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**STRUCTURAL DESIGN OF VIEWING PORTS
FOR OCEANOGRAPHIC VEHICLES**

by

James A. Nott

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ABSTRACT

A design procedure for the reinforcement of viewing port openings in spherical shells for deep-diving oceanographic vehicles is presented. This procedure is based on the concepts of membrane deflection and zero rotation of the spherical head at the viewing port intersection. A small model was tested to determine the validity of the developed procedure. Strain measurements indicated that no appreciable bending occurred and that the structure reacted in a highly favorable manner under hydrostatic loading.

INTRODUCTION

The exploration of the sea has long been of interest to the oceanographers both for gathering technological data and for simply observing sea life. This interest has created, therefore, a necessity for abyssal passenger-carrying vehicles. These vehicles must be designed to withstand the external hydrostatic pressure of the ocean's depths, and, since the weight of the vehicle's supporting structure must be kept at a minimum, efficient design techniques must be developed.

One of the problems in the design of an abyssal vehicle is that of determining the size of the reinforcement required to stiffen a viewing port, and, also, what effect the reinforcement has on the load-carrying capacity of the main structure of the vehicle. This report presents (1) a criterion for the design of a viewing port ring reinforcement and (2) an application of this criterion to the design of a viewing port for a proposed 15,000-ft-operating-depth vehicle. Test results of a small-scale machined model of the proposed design are also given and discussed.

PART I

DESIGN CRITERION FOR A VIEWING PORT

The relative geometry of a viewing port window for an oceanographic vehicle can be obtained from the early work of Piccard.¹ A sketch of a typical window designed in accordance with Piccard's finding is shown in Figure 1. The window is in the shape of a truncated cone with its small diameter facing the inside of the vehicle.

To arrive at a rational design of the reinforcement required for this type of viewing port window, a procedure developed in Reference 2 is applied. This method was initially applied to the reinforced juncture of cylindrical and conical shells, and also was recently applied to the design of cylindrical ring reinforcements in spherical shells.³ Essentially, the design procedure is based on the assumption that the optimum design of a juncture of a shell with a ring reinforcement is accomplished by eliminating resultant moments caused by the reinforcement and, also, by allowing only membrane deflection to occur in the shell. Thus the entire shell, including the portion adjacent to the ring reinforcement, behaves in a membrane manner as though the penetration does not exist. This is attained by (1) selecting a ring with the cross-sectional area required to yield the desired membrane radial deformations under the applied loads and, then, (2) positioning the ring in such a manner as to eliminate any resultant moments.

When a ring reinforcement of rectangular cross section is used to stiffen the opening in a spherical shell, the deformation of the ring reinforcement under radial loads can be determined accurately by Lamé's formula.⁴ The deflection of the sphere, in the direction radial to the ring and at the juncture of the ring and shell, can be determined by equilibrium and by simple geometric relationships.

¹References are listed on page 14.

Thus the area required to permit only membrane deformations to occur in the shell can be determined accurately if the behavior of the ring reinforcement follows the assumptions of Lamé, which are for very thick cylinders. Since the locations and magnitudes of the applied forces and reactions can be readily obtained for the cylindrical ring reinforcement, the location of the reinforcement which eliminates all resultant moments at the reinforcement-spherical shell juncture can also be accurately determined.

In designing a viewing port ring reinforcement similar to that shown in Figure 1, several departures from the above-described procedure are dictated. These changes must be introduced because of the asymmetric cross section of the ring reinforcement and the nature of the applied load through the plexiglass window. One departure is the introduction of a less accurate ring formula, since the Lamé formula is not applicable to nonsymmetrical geometries. Another departure is that the assumed distribution of loads through the plexiglass window must be based, to some extent, on engineering judgment. With these considerations, a typical viewing port for a deep-sea vehicle can be designed in the following manner.

The equations for the deflection and rotation of a thin ring are⁵

$$\Delta_R = \frac{(R_{CG})^2 F_R}{EA} \quad [1]$$

and

$$\theta_R = \frac{(R_{CG})^2 M}{EI} \quad [2]$$

respectively, where

- R_{CG} is the radius from the center of the window to the center of gravity of the cross section of the ring reinforcement,
- M is the total moment of the reinforcement,

E is Young's modulus,
 I is the moment of inertia of the cross section of the reinforcement,
 F_R is the resultant force applied to the reinforcement, and
 A is the cross-sectional area of the reinforcement.

Equations [1] and [2] represent expressions for the deflections and rotations of thin rings. The Lamé equations,⁴ which are for thick wall cylinders, are more realistic for the design of a heavy rectangular ring;³ however, in the case of a nonsymmetrical viewing port ring reinforcement, the Lamé thick wall theory does not apply. As a result, the more approximate expressions for the deflections and rotations are used in Equations [1] and [2].

To determine the required area A of the ring reinforcement, the reinforcement must be designed to (1) resist the total radial forces acting on the reinforcement and (2) deflect at the reinforcement-head juncture the same distance as a uniformly loaded complete spherical head would deflect if the reinforcement did not exist. Therefore, when the radial ring reinforcement deflection Δ_R of Equation [1] is equated to the transverse deflection of the sphere Δ_S , a membrane deflection will exist and the required area of the ring reinforcement can be determined. The transverse deflection of a spherical head Δ_S is

$$\Delta_S = \frac{p R_{so} R_{sm} (1-\nu) \sin \theta}{2 E h} \quad [3]$$

where p is pressure,
 R_{so} is the outside radius of the sphere,
 R_{sm} is the mean radius of the sphere,
 ν is Poisson's ratio,
 θ is the angle from the centerline of the window to ring-head juncture, and
 h is the thickness of the sphere.

The radii R_{so} and R_{sm} are used in Equation [3] so that equilibrium can be obtained on the viewing-port-ring reinforcement with the pressure acting on the outside surfaces of the ring reinforcement and spherical head. Figure 2 illustrates the dimensional notations of the ring and sphere together with the resultant forces which are applied to the ring. Setting $\Delta_R = \Delta_S$, the required area A can be expressed as

$$A = \frac{(R_{CG})^2}{R_{so} R_{sm}} \frac{2 F_R h}{p(1-\nu) \sin \theta} \quad [4]$$

The radial ring force F_R in Equation [4] is

$$F_R = \frac{R_{Ro}}{R_{CG}} (F_s \cos \theta + F_e - F_w \sin \phi) \quad [4a]$$

where F_s , F_e , and F_w are the forces acting on the ring reinforcement, as shown in Figure 2. It should be noted that the force F_w , acting on the surface ABCD, is that force which is transmitted to the ring from the transparent section of the window and that this force is assumed to act normally and along a circumferential line midway between lines AB and CD. The center of pressure of the surface ABCD is slightly closer to line AB; however, due to the absence of support of the plexiglass at the center of the viewing port, it appears that a greater unit pressure will exist in the neighborhood of line CD than will exist near line AB. Therefore, based on engineering judgment, the approximation is made that F_w acts at the midpoint. Also, the vertical component of the force F_w is equal to the magnitude of force on the outside of the window.

To determine the proper position of the ring reinforcement which will satisfy the initial assumptions of the problem for zero rotation, the location of the center of gravity of the ring with respect to the centerline of the spherical head must be such that the resultant forces

on the structure produce zero rotation at the reinforcement-head juncture. Therefore, from Equation [2]

$$\theta_R = 0 = \frac{(R_{CG})^2}{EI} \left\{ F_e(d - \bar{y} - \frac{e}{2}) - F_s(z \cos \theta - \bar{x} \sin \theta) \right. \\ \left. - F_b(\bar{x} - \frac{b}{2}) - F_w \left[\sin \phi(d - \bar{y} - \frac{l \tan \phi}{2}) - \cos \phi(\frac{l}{2} + b - \bar{x}) \right] \right\} \quad [5]$$

where the forces F_e , F_s , F_b , and F_w are determined from geometry (see Figure 2). From Equation [5], the distance z from the center of gravity of the ring reinforcement to the centerline of the spherical head at the reinforcement-head juncture can be expressed as

$$z = \frac{F_e(d - \bar{y} - \frac{e}{2}) + \bar{x} F_s \sin \theta - F_b(\bar{x} - \frac{b}{2})}{F_s \cos \theta} \\ + \frac{F_w \left[\cos \phi(\frac{l}{2} + b - \bar{x}) - \sin \phi(d - \bar{y} - \frac{l \tan \phi}{2}) \right]}{F_s \cos \theta} \quad [6]$$

It should be noted that the quantity e in Equation [6] is a function of z and, therefore, z is determined by numerical iteration. When the computed value of z in Equation [6] is equal to the assumed value of z in the structure, the initial assumption of zero rotation is satisfied.

With the use of Equations [4] and [6], the proper geometry of the ring reinforcement of a viewing port can be computed by the process of numerical iteration. To demonstrate the method, the following final set of calculations for a typical viewing port are presented.

The dimensions of the assumed geometry, utilizing the notations designated in Figure 2, are as follows:

$$R_{SO} = 48.000 \text{ in.}$$

$$h = 3.375 \text{ in.}$$

$$R_{RO} = 9.250 \text{ in.}$$

$$b = 1.750 \text{ in.}$$

$$\ell = 5.000 \text{ in.}$$

$$d = 7.000 \text{ in.}$$

$$e = 4.774 \text{ in.}$$

$$\alpha = \phi = 45 \text{ deg}$$

From geometry considerations,

$$\bar{x} = 2.558 \text{ in.}$$

$$\bar{y} = 2.973 \text{ in.}$$

$$A = 32.75 \text{ sq in.}$$

$$\theta = 11 \text{ deg } 31 \text{ min}$$

For a unit load of pressure on the structure ($p = 1 \text{ psi}$), the forces F_e and F_s acting on the unit width of the element (see Figure 2) are

$$F_e = pe = 4.774 \text{ lb}$$

$$F_s = p \frac{R_{sm}}{2} = 23.156 \text{ lb}$$

Also, the forces F_b and F_w , corrected in proportion to the ratio of their radii from the centerline of the window to the points of action of the forces and the radius from the centerline of the window to the unit width of the element, are

$$F_b = pb \left(\frac{R_{RO} - \frac{b}{2}}{R_{RO}} \right) = 1.584 \text{ lb}$$

$$F_w = p \frac{(R_{RO} - b)^2}{2 R_{RO} \cos \phi} = 4.300 \text{ lb}$$

The radial force of the ring F_R , corrected for its action at the centroid of the cross section of the element, can be determined from Equation [4a] to be

$$F_R = \frac{R_{Ro}}{R_{CG}} (F_s \cos \theta + F_e - F_w \sin \phi) = 33.76 \text{ lb}$$

The required area of the ring reinforcement needed to bring the structure into equilibrium can now be determined from Equation [4]; it is

$$A = \left(\frac{R_{CG}^2}{R_{so} R_{sm}} \right) \frac{2 F_R h}{p(1-\nu) \sin \theta} = 34.03 \text{ sq in.}$$

This required area of 34.03 sq in. is slightly greater than the assumed area, 32.75 sq in., of the initial geometry. The additional 1.28 sq in., which is required, can be added in the fairing radii above and below the hemispherical-head centerline at the reinforcement-head juncture.

The distance z , which defines the eccentricity that the force F_R must be offset for zero rotation of the ring reinforcement, as shown in Equation [6], can be determined as

$$z = \frac{1.640 F_e + 0.5109 F_s - 1.683 F_b + 0.1167 F_w}{0.9799 F_s} = 0.77 \text{ in.}$$

The value of z , as initially assumed, was

$$z = \bar{y} + e - d = 0.75 \text{ in.}$$

A final adjustment can be made in the geometry so that the assumed value of z is exactly the same as the computed value of z in Equation [6]; however, for design purposes a nominal value of 3/4 in. can be used.

PART II

EXPERIMENTAL PHASE OF DESIGN DEVELOPMENT

To establish a confirmation of the design criterion in Part I, a viewing port for a deep-diving oceanographic vehicle, designed for an operating depth of 15,000 ft and a weight displacement ratio of 63 percent, was developed. The pressure hull of the vehicle consists of a sandwich-type cylindrical shell with hemispherical heads on each end. Figure 3 illustrates the overall geometry of the vehicle. The structure is of a welded-type construction fabricated with HY-120 titanium. By use of the sample calculations in Part I, a scale model of the hemispherical head with a viewing port was designed, fabricated, and tested. This model, designated Model OV-5.1, has a shell thickness-radius ratio the same as that of the prototype and is machined from aluminum. The relative lower modulus of aluminum, as compared with titanium, will not affect the elastic stresses in the model if the rigidity of the window is insignificant. The required geometry of the viewing port is not a function of the mechanical properties of the material if it is assumed that the loads acting on the port are linear functions of the pressure only. This assumption was made in calculating the proportions of the model.

DESCRIPTION OF MODEL

Model OV-5.1 is a machined hemispherical shell with a viewing port at the apex of the hemisphere and a ring-framed cylindrical shell at the open end of the hemisphere. The geometry of the ring reinforcement for the viewing port was designed for membrane deflection and zero rotation, as discussed in Part I. Details for the geometry of Model OV-5.1 are shown in Figure 4. The juncture of the hemispherical head to the cylindrical shell was also designed for membrane deflection and zero rotation similar to the procedure given in Reference 2.

Because of its machinability qualities, 7075-T6 aluminum was used for the structure of the model. A shell thickness-diameter ratio for a 15,000-ft-operating-depth submarine was used so that equivalent strains and stresses could be determined for different structural materials. A plexiglass window in the form of a truncated cone with its small end facing the inside of the vehicle was used for the transparent section of the viewing port. The diameter of the smaller end is 0.20 in. (5.00 in. prototype, which is a size sufficient for visual observations with both eyes). The cone is truncated at an angle of 45 deg. The surfaces of both the plexiglass and the aluminum were machined to a fine finish to ensure a pressure-tight seal when the structure is subjected to external hydrostatic pressure. The thickness of the plexiglass was chosen to be the same as the diameter of the small end of the opening; i.e., $d_1/D = 1$ (see Figure 1). This relative thickness was 50 percent less than the thickness utilized by Piccard in the cabin of the TRIESTE; however, permanent deformation of the plexiglass windows of the TRIESTE was estimated to begin at depths greater than 63,000 ft.¹ From the results of model tests conducted by Piccard (see Reference 1), the estimated depth at which permanent deformation will begin in a window with a value of d_1/D equal to 1 is 32,000 ft. This value gives a safety factor greater than 2 for a 15,000-ft-operating depth.

Figure 5 shows Model OV-5.1 and the plexiglass window after fabrication.

INSTRUMENTATION

Model OV-5.1 was instrumented with electrical-resistance, foil-type strain gages. Both the inside and outside surfaces were instrumented at orientations from $6\frac{2}{3}$ to $22\frac{1}{2}$ deg from the centerline of the window to observe any bending strains which might occur on the window or on the hemispherical-head surfaces close to the window-head

junction. Also, gages were oriented at $33\frac{3}{4}$ and 45 deg from the center of the window to determine the normal behavior of the hemispherical head to be used for a direct comparison with the behavior of the structure around the viewing port penetration. Figure 6 illustrates the arrangement of instrumentation for Model OV-5.1. Each position represents one circumferential and one meridional gage. No gages were placed on the plexiglass.

PRESSURE TEST

Model OV-5.1 was subjected to external hydrostatic pressure in a 10-in. pressure tank with oil as a pressure median. Three pressure runs were made to 3000 psi and strains were observed at intermediate increments.

The open end of the model was sealed by a rigid closure bulk-head with an "O"-ring seal. The plexiglass window had no mechanical seal at its junction with the aluminum viewing-port-ring reinforcement. A thin film of silicone grease was applied between these two surfaces and the resultant load on the outside surface of the plexiglass was expected to make a pressure tight seal. Tape was used to hold the window in place while installing the model in the test tank.

TEST RESULTS

Elastic-strain sensitivities, defined as the slopes of the pressure-strain curves, were determined from the test data. These strain sensitivities, which represent a numerical average of the sensitivities obtained for the three pressure runs, are shown in Figure 6 and are given in microinches per inch per psi. All strain data were linear for all three pressure runs. The upper and lower numbers shown in the figure are the circumferential and meridional strain sensitivities, respectively.

Biaxial stress sensitivities were computed from the circumferential and meridional strain sensitivities.

Young's modulus, as determined by optical strain gage measurements on four specimens, varied between 10.6×10^6 psi and 11.3×10^6 psi depending upon the orientation of the specimen in the bar stock. A Young's modulus of 10.8×10^6 was assumed for all calculations. Poisson's ratio was not determined experimentally, but a value of 0.3 was utilized. The stress sensitivities are illustrated graphically in Figure 7. The ideal membrane stress, $p \frac{R_{so}^2}{2 R_{sm} h}$, is shown superimposed on the graph and illustrates a comparison between theoretical and experimental results.

DISCUSSION AND CONCLUSIONS

The test results indicate that no significant bending occurred in the area of the viewing port-hemispherical head juncture. If bending had occurred, due to rotation at this juncture, it would have been detected from strain readings at the 14- and 17-deg locations. Figures 6 and 7 both show, however, that bending did not occur at these locations. Also, these strains agree favorably with those values obtained at the 33 3/4- and 45-deg locations. This indicates that the entire viewing port structure acted as a continuous unit and deflected uniformly with the hemispherical head. Stresses for the inside surface were slightly higher than the ideal membrane stress; stresses for the outside surface were slightly lower than the ideal stress because of the difference in the inner and outer radii.

The response of the window under loading was favorable. A visual observation of the interior of the model after the test showed that no leakage occurred at the plexiglass-ring reinforcement juncture. The plexiglass itself was not structurally evaluated since the previous test results of Piccard demonstrated the feasibility of the material.

The safety factor of slightly greater than 2, for which the thickness of the plexiglass was designed, should be sufficient for a 15,000-ft-operating vehicle. That is, permanent deformation is not predicted to occur in the plexiglass until depths greater than 30,000 ft are reached.¹

The final results of the test indicate that for the geometry studied and tested, the design criterion presented in Part I produces a geometry in which the initial assumptions of membrane deflection and zero rotation are satisfied. The test of Model OV-5.1, however, was not a complete experimental analysis of the behavior of the structure. For example, stresses on the plexiglass window and, also, on the surface of the viewing port adjacent to the plexiglass were not determined due to the relatively small size of the model.

When using Equations [4] and [6] for the design of a viewing port of a new size and shape, it should be remembered that thin ring theory has to be utilized. Also, the location of the resultant load transmitted by the plexiglass to the ring reinforcement has been assumed. The position of this load could also possibly be affected by the relative moduli of the plexiglass and the ring reinforcement, since the two materials will deform unequally under a given load. Therefore, if a newly designed port has a ring reinforcement which is sufficiently different from that in Model OV-5.1, a model test of the resulting new viewing port is recommended. The investigation of Model OV-5.1 was a pilot study in which satisfactory results have been shown. These results, however, could be expanded, if desired, by the use of larger scale models or photoelastic studies.

ACKNOWLEDGMENTS

Appreciation is expressed to Mr. M. A. Krenzke for guidance throughout the course of this project. Thanks is also expressed to Mr. C. D. Bradfield for instrumenting the model and assisting in the conduct of the test.

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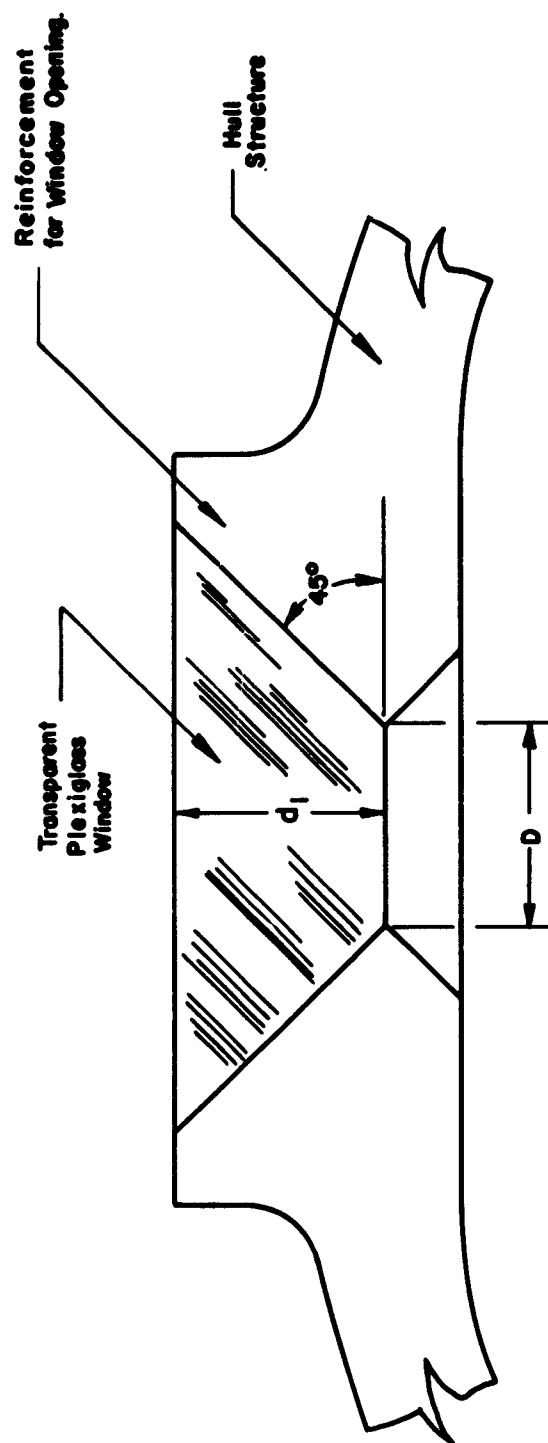


Figure 1 - Viewing Port for Oceanographic Vehicle

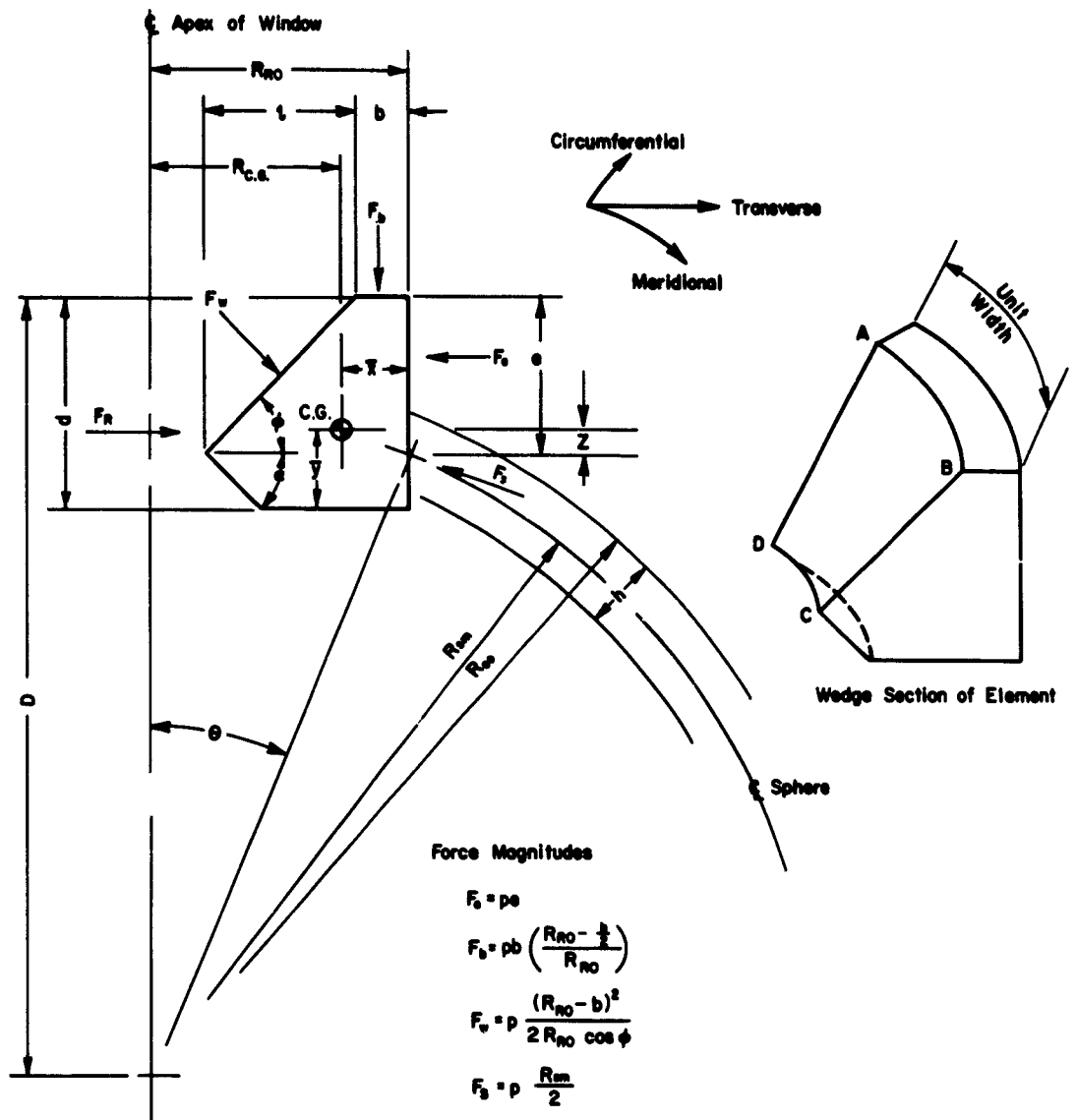


Figure 2 - Resultant Forces on Viewing Port Ring Reinforcement

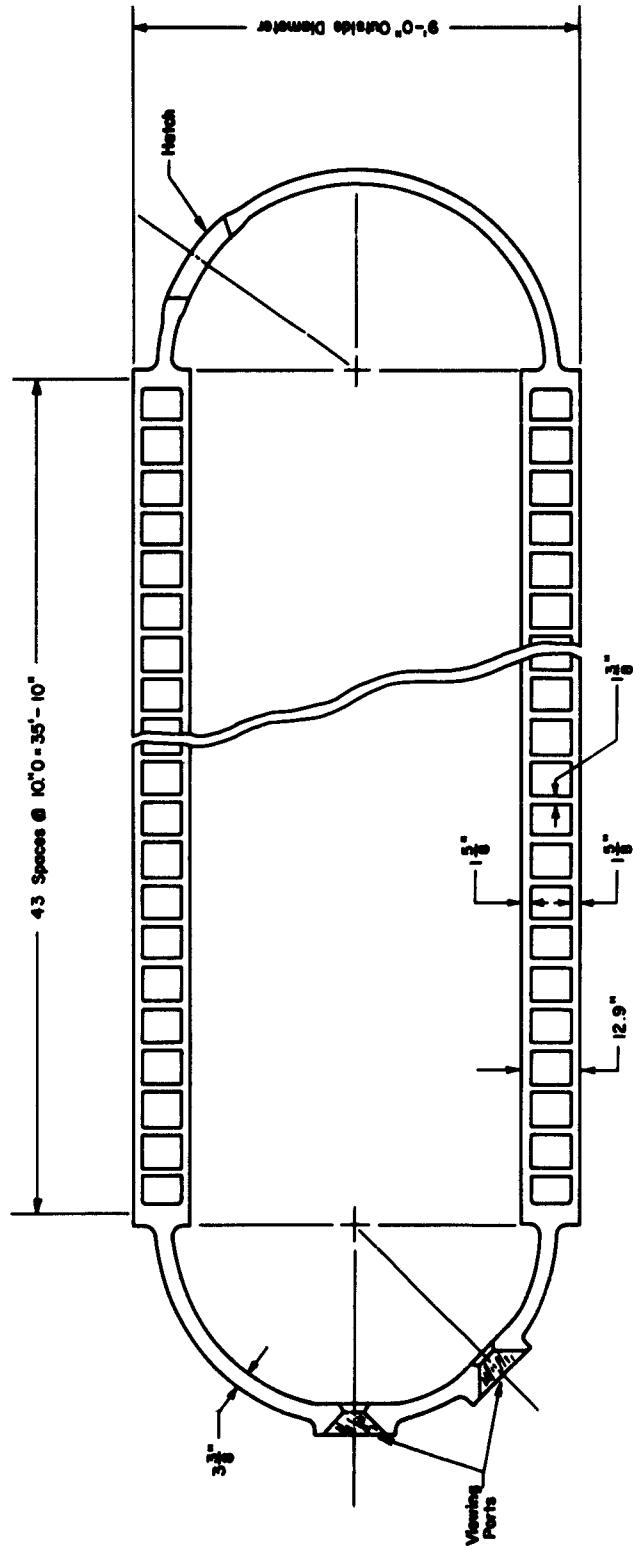


Figure 3 - Titanium Sandwich Pressure Hull for Proposed Oceanographic Vehicle

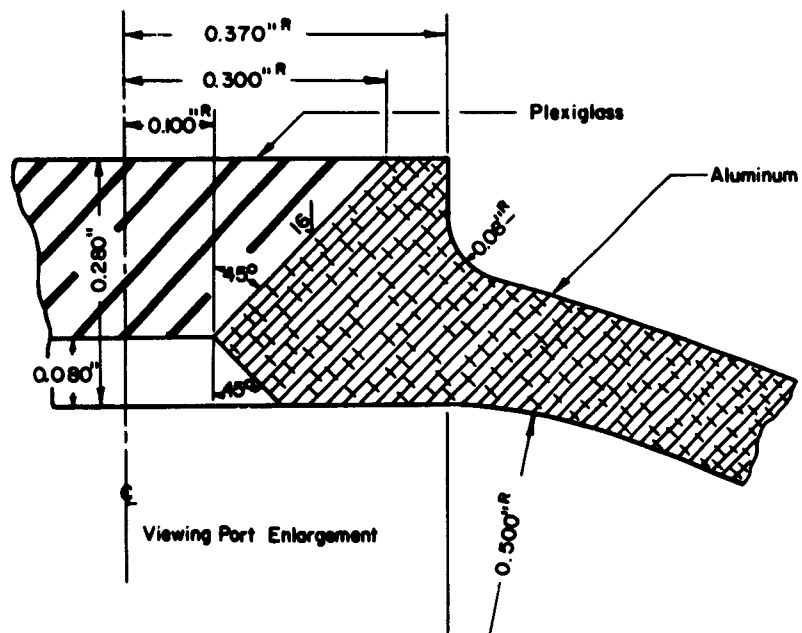
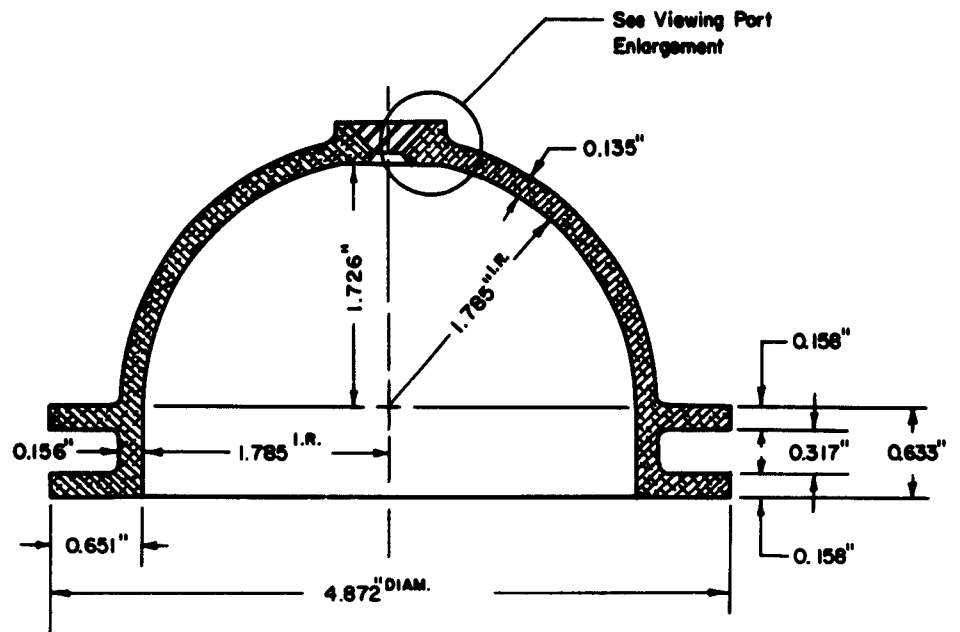


Figure 4 - Details of Model OV-5.1

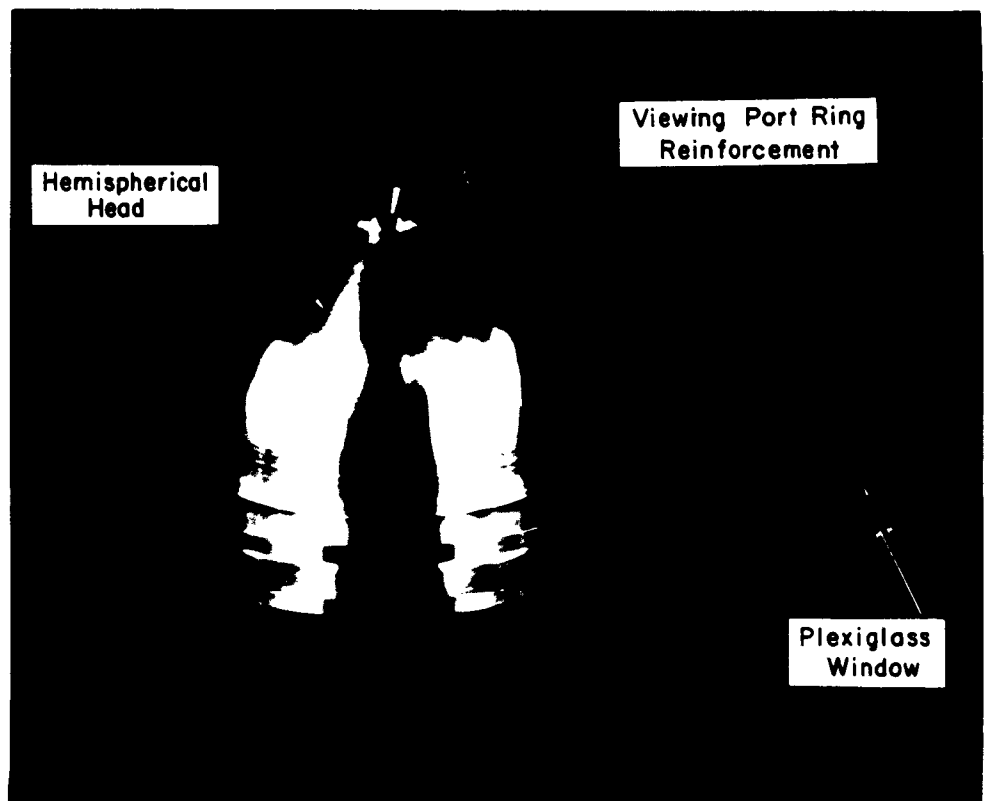


Figure 5 - Model OV-5.1

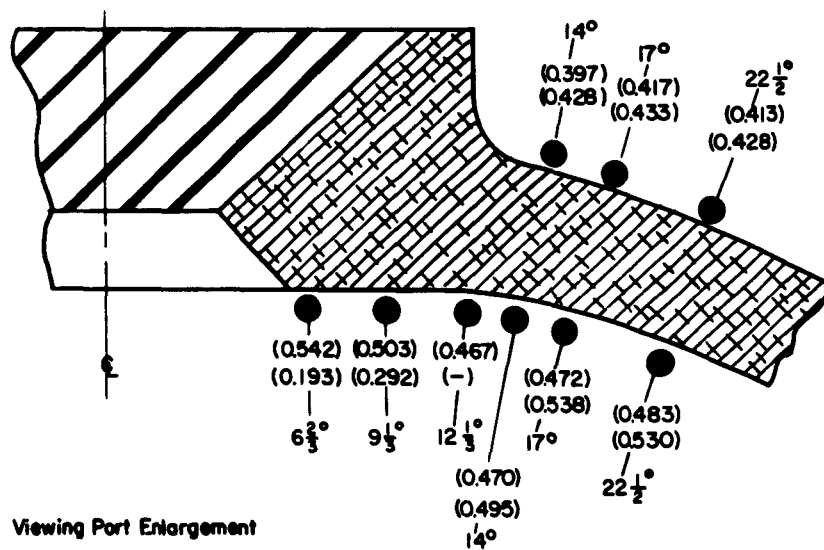
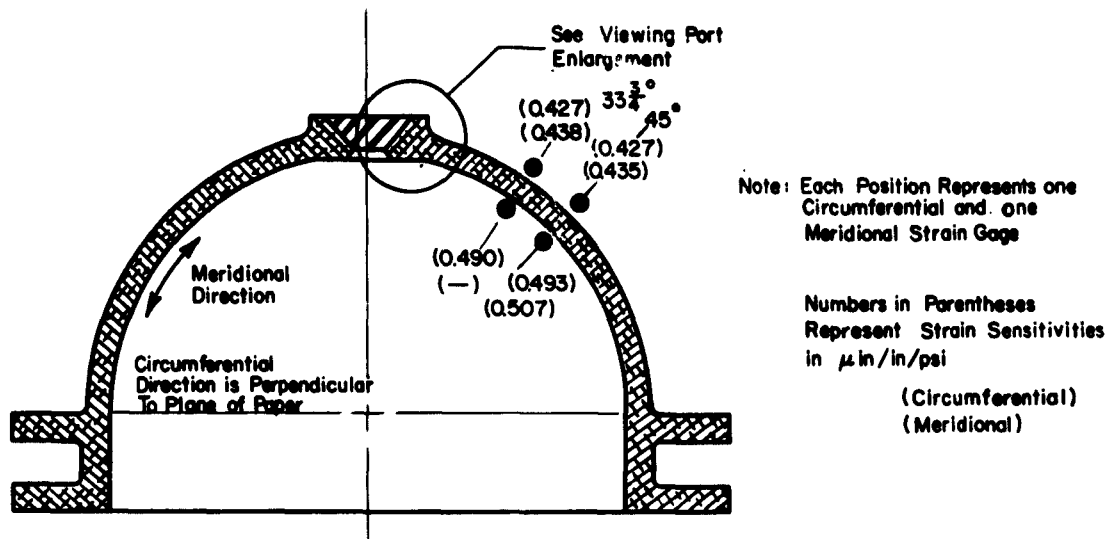


Figure 6 - Instrumentation and Strain Sensitivities for Model OV-5.1

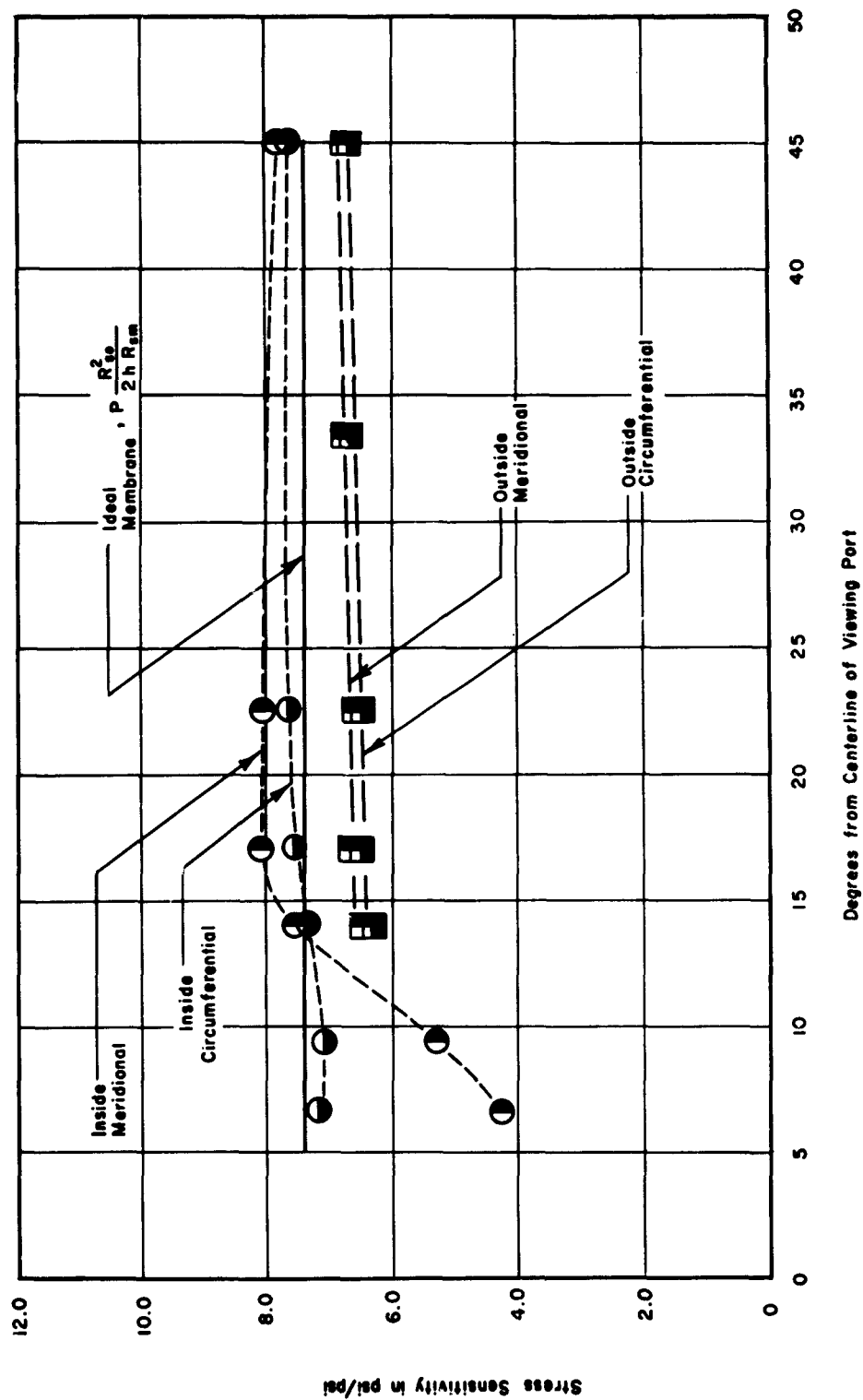


Figure 7 - Stress Sensitivities on Viewing Port and Hemisphere of Model OV-5.1

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STRUCTURAL DESIGN OF VIEWING PORTS FOR OCEANOGRAPHIC VEHICLES, by James A. Nott. Mar 1963. ii, 22p.
illus., photos., refs.

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Openings--Design
2. Surveying ships
3. Hemispherical shells--
Model OV-5.1
- I. Nott, James A.
- II. S-F013 01 08

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