Development of a Manned Transparent Capsule for Panoramic Marine Observation

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An underwater capsule with panoramic visibility, capable of carrying two men to a depth of 1000 ft, was designed and fabricated. The manned transparent capsule offers stereoscopic vision for optimum exploration of the marine environment. A modular approach was selected for the construction of a spherical pressure hull of 5.5-ft diameter from 2.5-in.-thick commercial-size acrylic sheets. Bonding twelve identical thermoformed and machined spherical pentagons, including two antipodes with steel inserts, created a "spherical dodecahedron." To evaluate the pressure hull for safety and certification to a depth of 1000 ft, hydrostatic testing was performed on 15-in.-diam scale models and the 5.5-ft hull. Acrylic plastic has demonstrated its capability to serve as a major pressure hull material at moderate depths. The collapse depth of the models was experimentally confirmed to be over 3000 ft, and no failure has been experienced in two years of cyclic loadings to 1000 ft.

Introduction

THE purpose of this paper is to review the development of an acrylic pressure hull that was designed to serve as a manned transparent capsule for gathering marine information by means of the human observer. For detailed exploration of the ocean environment, stereoscopic vision offers definition, acuity, and examination of events and conditions in the ocean simultaneously with data identified by traditional and indirect transducers. An underwater vehicle or facility possessing a 1-atm interior with unrestricted opportunity for visual observation of the hydrospace can enable anybody to investigate the ocean environment with a minimum of special training or sacrifice of comfort. convenience, and safety.

The Naval Missile Center (NAVMISCEN), Point Mugu, Calif., and Naval Civil Engineering Laboratory (NCEL), Port Hueneme, Calif., developed and evaluated a semimobile underwater laboratory concept featuring a 5.5-ft-diam acrylic sphere as the transparent capsule.[1–4] Although the capsule could be used for almost any underwater vehicle application at moderate depths, it was originally intended to meet the need for a compromise facility that would include the comfort and depth capability of submersibles with the unrestricted opportunity for observation from undersea habitats over long periods of time. Its function was to provide a means of inspection and observation of marine biological activity and physical phenomena of the ocean environment on the continental shelf. Another anticipated use for a manned transparent capsule with a 1-atm interior was as a platform for supervision of underwater project work by the supervisor, who would not need to be a trained scuba diver. The transparent capsule as an observatory-laboratory would give man the seagoing capability to match the requirements of his open ocean experiments without sacrifice of his field of view while submerged in the water. This is particularly applicable to research activity with marine mammals, as their exercises go beyond depths attainable by scuba-equipped training specialists.

Figure 1 is a sketch of one such application for a spherical transparent capsule, emphasizing unrestricted visual observation from one type of underwater structure. Although the capsule is not restricted to a particular type of underwater vehicle application, this paper will discuss the design, development, and evaluation of the transparent capsule as a key component in terms of its application to the underwater observatory-laboratory.

Capsule Design

General Design Parameters

Several primary design parameters had to be resolved before detail design of the transparent capsule could begin. The primary design parameters that needed resolving were 1)

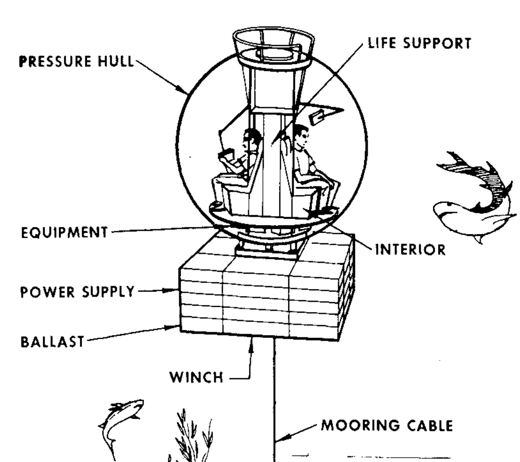

Fig. 1 Underwater observatory concept.

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selection of transparent material, 2) selection of hull configuration, and 3) selection of hull assembly technique.

Material selection

Material was selected on the basis of optical properties, mechanical properties, and commercial availability in large castings, forgings, or plates. After a brief review of existing transparent materials, the selection was narrowed down to the following materials: 1) glass, 2) acrylics, 3) epoxies, 4) polyesters, and 5) polycarbonates. The requirement for good optical properties eliminated the commercial quality polyesters, epoxies, and polycarbonates. The final choice narrowed down to only two materials: acrylic plastic[5] and glass. Both have excellent optical properties and acceptable mechanical properties. Glass has greater compressive strength than acrylic plastic, whereas acrylic plastic has a definite advantage over glass in its availability in large structural shapes, amenability to economical fabrication processes, and insensitivity to stress risers associated with penetrations and joints in pressure hulls of any shape. In a long-term research program, the previously mentioned advantages of acrylic plastic over glass could be largely neutralized by the discovery of techniques for economical casting of large glass shapes and the joining of several glass segments into a pressure hull without the presence of high-stress risers. However, within the limitations of this program, acrylic plastic was chosen since only a minimum of research and innovation was needed to build acrylic plastic pressure hulls immediately.

Hull configuration

Selection of a hull configuration was simple because only the spherical shape permitted the design of a complete acrylic pressure hull with 0.27 weight-to-displacement ratio for a 1000-ft operational depth, without any internal or external stiffeners that would unduly complicate the fabrication, as well as introduce unwanted tensile and shear stresses into the structure. All other hull shapes with 0.27 weight-to-displacement ratio required an elaborate system of stiffeners that, although technically feasible (Fig. 2), were definitely undesirable from the standpoint of stress distribution and economical fabrication.[6] Thus, a monocoque acrylic shell of spherical shape was chosen as the optimum hull configuration for the transparent capsule.

Hull assembly technique

The assembly technique selected for fabrication of the monocoque spherical shell had to result in the construction of a capsule that satisfied tight dimensional tolerances, required a minimum of special tooling, needed a minimum of machining time, wasted little material, and lent itself to mass production methods. Four possible ways of assembling the sphere were considered: 1) bonding of cast hemispherical shells, 2) bonding of hemispherical shells machined from cast acrylic, 3) bonding of hemispherical shells blown from acrylic sheets, and 4) bonding of spherical polygons that were thermoformed in molds from acrylic sheets and subsequently had their edges milled to the appropriate angle. Critical analysis of the four feasible construction techniques showed that only the bonding of spherical polygons satisfied the criteria considered desirable for construction of acrylic hulls, because only one concave mold of small size (less than $\frac{1}{5}$ of hemisphere surface) is needed, very little raw material is wasted as the polygons are thermoformed from acrylic sheets of desired thickness, and the machining is limited to turning of disks from sheet stock for thermoforming and subsequent milling of edges on the polygons. Since the polygons are relatively small, the thermoforming process does not appreciably alter the thickness of the acrylic, which depends on the chosen sheet stock of acrylic Type G. The fact that even a single spherical hull is made of 12 spherical pentagons allows mass production of polygon modules because the machining of pentagon edges is repeated 60 times and is adaptable to automated tools. Fabrication of more than one sphere at a time lowers the costs even further.

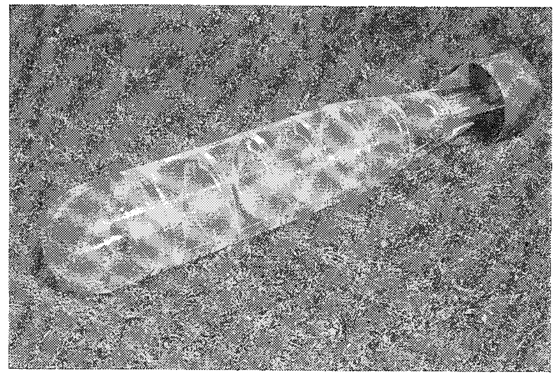

Fig. 2 Acrylic pressure hull for free diving instrumented oceanographic capsule.

Detail Design

Once the previously discussed main design parameters pertaining to material selection, shape configuration, and assembly technique had been selected, the detailed design of the capsule began. There were three focal points around which the detail design revolved: 1) means of ingress and egress to the capsule, 2) penetrations for electric and hydraulic conduits, and 3) attachment of the capsule to other structural systems.

Ingress and egress

Means of entering and leaving the capsule by its occupants could be achieved in several ways, but a hatch was placed in the top polar region of the sphere to permit egress even when the capsule was floating free on the ocean surface, after emergency ascent, without additional buoyancy systems. Although from a structural viewpoint an acrylic hatch was desirable because when closed it did not introduce any appreciable stress concentrations from material mismatch, a steel hatch system was chosen. The rationale behind this choice was that an acrylic hatch would soon lose its sealing ability because of scratch damage to the seal surface as equipment and personnel pass through the hatch opening. Since a steel hatch system constitutes a serious mismatch in rigidity with the acrylic hull, the thickness of the steel was kept to a minimum value corresponding in strength to the implosion pressure of the acrylic shell. Thus, the mismatch in rigidities of the two structural materials and the resulting stress riser effect could be minimized.

Penetrations

Penetrations for electrical and hydraulic conduits represented a serious problem, as they are also sources of serious stress concentrations. Rather than have them distributed throughout the pressure hull, they were centralized in the steel plate of the capsule and several design problems were eliminated. First, many penetrations in the acrylic hull, with the accompanying stress concentrations, were replaced by a single penetration with a stress concentration approximately equal to that of the hatch. Second, placing all the penetrations associated with electrical and hydraulic feedthroughs in a steel plate eliminated all problems associated with drilling and tapping acrylic. And third, placing the individual penetrations in the steel plate made the incorporation of reinforcements around the many penetrations a relatively easy problem by locating the penetrations in a thick section of the plate, where the effect of stress concentrations would not initiate the failure of the whole plate. The location of the plate itself was

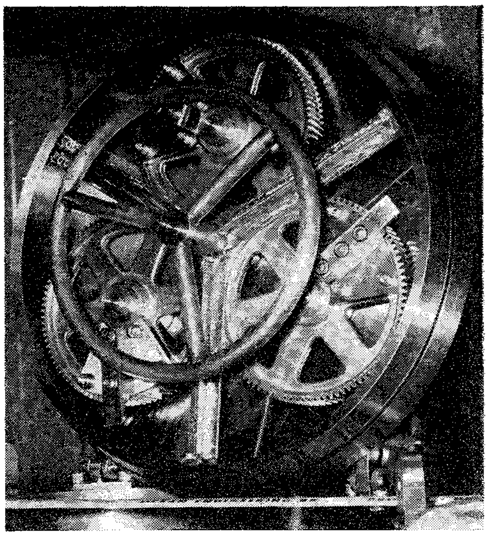

Fig. 3 Hatch closure mechanism for the 5.5-ft-diam capsule.

optional, depending on other factors, such as the method of attaching the capsule to other structural systems and components.

Attachment

Attachment of the capsule to other structural systems posed a difficult problem in several design and operational constraints that had to be satisfied by the proposed attachment method. The design constraints that had to be satisfied were 1) no direct attachment to the acrylic hull by means of metal bolts or screws, 2) no restraint to be imposed on the decrease of capsule diameter during pressure loading, and 3) the attachment to be stiff enough to prevent excessive movement of the whole capsule with respect to its structural framework during raising or lowering of the whole structure. Another operational constraint was the need for as much panoramic visibility as possible.

Only one type of attachment satisfied all the design and operational constraints. This attachment relied on bolts and screws fastened into the steel plate which did not decrease the field of view from inside the capsule, did not create additional stress risers in the acrylic hull, did not impose any restraint on hull compression during its deformation under hydrostatic pressure, and kept the capsule in a relatively fixed position with respect to the structural framework to which it was attached. Two other approaches of containing the capsule within either a net or a rigid cage were discarded because the net permitted too much relative movement between the capsule and the structural framework to which the capsule was attached, and the rigid cage obstructed the view.

The selected method of attachment dictated that the steel plate be located at the bottom of the capsule, to prevent the (upward) buoyant force acting on the capsule from applying a bending moment on the wall of the capsule. However, the attachment at the bottom steel plate imposed a point load on the capsule. The buoyancy of the capsule is constantly pulling upward while the attachment provides a downward pull on the bottom steel plate. The force applied to the steel plate by the attachment is transmitted to the acrylic hull by means of thin, flexible retaining flanges bolted to the inside face of the plate. The use of a flexible flange as well as rubber gaskets underneath them ameliorated any stress concentrations due to point loads. Because of these opposing forces, tensile as well as shear stresses are introduced into the acrylic hull when the capsule is submerged in water. However, when the submergence is deeper than 25 ft, compressive stresses generated in the hull by external hydrostatic pressure become larger than the tensile stresses caused by the action of the two opposing forces of upward pulling buoyancy and downward reaction by the plate, and the resulting principal stress in the hull becomes compressive. Thus, once the surface layer of the ocean has been penetrated, the effect of point loading on the stress distribution caused by the attachment of the capsule at the bottom plate becomes minimal. Furthermore, the magnitude of the point load can be considerably decreased if the weight of the equipment inside is distributed on the acrylic shell. Such distribution considerably decreases the net buoyant force of the capsule acting on the bottom plate attachment.

Working stresses

One of the crucial factors in sizing the hull components was the magnitude of the working stress. Since no reliable data were available in the literature on the response of acrylic plastic to long-term compression, an engineering judgment had to be made on what constitutes conceivably a safe working stress level in an acrylic hull subject to elastic, plastic, and creep instability under hydrostatic loading. Since the 5.5-ft-diam capsule will be subjected primarily to cyclical pressure loadings for durations of up to 10 hr, the stress range of 3000 to 3500 psi was selected as the average working stress level, although it was recognized that stress concentration factors of 1.5 to 3 magnitude would probably exist at the interface with steel plate penetrations in the acrylic hull. Because of its viscoelastic properties, acrylic under compressive stresses should locally readjust to excessive strain and avoid the initiation of stress cracks at penetrations.

The working stresses in the steel hatch and bottom polar plate were selected to be as high as safely feasible, in order to create metallic inserts that would be pliant and thus would counteract the stress riser effect generated by mismatch of modulus at steel plate and acrylic hull interface. Since the material selected for hatch and polar plate was stainless steel Type 316L, the average stress level chosen was in the 12,000–15,000 psi range. The hatch, of course, incorporates many other materials in the locking apparatus located on the interior side. The choice of less corrosion-proof materials in the experimental capsule was due to the availability of these components as off-the-shelf commercial items. For example, the hatch mechanism itself (Fig. 3) required a minimum of design time and machine shop time; the primary parts are standard gears. It is planned, however, in the future, to replace many of these standard commercial components with custom-made corrosion-resistant hardware.

Fabrication

The fabrication and assembly of the hulls were performed in the Technical Support Department facilities at Point Mugu, Calif. Approximately 18 of the hull models were built, and 14 were considered satisfactory for hydrostatic testing. Successful experience with the models led to construction of a 5.5-ft-diam capsule that has been exposed to shallow water tests to demonstrate its integrity against leakage, gravity loads, and buoyancy loads.

15-in.-Diam Hull Models

Each of the 15-in.-diam models was essentially hand-built except for the machining of pentagonal wall sections (modules), which were milled under control of a computer tape. The initial step was to thermoform a flat blank of 0.5-in.-thick and 10-in.-diam acrylic (Type G MIL-P-21105C) into a spherical shell of 7.5-in. radius between matched convex and concave spherical molds. To minimize mark-off from the mold to the acrylic, the surface texture of the aluminum molds was hand polished to 16 μ in. The molds were warmed to 150° F and the acrylic blank was separately preheated to 300° F before thermoforming by closing the two molds with 15 in. Hg vacuum in the concave mold to assure positive contact with the convex surface of the acrylic shell, which eventually becomes the outside surface of the hull model. Although the thermoforming could be accomplished without the convex mold, its use assured a uniform temperature effect at the risk of minor amounts of mark-off on the concave (inside) surface of the acrylic shell.

After the acrylic cooled to 180° F while held in the mold (approximately 5 min), the thermoformed shell was removed and allowed to cool in the ambient air. The spherical contour of the acrylic shell was compared against a curved template with a 7.500-in. radius, and shells whose curvature deviated more than 0.035 in. from the template were rejected.

The spherical shells of acrylic were band-sawed into pentagonal shape and milled accurately into pentagonal modules or sections. A vacuum-operated chuck was improvised to hold and index the spherical shell while a tape-controlled milling machine cut the shell to the desired pentagon shape.

Two of the twelve pentagon sections were bored for installation of steel end plates at the top and bottom of the hull model. The hole was bored conically with the imaginary apex of the cone at the origin of the radii to the spherical surfaces of the pentagon.

To assemble the twelve pentagonal sections into a dodecahedron-type sphere required the innovation of an assembly fixture, selection of a staisfactory cement, and manual skill to maintain the sphericity required while bonding all the joints uniformly. A special assembly fixture (Fig. 4) was constructed to preposition all twelve pentagons into a sphere and to hold and adjust each pentagon during bonding with polymerizing acrylic cement, PS-18. Bond thickness was between 0.010 and 0.015 in. After assembly, the hull was annealed at 160° F for 24 hr to help reduce stresses.

The assembled hull models were measured for diameter and spherical curvature with a micrometer and template, respectively. The mismatch in outside surface between adjacent pentagons was held to within 0.015 in. and the hull diameter measured 15.00 $\pm$0.02 in.

To complete the fabrication of the 15-in.-diam hull models, a stainless-steel end plate with spherical contour was installed in both polar pentagons and held in place by an elastic connection between the two opposed end plates (Fig. 5).

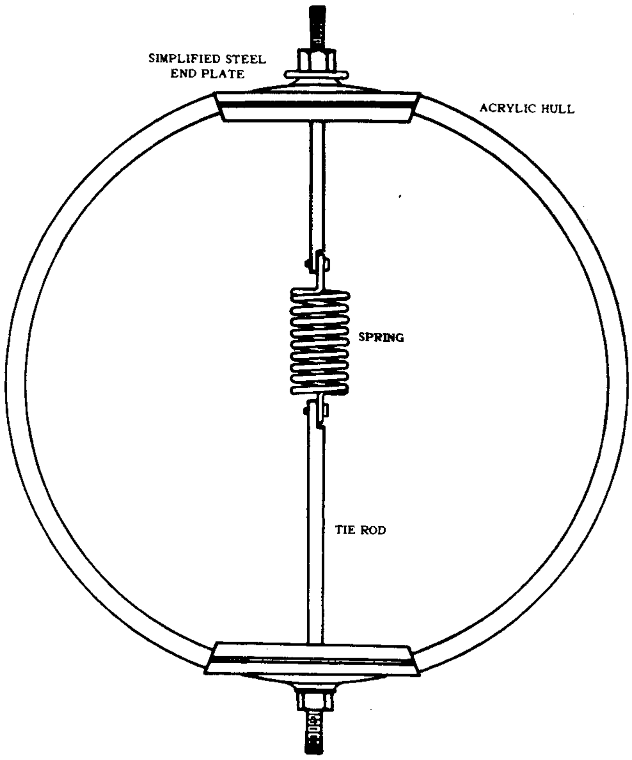

Fig. 5 15-in.-diam acrylic hull model with simple end plates.

5.5-ft-Diam Transparent Capsule

Fabrication of the 5.5-ft-diam spherical capsule was based on successful methods developed while working with the small models. Certain changes in handling procedures were required because of the size and weight involved. The thermoforming blank for each pentagonal section for the 5.5-ft-size capsule was a 46-in.-diam by 2.5-in.-thick disk which was cut from a 4 × 5 ft commercial sheet of Type G MIL-P-21105C acrylic. The thermoforming mold was a spherically concave aluminum dish with a surface texture of 32 μin. Four small vacuum holes were drilled at the apex of the mold. To assure a good vacuum seal between the heated blank and mold, the periphery of the acrylic disk was machined to a 200 μin. surface texture. After positioning the acrylic disk in the mold, they were placed in an oven and gradually heated to 300° F over a period of 24 hr. Thermoforming was accomplished at 300° F by applying a vacuum of 29 in. Hg to the mold cavity (Fig. 6). With the vacuum left on, the oven and contents were allowed to cool to 110° F in 16 hr, when the formed part was removed into ambient air. Since no matching convex mold was used, the concave surface of the acrylic shell was unmarred. The mark-off on the convex side of the shell will not be visible when the surface is submerged in water because of the similarity of refraction index between water and acrylic (1.33 and 1.49, respectively). The net result is a transparent wall with a minimum degradation of optical transmissivity in water.

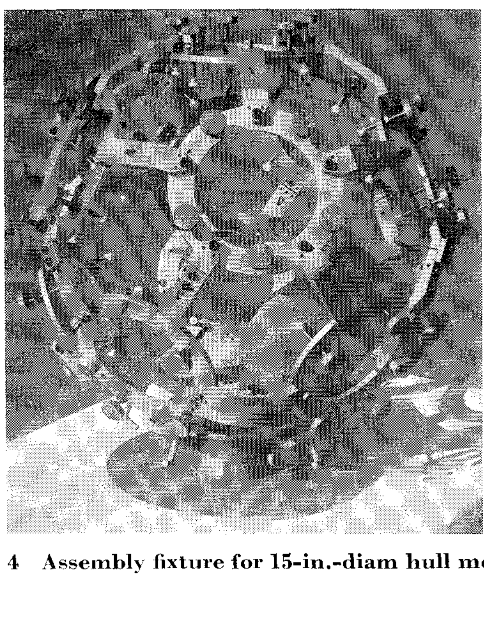

Fig. 4 Assembly fixture for 15-in.-diam hull model.

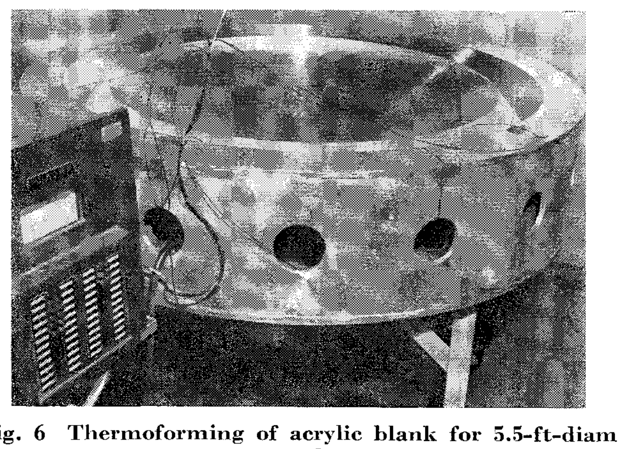

Fig. 6 Thermoforming of acrylic blank for 5.5-ft-diam capsule.

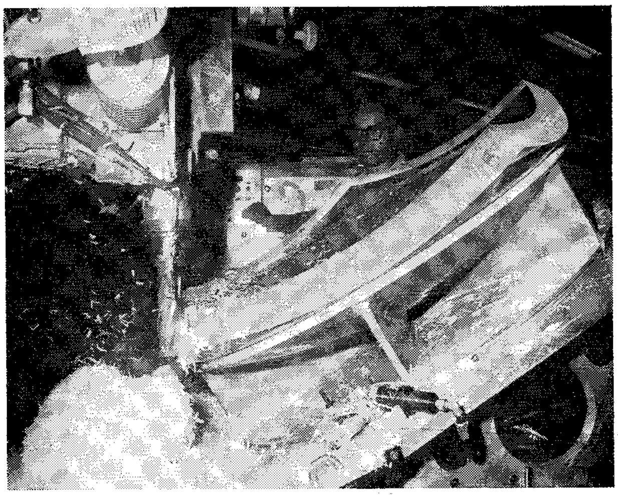

Fig. 7 Machining acrylic pentagon for 5.5-ft-diam capsule.

The spherical surface contour of the thermoformed acrylic shell was measured for dimensional conformity to a template with a 33.00-in. radius of curvature. Maximum departure from the correct curvature was 0.020 in. and occurred near the periphery where the shell tended to flare in. It was noted that the periphery of the shell flared in 0.012 in. with respect to the thermoforming mold. The expected spring-back by the acrylic did not occur.

Each thermoformed acrylic shell was stored in a contoured rubber-lined box where the unmarred inside surface was left protected from possible scratches. The same box was used subsequently to store machined pentagons before assembly.

An improved vacuum chuck was developed to hold and position the thermoformed acrylic shell for cutting and milling into pentagonal shape (Fig. 7). The milling was completely automatic once the part was loaded into the chuck. The milling cutter was a 2-in.-diam helical type rotating at 3800 rpm and fed at 10 in./min under control of a computerized tape program. Coolant was used during the machining operation, and the resultant surface texture of the finished edge was 63 μin. Two of the resulting pentagonal sections were conically bored to accommodate the steel end plates for the antipodes. The acrylic pentagons were annealed at 160° F for 24 hr after their machining, to reduce residual internal stresses. Changes in the spherical contour of the surface on the acrylic pentagons were hardly discernible after the machining and annealing operations.

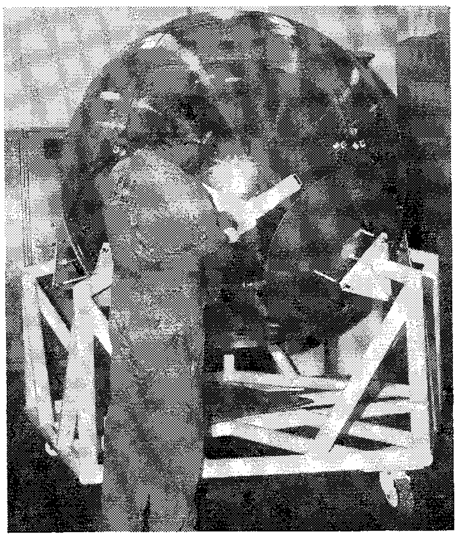

Fig. 8 Assembly of 5.5-ft-diam acrylic capsule.

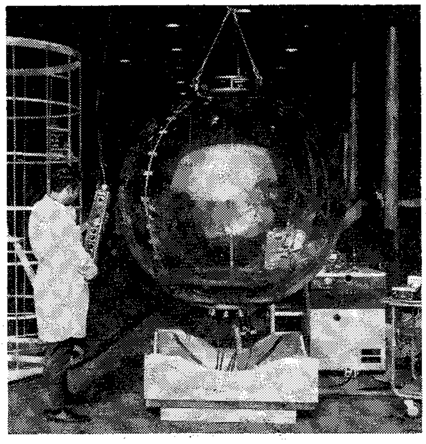

Fig. 9 5.5-ft-diam acrylic capsule.

Assembly of the 5.5-ft-diam acrylic hull was accomplished by a two-phase bonding technique where two quasi-hemispheres, made up of six bonded pentagonal sections each, were in turn bonded together into a sphere. A bonding procedure suggested by the Rohm and Haas Company for their PS-18 cement was used with a butt joint gap of $\frac{1}{8}$ in. between adjacent 2.5-in.-thick acrylic pentagons. The joints were tape masked to form a convex bead that could be polished away to assure a bonded joint flush with the spherical surface. A polyethylene squeeze bottle was used to apply the PS-18 cement over about 12 in. of joint per pour. The bonding operation was performed with the acrylic pentagons prepositioned in a holding fixture that was accurately preadjusted to a dummy hemisphere of 66.00-in. diameter. The final assembly into a sphere was performed by matching and bonding the two halves consisting of six bonded pentagons each (Fig. 8). The bonding resulted in about 10% random voids and bubbles in the joint because of shrinkage of the cement as it cured. The worst spots were mechanically removed and refilled with cement to reduce the vacant area to about 5%. The completed acrylic hull was annealed at 160° F for 24 hr.

Table 1 Summary of tests on 15-in.-diam models

Short-term tests

Collapse pressure, psi	Pressurization rate, psi/min	Comments
1600	100	End plate failed
1650	100	End plate intact
1525	100	End plate intact
1360	100	Had no steel zones

Long-term tests

Pressure, psi	Failure	Time at pressure, hr	Comments
1000	yes	10.5	Had several 500- and 750-psi cycles previously
1000	yes	14.75	No previous cycles
900	yes	35.5	Poorly constructed model
900	yes	124	Had a previous short cycle at 900 psi
750[a]	yes	4,488	Had 162 previous short cycles to 500 psi
500	no	12,400	Time as of 1 March 1968
250	no	12,500	Time as of 1 March 1968

Cycling tests, 100 psi/min

Pressure, psi	Time at pressure	Relaxation time	Number of cycles (as of 1 March 1968)
500	120	120	78
500	24	24	127
500	12	12	148
500[a]	$2\frac{1}{2}$	$2\frac{1}{2}$	162

[a] Same model used on both tests.

Diameter measurements were made between the centers and the corners of all the pentagons. The measurements indicate that the hull is probably slightly prolate with slight bulges centered on the pentagons.

The steel end plates for the 5.5-ft-diam capsule consisted of a $22\frac{1}{2}$-in.-diam bottom plate with nine feedthrough holes and attachment provision for external and internal point loads, and a two-piece top plate consisting of an $18\frac{1}{2}$-in.diam hatch and a $22\frac{1}{2}$-in.-diam mating ring. Both hatch and bottom plate were spherically shaped with an "O" ring seal on their conical edges. The bottom plate was attached to the acrylic hull by means of a split retaining flange that was bolted to the plate against a $\frac{1}{4}$-in.-thick rubber gasket placed on the acrylic. Thus, all differential radial loads were directly transferred across steel and acrylic hull interfaces.

The hatch closure mechanism was actuated by means of a pair of hand wheels mounted on a common shaft and located one inside and one outside the hatch. Through a 5 to 1 gear reduction, the hand wheel actuates three pawls which engage against the inner lip of the mating ring to effect a positive lock. "O" ring seals were used to seal hatch to ring and ring to acrylic pentagon. The mating ring, with its three suspension lugs, was held to the acrylic pentagon by a split retaining flange like the one used on the bottom plate. The man-size capsule assembly, including acrylic hull, stainless-steel bottom plate, and mating ring with hatch and a wooden floor mounted on the bottom plate, weighed 1526 lb (Fig. 9).

Capsule Evaluation

15-in.-Diam Hull Models

Description of tests

The 15-in.-diam models were subjected to hydrostatic tests at the facilities of NCEL, Port Hueneme, Calif. A model to be tested was held in a holder such that the buoyancy force was restrained through the bottom end plate. Figure 10 shows a typical test setup for installation in the pressure tank. Strain gage readings were periodically taken while the pressure and temperature of the pressurizing medium, sea water, were continuously recorded. A total of 14 models have been tested and of these, 9 were taken to destruction. Four models were pressurized to failure at a rate of 100 psi/min. Seven models have undergone long-term pressure loading. Three models are undergoing cycling tests to 500 psi.

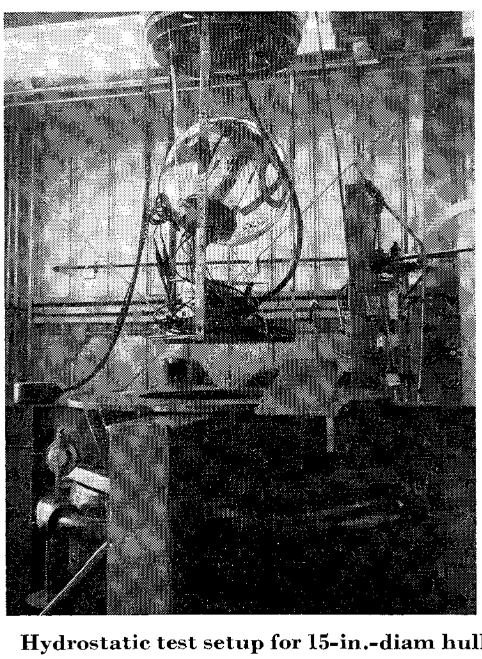

Fig. 10 Hydrostatic test setup for 15-in.-diam hull model.

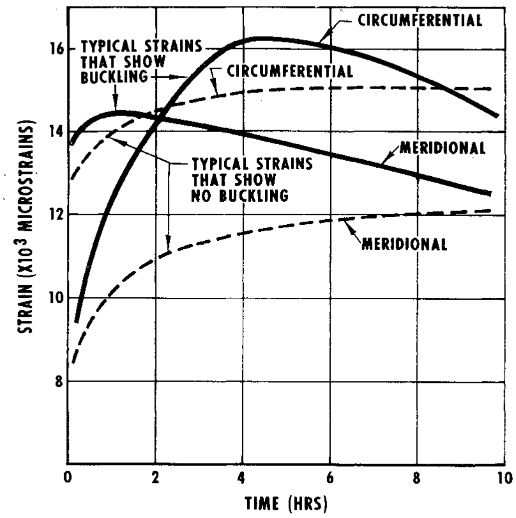

Fig. 11 Strain near equator at 1000 psi for 15-in.-diam hull model.

Results of tests

A summary of test results is given in Table 1. Strain readings from the test in which a model failed at 1000 psi after 10.5 hr are shown in Fig. 11. It includes a strain reading near the disintegrated area, which indicates decrease in compressive strain on the interior surface of the sphere prior to collapse. This phenomenon plus the inward fracture edges at the failure zone indicated local plastic buckling rather than simple compressive failure of the acrylic shell.

Figure 12a is a curve which has been drawn from the test results to show the time-dependent buckling characteristic of the models under constant hydrostatic load. Figure 12b is a series of curves indicating the time-dependent creep of the models under constant hydrostatic loading by measuring the decrease in the internal volume of the model using a liquid displacement technique.

5.5-ft-Diam Capsule

The assembled acrylic hull was tested to date only to certify its water-tightness and structural integrity with gravity and buoyancy point loads. Instrumentation was installed as shown in Fig. 13.

Leakage

Hull leak susceptibility was tested with Freon 12, helium, and water, with and without point loads. With a vacuum of 28.2 in. Hg inside the hull at rest, the Freon 12 leak-detector system did not indicate leak rates as low as 0.5 oz/yr. Using a helium leak-detector system with a vacuum of about 10^{-5} torr, a leak that was more than 10^{-10} cm^3 helium per second was detectable only in the area of the "O" ring seals. The application of a point gravity load by suspending the 1526-lb hull assembly by the three lugs on the mating ring at the top did not change the leakage pattern or rate. In an overnight period of 16 hr, the "locked in" vacuum reading in the hull went from 29.8 to 29.5 in. Hg during a temperature drop of 2° F.

The water leakage test was conducted during later buoyancy tests. A submersion of 24 hr in fresh water at 57° F and two more submersion cycles of 2 hr over a period of 48 hr did not cause a trace of water to appear inside the hull. The hygrometer, located inside the hull, read a steady value of about 35% relative humidity during the test.

Gravity point load

Strains in the acrylic wall caused by gravity loads were examined by means of eight strain gages installed on the inside

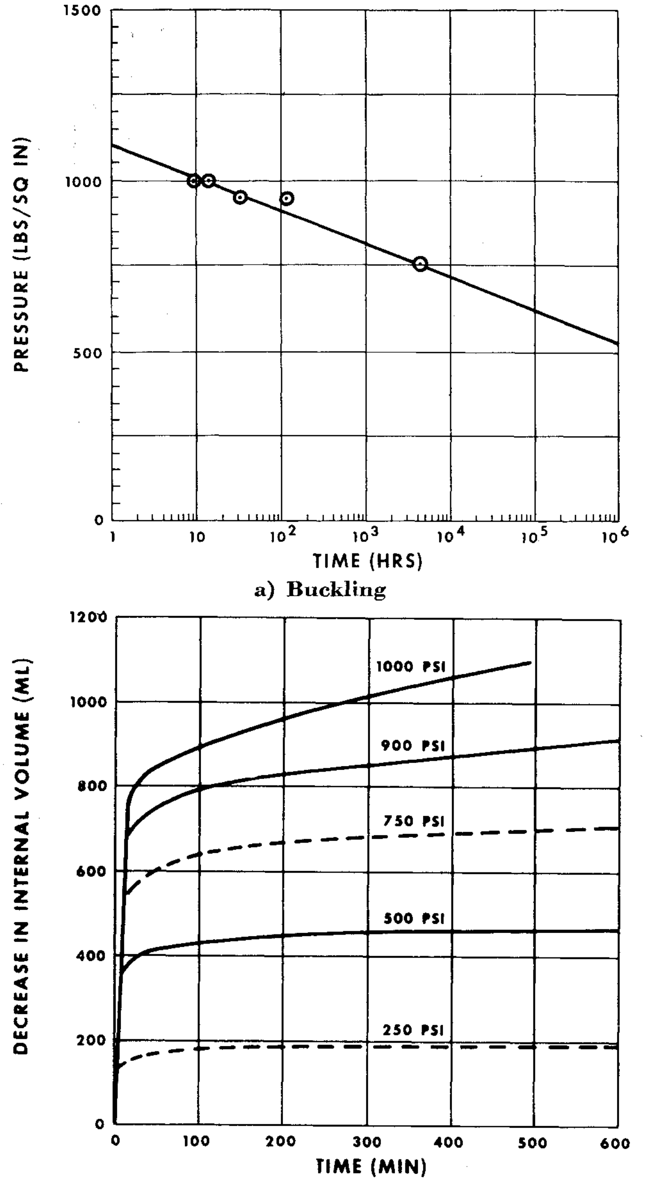

Fig. 12 Time-dependent characteristics of 15-in.-diam hull models under constant hydrostatic loading.

surface of the hull as shown in Fig. 13. A rectilinear potentiometer was installed between the top and bottom end plates to detect vertical diameter changes.

For gravity loads, the calculated uniaxial tensile stress in the acrylic wall near the end plate was of the order of 10 psi. Confirming this low value, the strain gages produced signal in their noise range for gravity loads as shown in Table 2. The potentiometer indicated a vertical diameter increase of 0.002 in. when gravity point load was applied to the hull.

Table 2 Strain gage readings for gravity point load of 1526 lb

Wall temperature: 64°F		Interior humidity: 50% RH	
Gage #[a]	At rest (microstrain)	Suspended (microstrain)	Change (microstrain)
1	11,050	11,040	−10
2	10,580	10,580	0
3	10,680	10,690	+10
4	10,510	10,490	−20
5	9,940	9,930	−10
6	9,480	9,440	−40
7	12,700	12,720	+20
8	12,380	12,360	−20

[a] See Fig. 13 for location of gage #.

Buoyancy point load

Buoyancy load tests were carried out by mounting the hull at the bottom plate to the support flange of a test cage for holding the 5.5-ft-diam capsule under water. The test cage was installed in a 15-ft-diam water tank (Fig. 14). The buoyancy point load was transferred through the bottom plate to the bottom area of the acrylic hull by means of the previously described rubber-lined retaining flange as a tensile and shear load on the acrylic. Water level on the hull was raised in 1-ft increments initially, but no undue strains were noted at each step.

Strain in the acrylic wall was examined by means of the eight strain gages mentioned before. Table 3 is a chart of reduced strain data and includes some other recorded parameters. The indicated strain due to buoyancy load was small and random. The potentiometer indicated a vertical increase in diameter of 0.10 in. The strain gages indicated a maximum rise in strain of approximately 200 microstrains near the bottom plate when adjusted for thermal expansion, which in acrylic equals 4×10^{-5} in./in./°F. No noticeable difference in strain regarding meridionally and circumferentially oriented gages could be established.

If uniaxial tension is assumed for simplicity at the strain gage and a modulus of elasticity of 400,000 psi is used for acrylic, the equivalent stress near the bottom plate is 80 psi compared to a predicted value of 25 psi based on a net buoyancy of 4100 lb for the hull assembly. Apparently the complex nature of stresses in the acrylic wall should not be interpreted in terms of simple uniaxial loads.

It is believed that the results of these preliminary tests demonstrate an acrylic hull assembly that is satisfactory against leakage and structural loads in shallow water. The hydrostatic proof testing of the 5.5-ft-diam acrylic hull for structural evaluation will be performed as soon as the installation of additional strain gages and other sensors on the capsule is completed. These tests will be carried out in the NCEL

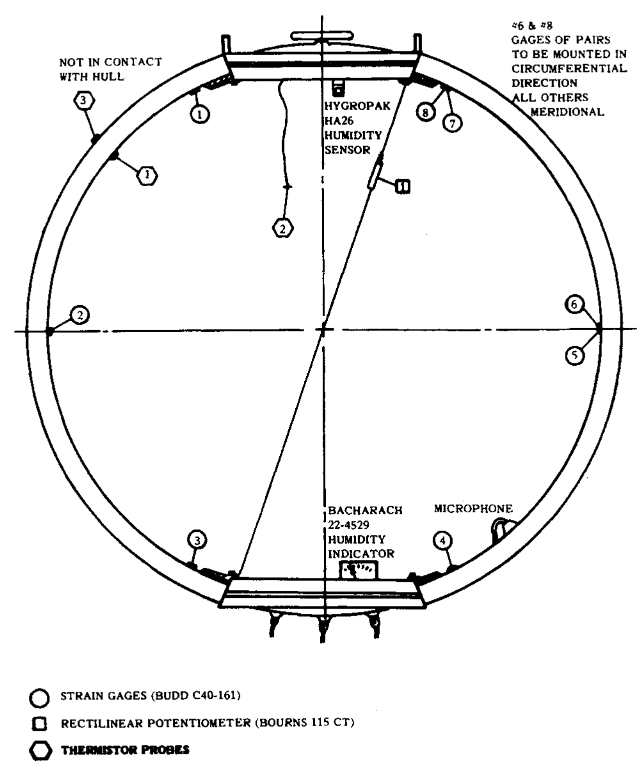

Fig. 13 Instrumentation layout for 5.5-ft-diam capsule.

Table 3 Test results for buoyancy loads on 5.5-ft-diam hull with buoyancy point load of 4100 lb

Test→ Load condition→	1 Submerged[b]	2 Unsubmerged	3 Submerged	4 Unsubmerged	5 Submerged	6 Unsubmerged
Strain gage #[a]	Reading changes between load and no load conditions in microstrains					
1	−115	−130	+150	−150	+240	−50
2	−115	+20	+70	−90	+200	−10
3	+30	−180	+220	−270	+20	−50
4	+95	−130	+180	−220	+330	−30
5	−80	−40	+90	−130	+210	−30
6	−130	+10	+60	−80	+170	+40
7	−40	+30	+20	−60	+110	+10
8	−105	+10	+60	−90	+110	+20
Change in wall temperature, °F	−4	+2	−2	+2	−2	+6
Change in vertical diameter, in.	+0.010	−0.009	+0.011	−0.011	+0.005	−0.003

[a] See Fig. 13 for strain gage location.
[b] Nominal submerged depth = 6 ft.

72-in.-diam, 5500-psi pressure vessel to generate information that would justify manned operation of similar acrylic pressure hulls in the ocean down to 1000 ft.

Summary and Conclusions

1) A 5.5-ft-diam transparent capsule was jointly developed by the Naval Missile Center and the Naval Civil Engineering Laboratory to explore and demonstrate the design feasibility of an acrylic plastic hull for manned undersea operations at continental shelf depths down to 1000 ft. It is intended for hydrospace applications where panoramic visual observation by man is required for correct interpretation and evaluation of the immediate prevailing situation.

2) The 5.5-ft-diam capsule was constructed from 2.5-in.-thick commercial sheets of Type G acrylic using a modular design which consisted of twelve bonded identical pentagons with a spherical surface contour.

3) A pair of diametrically opposed polar pentagons were equipped with 22.5-in.-diam end plates of contoured stainless steel, which included an 18.5-in.-diam personnel hatch at the top and a hull penetration zone with nine feedthroughs at the bottom. Internal and external point loads to the hull were successfully passed through either end plate and distributed evenly around the hole in the acrylic wall by means of a rubber-cushioned retaining flange.

4) The 5.5-ft-diam capsule has satisfactorily demonstrated its structural integrity against gravity and buoyancy loads without any water leakage. It is being prepared for extensive hydrostatic proof testing for man rating and certification purposes.

5) Fourteen scale models of the hull were hydrostatically tested at various pressures and loading cycles and the experimental collapse depth was well over 3000 ft. At 1000 ft the hull models showed no tendency to collapse after 12,400 hr (17 months) of test, nor did they indicate fatigue problems when cycled repeatedly (78 times) up to 1000 ft for 18,700 hr (26 months).

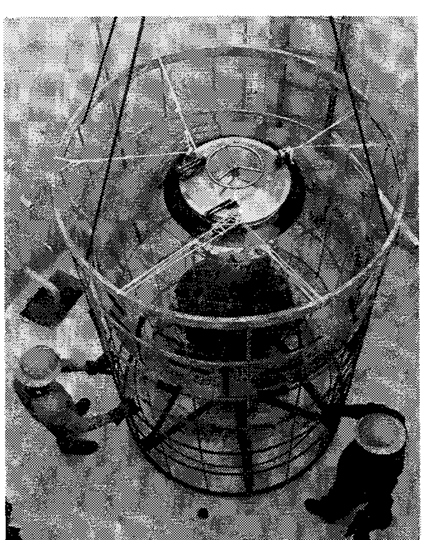

Fig. 14 5.5-ft-diam capsule in cage for buoyancy tests.

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