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Research paper

Concerning the Determination of the Geometrical Parameters of the Optical Element of a Deep-Water Surveillance System

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Abstract

This paper is devoted to the question of determining the geometrical parameters of the most common types of the optical element of the deep-sea porthole. The considered element is to be used as part of a lighting system operable at a depth of 5000 meters. The calculation was made without taking into account the influence of temperature.

Keywords: *Deep-water porthole, Schott N-BK7 glass, conical porthole, flat disk window, spherical shell windows with conical seat, optical element, deep-sea video surveillance system*

1. Introduction

The development of existing and the emergence of new technologies of research in various fields of human activity poses the task of modern science to create new tools for addressing problems facing modern society. The study of the ocean floor is essential for various areas of human activity. The substantial absorption of electromagnetic radiation in a wide spectral range from the ultraviolet to the infrared part of the spectrum limits the possibilities of surveillance in seawater at various depths. An integral part of both the surveillance system itself and the illumination system is the porthole. The requirements that are imposed on the material of the porthole of deep-sea surveillance systems should take into account its use at high and ultrahigh pressures, ultra-low temperatures, and gradients of these physical quantities. In addition, the selection of the material of the optical element should take into account: resistance to the chemical effects of seawater; the requirements for optical components, namely, high transparency in the spectral range used, the possibility of using known technologies for processing optical materials both for obtaining transparent surfaces (grinding hardness) and for imparting the required shape. Studies on this topic are presented in papers [1-6, 15-17].

2. Problem Statement

To create a reliable and, at the same time, effective for surveillance porthole, it is necessary to determine the optimal configuration of the optical element of the porthole, analyzing the dependence of key characteristics: strength at specified operating conditions (pressure, temperature, chemical aggressiveness of the environment), maintainability of the optical transparency of the element during operation. When choosing a material of an optical element such as - K8 (Schott N-BK7), the analysis of the influence of the temperature mode of operation, preservation of optical transparency, and even the resistance of chemical aggressiveness of the medium take a back seat compared with the definition of geometric characteristics.

The most popular configurations of the optical element are: a flat cylinder (disk, round thick plate) and a truncated cone, less popular - a truncated globular sector (“dome”). In order to increase the payload, as well as reduce the weight of the assembly, it is required to analyze the effect of geometrical parameters, such as the angle of taper and relative thickness, on the strength characteristics.

3. Strength Criterion

Depending on the intensity of loading, the same material may be in different mechanical states. A high level of external exposure can cause irreversible changes in the shape and size of the material structure.

When solving the problem, the following features will be taken into account: glass is a brittle material that works well only for compression, has significantly lower tensile and bending strength, axisymmetric body geometry and applied loads.

In the framework of the task, it is important to use the updated strength criteria of Pisarenko-Lebedev [7,8]:

$\sigma_{eq} = \chi\sigma_i + (1-\chi)\cdot\sigma_1 \leq \sigma_{max}$,

Where σ_i is the stress intensity, which is defined by the expression:

$\sigma_i = \sqrt{{\sigma_1}^2 + {\sigma_2}^2 - \sigma_1\sigma_2}$;

σ_1, σ_2 are the components of the main stress vector;

χ – the material plasticity factor, is found experimentally, numerically equal to:

$\chi = \frac{\sigma_t}{\sigma c}$;

σ_t, σ_c – ultimate tensile strength and compression, respectively;

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σ_{max} – the maximum permissible stress value, numerically coincides with σ_t.
For a fragile material, other strength criteria can be used, for example, the William-Warnke criterion, but it requires experimental data.

4. Original Data

The numerical determination of the stress-strain state of the optical element was carried out in conjunction with the porthole housing. The following are all the main characteristics used in modeling the stress-strain state and in analytical assessment of thickness of the element.
Physico-mechanical characteristics of the materials under consideration [9-12] are listed in the table:

Table 1: Mechanical properties

Material (analogue)	Elastic modulus *E*, GPa	Poisson's ratio ν	Density ρ, kg/m^3	σ_t, MPa	σ_c, MPa
K8 porthole glass (SchottN-BK7)	81	0.206	2510	88	377
Titanium BT1-0 porthole housing (Ti6Al4V)	110	0.32	4450	860	—

Loads and boundary conditions
Internal operating pressure P1 = 1 atm. (0.1 MPa); When modeling one must consider that water exerts pressure on the entire structure when immersed, and induces an increase in pressure inside the volume.
External hydrostatic pressure P2 = 630 atm. (63 MPa);
Rigid fastening of the frame in accordance with the drawing;
At the interface of media, the coupling condition is satisfied:
$\overline{\sigma_{1_n}} + \overline{\sigma_{2_n}} = 0; \overline{u_1} = \overline{u_2};$
In all design schemes, the diameter of the optical component is constant and equal to D=80 mm.
We consider that the quality of the processed surface of the optical component corresponds to the following characteristics (processing parameters are given according to GOST 11141-84 [14]):
working surfaces: roughness - RZ=0.05; surface finish - P=II; shape error - N=1 - 5; local error - DN=0.1 - 0.5
auxiliary surfaces: roughness - RZ=0.05–0.3; surface finish - P=IV; shape error - N=10 - 05; local error - not controlled.
Such quality of finishing eliminates surface defects.
An approximate conformity of the quality of the finished surface with US standards is presented below.
Proceeding data accordingly to MIL-O-13830 and MIL-G-174 standards:
1. Working surfaces: - roughness – polished fine;- surface quality – 60/40; - form error – 1 Fringe;- irregularity – DR/Fringe=0,002;
2. Other surfaces: - roughness – polished fine; 4. – surface quality – 80/50; 5. – form error – without control; 6. – irregularity – without control

5. Calculation of the Disc-Shape Deep-Water Porthole

Due to the immersion of the submersible to deep water, the source of destruction of porthole glasses are zones of force contact of dissimilar materials or maximum stresses acting in the area of the stress concentrator. The nature of the violation of the integrity of the glass element always has a similar appearance [7-10], the internal surface is destroyed first.
To find the optimal thickness, a series of numerical calculations of the porthole was carried out at various relative thicknesses h/D ($0.2 \leq {}^{h}/_{D} \leq 0.5$).
An important aspect in the design of a disk-shaped optical porthole is the selection of proper geometrical dimensions, as well as the optimal loading pattern. The following are the results of four load schemes (Figure 2, 1-4):

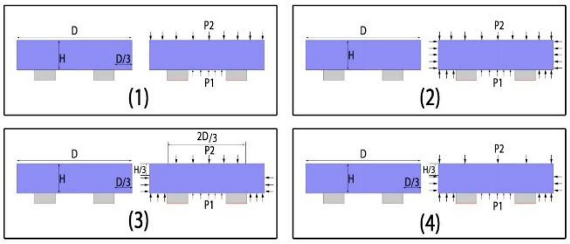

Fig. 1: Schemes of loads: 1) - High hydrostatic pressure is applied only to the upper surface of the porthole; 2) - Around the perimeter of the disc outside the annular support; 3) - Scheme of loads proposed by I.I. Diachkov; 4) - The proposed scheme of loads.

According to studies [1], an effective solution would be a variant with a partial distribution of external hydrostatic pressure over both bases and a cylindrical surface (Fig. 1.3). An o-ring is to be installed on the unloaded surfaces. However, this model is technically laborious, therefore, we shall consider a scheme with a uniform distribution of pressure over the entire upper surface and partial loading of the lateral and lower faces (Fig. 1.4).

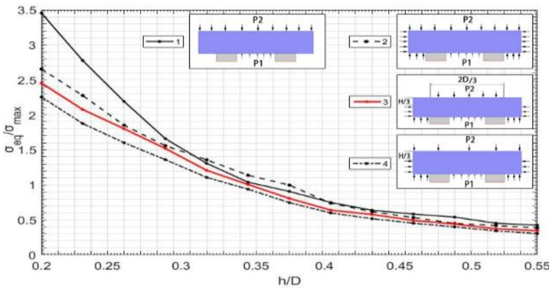

Fig.2: Dependence of σ_{eq}/σ_{max} on the relative thickness of the item (h/D) under various loading methods.

The thickness of the item varies on margin of safety used in the calculations. The paper [6] indicates that such problems use the minimum margin of safety equal to 2, which corresponds to the relation σ_{eq}/σ_{max} = 0.5. Then, with such a safety margin, we obtain the ratio h/D = 0.4, which corresponds to the thickness of the glass element h = 32 mm. However, in paper [1], based on experimental data, it was said that h/D = 0.52 ... 0.55 should be considered a good engineering solution, which corresponds to σ_{eq}/σ_{max} = 0.33. Therefore, based on numerical calculations, the relative thickness of the item, h/D should be chosen in the range from 0.4 to 0.55 (the height of the item cannot continuously increase due to the deterioration of optical properties).
Experimental results of similar studies are presented in the work by V.P. Lianzberg [4]. The author proposes to use leucosapphire as a material of the optical element, which has the best physicomechanical characteristics. However, this leads to a rise in price by 50-80 times.

6. Calculation of the Frustoconical Deep-Water Porthole

The second most common design of the porthole is a truncated cone. During operation, the relative thickness of the optical element varied within $0.2 \leq {}^{h}/_{D} \leq 0.5$, and the taper angle was $50° \leq \varphi \leq 90°$.

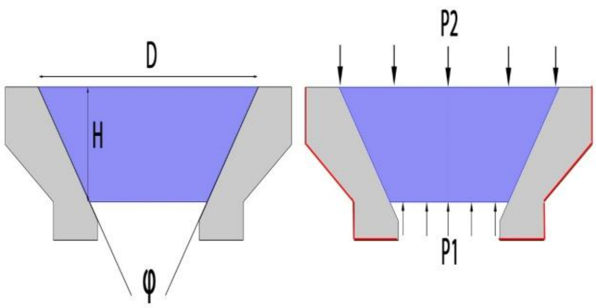

Fig. 3: Schemes of loads

Under hydrostatic pressure, the porthole is bent and crimped around a conical support surface. In this case, on the surface of small diameter, the area near its center will be under biaxial equal compression. Near the edges of this surface, in the area of contact of dissimilar materials, there will be a zone of maximum compressive stresses, the so-called stress concentrator.

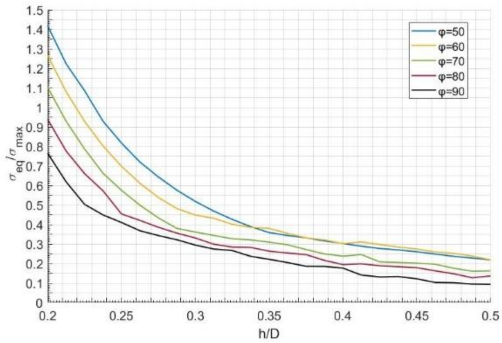

Fig. 4: Dependence of σ_{eq}/σ_{max} on the relative thickness of the item

Figure 4 shows the dependence of σ_{eq}/σ_{max} on the relative thickness of the item (h/D) at different angles of taper.

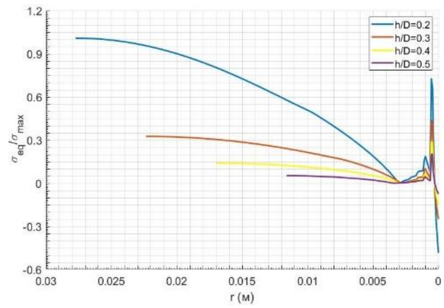

Fig. 5: Dependence of σ_{eq}/σ_{max} on the radius of the small-diameter surface.

Figure 5 shows a general picture of the ratio σ_{eq}/σ_{max} for different values of h/D. One can clearly see that the diagrams monotonously decrease to a certain value, after which the curve begins to oscillate when the stress concentration zone is reached. An increase in h/D is accompanied by a decrease in fluctuations.

We should understand that numerical calculations based on FEM give only an approximate solution with a certain degree of accuracy. Therefore, to assess the accuracy of the solution and the correctness of the specification of the boundary conditions, it is necessary to compare the numerical data with the analytical solution.

A.V. Lavrov [13] shows in his work the dependencies for calculating the circumferential and radial stresses of the conical porthole on the surface of small diameter:

$$\sigma_r = A - \frac{3P}{8}\left(\frac{r'}{h}\right)^2 (3+\nu)\left[1 - \left(\frac{r}{r'}\right)^2\right];$$

$$\sigma_\theta = A - \frac{3P}{8}\left(\frac{r'}{h}\right)^2 \left[(3+\nu) - (1+3\nu)\left(\frac{r}{r'}\right)^2\right];$$

where $A = \dfrac{R_0 P}{(R_0 + r)\, h} \cdot \dfrac{\cos\alpha - \mu \sin\alpha}{\cos\alpha - \mu \sin\alpha}$;

where P is the value of the external hydrostatic pressure; r is the current radius on the surface of small diameter d; r'= d/2; h is the porthole thickness; α is the opening angle of the cone; ν is Poisson's ratio; μ is the coefficient of sliding friction between the glass and the frame.

In view of the fact that an isotropic material is considered under biaxial compression, in this case $\sigma_1 = \sigma_r$, $\sigma_2 = \sigma_\theta$, then the Pisarenko-Lebedev strength criterion (1) taking into account ratios (2), (3) takes the form:

$$\sigma_{eq} = \chi\sqrt{\sigma_r^2 + \sigma_\theta^2 - \sigma_r\sigma_\theta} + (1-\chi)\cdot\sigma_r \le \sigma_{max}.$$

Unfortunately, the correlation between analytical evaluation and numerical results has a narrow operating range - the angle of taper must be within ϕ = 75..85°.

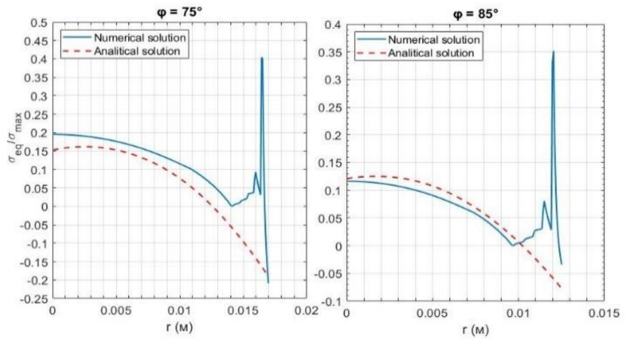

Fig. 6: Comparison of the FEM numerical solution and the analytical formula (4) for h = 30 mm at ϕ = 75° and ϕ = 85°.

For the considered geometric shape of the optical element, the relative thickness of the item should be within h/D = 0.31..0.44, with taper angles ranging from 70° to 85°.

7. Calculation of the Dome-Shaped Deep-Water Porthole

As we know from the course of structural mechanics, the dome-shaped roof structures hold the load better than flat ones. This is due to the fact that the domes have a large surface area and distribute loads more evenly. Further the object of research will be a porthole in the form of a "dome". The design is a complex model in the form of a spherical sector, characterized by: porthole diameter D, internal and external radii, R_a and R_b, respectively, and the opening angle of the cone ϕ.

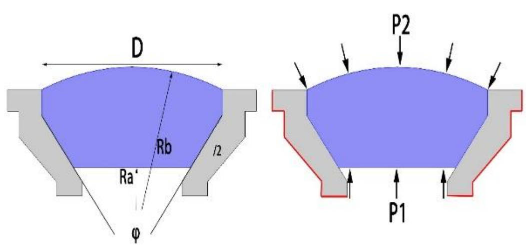

Fig. 7: Schemes of loads

Based on the Lame problem [14] of a hollow sphere, a formula was obtained for an approximate estimate of the thickness of the optical element. The thickness h can be expressed by the formula:

$$h = R_b - R_b\left|\frac{(-4P_a + 12P_b - 8\chi)(P_a - \gamma)^2}{8(\gamma - P_a)^3}\right|^{1/3} \quad (2)$$

Where P_a, P_b are internal and external pressure, respectively; R_a, R_b are the inner and outer radii; γ is the strength of the source material. A numerically calculated porthole according to the strength criterion of Pisarenko-Lebedev (1), showed satisfactory results.

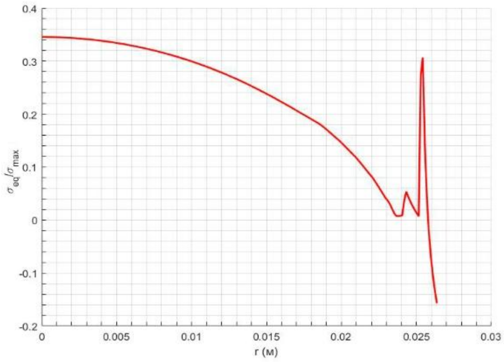

Fig. 8: A plot of σ_{eq}/σ_{max} against radius r(m).

Figure 8 shows σ_{eq}/σ_{max}, and their maximum is reached in the center of the base (bottom plane) of the porthole. It should be understood that formula (5) is rather rough and provides a threefold margin of safety for K8 glass.

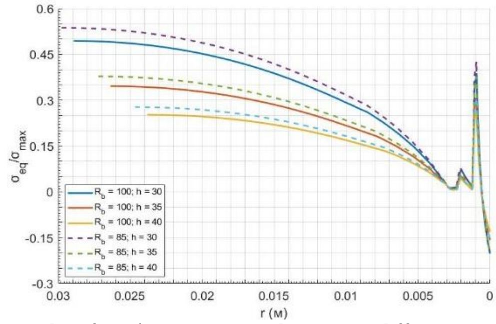

Fig. 9: A plot of σ_{eq}/σ_{max} against radius r(m) at different parameters

Figure 9 shows that the values of σ_{eq}/σ_{max} increase together with increasing thickness at a constant outer radius R_b. At a constant thickness h, with increasing radius R_b, the value σ_{eq}/σ_{max} decreases.

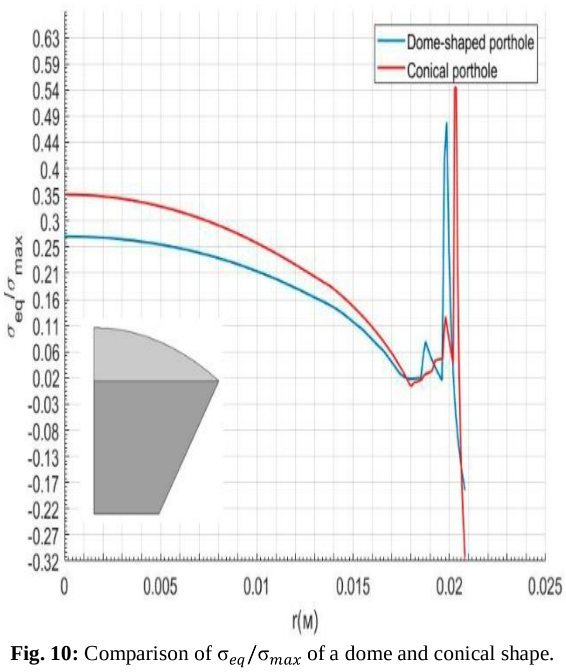

Fig. 10: Comparison of σ_{eq}/σ_{max} of a dome and conical shape.

Figure 10 shows the difference in the numerical values σ_{eq}/σ_{max} on the lower plane of the "dome" and the conical porthole with the same value of the radius of the light hole. We can see that the dome-shaped porthole has the boundary value σ_{eq}/σ_{max} 12% lower than that of the conical one. Using this fact with direct design will, at all other equal conditions, increase the immersion depth of the lighting system.

8. Conclusion

In the course of this research, existing models of the disc-shaped, frustoconical and dome-shaped portholes were considered. The Comsol Multiphysics finite element package allowed us to obtain numerical results well verified with analytical and experimental data. On the basis of the data obtained, the following recommendations can be formulated for the design of a deep-sea porthole of the lighting and surveillance system: "dome" shape, at D = 80 mm and h = 35 mm, Rb = 100 mm, $\phi = 75°$. A dome-shaped porthole with these geometrical parameters is clearly superior to the disc-shaped porthole in strength properties and has a lower weight. To roughly estimate the thickness of the dome-shaped element the formula (5) can be applied.

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