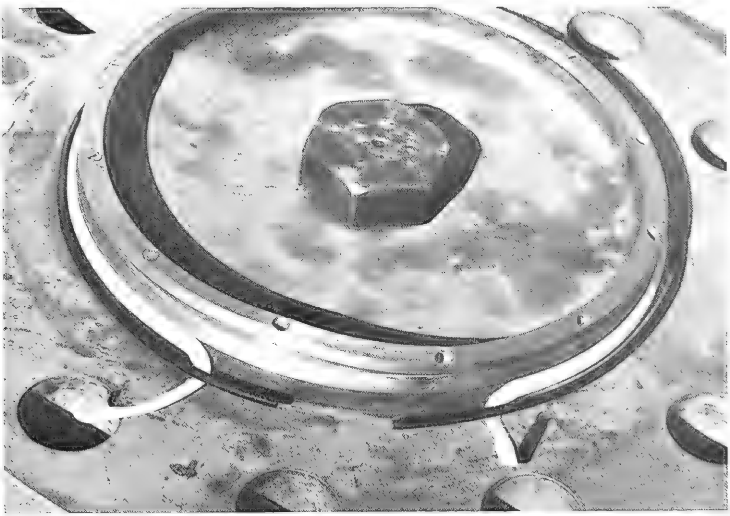


Figure B-5. Radial O-ring seal with a plastic antiextrusion backup after time-dependent creep failure at 10,000 psi.

Figure B-6. Self-energizing radial O-ring seal for high pressures in internally pressurized vessels.

O-Ring Seals in Elastic Follower Ring

The experimental evaluation of this seal arrangement has shown it to be markedly superior to all the other seal arrangements experimented with previously in this study. The superiority of this seal (Figure B-6) lies in its ability to seal out low and high pressures, as well as to follow the axial and radial dilation of the vessel without any loss in sealing ability. Its ability to accomplish all this lies in its use of hydrostatic pressure contained inside the pressure vessel to expand and translate the elastic follower ring so that it follows the radially dilating wall of the vessel and the axially displacing end closure. This self-energizing feature causes the seal to press harder against the end closure and wall as the pressure is raised. In this manner, it is assured that regardless of the magnitude of internal pressure or radial and axial displacement of vessel's interior surfaces, no extrusion will take place in O-rings even though they are soft elastomers.

Because of the self-energized elastic follower ring in which the O-rings are contained, no extrusion of the 70 shore-hardness O-rings took place even though the total radial clearance between the interior vessel wall and the end-closure skirt was more than 0.032 inch at 20,000 psi of internal hydrostatic pressure. When the internal pressure was released, the elastic follower ring returned to its original dimensions and no difficulty was encountered in removing the end closure. Upon examination of the O-rings, it was found that they were ready to be used again as a seal. The design and fabrication of the self-energized O-ring seal in the elastic follower ring is rather simple. These three elements are required:

(a) **Two O-rings.** One O-ring under radial and one under axial compression are required. The elastic follower ring must be so dimensioned that the O-rings are under sufficient compression at zero internal pressure to constitute a low-pressure radial and axial O-ring seal. The radially compressed O-ring must seal the inevitable small clearance between the vessel wall and the external radius of the elastic follower ring, while the axially compressed O-ring seals the clearance between the bottom of the vessel end closure and the top of the elastic follower ring. The radial O-ring is compressed at zero hydrostatic pressure by the close fit between the exterior surface of the elastic follower ring and the interior surface of the pressure vessel. The axial O-ring is compressed at zero pressure by bolts pushing a retainer ring against the elastic follower ring. When the pressure is raised inside the pressure vessel, it acts axially and radially upon the elastic follower causing it to push harder against the end closure and the cylinder, thus achieving zero clearance between the follower ring and the seal surfaces.

(b) **An elastic follower ring.** A ring sufficiently elastic to expand across the gap between the head and the vessel and subsequently to follow the radially dilating pressure vessel is required. For this application, the follower ring must be less stiff than the vessel wall whose dilating it is following. This is accomplished by making the follower ring either from material with a very low modulus of elasticity or by making it from the same material as the pressure vessel wall, but considerably thinner. Regardless of what material the follower ring is made, it must not yield during its radial dilation, or deform due to shearing stresses imposed on it while it is bridging the gap between the vessel end closure and the wall of the vessel. If either one occurred, the follower ring would have to be replaced after each pressurization, making this type seal uneconomical.

To provide sufficient radial and axial forces on the follower ring to maintain zero clearance between the ring and the seal surfaces on the end closure, the O-ring grooves (Figure B-7) must be machined at such locations in the follower ring that hydrostatic pressure causes radial and axial movement of the follower ring.

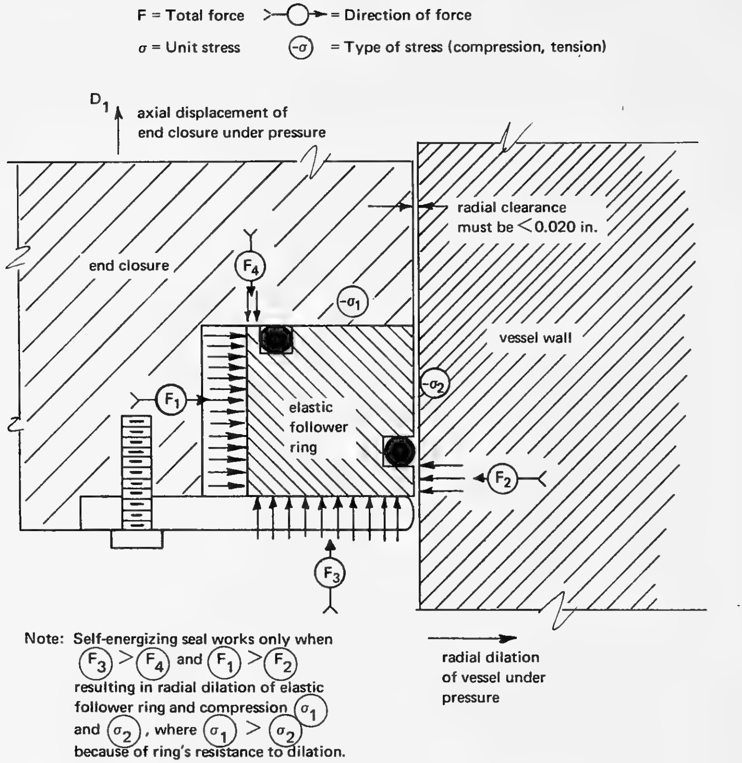
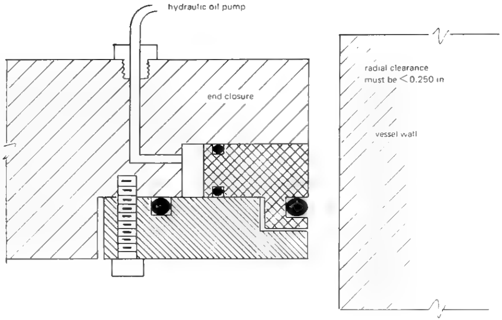
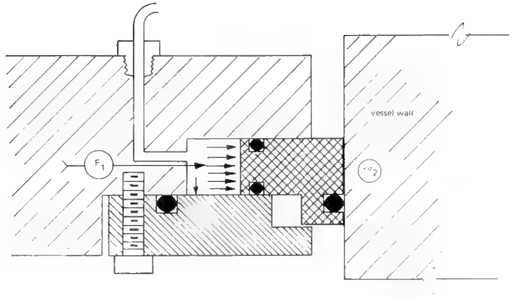


Figure B-7. Forces acting on the elastic follower ring containing the radial and axial O-rings.

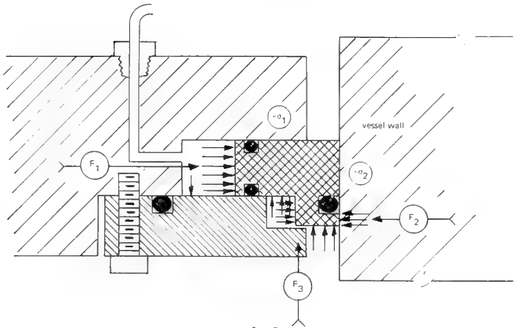
(c) **Radial precompression.** The only shortcoming of the self-energized radial seal is its requirement for sufficiently close (0.010 to < 0.020 inch) radial fit between the external radius of the follower ring and of the interior surface of the vessel to provide the initial compression of the radial O-ring so that it seals at low pressure and thus permits the self-energizing mechanism to function with increase in internal pressure. In its requirement for close radial fit, this seal is no different from the other seals investigated experimentally in this study. It appears, however, that modifying this design (Figure B-8) may permit greater clearance between the external radius of the follower ring and the vessel wall at zero pressure. Such a development would, of course, (1) make the fabrication



Step 1



Step 2



Step 3

Step 1 – $p_1 = p_2 = 0$. Pressures inside the vessel and in the hydraulic circuit are zero

Step 2 – $p_1 > 0$, but $p_2 = 0$. While pressure inside the vessel is still zero, the pressure inside the hydraulic circuit has been raised by an externally located hydraulic pump until the elastic follower ring has dilated radially to such an extent that it is contacting the vessel wall

Step 3 – $p_1 > 0$ and $p_2 > 0$, $F_1 > F_2$. The interior of the vessel is pressurized with an externally located separate seawater pump to the desired operational pressure

Step 4 – The interior of the vessel is depressurized at the conclusion of the test. When the pressure inside the vessel drops to zero, the hydraulic circuit is depressurized also, and the elastic follower ring radially contracts, breaking contact with the vessel wall.

Notes

p_2 = pressure inside the vessel

p_1 = pressure inside the hydraulic circuit

F = total force resulting from application of hydrostatic pressure on a given surface

—◯→ = direction of resulting force

σ = unit stress

($-\sigma$) = type of stress (compressive, tensile, etc.)

Figure B-8. Self-energizing radial O-ring seal with external pressure assist

of large-diameter vessels more economical, as tight machining tolerances of the radial seal surfaces on the follower ring and the interior of the vessel could be relaxed, and (2) facilitate opening and closing the end closure, since inserting the end closure with the elastic follower into the vessel would require less care.

CONCLUSIONS

The self-energizing radial seal from all the seals evaluated appears to be the most desirable seal from technological and operational viewpoint for containment of pressures in excess of 10,000 psi in vessels with diameters in excess of 120 inches.

RECOMMENDATIONS

The proposed modification of the self-energizing radial seal should be experimentally evaluated for possible incorporation into deep-ocean simulation chambers currently in construction or design stages. This modification may result in appreciable economies in fabrication and operation of large-diameter pressure vessels for containment of high pressures.

Appendix C

PHOTOELASTIC INVESTIGATION OF STRESS CONCENTRATIONS

INTRODUCTION

Since both the stacked-ring and the segmented-wall pressure vessel models failed at lower internal hydrostatic pressures than could be predicted by the nominal stress magnitude, it appeared desirable to investigate the magnitude of stress concentrations at locations where failures were initiated. To accomplish this, the magnitude of stresses and stress concentrations in these vessels had to be determined before meaningful recommendations could be formulated for redesigning the vessels. Two approaches were available: the analytical and the experimental. Although these approaches complement each other, the limited funding and time available for the determination of stress concentrations in the stacked-ring and structural-module (segmented-wall) vessels made two simultaneous investigations unfeasible. The experimental approach was chosen because it was felt that with the limited time and funding allowed for the hydrostatic pressure vessel study, experimentation would yield more exploratory engineering design data than would analysis.

BACKGROUND INFORMATION

Although many different methods are available for the measurement of strains in a structure with stress raisers, only one of them lends itself easily to quantitative interpretation. This method is the photoelastic strain-measuring technique.* Ideally, a three-dimensional photoelastic frozen-strain technique supplies the most detailed and accurate strain information for every part of a stressed structure. It is a cumbersome and expensive method requiring for its success not only an epoxy model of the vessel but also an oven for heating the vessel while it is internally pressurized. In addition, extremely fine slices must be taken out of the epoxy model after the strains have been frozen in; these slices are, after precision machining to a uniform thickness and polishing for uniform light transmissivity, photoelastically investigated under transmitted polarized light. The advantage of the frozen strain technique is, of course, its ability to present visually the distribution magnitude and orientation of strains

* For brevity, the materials, coatings, and techniques are all described as "photoelastic."

in every part of the structure, no matter where this part may be located on the structure, or how complex it may be. Because of the expensive model and equipment and the time required for machining of slices, it was decided instead to apply the two-dimensional photoelastic strain-investigation technique.

The two-dimensional photoelastic strain-investigation technique requires either photoelastic coatings on structural members under investigation, or biaxially loaded transparent structural members with surface boundaries at right angles to the polarized light source. In the first case, polarized light is reflected from the backside of the photoelastic coating, while in the second, light is transmitted through the structural member. In both cases, a camera records the number and location of photoelastic fringes in the photoelastically active material while it is stressed. The only severe limitation on the use of two-dimensional photoelastic technique is that it only provides information on the biaxial strains located in a plane perpendicular to the path of polarized light. This technique is incapable of detecting strains parallel to the light path and thus is somewhat limited in the evaluation of three-dimensional strains in a pressure vessel. It was felt, however, that by placement of photoelastic coatings on two-dimensional models of three-dimensional structural parts suspected of having stress concentrations, enough information could be obtained to alert the design engineer to the magnitude of stress concentrations that may be encountered in the vessel structure.

EXPERIMENTAL PROCEDURE

The two-dimensional photoelastic strain investigations were all conducted with reflected polarized light, but two kinds of test models were employed. The models consisted either of an epoxy-coated metallic shape, representing the cross section of the actual part, or of the actual structural part made out of epoxy painted on one side with a reflecting paint. The decision on whether to use the coatings on metallic models or actual structural parts made out of epoxy for investigation of strains in a particular part of the vessel structure was based primarily on the ease with which the particular structural part could be loaded sufficiently to generate a high number of photoelastic fringes to make the photoelastic analysis more reliable.

The structural parts of the vessel that lent themselves to the two-dimensional modeling without much trouble were the end-closure tie rods and flanges. For the strain investigation of the tie rods, special two-dimensional metallic models were made which represented the longitudinal cross section of the tie rod (Figure C-1). Since many different tie-rod heads can be used in pressure vessel fabrication, several kinds of heads were investigated besides the one actually used in the acrylic pressure vessel. After fabrication of the cross-sectional metallic models of different tie-rod heads, they were coated with photoelastic epoxy and subjected to tensile load tests (Figures C-2 through C-4) in a standard tensile load machine. The machine utilized a specially designed load applicator and the distribution of photoelastic fringes was recorded. Since the strains in the tie rods of the pressure vessel are uniaxial, it was felt that testing models (representing their longitudinal cross section and subjected to axial tensile loads) would adequately simulate the loading in the full-sized structural part.

For the investigation of strains in the closure flange, a metallic cross-sectional model was made. Since the flanges on the closures are subjected to three-dimensional strains when the interior of the vessel is pressurized, it is impossible to measure all of their triaxial components with simple biaxial cross-sectional models. However, it is known which load components generate the largest concentration of strains in the closure flange. Thus, cross-sectional models can be designed to show under biaxial loading the largest strain concentrations present in the actual closure flange.

To measure the strains in the meridional plane of the flange caused by both the shear, membrane, and flexural stresses in the closure under hydrostatic loading, a cross-sectional model was made that represented the cross section in the axial plane of the whole vessel closure (Figures C-5 and C-6). To load this cross-sectional model of the closure to simulate the hydrostatic loading imposed on the end closure by the fluid inside the pressure vessel, a hydrostatic loading jig was devised. This jig, utilizing hydraulic pressure acting on a laterally constrained O-ring mounted in a plate contoured to the internal radius of the vessel's hemispherical closure, simulated very effectively the hydrostatic loading acting on the actual vessel closure. The closure cross-sectional model was coated with epoxy prior to investigation under polarized light, since it was made of metal. During the application of simulated hydrostatic pressure with the hydraulic load application jig, photographs were taken of the photoelastic fringes at 50-psi intervals (Figure C-7). It is to be understood, however, that although the cross-sectional model gave a good representation of strains and strain concentrations in the closure adjacent to the flange caused by shear, flexure, and axial stresses in it, the model did not help in the determination of strain concentrations caused by hoop stresses in the flange and in the closure wall adjacent to it. These strain concentrations are caused by the abrupt change in the cross section of the closure wall. Since determination of the magnitude of this strain concentration would involve the use of a three-dimensional model for frozen photoelastic strain technique, this investigation was omitted. It was felt, however, that the strain concentrations caused by the shear, axial, and flexural stresses in meridional plane are much more severe than the one caused by hoop stresses.

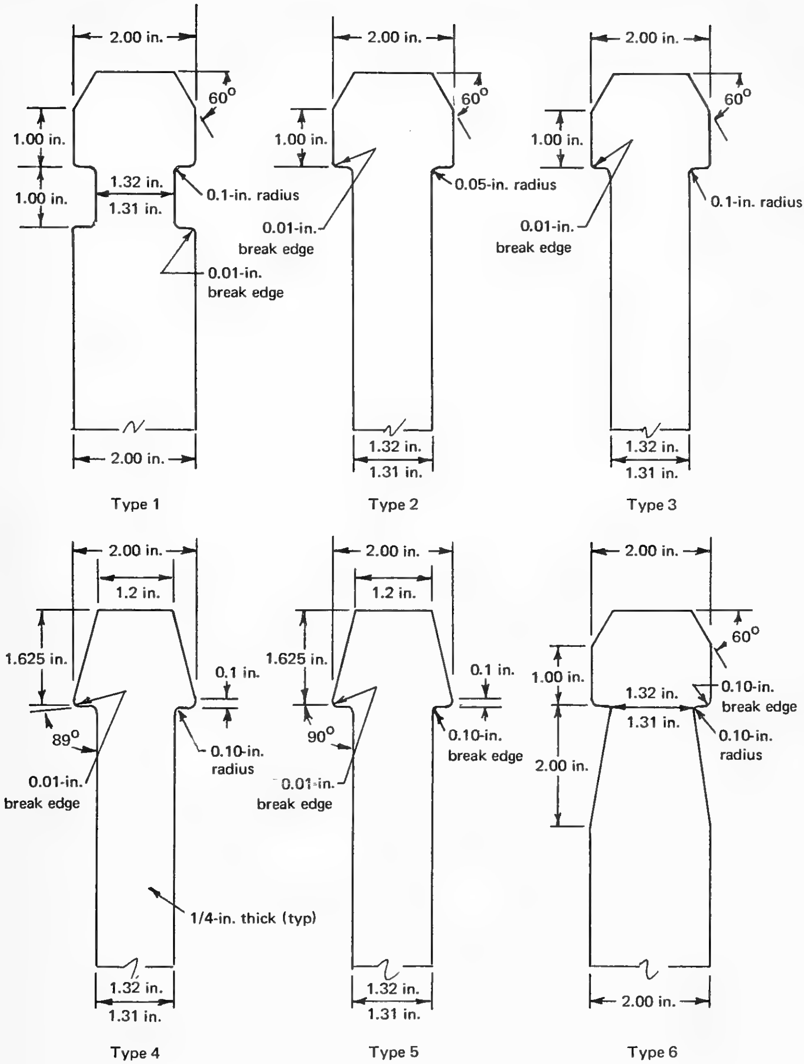


Figure C-1. Two-dimensional models of tie-rod heads for photoelastic investigation of stress concentrations.

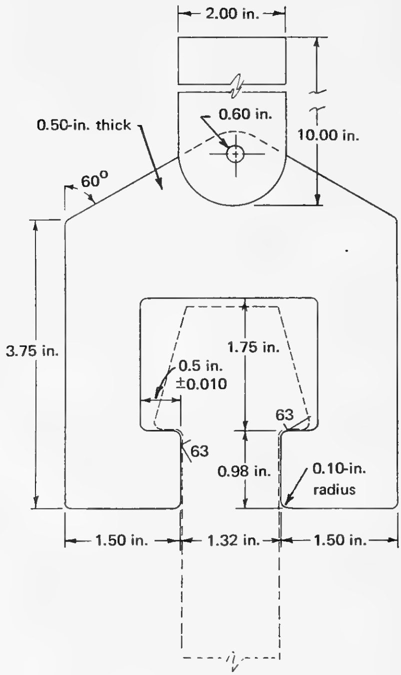


Figure C-2. Tensile load applicator for two-dimensional tie-rod head models investigated photoelastically for stress concentrations.

For this reason, no further efforts were made to determine the strain concentrations in the closure wall and flange caused by hoop stresses in the flange and adjacent closure wall.

The reflected light technique was also employed to measure the strains and strain concentrations in the segmented-vessel wall laminae, but instead of preparing a cross-sectional model for the determination of strains, scale segmented-vessel wall laminae were used (Figure C-8). To simulate the hydrostatic loading on a typical segmented-vessel wall laminae, several of them were assembled into a ring which was then placed over a hydraulic loading jig, similar to the one used in testing the vessel head flange (Figures C-9 and C-10). The modules in the top layer of the ring were made from epoxy sheets with a silvered back surface, and reflected circularly polarized light was used to determine the number and distribution of photoelastic fringes (Figure C-11). To observe the stress concentration better around the shear-pin holes, the nuts were removed for the test at locations where the fringes were to be photographed. The laminae in the other layers of the ring were fabricated from acrylic resin, a more economical and workable material. Since the modulus of elasticity of epoxy is comparable to that of acrylic resin, the distribution and magnitude of strains in the epoxy and acrylic resin laminae were approximately the same. The photoelastic fringes in the segment laminae were photographed at 50-psi load-level intervals until failure of the photoelastic model took place at slightly more than 220 psi (Figures C-12 and C-13).

Although both the stacked-ring vessel and the segmented vessel are known to possess other structural components in which strain concentrations occur that could not be analytically explored, it was impossible to evaluate them experimentally by means of reflected polarized light because of lack of time and funding.

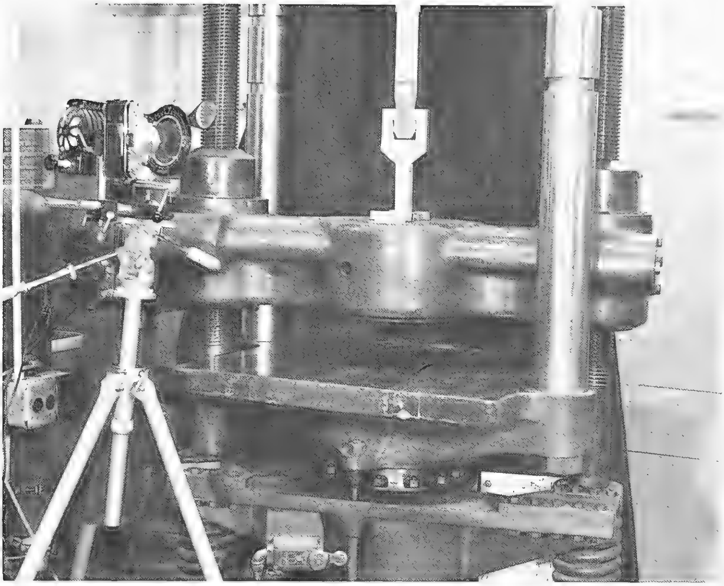


Figure C-3. Experimental setup for tensile testing of two-dimensional tie-rod head models.

FINDINGS

Tie-Rod Models

The exploratory analysis of the two-dimensional tie-rod models, coated with a photoelastically sensitive epoxy coating, indicates that the stress concentration (as compared to the average stress level that was observed in the tie rods) at the base of the tie-rod head was approximately 3 (model 2) based on the calculated nominal stresses at the smallest cross section of the tie rod. The stress concentrations in the other models representing feasible alternatives to the tie-rod head configuration used in the acrylic pressure vessel were 5 (model 1-2.1, model 3-2.0, model 4-3, model 5-2.5, and model 6-3.1). It appears that if the model 1 or 3 configuration had been substituted for the one used in this study, the stress raiser effect could have been substantially decreased.

Figure C-4. Typical birefringence in photoelastic coating on two-dimensional model of tie-rod head under a 3,000-pound tensile load.

End-Closure Model

The two-dimensional model of the end closure and end-closure flange when subjected to simulated hydrostatic pressure with the hydraulic loading jig indicated that a serious stress concentration does exist in the meridional plane of the end closure. The progress of the photoelastic fringes across the thickness of the model during loading indicates that the local stress concentration is caused primarily by flexure of the end closure at its flange. The magnitude of the stress concentration, based on the average membrane stress present in the model at locations distant from the stress raiser, is approximately 3.3 to 3.5.

Segmented-Wall Model

The testing of the segmented-wall laminae fabricated from photoelastically active epoxy showed that as previously predicted a serious stress concentration is generated by the presence of the stress raiser in the form of the shear pin holding the segmented-wall laminae together. Since the fit of the pins in the holes and the distance between holes in each individual segment laminae influence to a large degree the magnitude of stress concentrations both in the pin and in the segment laminae, the experimentally determined value of the stress concentration can be considered only a representative value. The magnitude of tensile stress concentration in the segment laminae around the shear-pin hole was found to be approximately 3.5, while the compressive stress concentration caused by the pin bearing against the edge of the hole was found to be approximately 6.5 in comparison to the nominal tensile stress in the narrowest cross section of the segment.

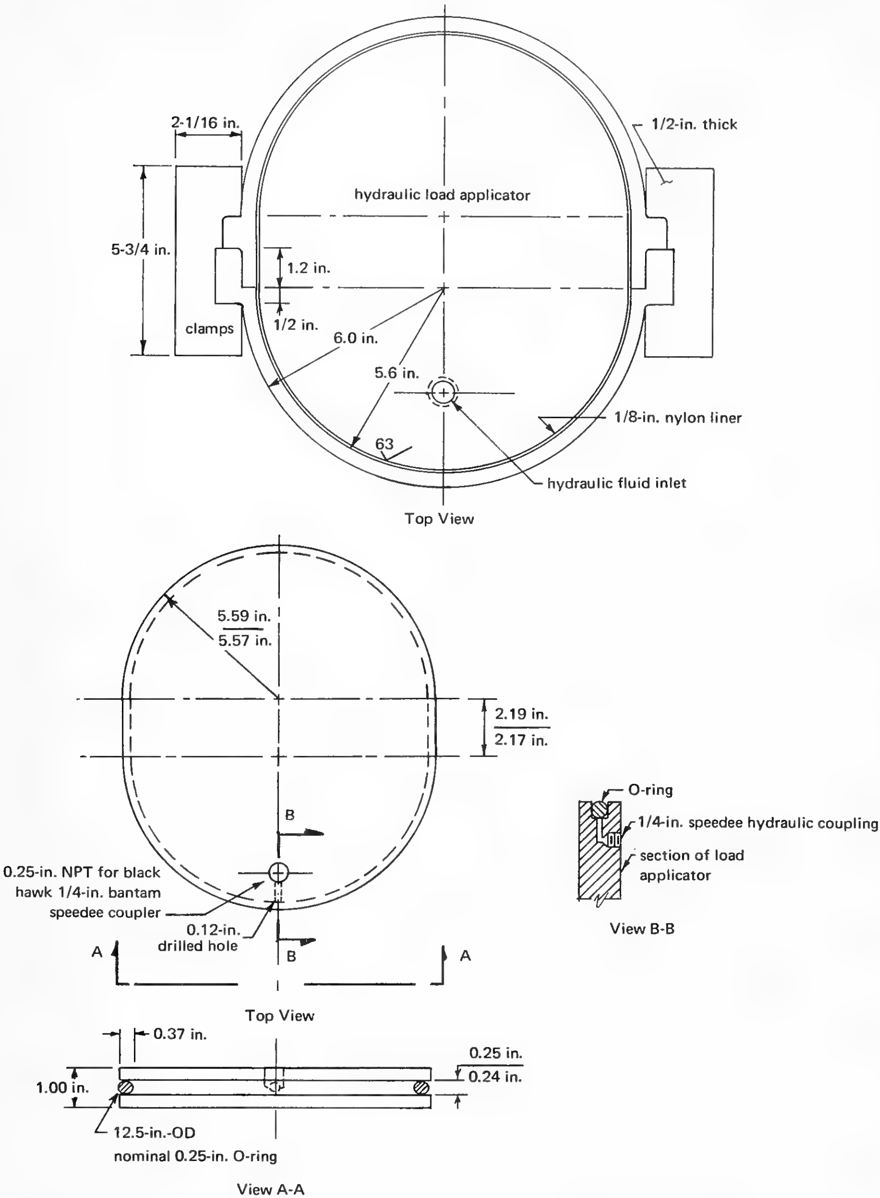


Figure C-5. Test assembly composed of two-dimensional model of hemispherical end closure mounted on hydraulic load applicator.

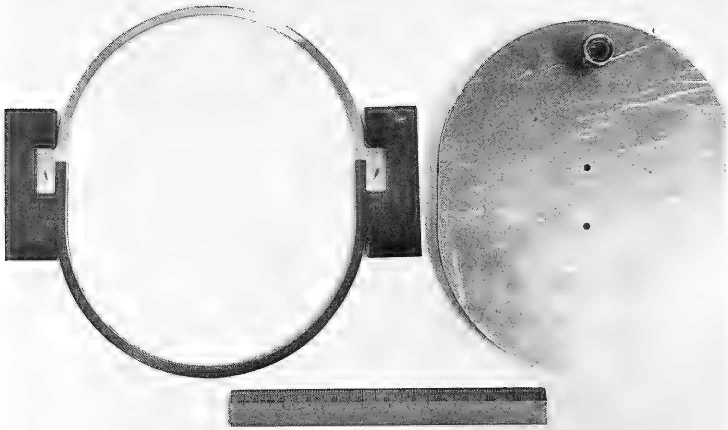


Figure C-6. Two-dimensional model of end closures and hydraulic load applicator.

CONCLUSIONS

Serious stress concentrations have been found (1) at the base of the heads of tie rods, (2) in the shape transition zone at the end-closure flange, and (3) around the shear-pin holes in the segmented-wall laminae. These stress concentrations occur at locations where failure was previously initiated in the acrylic pressure vessel models during hydrostatic testing. If full-scale pressure vessels of design similar to that of the models tested are built, these stress concentrations must be either eliminated or their severity taken into consideration during the vessel design.

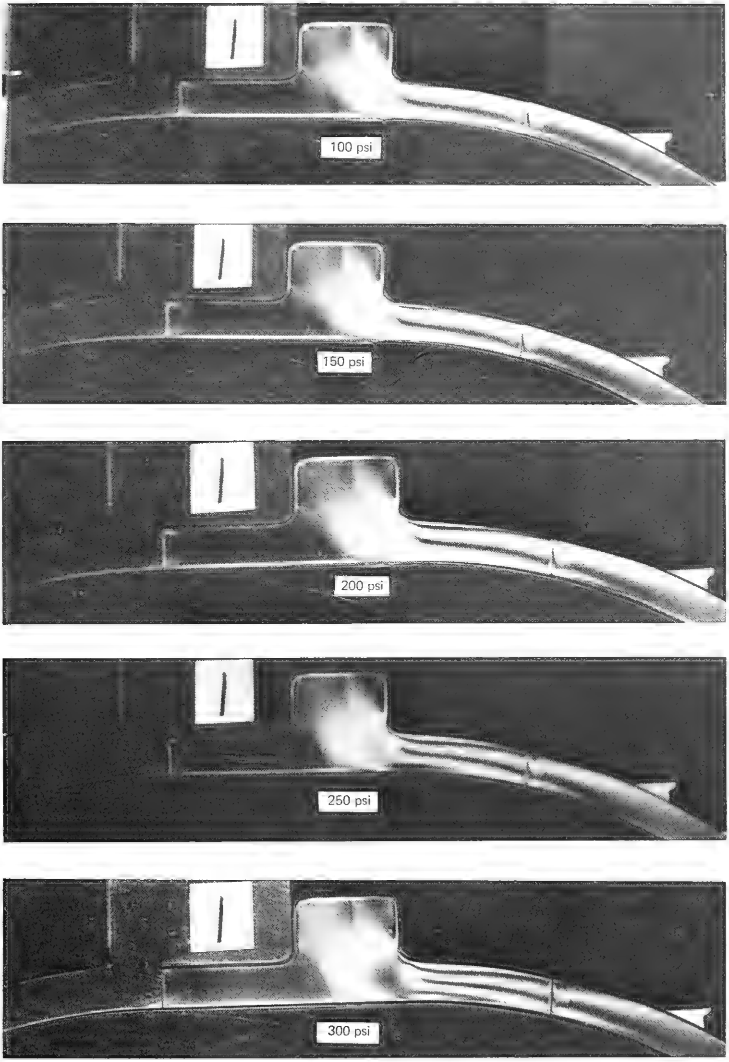


Figure C-7. Distribution of photoelastic fringes in two-dimensional end-closure model under different levels of loading.

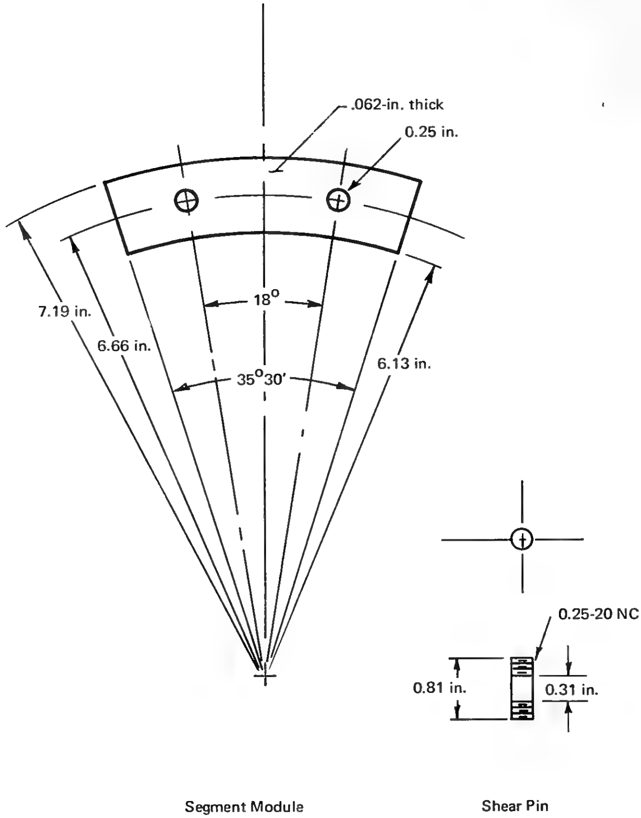


Figure C-8. Typical module from segmented pressure vessel fabricated from photoelastically active material.

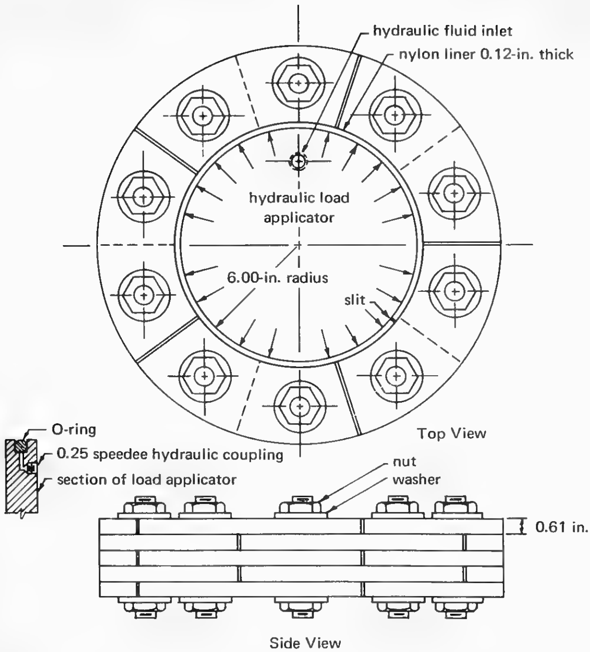


Figure C-9. Test assembly composed of five layers of segment modules mounted on hydraulic load applicator.

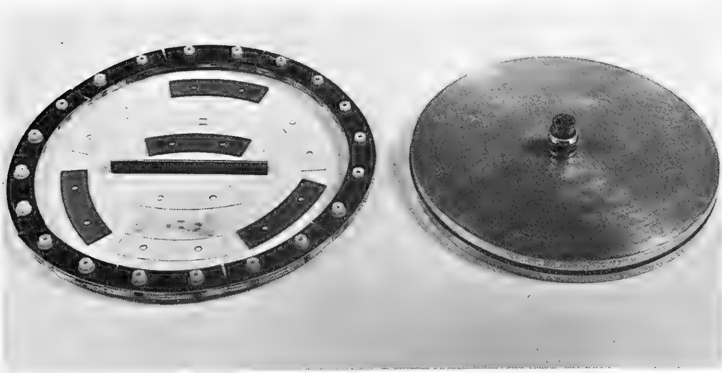


Figure C-10. Typical assembly of segment modules and the hydraulic load applicator. Only the segment modules in the top layer of segmented-wall assembly are of photoelastic material.

Figure C-11. Test setup for measurement of photoelastic fringes in the segment modules around shear pins.

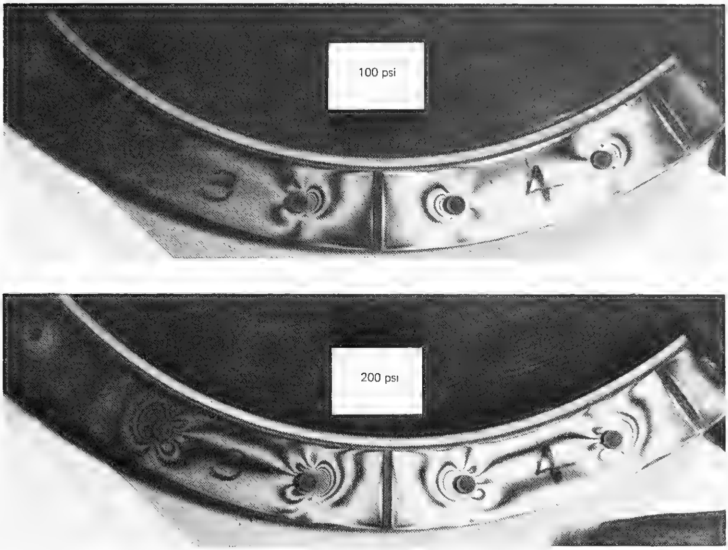


Figure C-12. Typical distribution of photoelastic fringes in the segment modules at different hydraulic loadings.

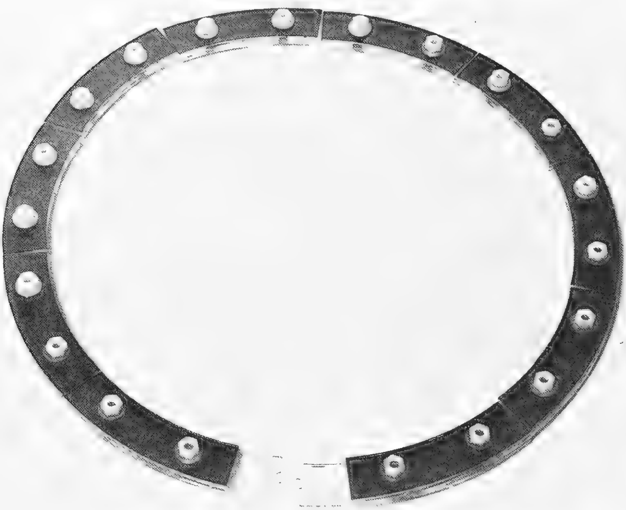


Figure C-13. Segmented-wall model after failure at 220 psi of hydraulic loading.

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TR-666 91 p. illus March 1970 Unclassified

1. Pressure vessels—Modular designs I. Y-R009-03-01-004

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Internal pressure						
Stacked ring						
Segmented design						
Circular polygon segments						
Tie-rod end closure						
Maraging steel						
Dome-shaped end closure						
Shear-pin holes						
Stress raisers						

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