Structural Research on Submarine Pressure Hulls at the David Taylor Model Basin

Martin A. Krenzke* and Thomas E. Reynolds†
David Taylor Model Basin, Washington, D.C.

For many years, there has been a need for improved pressure hulls for underwater vehicles because of the ever-present demands for better performance of military submarines. Growing interest in exploration and exploitation of the ocean depths has recently given impetus to the development of reliable high-strength, low-density hulls required to achieve deep operating depths while maintaining small vehicle size. The David Taylor Model Basin has been playing a key role in developing pressure-hull structures for various underwater applications. This paper summarizes some principal theoretical and experimental results of recent Model Basin studies. Specifically, it describes advances in the strength analysis of shell structures and investigations of the structural problems associated with the introduction of new hull materials. Preliminary results of studies aimed at developing a pressure hull and flotation system for the 20,000-ft search vehicle of the deep submergence systems project are also presented.

Introduction

FOR more than 35 years the David Taylor Model Basin has been involved in the development of pressure-hull technology for various underwater applications. This paper describes the principal results of recent theoretical and experimental investigations and reports on some of the current work not yet published. Space, time, and expedience limit this to a summary only of research at the Model Basin. The necessary omission of references to the work of others in this field is not meant to imply that other worthwhile work is not being done. The authors wish to emphasize that they are neither unaware nor unappreciative of the many excellent accomplishments and capabilities of industry, universities, and other government activities.

Structural Analysis

Various pressure-hull configurations are being used or have been proposed for use in underwater applications; several of these are shown in Fig. 1. Elements which, in combination, make up the pressure-tight envelope, include cylindrical, conical, spherical, and spheroidal shells. The approach has been to study each shell element in isolation before proceeding to the more difficult problem presented by the pressure-hull assembly. Reliance has been placed on first understanding the behavior of near-perfect structures and then considering the possible effects of initial imperfections and residual stresses which may be present as a result of the fabrication procedures employed. Normally, in these studies, there has been roughly an equal balance between analytical and experimental efforts. A comprehensive summary of this research, now in preparation, should be published within a year.

Presented as Preprint 66-706 at the AIAA/USN 2nd Marine Systems & ASW Conference, Los Angeles-Long Beach, Calif., August 8–10, 1966; submitted July 27, 1966; revision received November 7, 1966. This research has been primarily sponsored by the Naval Ship Engineering Center (formerly, the Bureau of Ships), the Special Projects Office, the Deep Submergence Systems Project, and the Model Basin Independent Research Program. Much of the work presented is being carried out by members of the Submarine Structures Division under the leadership of F. W. Dunham. [6.02, 6.14, 10.11]

* Head, Design Analysis Branch, Submarine Structures Division.

† Research Scientist, Submarine Structures Division.

Individual Shell Elements

Cylinders

Since the circular cylinder is presently the main structural element in submarine pressure hulls, much effort has gone into developing the most accurate theory possible for cylindrical structures. Most of the important results can be found in Refs. 1 and 2.

In the field of stress analysis, the basic nonlinear solutions[3–5] for the ring-stiffened cylinder have been substantiated by numerous tests. A later analysis[6] provides the appropriate corrections to be made for rings that are too deep to be handled adequately using thin-ring theory. Recent work by Lomacky deals with cases where irregularly spaced

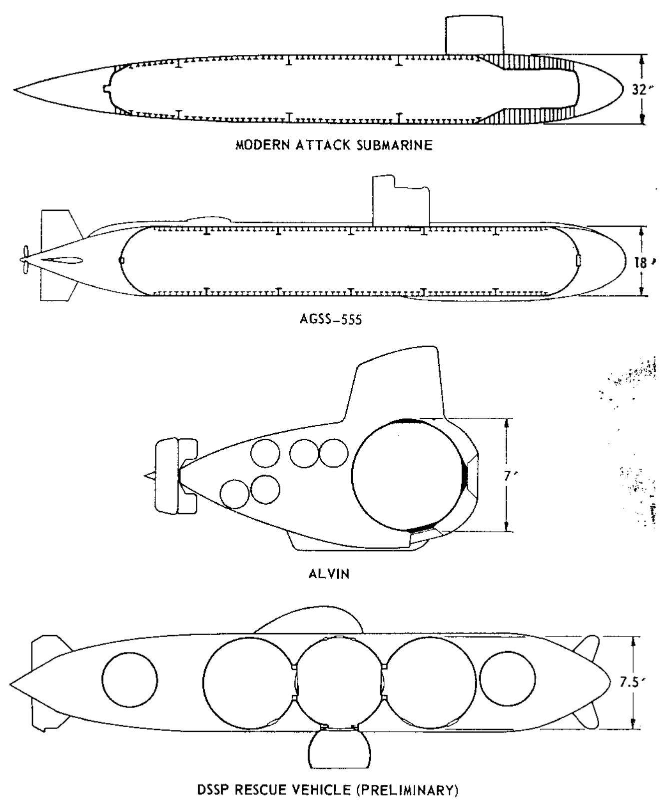
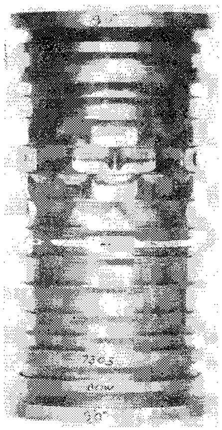
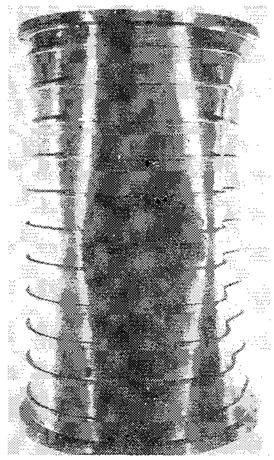


Fig. 1 Typical pressure hull structures.

a) Symmetric collapse mode



b) Asymmetric Collapse mode



c) Typical general-instability collapse mode

Fig. 2 Basic modes of collapse for a ring-stiffened cylinder loaded by a hydrostatic pressure.

stiffeners have different cross sections and there is a tapered thickening of the shell in the vicinity of a stiffener. The analysis is particularly suited for the case of "mixed frames," in which a regular stiffening system of shallow-depth rings is interrupted occasionally by rings of very deep cross section. It differs from other comparable analyses in that a series solution is employed. Problems introduced by nonsymmetric structures such as penetrations and nonuniform frames are also being studied.

The effects of imperfections on the stresses in ring-stiffened cylinders have also received attention. Symmetric imperfections in shape and associated residual stresses,[7,8] and the effects of nonsymmetric (lobar) imperfections[9,10] have been treated.

Other shell wall configurations have been studied in anticipation of their use in future deeper-diving vehicles. Of particular interest are the web-stiffened or box-type sandwich, the hoop-stiffened sandwich, and the layered shell with ring stiffeners. Recent analytical studies of stresses in these shells are summarized in Refs. 11–16. A recent analysis[17] of the stresses in a web-stiffened sandwich shell of orthotropic material has just been published; Hom has completed a comparable analysis of the single-walled ring-stiffened shell (as yet unpublished).

The elastic buckling of monocoque cylinders under hydrostatic pressure poses no great difficulty when the edges are simply supported.[18] However, when stiffeners are present or when the boundary conditions are not readily defined, the buckling problem can be greatly complicated. The various modes of buckling are illustrated in Fig. 2. Solutions for symmetric and nonsymmetric buckling between stiffeners are given in Refs. 3 and 19, where the effects of the stiffeners are included. Another possibility is over-all buckling (general instability) of the entire cylinder in which the stiffeners and shell deflect nonsymmetrically with a node at each end. For the case of simply supported ends, an analytical solution developed in England[20] was proved reliable with certain limits‡ through tests of machined cylinders at the Model Basin.[21] Later work showed that the results of the analysis could be approximated adequately by a simple two-term expression for the buckling pressure, with the first term representing the direct compressive strength of the shell, and the second, the bending strength of a stiffener plus an effective width of plating.[22]

The problem posed by a cylinder with mixed frames is inherently more difficult because of the possibility that additional buckling modes, heretofore not considered, may occur. If the deep stiffeners are not strong enough to localize buckling, the problem then is to determine the pressure associated with over-all buckling in which the deep stiffeners, as well as the shallow ones, deform radially. For this case, the easiest guess would be that one need merely add a third term (representing the bending strength of the deep stiffener) to the two-term expression. However, tests conducted with small aluminum cylinders[23] demonstrated that this can lead to an unsafe estimate of buckling strength. Subsequent efforts by Reynolds to correct this deficiency resulted in a new series solution to this buckling problem. Its main feature was to include in the assumed buckling shape additional configurations in which nodes appear at intermediate points (not at the stiffeners) along the length of the cylinder. The ends of the cylinder were taken to be simply supported. Another feature of the solution is that simple relations are found between the displacement coefficients; thus they are eliminated as unknowns and the problem of inverting a large matrix is avoided. A computer program has been written for this analysis; some results are compared with experiment in Fig. 3. The inadequacy of the three-term approximation is clearly shown in the figure, whereas the new solution appears to be fairly reliable.

The results of this new analysis raise the interesting point that, theoretically, for geometries representative of contemporary submarines, the critical buckling mode can be one in which more than one half-wave develops along the length of the cylinder, even when no deep frames are present. That is, the number of nodes is intermediate between one at each end (as occurs in over-all buckling) and one at each frame (as is the case for local interframe buckling). Tests have been initiated to investigate the phenomenon.

Inelastic buckling is the usual ultimate mechanism of collapse, and it has naturally been the subject of a good deal of

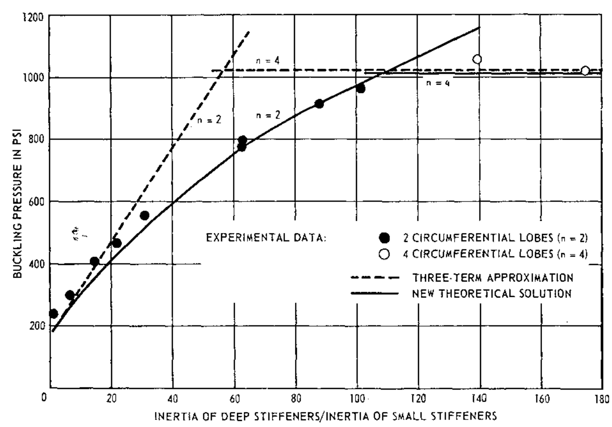


Fig. 3 Buckling of cylinders with mixed frames.

‡ The limits were those imposed by the material properties of the cylinders tested. Although high-strength steel was used, it was necessary that the stiffeners be relatively much smaller than those used in practical submarines in order to confine buckling to the elastic range.

study. Most of the analytical work to date has been based on the deformation theory of plasticity; this has proved fairly successful in explaining inelastic behavior of strain-hardening materials.

The inelastic buckling modes are generally of the same form as the elastic modes. Analyses for the local shell buckling modes have been developed[24,25] and evaluated experimentally.[26] The problem of over-all buckling has also been treated.[22,27] All of these analyses are for near-perfect, ring-stiffened cylinders made of strain-hardening materials. In general the results can be stated as follows. If the elastic buckling pressure for any of the modes of interest can be expressed as the sum of two terms

$$p_e = C_m + C_b \tag{1}$$

where C_m is a coefficient reflecting the resistance of the cylinder to membrane (or in-plane) deformations and C_b is another coefficient representing the resistance to bending, then the inelastic buckling pressure is given with quite good accuracy by:

$$p_i = [(E_sE_t)^{1/2}/E]C_m + (E_t/E)C_b \tag{2}$$

Here, E is Young's modulus and E_s and E_t are, respectively, the secant and tangent moduli as determined from uniaxial specimen tests. This general formula, as might be expected, has certain limitations.

Although residual stresses and imperfections in shape do not appear to affect significantly the elastic buckling strength of ring-stiffened cylinders, their influence on inelastic buckling is so serious that it warrants careful study. Since it is difficult to isolate and to measure these factors, a quantitative correlation between, say, imperfect circularity and buckling strength is practically impossible. What one can do, however, is to classify cylinders according to the processes used in their fabrication and reason that, on a statistical basis, cylinders made by the same process should generally have imperfections of the same magnitude. This approach at least enables distinguishing two classes of structures: accurately machined shells and shells fabricated by rolling and welding. Figure 4, obtained from Ref. 28, is a plot of results from numerous tests conducted at the Model Basin. One sees immediately that the factor which is all-important in determining imperfection sensitivity is the margin of stability p_e/p_i, the abscissa in the plot. The lower the margin of stability, the greater the sensitivity to imperfections.

Spherical shells

Most of the earlier results on spherical shells can be found in Refs. 29 and 30. Very briefly, these studies (largely experimental) established these salient facts: 1) The classical buckling equation for the elastic buckling of complete spheres

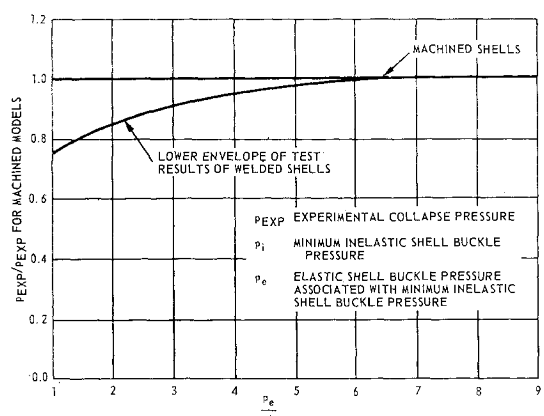


Fig. 4 A comparison of model basin data on machined ring-stiffened cylinders with welded cylinders.

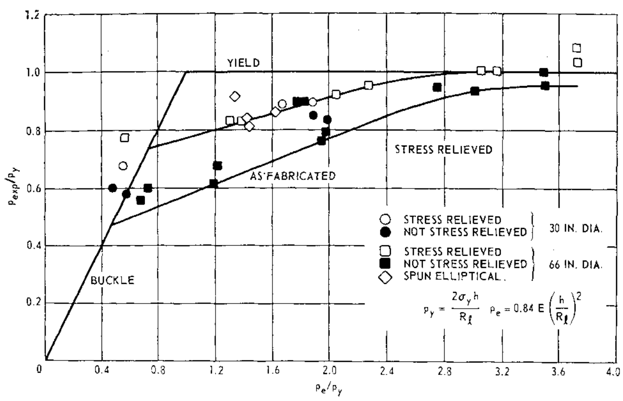


Fig. 5 Summary of HY-80 steel hemispherical and elliptical shell tests.

is probably correct, but the pressure is seldom, if ever, achieved because the slightest departures from sphericity greatly reduce the buckling strength. 2) An empirical correction to the classical equation enables precise calculations of buckling pressures for accurately machined, deep spherical shells; these pressures are about 70% of the classical value. 3) The buckling pressure for an imperfect shell can be adequately predicted provided local, rather than nominal, geometry is used in the equation. 4) Inelastic buckling strength can also be determined for shells made of strain-hardening materials by multiplying the elastic buckling pressure by $(E_sE_t)^{1/2}/E$. In addition, buckling tests[31] have been carried out to compare the collapse strengths of layered and single-walled spherical shells.

The most recent and by far the largest effort in spherical shells has been a program of buckling experiments with welded, segmented hemispheres constructed of HY-80 steel. These tests[32] examined the effects on elastic behavior and collapse strength of such factors as initial imperfections, residual stresses, and penetrations. Some shells were tested in the "as fabricated" condition; others were stress-relieved before testing. Prior to test, each shell was examined carefully for local departures from sphericity by means of a contour-measuring device. The resulting data were plotted on contour maps which were then analyzed to determine the worst flat spot for each shell defined over a critical arc length. These areas could then be defined in terms of a local radius of curvature and a local thickness, dimensions to be used in the empirical equation defining the buckling strength for each shell.

The results of these experiments are summarized in Fig. 5. All quantities are expressed in terms of the local geometry for the critical region, i.e., R_l, the local radius of curvature drawn to the midthickness of the shell, and h, the average local shell thickness. It is seen that only the most stable shells approach p_y, the pressure at which the theoretical membrane stress calculated for the critical region reaches the yield stress of the material. The data also show that significant gains in strength can be achieved through stress-relieving.

Spheroidal shells

Current interest in small craft for deep-sea operation has stimulated the consideration of prolate spheroidal shells (footballs). Unfortunately, they have received far less attention in the literature than have other shell structures. As a consequence, the Model Basin has initiated a fairly comprehensive program of study in this area. The accomplishments to date include a postbuckling analysis[33] in which the sensitivity to imperfections is assessed; an experimental buckling study[34] using plastic shells, stiffened as well as unstiffened; and a small deflection buckling analysis (as yet unpublished) carried out under contract by B. I. Hyman.

Results so far indicate that only very short spheroids (major-to-minor axis ratios of 1.25 or less) suffer from the kind of imperfection sensitivity associated with the sphere. Another important observation is that the spheroid possesses far more buckling strength than a comparable cylinder-closure dome structure of the same over-all dimensions, even for lengths as great as four diameters.

Collective Structure

Even when the behavior of each shell element is well established, the problem of determining the performance of a hull that is a composite of several elements remains a formidable one. Such a structure usually consists of ring-stiffened cylinders; conical or toroidal segments (where a diametral change is required); and, at each end, some sort of closure dome such as a hemisphere or an oblate spheroid.

Considerable effort has been spent at the Model Basin in developing methods for analyzing the stresses in such a hull resulting from axisymmetric deformations. In a recent analysis, Raetz has presented several methods for dealing with problems of this type. The emphasis is on shells of very short length where use of a "one-point" boundary approach eliminates the numerical difficulties which often arise when the edge-coefficient method is employed. The "one-point" scheme is shown to be numerically impractical for very long shells. In addition, an iterative technique is developed for obtaining solutions for shells of arbitrary shape. The coefficient of each applied boundary condition is given as a series of definite integrals, and it is shown that only one or two terms of the series are needed for short shells.

At the same time, efforts have been underway to make better use of computer technology in analyzing stresses in complicated structures. A number of the excellent computer programs in use here and abroad are being evaluated for possible application to structural hull problems. The first of these to be adapted for use with Model Basin computers is a British stress program.[35,36] The ability of this program to handle a complete axisymmetric pressure hull has been convincingly demonstrated.

The solution of the axisymmetric case is, of course, only the first step. Structural nonsymmetries pose stress problems of far greater complexity, but among the computer programs being evaluated there are several developed in this country that appear to have the capability of dealing with them. Nonsymmetric response is again involved in the determination of the over-all buckling strength of the structure, a problem that in some ways may be the most difficult of all. At the present time, a computer solution seems feasible, but much more work is still needed before this becomes an accomplished fact. In the meantime, experimental studies were initiated by the authors in the hope of establishing interim empirical buckling formulas. To determine how the elastic buckling strength of the cylinder is affected by the closure, buckling tests have been conducted with a large number of unstiffened and ring-stiffened cylindrical shells made of low-modulus plastic and terminated by spherical and spheroidal closure domes. The tests covered a variety of lengths, thicknesses, and dome configurations.

One of the most interesting and somewhat surprising results to date is that the buckling strength is insensitive to the thickness of the end closure. For example, a decrease of 40% in the thickness of a pair of hemispherical closures had no effect on buckling strength. On the other hand, the shape of the end closure was found to influence seriously the buckling strength. In one case, for example, a cylinder was more than 50% stronger when equipped with hemispherical ends than when equipped with spheroidal ends having a major-to-minor diameter ratio of 2.5.

Although these buckling studies appeared to promise eventual success in the establishment of empirical equations, it became clear that a great many tests would be needed to cover adequately the full range of interest. It was then that the authors turned to a more indirect approach in efforts to reduce the number of tests. This method, which utilizes the similarity between buckling and vibration modes, has proven to be highly successful. A brief description follows.

Since the buckling strength of a cylinder with closure domes is less than when it is equipped with simply-supporting end plates, an equivalent simply-supported cylinder (i.e., one having the same buckling strength) would be somewhat longer. The length of this equivalent cylinder is the effective length associated with the particular cylinder-dome configuration and the buckling mode in question. Similarly, the resonance frequency for the corresponding vibration mode is somewhat less than that for the simply-supported cylinder, and for this too, there is an associated effective length. The assumption was that if the vibration and buckling modes are sufficiently similar, the effective lengths for the two cases would be nearly identical. Vibration tests on small steel shells were carried out using an electromagnet and a phonograph pickup. An effective length for each mode was established by comparing the measured frequencies with the theoretical frequencies for a simply-supported cylinder.[37] The buckling pressure for the test structure was then determined by using these lengths in the buckling equation for a simply-supported cylinder.[18] In this way, a few vibration tests were used to predict the buckling pressures of the plastic shells. The effective lengths were found to depend on the over-all configuration and on the buckling mode, but not on the thickness of the shell. Thus vibration tests which used a single shell thickness were successful in predicting buckling pressures for similar shells which not only had different thicknesses but (for some) had ring-stiffeners as well. In the latter cases, the theory for the buckling of ring-stiffened cylinders was used.[20] So far, 28 buckling tests show an average error of 3.6% and a maximum of 11%. The results also show that the effective length L_e for any configuration can be adequately expressed as:

$$L_e = x + L \tag{3}$$

where L is simply the length of the cylindrical section and x is an additional length that depends only on the mode in question and on the shape of the end closure.

Deep-Depth Research Program

Research on structures for deep-depth applications was initiated at the Model Basin in 1957. It soon became evident that the basic approach to achieving greatly increased depths without corresponding increase in structural weight must be through the use of new hull materials with high strength-to-density ratios; see Fig. 6. Since such factors as imperfections, residual stresses, fatigue, stress corrosion, and toughness were not considered in developing Fig. 6, the depth potentials shown are highly optimistic.

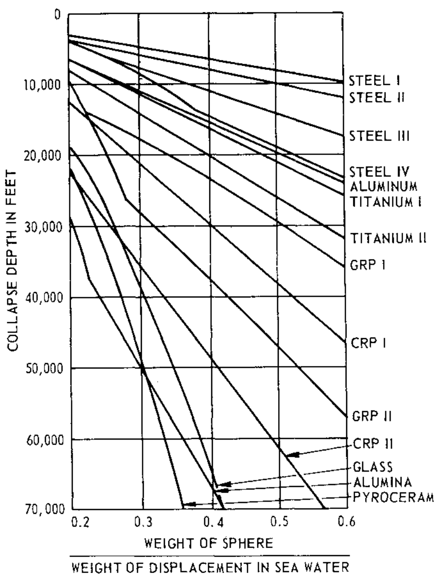
A fairly extensive experimental program aimed at developing a thorough understanding of the structural performance of deep-depth hulls of new materials is currently being conducted at the Model Basin. This research is aided to a considerable extent by the excellent high-pressure test facilities. The primary pressure tanks available have the following pressure capacity and diameter characteristics: 12-ft-diam, 1500 psi; 10-ft-diam, 12,000 psi; 6-ft-diam, 6000 psi; 4-ft-diam, 15,000 psi; and $1\frac{1}{2}$-ft-diam, 25,000 psi. These tanks are equipped for applying cyclic as well as static pressure.

The deep-depth research program can be divided into three general areas: metallic hulls, fiber-reinforced plastic (FRP) hulls, and massive glass and ceramic hulls. The following is a brief summary of the research findings in these areas.

Metallic Hulls

Present studies of deep-depth metallic hulls are limited almost entirely to high-strength steel and titanium alloys. From a structural viewpoint, the high-strength steels can be grouped into two general categories. The first group involves hull materials which possess essentially elastic, ideally plastic, stress-strain curves. At present, HY-80 and HY-100 are the only steels in this category which are receiving serious consideration. The static performance of deep-depth hulls of these steels can be predicted with confidence using the analyses described in the previous section.

The second group involves hull materials which have curvilinear stress-strain diagrams; specific materials under consideration include HY-130/150, HP-150, 9425, 12% nickel maraging, and 18% nickel maraging steel.[38] A fairly extensive program has been initiated to study the effects of fabrication procedures on the collapse strength of hulls incorporating high-strength steels of this second group. A number of cylindrical and spherical, welded HY-130/150 steel models have been designed by Jones and Nishida and are presently being tested. The preliminary results indicate that their collapse strength can be adequately calculated by modifying the procedures used to calculate the strength of HY-80 hulls. This modification simply introduces the inelastic buckling equations. Results to date indicate that the effect of residual forming and welding stresses on collapse strength is about the same for both HY-130 and HY-80/100 steel hulls despite the different chemical compositions, yield strengths, and stress-strain characteristics.



ASSUMED MATERIAL PROPERTIES			
MATERIAL	YIELD STRENGTH PSI × 10³	MODULUS PSI × 10⁶	DENSITY LB/FT³
STEEL I	80	30.0	490
STEEL II	100	30.0	490
STEEL III	150	30.0	490
STEEL IV	200	30.0	490
TITANIUM I	120	17.5	276
TITANIUM II	150	17.5	276
ALUMINUM	60	10.8	173
GLASS-REINFORCED PLASTICS (GRP) I	75	5.0	130
GLASS-REINFORCED PLASTICS (GRP) II	120	5.0	130
CARBON-REINFORCED PLASTICS (CRP) I	60	3.0	84
CARBON-REINFORCED PLASTICS (CRP) II	100	10.0	84
GLASS	≥ 300	9.0	139
PYROCERAM	≥ 300	17.3	163
ALUMINA	≥ 400	45.0	233

Fig. 6 Strength-weight potential for near-perfect spheres.

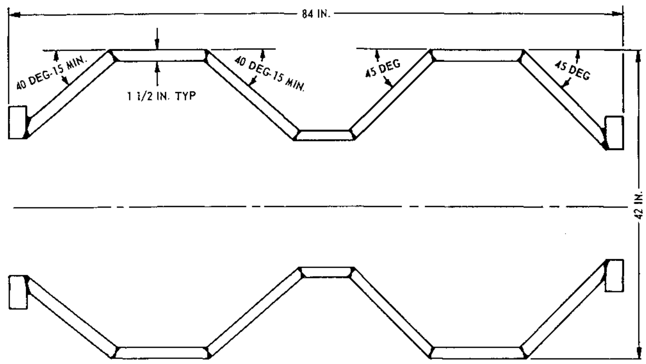


Fig. 7 Structural fatigue model of HY-80 steel.

Two geometrically similar ring-stiffened cylinders of 12% nickel maraging steel have been fabricated and soon will be tested to determine the effects of aging on collapse strength. One model will be tested in the aged condition and the second in the as-welded condition. It is anticipated that strength can be significantly improved by removing most of the residual fabrication stresses and by using aging to bring the welds and heat-affected zones up to the specified yield strength. In fact, the superior static strength performance of welded hulls after aging or stress-relieving is one of the inherent advantages in using maraging steels or high-strength martensitic steels which can be stress-relieved.

The performance of high-strength steel hulls under low-cycle fatigue loading is also important. Test programs in this area have been initiated, but no results have been obtained to date. Models of HY-100, HY-130/150, and maraging steels have been or are being fabricated similar to those with which Palermo successfully simulated the fatigue performance of welded HY-80 hulls. These models, usually about 4 ft in diameter and 1 to 2 in. thick, consist of sections of cones and cylinders welded together to form an hour-glass shape (see Fig. 7). The desired stresses can be produced through proper selection of the external cyclic pressure, thickness-to-radius ratio, and cone angle. Since hull-plating and fabrication procedures play a significant role in the fatigue performance of full-scale submarines, it is a distinct advantage to be able to represent them in a relatively small-diameter model. The main disadvantage of the hour-glass models is the time and cost involved in fabrication and testing. Efforts are currently underway to develop a simpler structural fatigue test which can also be used in selecting new hull materials.

Preliminary results obtained by Kiernan and Fishlowitz from tests of two welded 721-titanium models indicate behavior similar to that observed in the HY-130/150 steel models. It does appear, however, that creep of most titanium alloys under high sustained loads is such that it probably must be considered in order to adequately predict inelastic collapse strength under long-term loading. For the most part, creep has not been considered to date, since the models were loaded at approximately the same stress rate as used to establish the uniaxial stress-strain curves.

The discovery of stress-corrosion cracking in 721-titanium by Brown at the Naval Research Laboratory has virtually ended any interest in this particular alloy for deep-submergence applications. However, it is felt that the results of the collapse tests of the 721-titanium models can be applied in general to welded hulls of other titanium alloys.

The low-cycle fatigue performance of high-strength steel and titanium pressure hulls will most likely be a critical area. Use of structural details which limit the applied stresses to a level somewhat below yield and which minimize tensile stresses will undoubtedly be mandatory for most applications. Improved cyclic performance should result from aging or stress-relieving the completed structure. This is practical at least for the small research vehicles, providing the selected hull material lends itself to post-weld treatment.

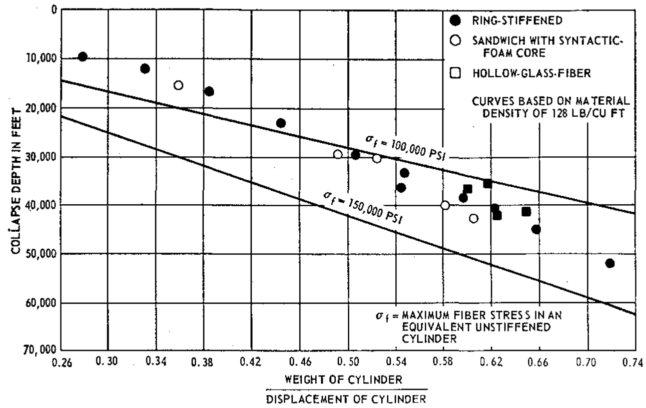


Fig. 8 Test results for glass-reinforced plastic cylinders.

Fiber-Reinforced Plastic Hulls

To date, most Model Basin studies on FRP hulls have involved solid and hollow glass fibers. This work has recently been summarized in Ref. 39. Investigations involving both low- and high-modulus carbon-fiber-reinforced plastic (CRP) are presently being initiated. Boron, alumina, and other fibers which also possess promising properties for deep-submergence applications have not been incorporated into the program to date because of their extreme cost and the lack of technology for fabricating composite shells of these materials.

The best strength-weight performances obtained to date for glass-reinforced plastic (GRP) cylinders are shown in Fig. 8. Effective strength levels on the order of 115 to 120 ksi have been developed in ring-stiffened cylinders. In contrast, the Naval Applied Science Laboratory has attained stress levels of 170 ksi and over on 2:1 orthogonal laminates under unidirectional compressive loading. Similar results have been observed in extremely thick-walled, unstiffened cylinders under hydrostatic pressure.[40] The lower strength levels observed in ring-stiffened cylinders are a result of localized bending and shearing stresses.

The presence of ring stiffeners and other discontinuities not only degrades the static collapse strength of GRP cylinders but also impairs their cyclic performance. It has been necessary to restrict the maximum cyclic pressure to about one half of the static collapse pressure in order to achieve a cyclic life of 5000 to 10,000 cycles. Thus, the importance of reducing localized shear and bending cannot be overemphasized.

Several approaches to this problem are being investigated. The use of an unstiffened cylinder is the simplest method. Development of lower density, hollow-fiber GRP has made this concept practical for hulls having a collapse depth in excess of 35,000 ft. The increase in thickness of the cylinder afforded by the lighter-weight hollow fibers more than offsets the loss of stability due to lower elastic modulus of the material. In addition, better compatibility exists between the glass reinforcement and resin binder. To date, little or no static strength-weight advantage has been found for the hollow-glass cylinder over the ring-stiffened cylinder.[41] However, improved cyclic performance is anticipated.

Another concept which offers reduced shear and bending is the sandwich cylinder with a uniform core of syntactic foam. Ward has recently tested shells of this type in which stress levels of 160,000 psi were developed in the GRP facings. However, only a marginal increase in static strength performance over that of the ring-stiffened cylinders was achieved due to the relatively low strength-weight characteristics of present foams.

Still another potential method for alleviating the stiffener problem in cylindrical hulls is the use of fibers with a higher modulus and/or a lower density than glass fibers. At present, the high-modulus carbon fibers appear the most promising and cylinders incorporating them are presently being wound.

Also under investigation are doubly curved shells, such as spheroids and spheres, which offer improved stability characteristics and associated improvements in shear stress levels. Considerable difficulty has been encountered in obtaining optimum fiber distribution in filament-wound structures of this type and consequently, most of the test results to date have been somewhat less than desired. However, the performance of filament-wound GRP spheres recently fabricated by the H. I. Thompson Fiber Glass Company was equal to that of the best cylinder on a static strength-weight basis.

Extremely encouraging results have been observed in tests of GRP spheres fabricated by the United States Rubber Company. A unique fabrication technique is used to achieve radial orientation of each fiber The observed strength-weight characteristics of these shells have been more than double those obtained for GRP cylinders.[42] These results serve to demonstrate further that hull concepts and material disposition play a very important role in realizing the full strength potential of FRP materials.

Studies have also revealed the importance of maintaining the integrity of a structure subjected to high-pressure water environment.[39] Water absorption can be a serious problem when cut fibers are exposed to the pressure medium. The use of a watertight metallic jacket is one possible method of overcoming this problem. A high-modulus carbon-fiber composite with a titanium sheath appears particularly attractive since it would permit stressing the primary structural elements to an ultimate composite stress without overstressing the metallic jacket.

Massive Glass and Ceramic Hulls

Tests initiated in 1959 indicated that spheres of certain glass and ceramic materials have potentially higher strength-weight characteristics than attainable by any other hull using currently available materials. Since that time, however, the difficulty of developing *reliabie* hulls of these materials has become evident.[43]

The various glass and ceramic materials include annealed and chemically strengthened glasses, glass ceramics, and high-density alumina. Recent tests have been limited primarily to uniaxial compressive tests of rods and to collapse, cyclic, and impact tests of spheres.

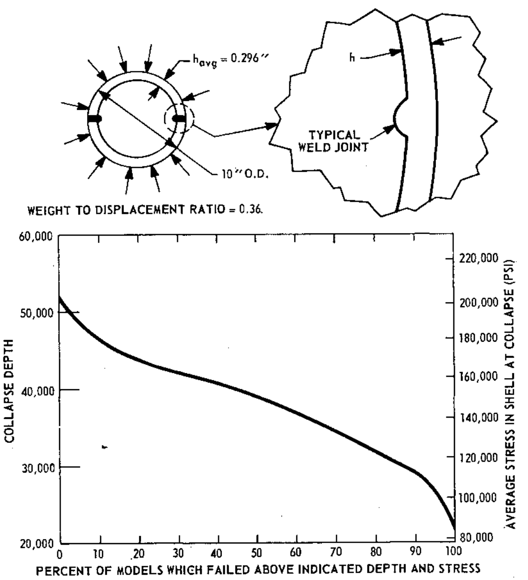


Fig. 9 Test results for fusion-sealed, annealed glass spheres.

A major portion of the Model Basin program has concentrated on compression joints since these constitute the greatest source of trouble in developing glass and ceramic hulls. In structural configurations as simple as a sphere, the presence of a single circumferential joint can degrade strength considerably.

Uniaxial compression tests of rods have proven very useful in screening possible joint details for shell structures. Moreno is in the process of studying the effects of gasket material, surface finish, bearing surface geometry, specimen size, environment, and loading rate, and has so far completed approximately 500 tests.

Fairly good correlation has been obtained between the uniaxial compression strength and the strength of the compression joint in 10-in.-diam spheres composed of two hemispheres placed together and collapsed under external oil pressure.[43] The chemically strengthened glass spheres were, on the average, stronger than the annealed glass spheres, but with considerable scatter in the test results. Maximum membrane stresses of 200,000 and 170,000 psi were developed in tests of chemically strengthened and annealed glass spheres, respectively. However, the minimum membrane stresses at fracture were as low as 84,000 and 53,000 psi. As was the case in the uniaxial compression tests, no gasket materials were found which offered any improvement and no effects of bearing surface finish were detected. The membrane stresses of four alumina spheres at collapse ranged between 210,000 and 270,000 psi, once again indicating that the bearing strength of ceramics may be considerably better than that of the glasses tested to date.

The most successful joints found in glass have been the fusion-sealed joints of Corning's annealed glass spheres.[43] Even in this case, collapse strength of nominally identical spheres was observed to vary by a factor of $2\frac{1}{2}$ in early tests; see Fig. 9. A considerable amount of this scatter is attributed to flaking of bead on the inside of the sphere, frequently at pressures as low as one half of the collapse pressure. However, the better results of very recent tests indicate that improved fusion-sealing techniques may upgrade this performance considerably.

By far the highest stress levels (260,000 to 400,000 psi) achieved in this program have been in tests of Coor's 10-in.-diam monolithic alumina spheres.[43] The maximum stresses were considerably greater because of variations in local radii and thickness.

Static and cyclic fatigue performance is also critical. Most of the tests in this area have been conducted on annealed, fusion-sealed glass spheres under the search vehicle program, which will be discussed later. The limited tests conducted on chemically strengthened glass spheres with a single circumferential bearing joint have not been very encouraging. Premature failures have occurred after as few as 10 cycles to 75% of the proof pressure under simulated salt water environment. Preliminary results from tests of similar models of annealed glass indicate that their hydrostatic collapse strength is a function of both the rate at which load is applied and the environment. No cyclic tests have been conducted on annealed glass models with bearing joints because of their poor performance under hydrostatic loading. Studies of the static and cyclic fatigue performance of alumina hulls have been initiated, but no results are available at this time.

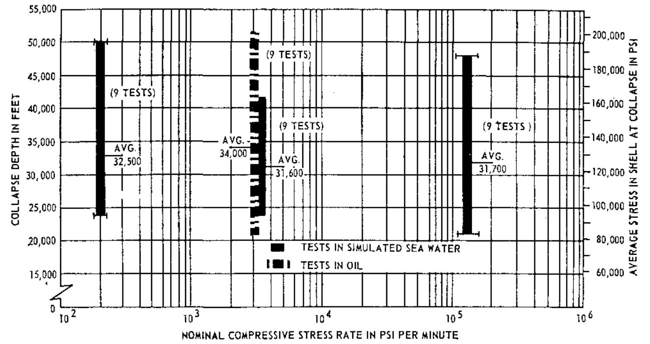


Fig. 11 Static fatigue tests of 10-in.-diam fusion-sealed, annealed glass spheres.

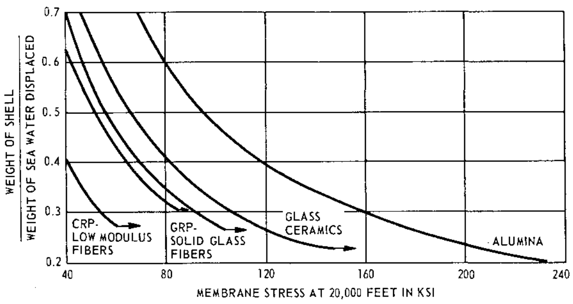


Fig. 10 Potential nonmetallic spheres for the search vehicle.

Search Vehicle

The United States Navy's deep submergence systems project is developing a search vehicle which can search the ocean bottom to depths of 20,000 ft and recover small objects from the ocean floor. Although not necessarily a requirement at this time, it is desirable that the size of the vehicle be limited to 50,000 lb to permit transportation by air and to minimize the problems of handling at sea. Pressure hulls and/or floatation systems involving shells of nonmetallic materials must be developed in order for a search vehicle to operate at 20,000 ft while maintaining small vehicle size.[28]

During the past year, the Model Basin has been exploring the potential of several nonmetallic spheres for use in the search vehicle. The weight-to-displacement (W/D) ratios for near-perfect spheres of a number of potential nonmetallic material candidates are shown in Fig. 10. A W/D ratio of 0.4 has been set as a target goal for the program.

Somewhat promising results have been achieved thus far in tests of fusion-sealed, annealed glass and radially oriented, glass-reinforced plastic spheres. These shells, having no penetrations or bearing joints, are being considered only for use as structural floats and not, at this time, as the main pressure hull. Tests of other materials and more complex configurations have not progressed sufficiently to be summarized at this time.

The scatter in earlier test results on fusion-sealed glass spheres suggested that the only foreseeable hope for developing reliable pressure hulls of this type was in the successful introduction of a proof-test approach. The effects of static and cyclic fatigue are of utmost importance in establishing the validity of such a technique for glass structures, since it is well known that at least the tensile strength of glass is a function of the environment and loading rate. To study the feasibility of developing reliable glass floats through a proof-test approach, 40 fusion-sealed, annealed Pyrex glass spheres with 10-in. diameters and a W/D ratio of about 0.36 were tested under static and cyclic pressure.[44] Three series of models tested in sea water showed no apparent effect of loading rates on collapse strength. A fourth series of models tested in oil showed no appreciable difference in strength when compared to those tested in seawater; see Fig. 11. Four models initially proof-tested to a depth of 30,000 ft survived 5000 cycles to a depth of 22,500 ft. Each model was then tested to destruction at depths in excess of 30,000 ft. A group of 44-in.-diam spheres are presently being procured from Corning Glass Works and will be tested to determine whether these promising results can be repeated in larger-diameter models.

Since environment and loading rate had a negligible effect on collapse strength, the preceding results suggest that the failure of the fusion-sealed, annealed glass spheres originates at flaws in the interior of the glass. This probably also explains the difference between the performance of fusion-sealed and bearing joints. Failure of the spheres with bearing joints probably initiates at flaws in the bearing surface which are exposed to the environment.

Although work on radially oriented GRP spheres has not progressed as far as that on fusion-sealed, annealed glass spheres, the preliminary results obtained by Blumenberg are also very encouraging. Static collapse tests have demonstrated excellent and highly repeatable strength-weight characteristics. An 11-in.-diam sphere with a 0.39 W/D ratio collapsed at a pressure of 20,000 psi, with a corresponding membrane stress of 180,000 psi. A similar model has successfully withstood an external pressure of 11,600 psi for three weeks with no apparent damage. Fatigue tests have been initiated, but only several hundred cycles have been applied to date. However, earlier tests of a smaller model were very encouraging.[42]

Future tests of radially oriented GRP spheres with and without coatings will concetrate on cyclic and long-term loading. A 32-in.-diam sphere is presently being procured from the United States Rubber Company to study the scale-up effects on fabrication, cyclic performance, and collapse strength.

Although the preceding tests of glass and GRP spheres have been encouraging, they represent only a first attempt to develop reliable, light-weight hulls for the search vehicle. It will probably be at least two years before the feasibility of nonmetallic spheres may be properly assessed for this application, particularly if prior demonstration of performance at full-scale is required.

References

[1] Pulos, J. G., "Structural analysis and design considerations for cylindrical pressure hulls," David Taylor Model Basin Rept. 1639 (April 1963).

[2] Pulos, J. G. and Krenzke, M. A., "Recent developments in pressure hull structures and materials for hydrospace vehicles," David Taylor Model Basin Rept. 2137 (December 1965).

[3] Pulos, J. G. and Salerno, V. L., "Axisymmetric elastic deformations and stresses in a ring-stiffened, perfectly circular cylindrical shell under external hydrostatic pressure," David Taylor Model Basin Rept. 1497 (September 1961).

[4] Krenzke, M. A. and Short, R. D., "Graphical method for determining maximum stresses in ring-stiffened cylinders under external hydrostatic pressure," David Taylor Model Basin Rept. 1348 (October 1959).

[5] Short, R. D. and Bart, R., "Analysis for determining stresses in stiffened cylindrical shells near structural discontinuities," David Taylor Model Basin Rept. 1065 (June 1959).

[6] Short, R. D., "Effective area of ring stiffeners for axially symmetric shells," David Taylor Model Basin Rept. 1894 (March 1964).

[7] Lunchick, M. E. and Short, R. D., "Behavior of cylinders with initial shell deflections," David Taylor Model Basin Rept. 1150 (July 1957).

[8] Krenzke, M. A., "Effect of initial deflections and residual welding stresses on elastic behavior and collapse pressure of stiffened cylinders subjected to external hydrostatic pressure," David Taylor Model Basin Rept. 1327 (April 1960).

[9] Galletly, G. D. and Bart, R., "Effects of boundary conditions and initial out-of-roundness on the strength of thin-walled cylinders subject to external hydrostatic pressure," David Taylor Model Basin Rept. 1066 (November 1957).

[10] Hom, K., "Elastic stresses in ring frames of imperfectly circular cylindrical shells under external pressure loading," David Taylor Model Basin Rept. 1505 (May 1962).

[11] Pulos, J. G., "Axisymmetric elastic deformations and stresses in a web-stiffened sandwich cylinder under external hydrostatic pressure," David Taylor Model Basin Rept. 1543 (November 1961).

[12] Nott, J. A., "Graphical analysis for maximum stresses in sandwich cylinders under external uniform pressure," David Taylor Model Basin Rept. 1817 (May 1964).

[13] Nott, J. A. and Ward, G. D., "Evaluation of stresses in web-stiffened cylindrical sandwich shells subjected to uniform external pressure," David Taylor Model Basin Rept. 2092 (September 1965).

[14] Raetz, R. V., "Analysis of stresses in axisymmetric shell structures utilizing toroidal shells as reinforcing rings," David Taylor Model Basin Rept. 1569 (January 1962).

[15] Short, R. D., "Membrane design for stiffened cylindrical shells under uniform pressure," David Taylor Model Basin Rept. 1898 (April 1965).

[16] Raetz, R. V., "Tests of fabricated multilayered ring-stiffened cylindrical models under external hydrostatic pressure," David Taylor Model Basin Rept. 2173 (April 1966).

[17] Nott, J. A., "Axisymmetric stresses in orthotropic, web-stiffened sandwich cylinders loaded with uniform external pressure," David Taylor Model Basin Rept. 1859 (April 1966).

[18] Von Mises, R., "The critical external pressure of cylindrical tubes," David Taylor Model Basin Translation 5 (August 1931).

[19] Reynolds, T. E., "Elastic lobar buckling of ring-supported cylindrical shells under hydrostatic pressure," David Taylor Model Basin Rept. 1614 (September 1962).

[20] Kendrick, S., "The buckling under external pressure of circular cylindrical shells with evenly spaced, equal strength circular ring frames—Part III," Naval Construction Research Establishment Rept. NCRE/R-244 (September 1953).

[21] Reynolds, T. E. and Blumenberg, W. F., "General instability of ring-stiffened cylindrical shells subject to external hydrostatic pressure," David Taylor Model Basin Rept. 1324 (June 1959).

[22] Krenzke, M. A. and Kiernan, T. J., "Structural development of a 15,000- to 20,000-foot titanium oceanographic vehicle," David Taylor Model Basin Rept. 1677 (September 1963).

[23] Blumenberg, W. F., "The effect of intermediate heavy frames on the elastic general-instability strength of ring-stiffened cylinders under external hydrostatic pressure," David Taylor Model Basin Rept. 1844 (February 1965).

[24] Lunchick, M. E., "Plastic axisymmetric buckling of ring-stiffened cylindrical shells fabricated from strain-hardening materials and subjected to external hydrostatic pressure," David Taylor Model Basin Rept. 1393 (January 1961).

[25] Reynolds, T. E., "Inelastic lobar buckling of cylindrical shells under external hydrostatic pressure," David Taylor Model Basin Rept. 1392 (August 1960).

[26] Boichot, L. and Reynolds, T. E., "Inelastic buckling tests of ring-stiffened cylinders under hydrostatic pressure," David Taylor Model Basin Rept. 1992 (May 1965).

[27] Lunchick, M. E., "Plastic general instability pressure of ring-stiffened cylindrical shells," David Taylor Model Basin Rept. 1587 (September 1963).

[28] Krenzke, M., Hom, K., and Proffitt, J., "Potential hull structures for rescue and search vehicles of the deep submergence systems project," David Taylor Model Basin Rept. 1985 (March 1965).

[29] Krenzke, M. A. and Kiernan, T. J., "Tests of stiffened and unstiffened machined spherical shells under external hydrostatic pressure," David Taylor Model Basin Rept. 1741 (August 1963).

[30] Krenzke, M. A. and Kiernan, T. J., "The effect of initial imperfections on the collapse strength of deep spherical shells," David Taylor Model Basin Rept. 1757 (February 1965).

[31] Nishida, K., "Tests of machined multilayered spherical shells with clamped boundaries under external hydrostatic pressure," David Taylor Model Basin Rept. 2012 (August 1965).

[32] Kiernan, T. J. and Nishida, K., "The buckling strength of fabricated HY-80 steel spherical shells," David Taylor Model Basin Rept. 1721 (July 1966).

[33] Hyman, B. I., "Elastic instability of prolate spheroidal shells under uniform external pressure," David Taylor Model Basin Rept. 2105 (December 1965).

[34] Healey, J. J., "Parametric study of unstiffened and stiffened prolate spheroidal shells under external hydrostatic pressure," David Taylor Model Basin Rept. 2018 (August 1965).

[35] Kendrick, S. and McKeeman, J. L., "Pegasus computer specifications– axisymmetric stress analysis," Naval Construction Research Establishment Rept. R-452 (March 1961).

[36] Kendrick, S. and McKeeman, J. L., "Pegasus computer specifications—axisymmetric stress analysis—Part II," Naval Construction Research Establishment Rept. R-477 (May 1963).

[37] Arnold, R. N. and Warburton, G. B., "Flexural vibrations of the walls of thin cylindrical shells having freely supported ends," Proc. Roy. Soc. London, **A197,** 238–256 (1949).

[38] Willner, A. R. and Salive, M. L., "Materials survey for the rescue and search vehicles of the deep submergence systems project," David Taylor Model Basin Rept. 1987 (March 1965).

[39] Hom, K. and Couch, W. P., "Investigation of glass-filament reinforced plastic for deep-submergence applications," David Taylor Model Basin Rept. 1824 (September 1966).

[40] Meyers, N. C. and Fink, B., "Filament wound structural model studies for deep submergence vessels," Naval Engrs. J. **77,** 275–290 (April 1965).

[41] Proffitt, J. L., "Hydrostatic pressure tests of cylinders fabricated from hollow-filament, glass-reinforced plastic," David Taylor Model Basin Rept. 2132 (December 1965).

[42] Couch, W. P., "Hydrostatic, creep and cyclic tests of radially oriented glass-fiber reinforced plastic spheres," David Taylor Model Basin Rept. 2089 (September 1965).

[43] Kiernan, T. J., "An exploratory study of the feasibility of glass and ceramic pressure vessels for naval applications," David Taylor Model Basin Rept. 2243 (August 1966).

[44] Nishida, K., "Static and cyclic fatigue tests of fusion-sealed glass spheres," David Taylor Model Basin Rept. 2246 (August 1966).

JULY 1967 J. HYDRONAUTICS VOL. 1, NO. 1

Behavioral Cybernetic Theory Applied to Ship/Manipulator Control in Small Submarines

ALAN J. PESCH*
General Dynamics, Groton, Conn.

This paper discusses some of the human factors design problems in man-multiloop control systems from the standpoint of behavioral cybernetic theory. Specifically, it addresses itself to problems associated with small submersibles where an operator is required to control the vehicle's orientation in a mobile medium. This refers to control of hovering, trim angles, etc., while interacting with the external environment via manipulator arms and visual observation operations. An experiment is described which was conducted to evaluate the effects of three-dimensional television systems on the performance of manipulator tasks. Also covered in this paper are some of the problems encountered in the design of manipulator systems. Several manipulator control systems were researched and are described. The problem of ship control of small submersibles such as the Deep Submergence Rescue Vehicle (DSRV) is treated, including use of a simulating facility to study hovering and the mating of the vehicle with a sunken submarine.

Introduction

THE problem of optimally extending a man into the sea with TV cameras for vision, with manipulator arms for limbs, and with a controllable vehicle for movement and orientation is the current subject of research at Electric Boat. The optimization of a control system for small submersibles requires research guided by a total systems concept and by sound behavioral theory.

K. U. Smith at the University of Wisconsin is the head of a continuing research program, the results of which are being formulated into comprehensive theory of behavioral control and learning. Dr. Smith refers to this theory as behavioral cybernetic theory. The research work in this area by Smith and his associates directly applies to the design of devices and control systems, which require behavioral extensions of man's capabilities (vision, manipulation, locomotion) into an external environment.

The basis for behavioral cybernetic theory is the assumption that the organization of motor performance and learning is based on the properties of closed-loop feedback control. The theory describes man as a continuous controller using self-integrated and differentiated information from multiloop feedback circuits to guide sensing and response to a dynamic environment. This theory provides a reference from which it is possible to describe and define transformations and perturbations of feedback and their effect on control performance. More importantly, it does not attempt to model mathematically the control process but tends to lay a format for understanding the control system of man and defining the requirements of machine design in respect to his multidimensional control response. Research in this area at Electric Boat has attempted to define the normal and breakdown ranges for feedback perturbation relative to our problems of optimally designing extensions of man under the sea.

Underwater Television Research

The question of where to place a TV camera for optimum observation and manipulation by the operator of a small submarine is the object of a current research project at Electric Boat. The placement must be integrated into a systems configuration, considering lighting, operator placement, manipulator placement, and the specific functional requirements of the submarine. This magnifies the placement problem, since each element of the system requires optimization of its characteristics. Placement requirements also vary directly with the intended use of the visual loop. For instance, simple observation and "ball park" ship control do not require special or anthropomorphic placements; however, manipulator and precise ship control, as in a DSRV mating sequence, do require special camera placements and angles.

The decision to follow on the anthropomorphic TV camera placement scheme for manipulator operation was based on the results of studies examining experimentally displaced vision. Work by Smith at the University of Wisconsin particularly applies to our problems as his experiments used television techniques to displace the visual field. Specifically,

Presented as Preprint 66-712 at the AIAA/USN 2nd Marine Systems and ASW Conference, Los Angeles-Long Beach Calif., August 8–10, 1966; submitted July 27, 1966. [12.07, 12.11]

* Research Scientist, Human Factors Section, Research and Development Department, Electric Boat Division.