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Pressure Resistant Ceramic Housings for Deep Submergence Systems

Jerry D. Stachiw

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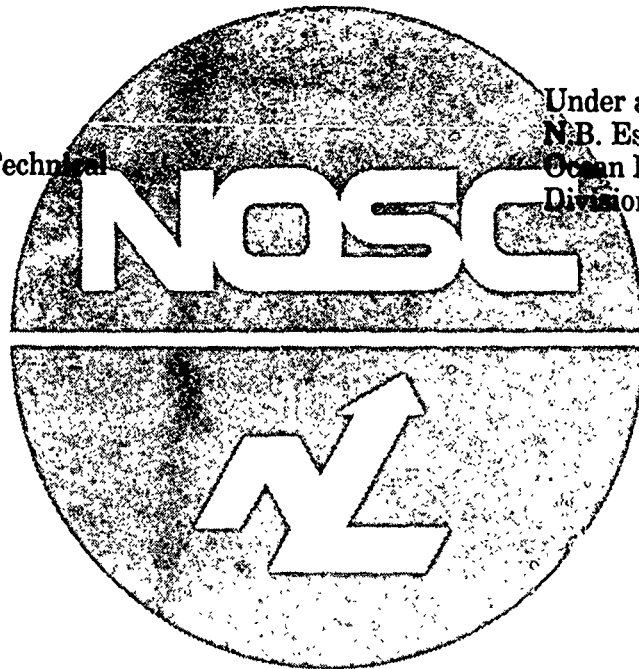
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ABSTRACT

Model-scale and full-scale ceramic housings for deep submergence service have been designed, fabricated from 99- and 94-percent alumina ceramic, proof-tested to 10,000 psi, and pressure cycled to 9000 psi without catastrophic failure. The ceramic 6- and 12-inch diameter cylinders with $t/D_0 = 0.0344$ and $L/D_0 = 1.5$ dimensions were joined together and to ceramic hemispherical bulkheads with $t/R_0 = 0.0346$ by titanium coupling rings that also serve as radial stiffeners. The payload of the pressure housing varied with the number of cylinders joined by coupling rings.

The weight-to-displacement ratio of 0.60 achieved in the NOSC ceramic materials program by a ceramic housing assembly with a 12-inch diameter and 84-inch length is the lowest ratio achieved to date with a ceramic or glass housing for 20,000-foot operational depth with a payload capability in excess of 100 pounds. This ratio can be reduced further to 0.5 for this housing design by substituting 7150-T7E97 aluminum alloy for titanium joint housing components and raising the design stress in ceramic from 150,000 to 175,000 psi.

GENERAL

This paper has two purposes. The first and general purpose is to suggest that ceramics should be considered as appropriate materials for the potential fabrication of ceramic buoys, submersibles, remotely operated vehicles (ROVs), buoyancy propelled vehicles (BPVs), and autonomous underwater vehicles (AUVs) to be used in deep submergence applications. The second and more specific purpose is to present the results of the design, fabrication, testing, and evaluation of NOSC's ceramic housing assemblies for deep submergence service.

INTRODUCTION

The Navy, among other organizations and institutions, is very interested in acquiring the most operationally effective and cost efficient manned submersibles, remotely operated vehicles, buoyancy propelled vehicles, autonomous underwater vehicles, and buoys for deep submergence operations. Three factors determine if such submersibles or vehicles meet mission standards: payload, operational range, and speed. Each of these factors is a direct function of the system's buoyancy.

Clearly, buoyancy is the critical issue. Optimally, buoyancy is provided by a well-designed pressure hull. However, if the buoyancy provided by the pressure hull is inadequate, there are palliative measures available to increase it, but these usually reduce the effectiveness of the underwater vehicle in fulfilling its mission task. For instance, additional buoyancy can be provided by attaching blocks of syntactic foam or soft shell tanks filled with lighter-than-water fluids to the pressure hull. This approach has an overall negative impact on system cost and operational effectiveness.

Thus, the design and materials used in fabricating pressure hulls are of critical concern. The optimization of shape and the use of premium material in the construction of the hull is required to obtain a pressure hull with low weight-to-displacement ratio (a large positive buoyancy). The reason for seeking the low weight-to-displacement ratio is to maximize payload, while minimizing hydrodynamic drag, and, thus, achieving optimum range and speed.

The choice of materials is more limited than the optimization of shape options as only a few materials are lightweight, corrosion resistant, and strong in compression. The characteristics of premium structural materials for external pressure housing are shown in Table I, and housing assembly material characteristics in terms of design pressure and of weight-to-displacement ratio are presented in Fig. 1.

A weight-to-displacement ratio less than, or equal to 0.5 has been found by operational experience to be desirable for the pressure housing assembly, so it may provide the vehicle with adequate buoyancy for its propulsion, guidance, and work subsystems. Pressure housing assemblies with a weight-to-displacement ratio greater than 0.5 provide inadequate buoyancy that must be augmented at a great expense and increase in vehicle's displacement by the addition of syntactic foam blocks or an increase in hull's volume.

A quick glance at the numbers is sufficient for recognizing that high-strength steel does not meet the rigid buoyancy requirements (Table I). Stated simply, the poor weight-to-strength ratio will sink deep submergence vehicles constructed from steel. Titanium and high-strength aluminum hulls provide some buoyancy, which, however, have to be augmented with syntactic foam to support the operationally specified payload.

Table 1. Premium Structural Materials for External Pressure Housings.

Material	Weight (lbs/in ³)	Compressive Strength (kpsi)	Strength Weight	Safety Factor
Steel (HY80)	0.283	80	280	1.25
Steel (HY130)	0.283	130	460	1.25
Aluminum (7075-T6)	0.10	73	730	1.25
Titanium (6AL-4V)	0.16	125	780	1.25
Glass (Pyrex)	0.08	100	1250	2
Glass Composite	0.075	100	1330	2
Graphite Composite	0.057	100	1750	2
Beryllia Ceramic 96%	0.104	225	2160	2
Alumina Ceramic 94%	0.130	300	2310	2
Glass Ceramic (Pyroceram 9606)	0.093	350	3760	2

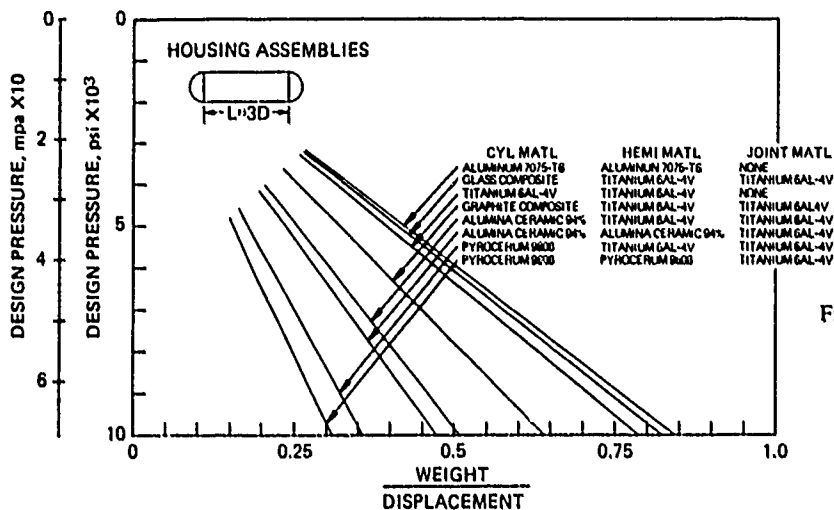


Fig. 1. Design Pressure versus Weight-to-Displacement Ratio of External Pressure Housings Assembled from Components Fabricated from Different Materials. The Design Stresses Are Based on Compressive Strength of Materials and Safety Factors Shown in Table 1.

While graphite fiber-reinforced plastic (GFRP) is certainly an acceptable material for construction of deep submergence pressure hulls, the high intrinsic cost of graphite fibers, tooling, and labor intensive fabrication process drive the price of custom-built GFRP housings beyond the reach of most ocean science research programs. In addition, the permeability of composite materials makes the application of GFRP pressure housings inappropriate for containment of sensitive electronics in equipment submerged in the ocean for a long time. Furthermore, the composite's low heat conductivity precludes the use of GFRP pressure housings for containment of propulsors generating a great quantity of heat.

Ceramics, on the other hand, not only possess the required structural properties for construction of external pressure housings with ≤ 0.5 weight-to-displacement ratio for service to 20,000 feet, but are also impermeable, corrosion resistant, and are good conductors of heat. Their sole shortcoming is brittleness, which caused many pressure housings to fail unexpectedly in service. To arrive at an operationally usable external pressure housing of ceramic material, several fabrication and design problems needed to be solved that have in the past worked against the acceptance of such housings by the ocean engineering community. These problems were economical fabrication of large ceramic cylinders, reliable mechanical joining of several ceramic cylinders into a cylindrical pressure housing of desired length, elimination of stress risers on the ceramic bearing surfaces between individual housing assembly components, secure mounting of payload components inside the ceramic housing, and protection against impact.

NOSC set out to demonstrate, during its program on the feasibility of utilizing ceramics for construction of deep submergence housings, that the problems of fabrication economics, cylinder joining techniques, elimination of stress risers, internal mounting of payload components, and protection against point impact were addressed and solved to varying degrees. These solutions were instrumental in the recommendation that the Navy seriously consider pressure housings fabricated from 94-percent alumina ceramics (Table II) for its deep submergence systems in the form of remotely operated vehicles, autonomous underwater vehicles, buoyancy propelled vehicles, and deep submergence oceanographic buoys.

PAST EXPERIENCES WITH GLASS AND CERAMIC PRESSURE HOUSINGS

The potential for using glass and ceramic materials for deep submergence vehicles has been discussed at technical symposia for more than 30 years. During that time, a sufficient number of papers, articles, and reports have been published to convince even the most conservative scientist or engineer that glass and ceramics do indeed possess the highest compressive strength and compressive strength-to-weight ratio of all commercially available structural materials.¹⁻¹⁶ The superior ability of these materials to withstand compressive stresses has been described and experimentally proven to such a point that it can be considered axiomatic and does not require any further discussion. This historical overview summarizes the progress made in finding applications for glass and ceramics in the deep submergence structures field and serves as a background for the NOSC research program in this field.

After the discovery in the 1950s that glass and ceramics possess outstanding compressive strength properties and, thus, would seem to be ideal choices for application in deep submergence structures, their use in that field seemed unlimited. They could be used for the hulls of deep-diving submersibles, torpedoes, mines, buoys, instrumentation capsules, remotely operated vehicles, permanent ocean-bottom installations, and as components of other deep submergence equipment where their high compressive strength and/or optical transparency make them desirable. No matter which application was considered, the problems of designing with unfamiliar materials, of fabricating complex and large structures, of joining glass and ceramics to each other and to dissimilar materials, of protecting against point and shock loading, of mounting securely components of payload on ceramic surfaces, and of establishing adequate quality control had to be tackled and, at least partially, solved.

From the very beginning, three distinct foci for research in the use of glass and ceramics for deep submergence applications became discernible. *First*, there was the research focused on the utilization of these materials for deep submergence buoys; *second*, the research aimed on the utilization of these materials for deep submergence antisubmarine warfare (ASW) torpedoes, BPVs, ROVs, and AUVs; and *third*, the research dedicated to the solution of the engineering problems standing in the way of using glass in manned submersible's hulls.

BUOYS

The first deep submergence use of glass and ceramics was in the construction of buoys. Buoys, generally made in spherical shapes, are very simple structures, as their main application is to provide buoyancy for diverse oceanographic systems. Since their main application is simply providing buoyancy to other oceanographic systems, the structures consist only of a hull, which is either a monolithic sphere or is made up of two hemispheres held together mechanically, by vacuum, or with the help of adhesives. Up to the present time, the spherical, deep submergence buoys have been fabricated commercially only in alumina ceramic or borosilicate glass with approximate weight-to-displacement ratios of .27 and .37 respectively. The buoys were made in diameters of 4 to 12 inches (alumina ceramic) and in 2 to 20 inches (glass). Both the alumina and glass buoys have an operational depth of 20,000 feet, while their implosion depths vary from 25,000 to 40,000 feet, depending on the buoy's size, type of joint, gasket material in the joint, and deviation from nominal diameter, wall thickness, and sphericity. Electrical bulkhead penetrators have been incorporated successfully into some of the glass buoys that serve as oceanographic instrumentation housings.

In addition to commercially made, off-the-shelf items, there were various custom-made items which have proven interesting in experiments. Chemically strengthened glass hemispheres, fabricated from HERCUGLASS^R have been found capable of withstanding somewhat higher compressive bearing stresses than semitempered or annealed borosilicate hemispheres. Also, custom-made spheres of PYROCERAM 9606^R glass ceramic have been found to withstand approximately the same compressive bearing stresses in lapped ceramic-to-ceramic joints as the alumina ceramic spheres.

There is no doubt that the future of glass and ceramics in applications to deep submergence buoys is assured. They are economical structural materials without peer in compressive strength and corrosion resistance, lending themselves to economical fabrication processes. With proper economic incentive, the glass and ceramic fabricators could produce spheres up to 5 feet in diameter. The technology for such sizes already exists; 44-inch buoys were already cast in 1968 by Corning Glass.

SUBMERSIBLES

The most glamorous application for which glass and ceramics have been considered by the Navy^{1,6,11} is that of a hull for deep submergence submersibles. It would provide a pressure hull that permits a man to descend to the very bottom of the oceans without recourse to flotation buoys, syntactic foam, or liquid-filled dirigibles for support of the pressure hull containing the crew. Glass, in particular, is desirable for such an application because it would not only provide the necessary strength for the hull, but would also provide almost unlimited visibility for the crew—a dream long cherished by submariners and undersea explorers.

Work initiated in 1962¹⁴ in this area has been unsuccessful. Models of surface-compressed glass spherical hulls with a joint, glass entry hatch, and metallic feedthrough were tested to more than a 20,000-foot depth without failure, while the compressive membrane stress in the hull was approximately 65,000 psi. The highlight of the application of glass to submersible hulls was the fabrication by Corning Glass of 56-inch-diameter, 1.5-inch-thick borosilicate glass hemispheres joined with a mechanical, watertight joint designed and machined from titanium alloy. The pressure hull was designed for a 20,000-foot depth, but cracks developed in the glass bearing surfaces after 100 pressure cycles to a 2000-foot depth. The disappointing test results and the lack of further funding terminated this ambitious program at Naval Undersea Center (NUC), Hawaii Laboratory.⁷

The demise of this glass submersible program was followed soon by the termination of another ambitious program at NUC, where the submersible DEEPVIEW was being assembled from a steel cylinder capped at one end with a 44.5-inch-diameter borosilicate glass hemisphere of 1.25-inch thickness. Based on successful pressure cycling to 5,000 psi with a 10-inch-diameter, 0.28-inch-thick borosilicate glass hemisphere on a concave steel ring covered with Fairprene gasket, the DEEPVIEW submersible was designed for a 5,000-foot depth.¹⁵ Unfortunately, the 44.5-inch-diameter hemispheres developed cracks in the equatorial bearing surface during pressure cycling to 2500 psi that resulted in spalling and, in some cases, in catastrophic failure.¹⁶

The inability of large borosilicate glass hemispheres to withstand repeatedly nominal membrane stresses of 20,000-psi magnitude without spalling or cracking at the joint does not indicate that this is a general physical limitation of glass. Rather it indicates that this is the result of poor quality control in glass formulation, casting, and grinding of bearing surfaces. Subsequent tests at NOSC have confirmed this. Optical quality, model-scale, glass 150-degree spherical sectors with optically plane, polished bearing surfaces resting on epoxy impregnated Kevlar 49 cloth gaskets have been shown to have a crack-free cyclic fatigue life in excess of 1000 cycles at -40,000-psi nominal membrane stress loading.¹⁷ The projected costs of fabricating such precision-ground optical glass full-scale spherical sectors would be, however, an order of magnitude higher than what was expended on the DEEPVIEW dome and joint assembly.

In general, the progress shown in applying glass to submarine hull construction has been exceedingly slow and has not yet passed the feasibility study state. A large amount of design, engineering, quality control, and production experience will have to be accumulated before glass and ceramic pressure hulls for manned submersibles can be considered reliable and safe structures.

HOUSINGS FOR ROVS, AUVs, AND BPVS

The fact that BPVs, torpedoes, AUVs, and ROVs are generally smaller, less costly, and unmanned explains why experimentation with these structures has been more extensive. The small size of the housings required by these vehicles places them definitely in the present state of the fabrication art so that many relatively inexpensive units can be built and experimented with for the same cost that would be involved in building even one small manned submersible hull. The fact that they are unmanned imposes fewer requirements on the reliability of the experimental design or fabrication method.

So far, the design and fabrication of pressure housings have pivoted around two dissimilar approaches to the problem of providing more buoyancy for the carrying of payloads. One approach relies on the design and fabrication of progressively larger hemispheres which, when secured together, provide increasingly greater payload-carrying buoyance. The other approach is to provide increasing amounts of buoyancy, not just by building increasingly larger monolithic hemispheres whose size is severely limited by the fabrication capability of the glass and ceramic industry, but by joining many cylindrical shell sections of size within the scope of industry's fabrication capability into longer cylindrical hulls.

The well-established tradition of using cylindrical shapes in torpedo hulls, BPVs and AUVs is grounded in the well-known property of cylindrical hulls to cause less hydrodynamic drag than spherical hulls of equal displacement. Additionally, their payload capacity can be varied according to payload requirements by varying the total number of cylindrical shell sections in the hull. As far as fabrication is concerned, it is technically more feasible and economical to fabricate many small-diameter cylindrical shell sections than few large-diameter spherical shell sections with equal payload.

There are, however, some problems associated with the use of cylindrical shell sections. The two major problems are the need to incorporate ribs into the cylindrical shells to make them elastically stable for abyssal depths and the need to provide the shell sections in the hull with mechanically reliable and structurally strong metallic joints. It requires the design of a joint that is not only compatible with the glass or ceramic that it joins, but that is also capable (1) of withstanding the large axial and radial stresses generated by hydrostatic pressure loading at operational depths, and (2) the high bending moments imposed on the hull during launching or retrieval operations at sea.

To prove the feasibility of the cylindrical housings for deep submergence service, it was necessary first to establish the feasibility of stiffening glass and ceramic shells with integral ribs and their fabrication (Fig. 2). For this purpose, a series of 8-inch-diameter ceramic rib-stiffened shells (Fig. 3) was designed and tested² in 1962. Alumina and PYROCERAM 9606R were chosen because of their high moduli of elasticity, high compressive strength, and above average flexural strength. It was found that the analytical formulas for calculation of elastic stability and stress distribution developed previously for rib-stiffened metallic cylinders apply in an identical manner to ceramic cylinders, except that in the ceramic cylinders all of the strains take place in the elastic strain region of the material, resulting in very high peak stresses at stress riser locations. The implosion resistance of integrally rib-stiffened ceramic cylinders to underwater shock waves and point impact was found to increase with depth, while that of metallic cylinders of identical dimensions decreased.²

The ceramic cylinders tested were only 8 inches in diameter; however, their buoyancy was of sufficient magnitude that if several cylindrical shell sections were assembled into a long hull¹ and fastened by mechanical joints, this hull could support oceanographic instrumentation in a buoyancy propelled vehicle. A breech-lock joint was developed for such purpose that satisfied both hydrodynamic and structural requirements of a deep submergence BPV (Fig. 4). A clamp-band joint was also developed for ceramic hulls that could be fitted with an external fairing or protective jacket to fair in the protruding clamp band (Fig. 5). Although experimental data generated by the author² and others⁴ show that highest bearing stresses are carried only by a ceramic-to-ceramic joint interface, its

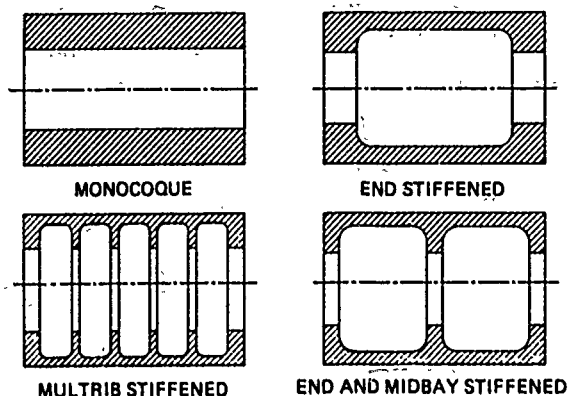


Fig. 2. Different Approaches to Providing the Desired Elastic Stability for Cylinders Under External Pressure Loading. The Monocoque Design Generates the Least Buoyancy; While the Multirib Stiffened Design Generates the Most Buoyancy for Cylinders With the Same External Dimensions and Elastic Stability; Fabricated From the Same Material.



Fig. 3. Multirib Stiffened 8-Inch-Diameter Cylinders Fabricated from PYROCERAM 9606R Glass Ceramic. Maximum Recorded Stress at 10,000 psi Pressure Was 185,000 psi. The Cylinder Imploded at 11,000 psi After First Being Subjected to 2000 Pressure Cycles from 0 to 9000 psi.²

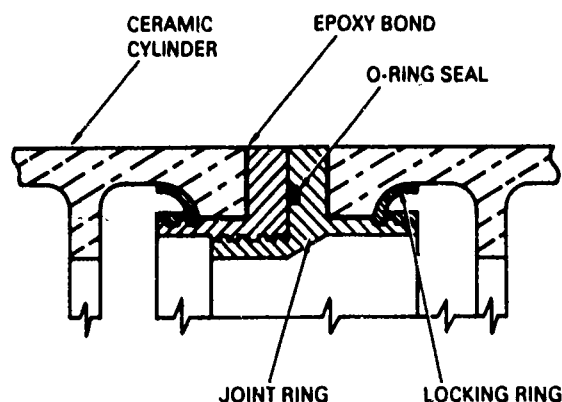


Fig. 4. Breech Lock Joint for Ceramic Cylinders Developed by the Applied Research Laboratory, PSU.²

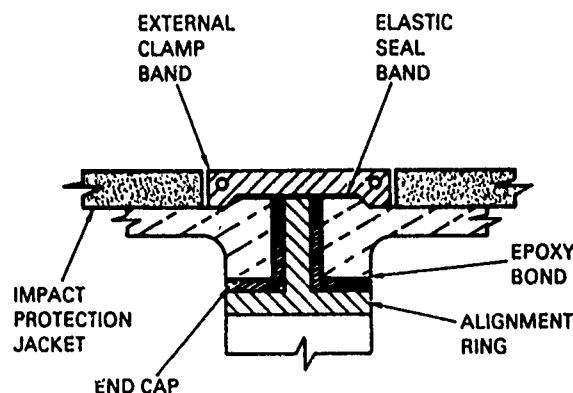


Fig. 5. Wedge Clamp Metallic Joint for Ceramic Cylinders Developed by the Applied Research Laboratory, PSU.²

application is limited as it requires plane, beveled, perfectly polished, hand-lapped mating surfaces for its successful operation. If ceramic hulls are ever to be fabricated economically, they must be mass produced and assembled on an interchangeable basis, where hand-lapped and hand-fitted ceramic-to-ceramic joints cannot be considered acceptable, although their bearing stress-carrying ability may be outstanding.

The mechanical joints found workable for deep submergence cylindrical ceramic hulls incorporate the bearing surfaces into a structural ring member of the joint, so the whole ring material can be considered as a gasket. Since the bearing stresses encountered at the ends of monocoque ceramic cylinders amount, at most, to only one-half of hoop stresses in the cylinder, the largest bearing stresses that can be expected will be less than 170,000 psi. In most designs, however, the maximum bearing stresses that generally have to be contended with are less than 100,000 psi. Such a bearing stress is low enough that a variety of metallic joint materials can withstand it. Unfortunately, under cyclic loading, fretting occurs due to relative motion between ceramic cylinder and metallic joint, causing the ceramic bearing surface to chip and subsequently crack. For this reason, some kind of a compliant but nonextruding gasket is required between the ceramic and metallic bearing surfaces.

Although different metals can be used in the joints for different glasses and ceramics to match their moduli of elasticity and/or coefficients of thermal expansion, titanium emerges as the optimum joint material for ceramic shells. Other factors, however, such as the need for lowest weight or economical fabrication of joints may dictate the use of high-strength aluminum.

Several gasket materials were also evaluated for placement between the ceramic bearing surfaces and metallic joints. Thin Nylon fiber-reinforced Neoprene and Kevlar fiber-reinforced epoxy gaskets have been found to perform satisfactorily only at 20,000- and 40,000-psi bearing stress levels, respectively. At higher compressive stress levels, the gaskets wear out rapidly, sometimes in as little as two or three pressure cycles. At bearing stresses above 40,000 psi, the only viable gasket materials are a cast in place thin layer of epoxy adhesive that is restrained from extruding by the adherence of epoxy to the mating surfaces of the joint, or a metallic washer brazed to the metalized ceramic bearing surface.

Once the design and fabrication of cylindrical rib-stiffened shells and their mechanical joints were proven feasible, the design and fabrication of glass and ceramic capsules capable of carrying commercially available oceanographic instrumentation to abyssal depth could proceed. In the design for external pressure shells, there are basically two approaches to the utilization of a given structural material. One approach, generally applied to vehicles where strength-to-weight ratio of the hull, regardless of cost, is the overriding requirement, relies on exhaustive quality control of the material and painstaking adherence to very tight dimensional tolerances. Assured of the quality of the material and tolerances in regard to roundness and uniformity of wall thickness, the designer can set the operational depth of the structure as close to its theoretical collapse pressure as he desires, in which case the difference between the operational depth and the collapse depth is a small, but true safety margin. The second approach is a compromise between fabrication cost and the desirable strength-to-weight ratio of the structure. In such a case, quality control and dimensional tolerances are relaxed sufficiently to bring the cost of fabrication down to be competitive with metallic or GFRP housings, and yet the tolerances cannot be set so loose as to divest the structure of the attractive strength-to-weight ratio imparted to it by the good mechanical properties of the material.

Both approaches to the design of external pressure vessels have their place in engineering practice, and both have been applied to the design and fabrication of the first experimental glass and ceramic oceanographic buoyancy propelled vehicles.

DIVEAR. The hull of the DIVEAR (Diving Instrumentation Vehicle for Environmental and Acoustic Research) was designed for a low nominal stress level of 87,000 psi at 20,000-foot maximum depth so that considerable deviation from nominal dimensions could be tolerated without failure of the hull at its operational depth.⁸ The hull of DIVEAR was a borosilicate glass, integrally rib-stiffened cylinder (Fig. 6) with an outside diameter (OD) of 16 inches and 58 inches in length. The glass hull was fabricated inexpensively by fusing several tubular glass sections that had been hand-blown separately within an external cylindrical mold. Since each of the tubular glass sections was actually a flat-bottom jar, whose bottom had been only partially removed, the remaining bottom rim served, after extensive flame working on lathe, as a rib for stiffening the hull.

After all of the glass sections were fused together, the ends of the hull were ground and the whole structure annealed. No special emphasis was placed on the elimination of air bubbles or external ridges in the glass welds. Thus, the finished glass structure had pronounced ridges and some trapped air bubbles in the areas of the fusion welds. The ends of the hull were capped with 7075-T6 aluminum hemispheres machined to accept internal and external attachments, plus several feedthroughs. To keep the fabrication cost down to an absolute minimum, extremely generous dimensional tolerances were permitted, which varied from one-eighth to three-eighths inch, depending on the dimension described.

During hydrostatic testing, the glass cylinder failed at 4,600-psi hydrostatic pressure due to general elastic instability. Posttest failure analysis discovered that the manufacturer exceeded the dimensional tolerances by making the ribs too small. In future designs of hand-blown cylinders, the height and width of circumferential stiffeners inside cylinders shown on the drawing should exceed by 100 percent the calculated nominal values. If this precaution is taken, the hand-blown fabrication process can produce inexpensively cylindrical pressure hulls of 12- to 24-inches in diameter and 1/2- to 1-inch wall thickness for service to 20,000 feet. The weight-to-displacement ratios of cylinders dimensioned and fabricated in this way are, however, only marginally better than that of 7075-T6 aluminum. In this case the only incentive for using glass is its corrosion resistance and its significantly lower cost than that of a tubular aluminum forging with subsequently machined ribs. The higher cost of aluminum is offset here, however, by absence of structural failure risk generally associated with glass housings, and, thus, aluminum is preferred over glass for external pressure housings with weight-to-displacement ratios of ≥ 0.75 .

BENTHOS. The approach to the design and fabrication of ceramic hull for BENTHOS buoyancy propelled vehicle⁸ has been just the opposite to that used in the design and fabrication of DIVEAR. The requirements were, in this case, for a high-speed, hydrodynamically streamlined, buoyancy propelled vehicle having the lowest attainable weight-to-displacement ratio capable of sustaining oceanographic missions to depths of 20,000 feet. To accomplish these requirements, the hull shape and finish had to present minimum hydrodynamic drag, while the design stresses had to be of maximum allowable magnitude to give the vehicle the utmost in payload-carrying capability for its displacement.

To meet the low hydrodynamic drag requirements, a cylindrical shape was chosen. It was capped at the front with a hemispherical nose and at the rear with a gradually tapered afterbody, terminating in shrouded cruciform fins (Fig. 7). Both the ceramic and metallic sections of the hull were fabricated to 32 RMS finish requirements to insure low hydrodynamic drag. To make the interior of the vehicle accessible and the fabrication costs more economical, the hull was conceived as an assembly of five ceramic and one metallic shell sections joined together by titanium breech-lock joints. The fact that BENTHOS was to provide maximum buoyancy for a given



Fig. 6. The 16-Inch-OD by 14.75-Inches-ID by 59.5-Inches-Long Borosilicate Cylinder with 0.875 High Integral Ribs Spaced 7 Inches Apart, Hand-Blown from Borosilicate Glass by Corning Glass Works in 1962.⁸

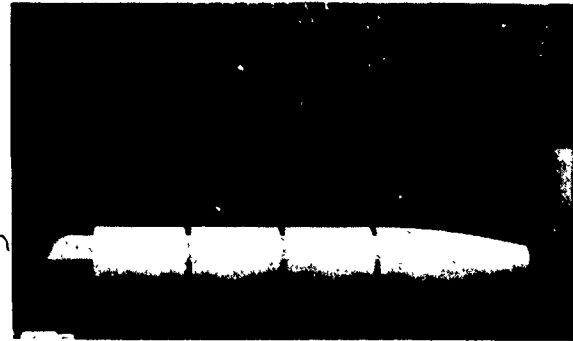


Fig. 7. BENTHOS Buoyancy Propelled Vehicle with 12-Inch OD and 97 Inches in Length. The Hull Assembly Consists of Four Cylindrical Sections and a Single Tapered Section.⁸

displacement at 20,000 feet dictated that only materials with high compressive strength-to-weight ratios and reliable fabrication methods with tight dimensional control could be employed in the construction of this vehicle. Only PYROCERAM 9606R, 99-percent alumina, and 99-percent beryllia ceramics were found capable of withstanding consistently compressive design stresses at the 200,000-psi level. Since the specific gravity of PYROCERAM 9606R is 2.61 versus 3.85 of alumina ceramic, it was chosen for fabrication of BENTHOS. Beryllium oxide ceramics were not selected for this application because of their high cost and the lack of adequate facilities for fabricating such large cylindrical sections economically.

During the fabrication of the shell sections, good quality material control was exercised. Unlike any other ceramic material, PYROCERAM ceramics are transparent in the initial stages of fabrication, and for this reason visual inspection methods were used to examine them for internal defects. Optical glass-melting techniques were used to assure uniform composition, constant density, freedom from bubbles and striations, and uniform electrical properties. Subsequent controlled heat-treating cycles activated special nucleating agents in the body to produce a fine-grained, uniform crystal growth. No problems were encountered in the firing of the thick-walled cylindrical, hemispherical, and conical blanks, from which excess material was subsequently removed by grinding. By means of a tracer lathe and proper guide templates, the excess material was removed until a rib-stiffened structure with integral ribs resulted.

After fabrication of Benthos components, one of the cylindrical sections with 0.34 weight-to-displacement ratio (Fig. 8) was strain-gaged and tested to destruction under external hydrostatic loading. For this test the ends of the cylinder were closed off with thick 7075-T6 aluminum alloy plates. Neoprene-impregnated nylon cloth served as bearing gaskets between the ceramic and metallic plates. Maximum stress of ~283,000 psi was recorded at 13,000-psi implosion pressure. The high compressive stress and implosion pressure validated the design, PYROCERAM 9606R material, and fabrication approach. The remainder of the BENTHOS program did not fare so well. When the assembled vehicle was hydrostatically proof tested, the end ribs in some of the cylinders cracked prior to reaching design depth. Inspection of cracked components revealed that the ceramic bearing surfaces on some cylindrical sections deviated from a true plane, resulting in unacceptably high bearing stresses at the circular line of contact between the ceramic and metallic joints. Thus ended the most ambitious construction program of ceramic vehicles undertaken to date.

Unfortunately, the DIVEAR and BENTHOS housings did not reach their most important goal: to become operational buoyancy propelled vehicles with 20,000-foot design depth. Both programs, however, made very valuable contributions to the engineering knowledge and fabrication technology of large glass and ceramic housings.

SUMMARY OF PAST EXPERIENCE

1. Ceramics and glass appear to be suitable materials for construction of external pressure housings with low weight-to-displacement ratios. Weight-to-displacement ratios of less than 0.5 have been achieved with cylindrical and 0.3 with spherical housings. Ceramic (high-purity alumina or PYROCERAM 9606) and glass housings have withstood peak compressive stresses of 300,000- and 100,000-psi magnitude, respectively, without fracture initiation.

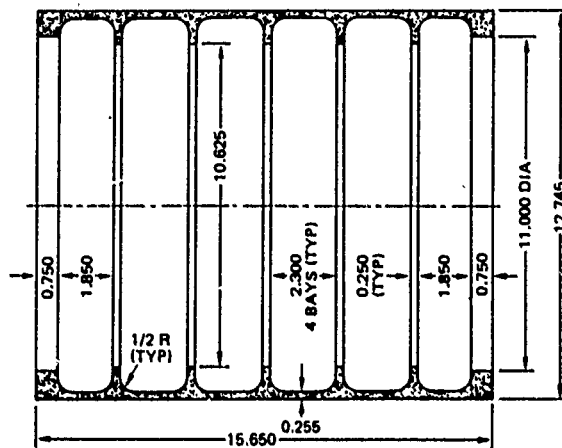


Fig. 8. Typical Cylindrical Shell Section of BENTHOS BPV Fabricated from PYROCERAM 9606R by Corning Glass Works in 1964. The 0.32 Weight-to-Displacement Ratio of This Shell Section Represents the Lowest Ratio Ever Achieved by a Cylinder with Critical Pressure of 13,000 psi.⁹

2. High-purity alumina and beryllia as well as glass ceramics (i.e., PYROCERAM and CERVIT) have consistently demonstrated a tolerance to higher bearing stresses at joints than borosilicate glass. Bearing stresses of up to 200,000 psi on surfaces in contact with bare metallic joints have been withstood by ceramics and only 100,000 psi by glass prior to failure during a single pressurization. Cyclic fatigue life in excess of 1,000 pressurizations was achieved only when the compressive bearing stresses at the joint were less than 50,000 psi for ceramics and 25,000 psi for glass. The bearing stress for glass in cyclic applications can be increased to 50,000 psi if the surfaces on the glass housing are chemically pre-compressed by ion exchange.
3. Bonding of ceramic components has been successfully achieved with cyanoacrylate and epoxy adhesives. Compressive bearing stresses in excess of 300,000 psi have been carried by the bonded joints without delamination or squeezing out. Similar results have been obtained by the brazing of metalized bearing surfaces on ceramics.
4. The fracture resistance of ceramics and glass to underwater shock waves and point impacts increases with depth.¹²
5. Both glass and ceramics exhibit static and cyclic fatigue under tensile strain.
6. Penetrations through glass or ceramic hemispheres do not require reinforcement around the opening, provided that the magnitude of compressive stress does not exceed 150,000 psi.

NOSC CERAMIC HOUSING PROGRAM

OBJECTIVES

The deep submergence ceramic housing feasibility study at NOSC that was initiated in FY 87 has two primary objectives:

- *Demonstrate* the feasibility of massive ceramic housings for deep submergence service that can be fabricated economically by existing manufacturing technology in sizes applicable to BPVs, ROVs, and AUVs.
- *Provide* the Navy with at least one experimentally validated design of ceramic pressure housing that is economical to fabricate and can be readily extrapolated to larger diameter and/or expanded in length to accommodate larger payloads.

APPROACH

The objectives of the program were defined analytically and validated experimentally by fabrication and testing of model-scale and full-scale ceramic components of the pressure housing. Analytical calculations were used to size the components, finite element computer programs to predict the structure response of the housing assembly to external pressure, and strain-gage data analysis to confirm the predicted magnitude of stresses.

SCOPE

The scope of the program was limited to:

- Monocoque cylinder design
- 6- and 12-inch diameter sizes
- Monolithic and polyolithic constructions of cylinders
- 99- and 94-percent alumina compositions
- Titanium and aluminum joint rings
- Ceramic and titanium hemispherical bulkheads with penetrations.

DESIGN DEFINITION

The design selected for the NOSC ceramic housing differed radically from previous designs described in technical literature. The glass and ceramic cylindrical hull concepts studied in the past and described in technical literature relied on integral ribs for providing the cylindrical housing with elastic stability at design depth without increasing its wall thickness and the associated weight. By proper dimensioning and spacing of circular ribs, the stresses in the rib-stiffened cylinders could be raised to any selected design stress level without reducing the critical instability pressure of the cylinders below acceptable levels.

The primary advantage of *integral ribs* for stiffening of cylinders is optimization of structural performance with associated reduction of weight. The disadvantages of integral ribs are (1) high fabrication costs due to pressing and firing of extra heavy walled cylinders, followed by extensive grinding away of excess ceramic between the ribs, and (2) reduction of interior payload envelope diameter/volume by protruding ring stiffeners. Because of these disadvantages inherent to integral rib-stiffened cylinders, this design can be economically and operationally justified only if the mission scenario of the vehicle calls for a ≤ 0.4 weight-to-displacement ratio.

The design selected for the NOSC ceramic housing utilizes monocoque cylinders with $1.5 D_0$ length supported radially at the ends by hemispherical bulkheads, or *removable* metallic ring stiffeners (Fig. 9). Monocoque cylinders with $3D$ length are also supported at midbay with a metallic stiffener held in place by epoxy adhesive. The removable ring stiffeners at the ends of cylinders align the ends of the cylinders during assembly, contain the o-ring seals, and serve as components of mechanical joints for fastening together individual cylinders and hemispheres into a single housing assembly.

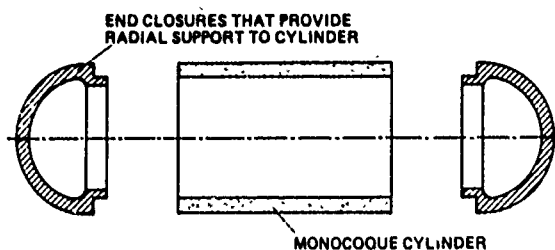


Fig. 9a. Typical Approach to Raising the Elastic Stability of an External Pressure Housing Assembly Consisting of a *Single* Monocoque Cylinder Utilizes Radial Support Provided to the Cylinder Ends by Compliant End Closures.

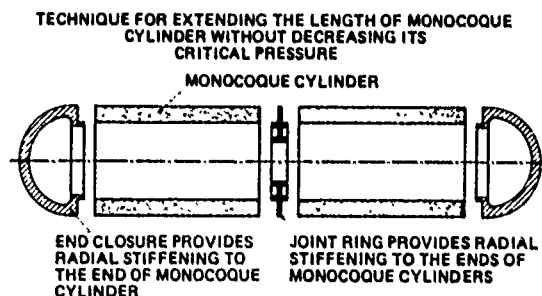


Fig. 9b. Design Approach Developed by NOSC to Raising the Elastic Stability of External Housing Assemblies Made Up of *Many* Monocoque Cylinders Utilizes Radial Support Provided by Compliant Joint Stiffeners Between Individual Cylinders.

The advantages of monocoque cylinders supported radially at the ends by removable metallic stiffeners are (1) *reduced fabrication cost* resulting from pressing and firing cylindrical castings with thinner walls, followed by only minor removal of material from the exterior and interior surfaces of the cylinder to specified diameters, and (2) *increase in the internal diameter* and volume of internal payload envelope.

Alumina ceramics were selected for fabrication of ceramic components because of their low intrinsic cost, high modulus of elasticity, high compressive strength, and/or modulus of rupture that exceed that of glass. The thickness, diameter, and length of individual monocoque cylinders were chosen on the basis of maximum allowable stress and critical buckling pressure (Fig. 10). The maximum compressive design stress was *not* to exceed $\sim 150,000$ psi, and the critical pressure due to general buckling *between* joint supports was to exceed $18,000$ psi. The thickness of the ceramic hemispheres was based on maximum allowable stress of $\sim 175,000$ psi around penetrations and critical pressure in excess of $18,000$ psi. Model-scale ceramic cylinders were fabricated from both 99- and 94-percent alumina, while the full-scale sections only from the more economical 94-percent alumina.

The fretting of ceramic bearing surface at the joints present in some past designs described in the literature¹⁸ was eliminated by bonding the ends of cylinders and ceramic hemispheres to metallic caps (Fig. 11). In this arrangement the direct radial and axial contacts between the ceramic cylinder and the metallic joint stiffener ring still take place, but now it is the metallic cap that slides upon

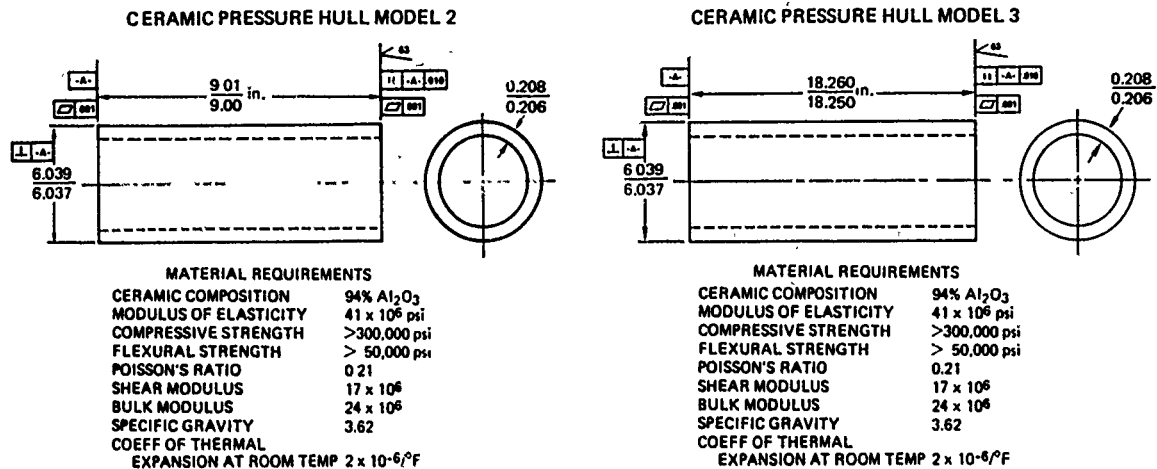


Fig. 10. Model-Scale Monocoque Cylinders of 94-Percent Alumina Ceramic Utilized in NOSC Ceramic Materials Program.

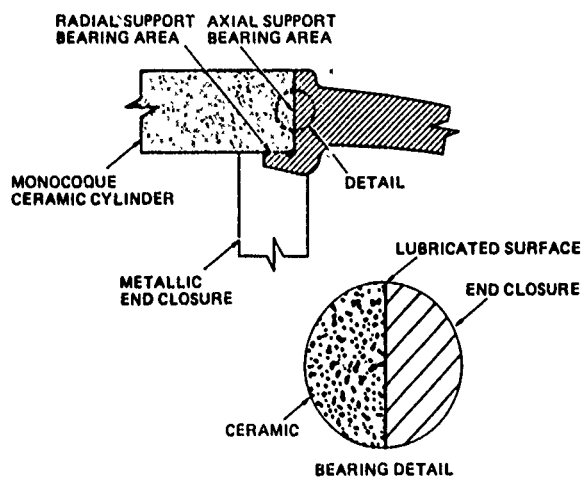


Fig. 11a. *Direct Contact* Between Ceramic and Metallic Components of External Pressure Housing Utilized in the First-Generation NOSC Designs Produced Fretting of Ceramic Bearing Surfaces After Less Than 50 Pressure Cycles.¹⁸

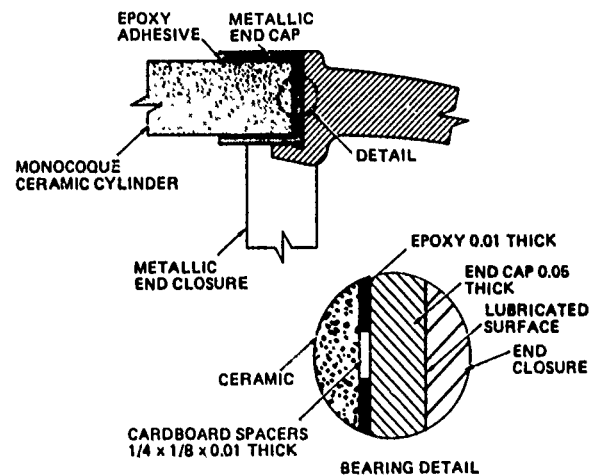


Fig. 11b. *Encapsulation* of Ceramic Bearing Surfaces Performed by NOSC on Their Second-Generation Ceramic Housing Designs. Encapsulation Extended the Fatigue Life of Ceramic Bearing Surfaces Beyond 2000 Pressure Cycles.¹⁹

the metallic joint stiffener and not the ceramic (Fig. 12). Since the maximum compressive bearing stress acting on the cap does not exceed -70,000-psi, the cap can be machined from either high-strength aluminum or titanium alloy. Caps were machined from both materials and bonded with epoxy resin to model-scale ceramic cylinders for subsequent pressure testing. Only titanium caps were utilized with full-scale ceramic housing to prevent corrosion when the ceramic housing is incorporated into an operational vehicle.

The joint stiffeners were designed to raise the 3900-psi critical pressure of an infinitely long monocoque cylinder with $t/D_0 = 0.034$ to the 11,250-psi minimum critical design pressure specified for the whole housing assembly by radially supporting the ends of cylinders encased in metallic caps. Since the design stresses in the joint stiffener do not exceed at any location compressive stress of -70,000 psi or tensile stress of +30,000 psi, the joint stiffener could be designed for and machined from either high-strength aluminum or titanium. To evaluate both materials for this application, model- and full-scale joint stiffeners were designed and fabricated in 7075 T6 aluminum and Ti-6AL-4VA titanium alloys (Figs. 13 and 14). Ceramic housings incorporating these stiffeners were subsequently proof tested and pressure cycled to design depth. To insure uniformity of radial contraction by ceramic cylinders under hydrostatic loading, the radial clearance between the joint stiffener flanges and the metallic end caps on the cylinder was specified *not to exceed* 0.001 R_0 inches.

The stiffeners for supporting ceramic cylinders with $3D_0$ length at midbay were similar to the joint stiffeners (Fig. 14). These midbay stiffeners were designed only in high-strength aluminum, as titanium did not offer any advantages over aluminum for this application where the stiffener is not exposed to seawater. Epoxy adhesive was used to keep the midbay stiffener in place and to serve as cast in place bearing gasket between metallic stiffener and ceramic cylinder. (Fig. 15).

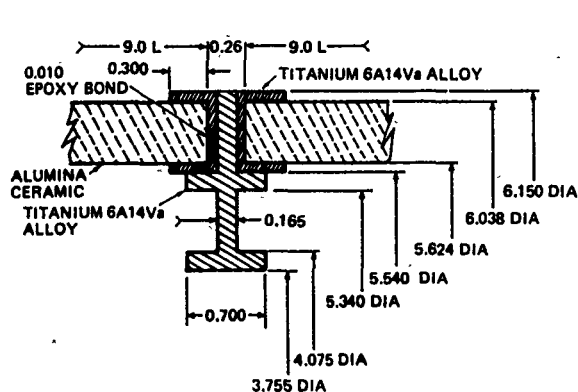


Fig. 12. Components of NOSC Joint Ring Stiffener Design for Model-Scale Ceramic Housing Assembly.

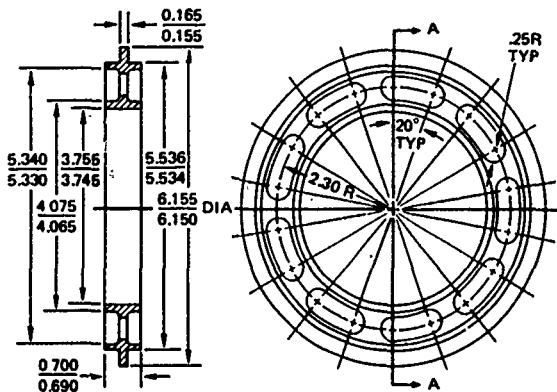


Fig. 13. Titanium Joint Ring Stiffener for Providing Radial Support to NOSC Model-Scale Monocoque Ceramic Cylinders of 1.5D Length.

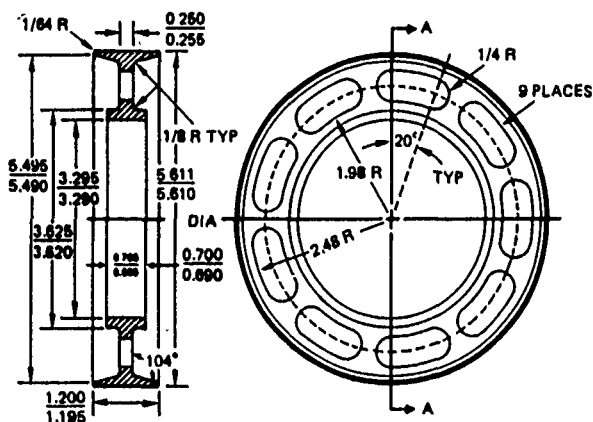


Fig. 14. Aluminum Midbay Stiffener for Providing Radial Support to NOSC Model-Scale Monocoque Ceramic Cylinders of 3D₀ Length.

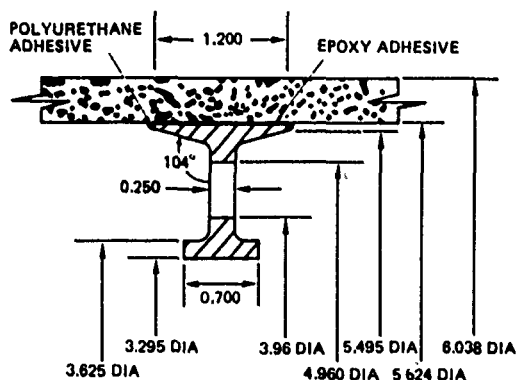


Fig. 15. Typical Installation of Midbay Stiffener Inside a NOSC Model-Scale Ceramic Cylinder.

The bulkheads, which both provide radial support and close off the ends of ceramic cylinders, were designed in titanium and in ceramic to meet the 11,250-psi minimum critical pressure requirement of the housing assembly (Fig. 16). Only titanium bulkheads were provided for the 6-inch ceramic cylinders, while the 12-inch ceramic cylinders were equipped with both titanium and ceramic bulkheads. The weight of ceramic hemisphere assemblies (ceramic hemisphere with titanium equatorial joint) was approximately 50 percent less than of titanium hemispheres.

TEST SPECIMENS

As test specimens served the following pressure housing components:

MODEL SCALE

- Two 6-inch-diameter monocoque *monolithic* cylinders of 99-percent alumina with $t/D_0 = 0.031$
- Two 6-inch-diameter monocoque *polyolithic* cylinders of 99-percent alumina with $t/D_0 = 0.031$
- Two 6.037-inch-diameter monocoque *monolithic* cylinders of 94-percent alumina with $t/D_0 = 0.034$
- Two 2.951-inch spherical radius hemispheres of Ti-6Al-4VA alloy with $t/R_0 = 0.046$
- Joint stiffeners, midbay stiffeners, and end caps fabricated either from aluminum or titanium.

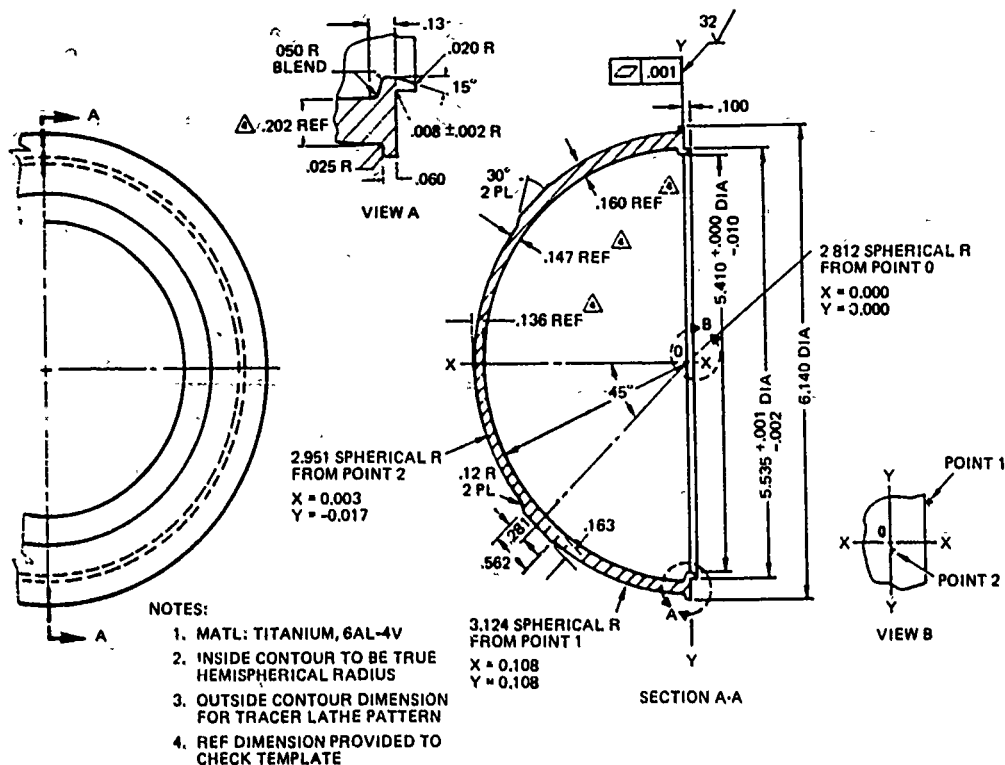


Fig. 16a. Titanium End Closure for NOSC Model-Scale Housings.

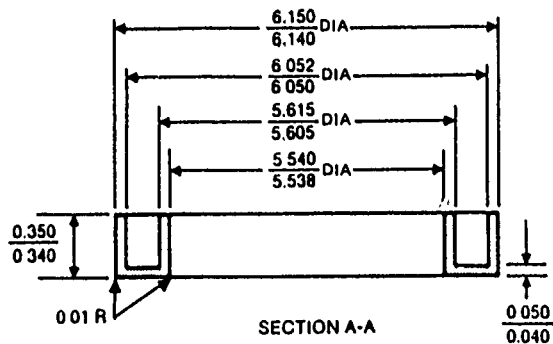


Fig. 16b. Titanium End Cap Ring for Protection of Ceramic Bearing Surfaces Against Contact with Metallic End Closure or Joint Stiffeners. The Cap Is Filled with Epoxy Resin Prior to Placing It Over the End of Ceramic Cylinder.

FULL SCALE

- Four 12-inch-diameter monocoque *monolithic* cylinders of 94-percent alumina with $t/D_0 = 0.034$
- Four 5.9-inch spherical radius hemispheres of 94-percent alumina with $t/R_0 = 0.034$
- Two 5.9 spherical radius hemispheres of Ti-6Al-4V alloy with $t/R_0 = .05$
- Joint stiffeners, end caps, split clamps, penetration inserts, and bulkhead connectors fabricated either from aluminum or titanium.

TEST PLAN

The test plan called for *validating the NOSC housing design concept* and *qualifying the materials of construction* by fabrication and extensive pressure cycling of 6-inch-diameter housings. Only if the model-scale housings passed the pressure cycling tests satisfactorily was the housing design and the materials from which it was fabricated considered acceptable for application to larger diameter full-scale housings.

The scalability of *NOSC housing components* without any reduction of their structural performance was to be established by fabricating 12-inch pressure housings that were scaled-up versions of the 6-inch housings and subjecting them to the same pressure tests as the 6-inch housings. If the structural response of the 12-inch-diameter housings was found to be identical to that of 6-inch-diameter housings, NOSC would recommend applying this housing design to pressure hulls with diameters in the 20- to 30-inch range.

TEST SCHEDULE

To establish confidence in the design and materials of the pressure housing components, their structural performance was evaluated by testing of complete pressure housings and not of individual components. Only by testing of complete assemblies could the structural interaction at the joints between individual components under hydrostatic loading be realistically evaluated.

To minimize damage to assembled housing components and financial loss to the program due to unexpected implosion of a single component during testing, the preliminary evaluation of component design was preferably performed on a model scale 6-inch housing containing only one previously untested component. As a result of this approach to testing, the complexity of test assemblies progressively increased from single model-scale monolithic or polyolithic cylinders closed off with titanium end closures, to four model-scale monolithic cylinders joined by three titanium or aluminum joint stiffeners (Figs. 17 through 20).



Fig. 17. Test Assembly for Pressure Testing Model-Scale Monocoque *Monolithic* Ceramic Cylinders.



Fig. 18. Test Assembly for Pressure Testing Model-Scale Monocoque *Polyolithic* Ceramic Cylinders.

A similar procedure was followed when testing the components of the full-scale 12-inch-diameter assembly. The first housing in the full-size test series consisted of only one 12-inch OD by 18-inch length cylinder closed off on one end with a titanium and on the other end with a ceramic hemisphere (Figs. 21, 22, and 23). The last housing incorporated four ceramic cylinders and two ceramic hemispheres supported by three removable joint stiffeners and held together by four split clamp bands (Figs. 24, 25, 26, and 27). Titanium penetration liners supported by plastic bearing gaskets were placed in each penetration through ceramic hemispheres to protect the ceramic from contact with steel bulkhead penetrators that could initiate fracture in the ceramic.

TEST PROCEDURES

The evaluation of each 6-inch model-scale housing consisted of instrumenting it with strain gages and subjecting it to 10,000-psi proof pressure, followed by 1000 pressure cycles to 9000-psi service pressure. The proof test consisted of 4 hours of sustained pressure, while each service cycle consisted of 15 minutes of sustained pressurization followed by 15 minutes of relaxation.

The 12-inch full-size housings were tested in identical manner as the model-scale housings, except that they were each, after proof testing, subjected individually to only 50 pressure cycles. After completion of the testing, each housing component was inspected visually for wear, fretting, or incipient cracks. Following the visual inspection, a dye penetrant test was performed on all edges of penetrations in the ceramic hemispheres.



Fig. 19. Components of the Longest NOSC Model-Scale Ceramic Housing Test Assembly. Note the Joint Stiffeners That Are Used in Each Joint Between Individual Cylinders.



Fig. 20. The 6-Inch-OD by 42.5-Inch Length Model-Scale Pressure Housing Test Assembly Made Up of Components Shown in Fig. 19.

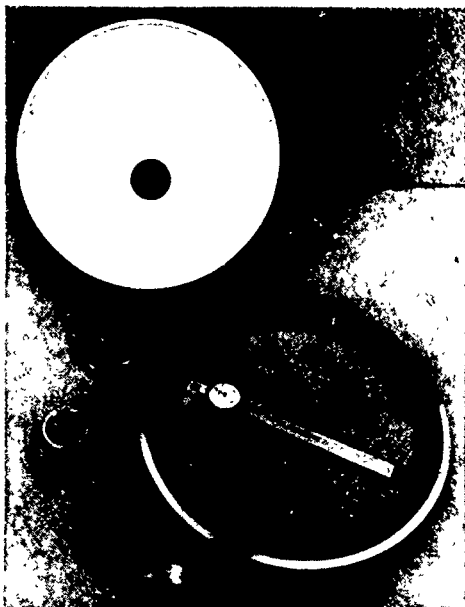


Fig. 21. Components of the Ceramic End Closure for NOSC Full-Scale Ceramic Housing.

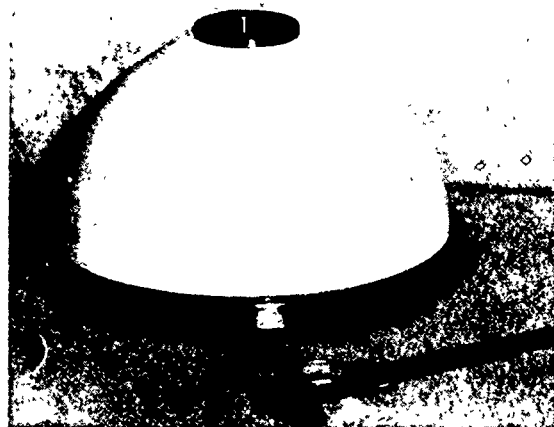


Fig. 22. Full-Scale Ceramic End Closure Assembled from Components shown in Fig. 21.



Fig. 23. Full-Scale Test Assembly Equipped with Ceramic End Closure of Fig. 27 After Being Instrumented with Electric Resistance Strain gages for Measurement of Strains During Pressure Testing.

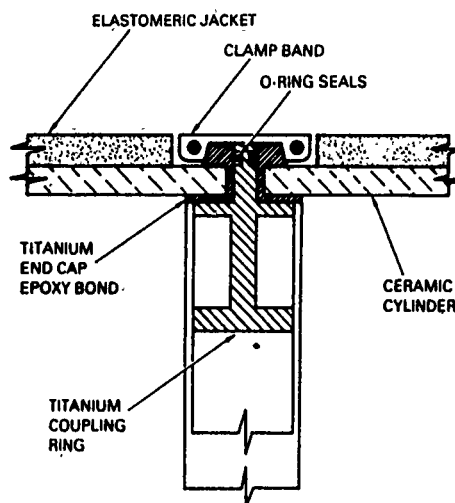


Fig. 24. Mechanical Joint Developed by NOSC For (1) Alignment, (2) Radial Support, and (3) Clamping of Adjacent Ceramic Cylinders in the 12-Inch OD Housing.

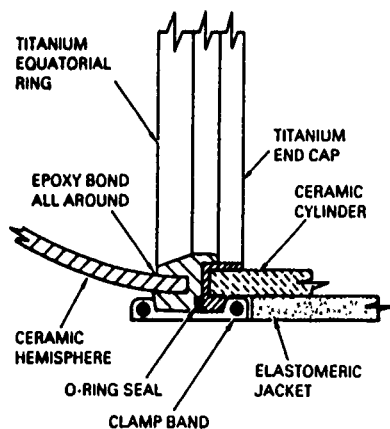


Fig. 25. Modified NOSC Mechanical Joint for (1) Alignment, (2) Radial Support, and (3) Clamping of the Ceramic Cylinder to Ceramic Hemisphere in the 12-inch-OD Housing.

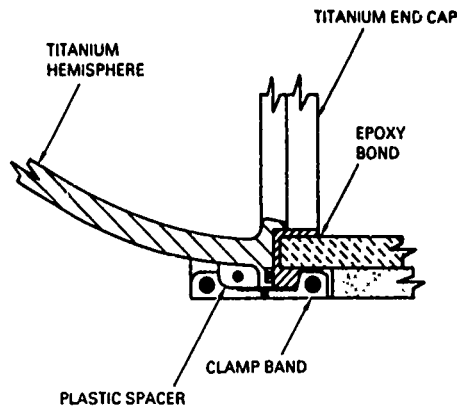


Fig. 26. Modified NOSC Mechanical Joint for (1) Alignment, (2) Radial Support, and (3) Clamping of the Ceramic Cylinder to Titanium End Closure in the 12-Inch-OD Housing.

TEST RESULTS

1. All of the 6-inch-diameter, 99- and 94-percent alumina ceramic cylinders with $1.5 D_0$ length withstood individually proof tests to 10,000 psi and 1000 pressure cycles to 9000-psi design pressure, when radially supported at the ends by titanium hemispherical bulkheads. These ceramic cylinders withstood, subsequently, the same pressure tests when assembled into housings made up of two, three, or four sections joined together by removable metallic joint ring stiffeners and closed off with titanium hemispherical bulkheads.¹⁹
2. The 6-inch-diameter alumina ceramic cylinders with $3 D_0$ length also withstood the proof and cyclic pressure tests when, in addition to the removable joint stiffeners, they were supported at midbay by metallic ring stiffeners bonded to the interior surface of the cylinders with epoxy adhesive.

3. The 6-inch-diameter ceramic cylinders of *polyolithic* construction performed structurally under hydrostatic loading in the same manner as *monolithic* cylinders with identical dimensions and ceramic composition. There was no separation at the brazed joints between mating ceramic rings.¹⁹
4. The 12-inch-diameter, 94-percent alumina ceramic cylinders with 1.5 D_0 length withstood proof tests to 10,000 psi and cyclic pressure tests to 9000 psi when tested individually, or in housing assemblies made up of two, three, or four sections supported by removable metallic joint stiffeners and secured by metallic clamp bands. As bulkheads served hemispheres fabricated either from titanium or ceramic.
5. Stresses in ceramic and metallic components of the housings did not exceed design stresses. The maximum stresses recorded during proof testing to 10,000 psi were -138,000 psi on interior surface of ceramic cylinder at midbay, -141,000 psi on interior surface of ceramic hemisphere, -100,000 psi on interior surface of titanium hemisphere, and -69,000/+26,000 on titanium joint stiffener. Around penetrations in the hemispheres, the hoop stresses were -156,000 psi at the openings reinforced by increased wall thickness.
6. The weight-to-displacement ratio of ceramic housings composed of 94-percent alumina ceramic cylinders and hemispheres joined by titanium ring stiffeners and fastened together by aluminum split wedge clamp band is 0.58. When titanium hemispheres are substituted for ceramic hemisphere, the ratio increases to 0.6.

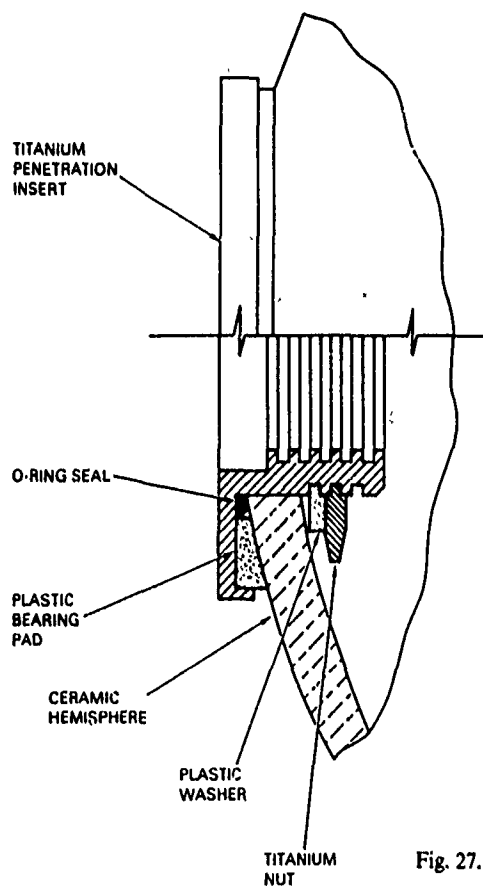


Fig. 27. Penetration Protectors for Ceramic End Closures.

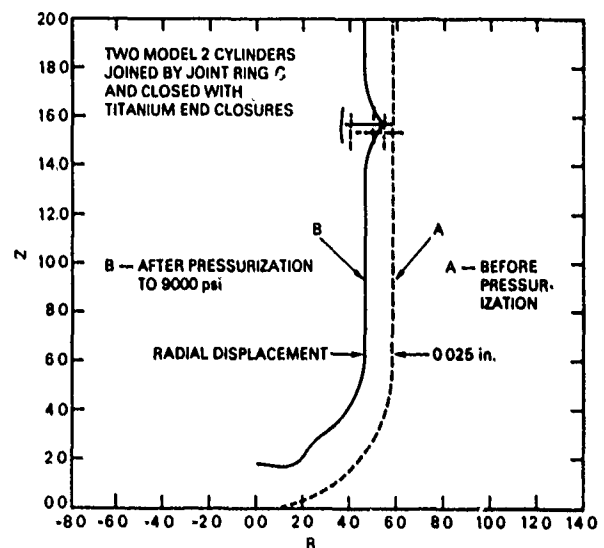


Fig. 28. Radial Support Provided to the Monolithic Ceramic Cylinders by Titanium Joint Stiffeners. Note the Magnitude of Radial Displacements.

FINDINGS

Design. NOSC pressure housings assembled from (1) ceramic monocoque cylinders individually supported at midbay and/or at the ends by metallic ring stiffeners and (2) ceramic or metallic hemispheres attached to the cylinders by split wedge band clamps perform like a monolithic, rib-stiffened pressure housing.

Metallic ring stiffeners provide adequate radial support for monocoque ceramic cylinders. A ring stiffener bonded to the interior surface of monocoque ceramic cylinder at midbay allows unrestricted access to one half of the cylinder interior from either end. Removable joint stiffeners allow unrestricted access to the full length of cylindrical housing's interior from either end for placement, adjustment, maintenance, and repair of electronic or hydraulic components of the payload.

Encasing of ceramic bearing surfaces on cylinders and hemispheres with closely fitting metallic cap rings that are securely bonded to ceramic surfaces with epoxy resin prevents initiation of cracks in these surfaces during cyclic application of *bearing stress* in the 0- to -70,000-psi range.

Aluminum ceramic components with 94- or 99-percent composition have a cyclic fatigue life in excess of 2000 load applications when the maximum value of *compressive stress* in the ceramic component is < 160,000 psi.

Penetrations can be incorporated into the titanium or ceramic hemispheres, providing that the openings are reinforced by local increase in wall thickness. Plastic bearing gaskets must be utilized with metallic bulkhead penetrators in ceramic hemispheres to prevent direct bearing contact between the metallic head of the penetrator and the external surface of ceramic hemisphere.

Fabrication. The acquisition cost of the custom-built NOSC monocoque ceramic housing for 20,000-foot-depth service with 12 inches OD and 86 inches overall length (assembled from four monolithic, monocoque, 94-percent alumina ceramic cylinders, two ceramic hemispheres, three titanium joint stiffeners, and eight titanium end caps and split band clamps) is *significantly less* than the cost of a custom-built housing with *same* depth rating and positive buoyancy fabricated from any other premium structural material (i.e., graphite or glass fiber reinforced plastic, titanium, aluminum, or glass ceramic).

A full-scale 12-inch-OD by 18-inch-long ceramic cylinder is shown in figure 29. The properties of alumina ceramics used in structural applications are presented in Table 2.



Fig. 29. Full-scale 12-inch-OD by 18-inch-long ceramic cylinder prior to attachment of end cap rings.

Table 2. Alumina Ceramics for Structural Applications.

PROPERTIES*	UNITS	AD-94 Nom 94% Al ₂ O ₃	AD-96 Nom 96% Al ₂ O ₃	AD-99.5 Nom 99.5% Al ₂ O ₃
SPECIFIC GRAVITY		3.82	3.72	3.89
HARDNESS ROCKWELL KNOOP	R45N GPa	78 11.1	75 11.1	83 14.7
SURFACE FINISH AS FUSED GROUND POLISHED	MICROMETRES (MICROINCHES)	1.6 (63) 1.3 (51) 0.3 (12)	1.6 (63) 1.3 (51) 0.3 (12)	0.9 (35) 0.5 (20) 0.1 (3.9)
CRYSTAL SIZE RANGE AVERAGE	MICROMETRES (MICROINCHES)	2-25 (79-985) 12 (473)	2-20 (79-788) 11 (433)	5-50 (197-1970) 16 (670)
WATER ABSORP.		NONE	NONE	NONE
GAS PERM.**		NONE	NONE	NONE
COLOR		WHITE	WHITE	IVORY
COMPRESSIVE STRENGTH 25°C 1000°C	MPa (kpsi)	2103 (305) 345 (50)	2068 (300) — (—)	2620 (380) — (—)
FLEXURAL STRENGTH TYP., 25°C MIN., 25°C*** TYP., 1000°C MIN., 1000°C***	MPa (kpsi)	352 (51) 317 (46) 138 (20) 117 (17)	358 (52) 324 (47) 172 (25) 138 (20)	379 (55) — (—) — (—) — (—)
TENSILE STRENGTH 25°C 1000°C	MPa (kpsi)	193 (28) 103 (15)	193 (28) 96 (14)	262 (38) — (—)
MOD. OF ELAST. SHEAR MODULUS BULK MODULUS TRANS. SONIC VEL. POISSON'S RATIO	GPa (10 ⁶ psi) GPa (10 ⁶ psi) GPa (10 ⁶ psi) m/sec (ft/sec)	283 (41) 117 (17) 165 (24) 8.9 (29) 10 ³ 0.21	303 (44) 124 (18) 172 (25) 9.1 (30) 10 ³ 0.21	372 (54) 152 (22) 228 (33) 9.8 (32) 10 ³ 0.22
MAX. USE TEMP. (No load condn)	°C (°F)	1700 (3100)	1700 (3100)	1750 (3180)
COEFFICIENT OF LINEAR THERMAL EXPANSION 25-200°C 25-200°C 25-500°C 25-800°C 25-1000°C 25-1200°C	10 ⁻⁶ /°C (10 ⁻⁶ /°F)	3.4 (1.9) 6.3 (3.5) 7.1 (4.0) 7.6 (4.3) 7.9 (4.4) 8.1 (4.5)	3.4 (1.9) 6.0 (3.4) 7.4 (4.1) 8.0 (4.5) 8.2 (4.6) 8.4 (4.7)	3.4 (1.9) 7.1 (4.0) 7.6 (4.3) 8.0 (4.5) 8.3 (4.6) —
THERMAL CONDUCTIVITY 25°C 100°C 400°C 800°C	W/m-K (g-cal/(sec)(cm) ²) (°C/cm)	18.0 (0.043) 14.2 (0.035) 7.9 (0.017) 5.0 (0.010)	24.7 (0.059) 18.8 (0.045) 10.0 (0.024) 5.4 (0.013)	35.6 (0.085) 25.9 (0.062) 12.1 (0.028) 6.3 (0.015)
SPECIFIC HEAT 100°C	J/kg-K (cal/g/°C)	880 (0.21)	880 (0.21)	880 (0.21)

(NOTE: The ceramic compositions shown above are the den, Colorado 80401.

Ceramics of 94-percent alumina composition appear to be structurally acceptable and more cost effective than 96-, 99-, and 99.9-percent alumina oxide compositions for fabrication of large cylinders and hemispheres. The dynamics of sintering process associated with ceramics of > 94-percent alumina compositions preclude their use in cylinders and hemispheres with diameters > 12 inches.

The size of alumina ceramic cylinders and hemispheres for deep submergence service is limited by five factors: diameter and length of largest available isostatic press (i.e., pressure vessel with working pressure capability > 10,000 psi), internal dimensions of largest available kiln, sintering process dynamics, composition of ceramic, and grinding machine capacity. These factors combine to *limit* the diameter of monolithic ceramic cylinders and hemispheres to less than 96 inches. The same limitation similarly applies also to length of monolithic cylinders. The fabrication of cylinders and hemispheres of 94-percent alumina composition with diameters > 24 inches requires, however, at the present time extensive experimentation with the sintering process to minimize residual stresses in the thick ceramic shells that would cause them to crack during sintering or grinding.

Monocoque cylinders have been successfully fabricated by brazing together 94-percent alumina ceramic rings that have been individually pressed, sintered, ground flat, and metalized on the flat bearing surfaces. This fabrication method has the potential of producing economically polyolithic cylinders in sizes beyond the scope of fabrication method for monolithic cylinders.

Ceramic cylindrical pressure housings assembled from monolithic or polyolithic monocoque cylinders radially supported by metallic ring stiffeners are *less expensive* to fabricate than housings of same dimensions assembled from monolithic cylinders with integral ribs. Furthermore, there is less breakage during fabrication, as monocoque cylinders require significantly less grinding of the fired cylinder body than cylinders with integral ribs.

CONCLUSIONS

Alumina ceramics have been shown to be economical, structurally reliable materials for fabrication of pressure housings in any desirable shape with < 0.6 weight-to-displacement ratio for 20,000-foot design depth. The 94-percent purity alumina ceramic, in particular, represents a good compromise between fabrication costs of the housing and structural properties of material.

Although at the present time the industrial fabrication process know-how limits the size of alumina ceramic cylinders to 24 inches in diameter and 48 inches in length, this does not constitute the present limit of payload capability for ceramic pressure housings. The payload capability of a ceramic pressure housing is limited instead by the number of cylindrical sections that can be securely held together by mechanical joint stiffeners without generation of unacceptably high stress in ceramic components during handling of the housing on a deck of a ship at sea. The length limitation on alumina ceramic cylindrical assembly of NOSC design is in excess of 6D_o. Thus, the *present limit* on the payload of alumina ceramic pressure housings for 20,000-foot design depth is approximately 1000 lbs.

Ceramic and titanium spherical bulkheads can be used interchangeably to support and close off the ends of the ceramic cylinders. Bulkhead penetrators can be incorporated into spherical bulkheads of both materials, providing their diameter is ≤ 0.35 spherical radius of the bulkhead.

RECOMMENDATIONS

1. Alumina ceramic should receive serious consideration by designers for construction of cylindrical pressure housings with payload requirements in the 100- to 200-pound range for service on present generation of ROVs, BPVs, and AUVs with 20,000-foot service depth.
2. The exploratory development of alumina ceramic housing technology successfully completed at NOSC should be expanded to include cylindrical housings with payload capability in the 500- to 1000-pound range.
3. Other ceramics should also be considered for potential application to construction of pressure housings with lower weight-to-displacement ratio than can be provided by 94-percent alumina ceramic. Housings fabricated from glass ceramics (Cervit[®] and PYROCERAM[®]) promise to provide a weight-to-displacement ratio of ≤ 0.4 .

REFERENCES

1. Perry, H.A., "Feasibility of Transparent Hulls for Deep Submergence," Paper No. 63-WA-219, *American Society of Mechanical Engineers, Winter Meeting*, Philadelphia, Pennsylvania, November 1963.
2. Pennsylvania State University, Ordnance Research Laboratory, External Report No. 63-0209-C-2, *Solid Glass and Ceramic External-Pressure Vessels*, by J.D. Stachiw, 1964.
3. Stachiw, J.D., "Glass and Ceramics for Underwater Structures," *Ceramic Age*, July 1964.
4. David Taylor Model Basin, Report No. 1759, *The Elastic Buckling Strength of Spherical Glass Shells*, by M.A. Krenzke and R.J. Charles, September 1963.
5. David Taylor Model Basin, Report No. 1641, *Exploratory Tests of Long Glass Cylinders Under External Hydrostatic Pressure*, by M.A. Krenzke, 1963.
6. Perry, H.A., "The Argument for Glass Submersibles," *Undersea Technology*, September 1964.
7. Murphy, D.W., "Development and Testing of a 56-in-diameter Jointed Glass Pressure Hull," *Trans. ASME/J. Eng. for Industry*, Vol. 32, No. 3, Aug 1970.
8. Stachiw, J.D. and R.F. Snyder, "The Design and Fabrication of Glass and Ceramic Deep Submergence Free-Diving Instrumentation Capsules," *National Underwater Technology Conference*, New London, Connecticut, May 5-7, 1965, ASME Paper 65-UNT-1.
9. Stachiw, J.D., "Hulls for Deep Submergence Capsules," *American Ceramic Society Bulletin*, Vol. 47, No. 2, February 7, 1968.
10. U.S. Naval Ordnance Test Station, Internal Report IDP2703, *Monolithic Glass As Structural Material*, by N.B. Estabrook, 1967.
11. Coffman, W.B., et al., *Feasibility Study of Plastic-Clad Glass Capsules for Deep Diving Submersibles*, NOLTR 65-76, US Naval Ordnance Laboratory, White Oak, MD, June 1965.
12. Faux, W.H., and C.R. Niffenegger, *The Resistance of Hollow Glass Models to Underwater Explosions at Great Depths*, NOLTR 65-145, US Naval Ordnance Laboratory, White Oak, MD, September 1965.
13. Gray, K.O., and J.D. Stachiw, "Glass Housings for Hydrospace Lights and Instruments," *Trans. ASME/J. Eng. for Industry*, Vol. 92, Series B, No. 1, February 1970.
14. Forman, W., "Submersibles With Transparent Structural Hulls," *Astronautics and Aeronautics*, April 1969.
15. Forman, W. and R. DeHart, "Submersible Deep View Pioneers Glass/Metal Bonding," *Undersea Technology*, December 1967.
16. Southwest Research Institute Final Report, contract No. N66001-73-C-0285, *Hydrostatic Evaluation of Deep View Glass Hemisphere Assembly and Hull Segment*, by R.C. DeHart, E.M. Briggs, and J.J. Jones, March 1973.
17. Naval Undersea Center, Technical Publication 393, *Glass or Ceramic Spherical Shell Window Assembly for 20,000 psi Operational Pressure*, by J.D. Stachiw, 1974.
18. Naval Ocean Systems Center, Technical Report 951, *Exploratory Evaluation of Beryllia Ceramic Cylindrical Housing for Deep Submergence Service*, by J.D. Stachiw, 1984.
19. Naval Ocean Systems Center, Technical Report 1176, *Exploratory Evaluation of Alumina Ceramic Cylindrical Housings for Deep Submergence Service: The Second Generation NOSC Ceramic Housings*, by J.D. Stachiw, 1987.