OPTIMIZATION OF PRESSURE HULLS OF COMPOSITE MATERIALS

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Keywords: Pressure hulls, composite material, buckling, optimization.

Abstract

The conventional submarines are made up of high strength steel, allowing them to go to great depths in spite of their large dead weight. In the present work, the pressure hull of a submarine is considered both in steel and in composite material. Materials with high strength to weight ratio include carbon-fiber composites. Carbon-fiber reinforced polymer (CFRP) is a material containing carbon-fiber on various orientations. A parametric study is conducted to find the optimum thickness of ply by employing a Finite Element Analysis Software package, ANSYS. Linear buckling analysis is used to predict the feasibility of CFRP submarine pressure hull at deep waters. From the studies conducted regarding the weight reduction, it is estimated that replacing steel by CFRP results saves up to 60% in the structural weight.

1. Introduction

The development of submarines in the future will keep on requiring designs to allow them to reach increasingly greater depths. Pressure hulls are one of the keys in the design of these vehicles, and this paper is focused on this matter, figure 1. The problem is similar to other fields, like a vessel subjected to external pressure, which has to resist that load with the best possible design.

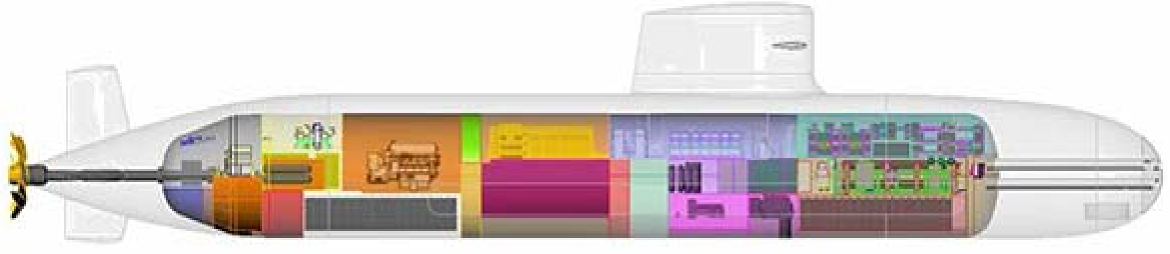

Figure 1. Example of pressure vessel in a submarine.

The problem with vessels under external pressure is their failure due to shell instability, in other words, the buckling. Maximum external pressure to be considered during designing in vacuum operating conditions in industrial equipment is 1 Kg/cm^2, but in submarines design the external pressure reaches very high values, in the order of 100 Kg/cm^2, depending on the immersion depth limit. Also, the size of these pressure vessels have attracted the interest of many researchers since the initial developments of these vehicles during S. XIX, the forgotten I. Peral being among them [1].

The problem with buckling of cylindrical shells under external pressure has been studied by many authors from Bresse [2] and Timoshenko [3], up to specialists in this field as Ross [4]. The usage of composite materials to improve the buckling of pressure vessels has been largely reviewed by Ross [5], Mathai [6], and others, who have been doing a great investigation effort for understanding both simulations and laboratory tests results.

The interest of this matter has a renewed importance due to the issues detected in the S-80 submarine, currently being constructed at the Navantia shipyard and precisely named “Isaac Peral”. The weight of this pressure shell has exceeded the expected, compromising the buoyancy of the submarine. The report from the consultants has determined the only solution for this issue is to enlarge the length of the submarine, figure 2.

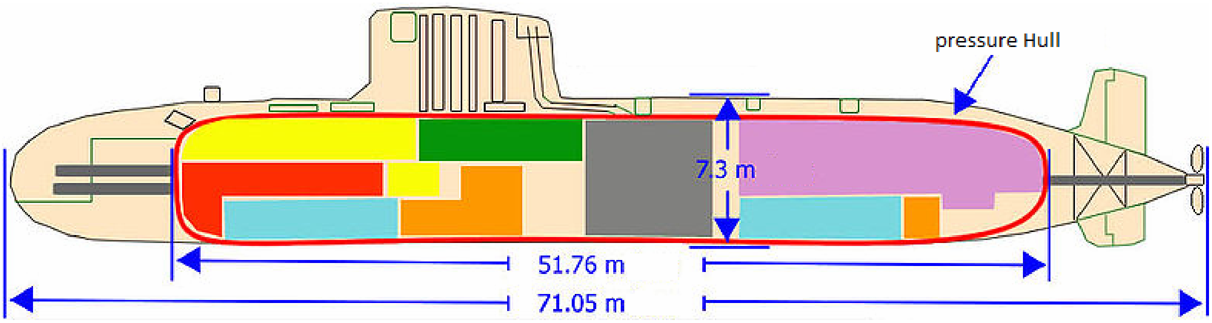

Figure 2. Submarine S.80 (Navantia). High strength Steel

In this study, the usage of composites in addition to FEM optimization techniques are the tools we have proposed in order to assess if the mistakes that may happen in these elements may have a better solution with composites instead of steel, avoiding costly modifications as in the case exposed.

2. Structural design of the pressure vessel

The pressure hull is a pressure vessel that operates in deep waters. This hull type is designed to withstand safely the hydrostatic pressure at the operational deep. The design of these pressure vessels is made with reduced thickness walls and ring stiffeners joined to the walls, with an arrangement as shown at figure 3.

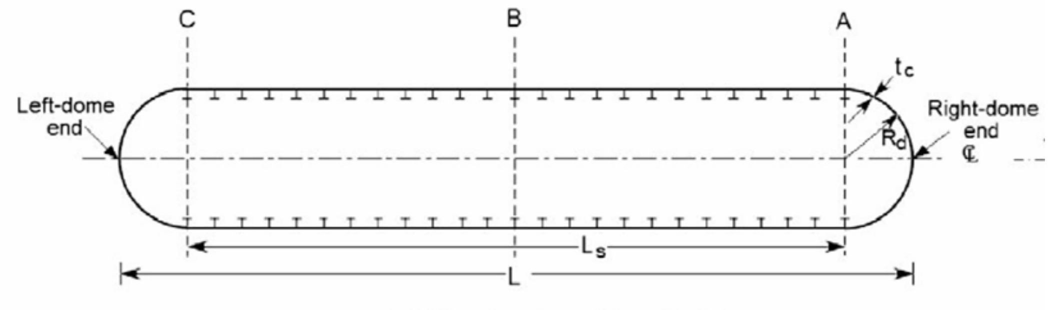

Figure 3. Arrangement of a Steel Submarine Hull

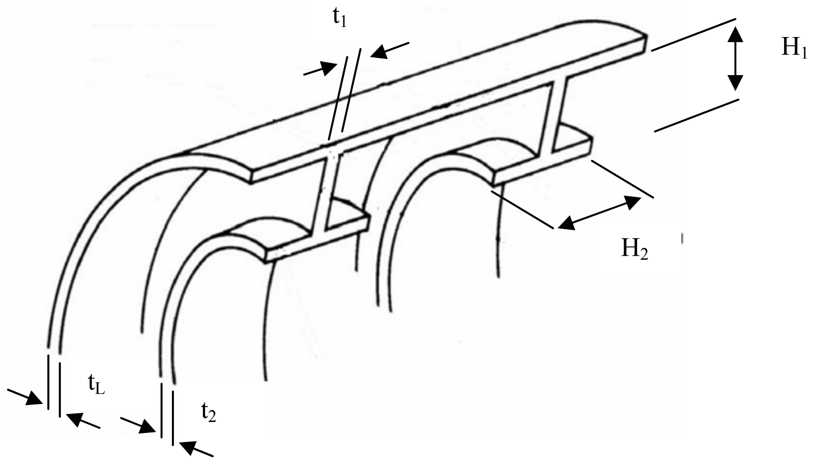

Figure 4. Configuration of Steel Submarine Hull

2.1. Steel model

A Finite Element Model has been developed with the software ANSYS [7], according to the following initial parameters, table 1.

Ls	N_r	R	t_l	t_1	t_2	H_1	H_2	P_d	Weight	Volume
10	20	3	0.1	0.08	0.1	0.3	0.3	25E+5	293,202	283

Table 1. Configuration of Steel Submarine Hull model

In this table, N_r is the number of stiffeners rings, and P_d is the design pressure, in N/m^2, considering a safety coefficient, $\eta=2$, over the linear buckling pressure. The remaining parameters are shown at figures 3 and 4, in meters. The elastic module of the steel is E = 2.1E11 N/m^2 and a Poisson's ratio $\nu = 0.3$. A Finite Element Model has been developed with these parameters using the element Shell181 [7]. A view of the model is shown in figure 5.

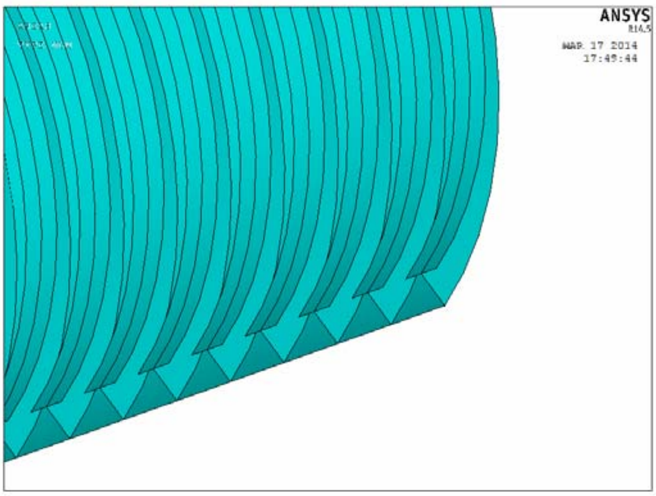

Figure 5. FEM Steel Submarine Hull

2.2. Optimization of the steel model

The optimization problem is presented as a classic gradient method [8], according to the formula (1)

$$\begin{aligned} &Minimize: \; Vol = V(h_i, t_i, t_l) \\ &subject\ to: \quad p_{mx} \le p_{cr} \\ &\qquad \sigma_{mx} \le \sigma_{cr} \\ &\qquad h_{il} \le h_i \le h_{ih} \qquad i = 1,2 \\ &\qquad t_{il} \le t_i \le t_{ih} \qquad i = 1,2 \\ &\qquad t_{ll} \le t_l \le t_{lh} \end{aligned} \tag{1}$$

Where we have applied the restrictions of buckling and the Von Mises stress. The design variables between its upper and lower limits have also been limited.

The evolution during the material volume optimization in this section of the shell is shown at figure 6, and multiplying it by the density of steel, 7850 Kg/m^3, the optimized total weight is obtained.

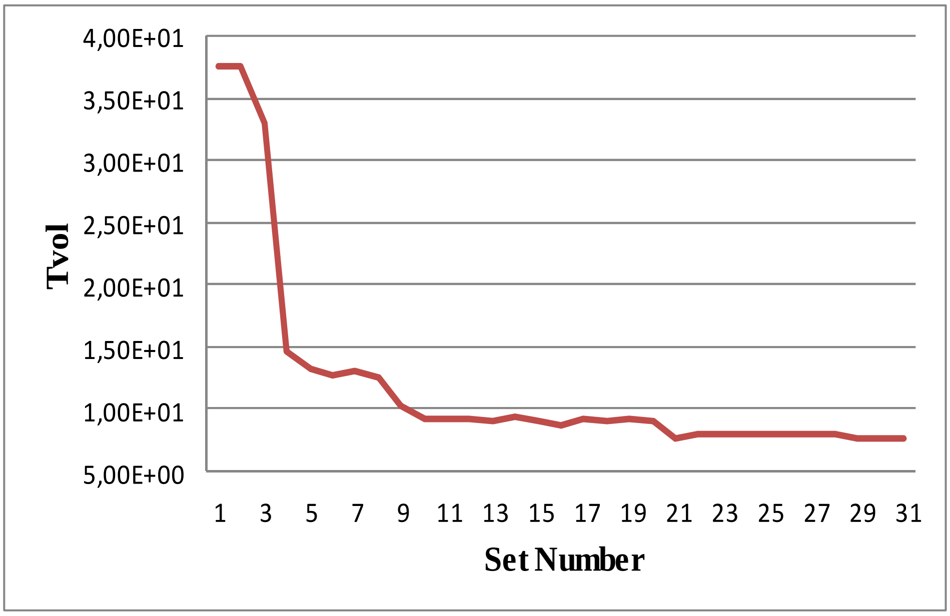

Figure 6. Total volume optimized for Steel material

The optimization final result for 250 m of operational deep is shown at table 2:

t_L	t_1	t_2	H_1	H_2	Weight
0.3E-1	0.25E-1	0.1E-1	0.16	0.16	60,445

Table 2. Configuration of Steel Submarine Hull model

The first buckling form obtained is composed of three lobes, as shown at figure 7.

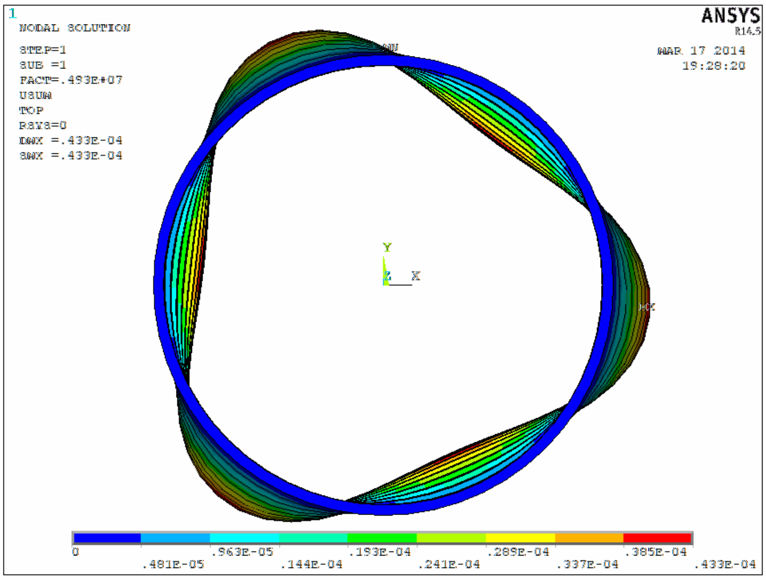

Figure 7. Buckling form

The following table is obtained by repeating the calculations with increasingly larger operating depths of 500, 1000 and 2000 meters of water column:

Depth	250	500	750	1000	1250	1500
Weight	60,445	105,033	145,225	188,400	238,326	289,665

Table 3. Depth and weight for Steel Submarine Hull model

2.3. Composite model

Now the composite model is reviewed, with a similar geometry to the steel model, and a laminate stacking for the pressure vessel shell [90º/0º/-45º/+45º/+45º/-45º/0º/90º] as shown at figure 8.

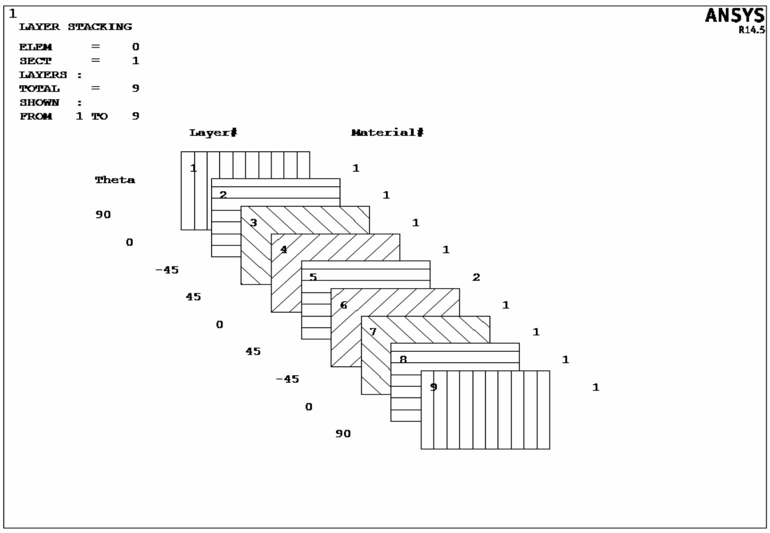

Figure 8. Laminated pressure hull designed for operational depth of 250 m

The element Shell181 has been used, and the initial parameters for this case are

t_1	t_2	t_3	t_4	H_1	H_2	Weight
0.01	0.01	0.01	0.01	0.3	0.3	47,700

Table 4. Configuration of Composite Submarine Hull model

The elastic constants of CFRP, that is the material used for the layers of the laminate, are in table 5:

Ex	Ey	Ez	Gxy	Gxz	Gyz	□xy	□xz	□yz	dens
120.7E9	8.5E9	8.5E9	3.4E9	2.7E9	2.7E9	0.253	0.421	0.421	1,500

Table 5. Material properties of CFRP used

The stress parameters for obtaining the Tsai-Wu criterion are in table 6:

Xten	Yten	Zten	XY	XZ	YZ
120.7E9	8.5E9	8.5E9	3.4E9	2.7E9	2.7E9

Table 6. Failure properties of CFRP used

2.4. Optimization of the composite model

The optimization problem is presented as a classic gradient method [8], according to the formula (2), where we have applied the restrictions of buckling, and the Tsai-Wu criterion. The design variables between its upper and lower limits have also been limited.

$$
\begin{aligned}
&Minimize: \; Vol = V(h_i, t_i) \\
&subject\ to: \; p_{mx} \leq p_{cr} \\
&\qquad ITW \leq 1 \\
&\qquad h_{il} \leq h_i \leq h_{ih} \qquad i = 1,2 \\
&\qquad t_{il} \leq t_i \leq t_{ih} \qquad i = 1,4
\end{aligned}
\tag{2}
$$

Where t_i are the thicknesses of the layers, h_i are the dimensions of the ring stiffeners, as in the case of the steel model, and ITW is the Tsai-Wu index.

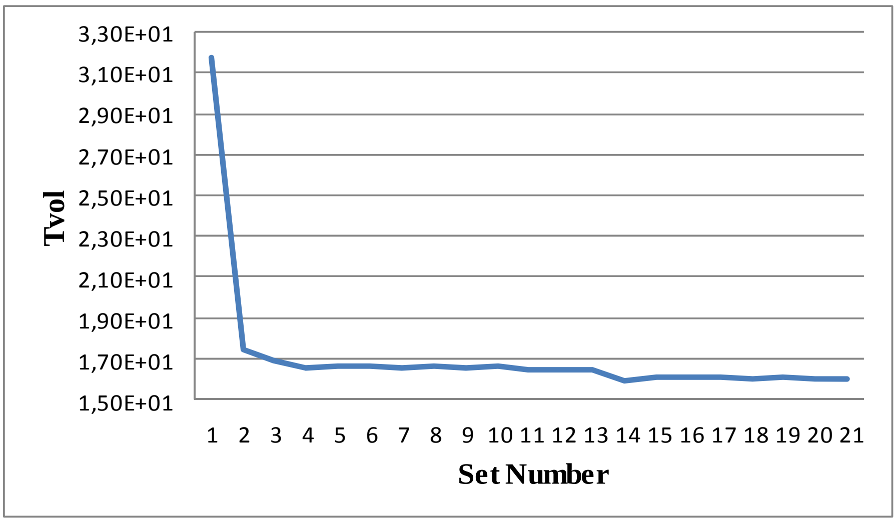

Figure 9. Total volume optimized for composite material

The evolution during the material volume optimization in this case is shown at figure 9.

t_1	t_2	t_3	t_4	H_1	H_2	Weight
0.50E-2	0.58E-2	0.51E-2	0.51E-2	0.29	0.26	24,800

Table 7. Configuration of Composite Submarine Hull model

The following table is obtained by repeating the calculations with increasingly larger operating depths of 500, 1000 and 2000 meters of water column:

Depth	250	500	750	1000	1250	1500
Weight	24,800	34,100	49,600	68,200	85,250	105,400

Table 8. Depth and weight for Composite Submarine Hull model

3. Results and conclusions.

The displacement is the same in both cases and is only a function of the length L and diameter D. Thus, considering a density of 1000 kg/m3 water, we obtain:

$$\begin{aligned} Displ. &= \frac{\pi D^2}{4} L\rho = 282.744\ \mathrm{Kg} \\ Displ. &= 282.744\ Kg/10m \end{aligned} \tag{3}$$

The quotient between the weight and the displacement, results in a ratio indicating the percentage dedicated to buoyancy the hull, and to the payload allowance. If we compare the effect of using steel and the effect of using composite material, two design curves are obtained, as shown at figure 10

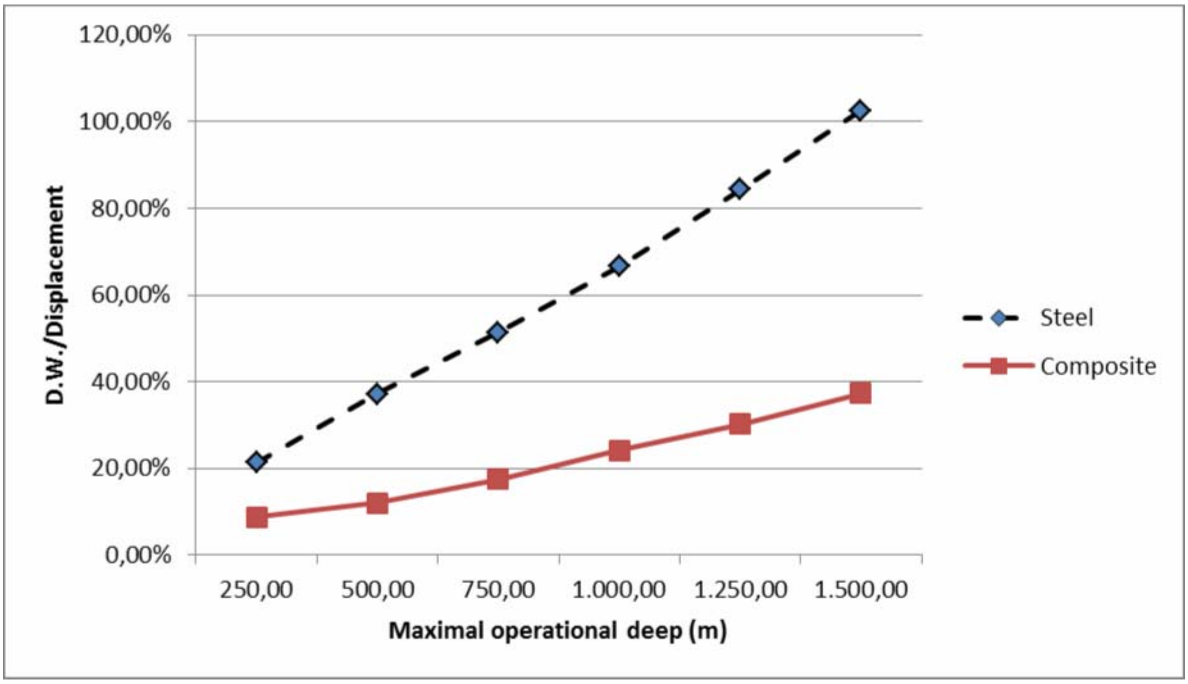

Figure 10. (DW/Desp) – Depth Ratio

If we consider the particular case of the S-80, for an operating depth of 500 m, the difference of composite pressure hull versus of steel is:

$$\Delta W = 5*(105{,}033 - 34{,}100) = 5*70{,}933 = 354{,}665 \;\; Kg \approx 355 \;\; Tn \qquad (4)$$

The submarines are designed to operate near this depth, 400 m to 600 m. As you can see in figure 11, the difference in payload in this depth is more than 60%.

As a result of these important differences between the pressure hull made in steel, and pressure hull in composite, we may conclude:

-The use of Composite allows a great increase of the payload.

-The depth that can be achieved with the pressure hull made of composite is higher than the one achieved with the steel hull.

-An extra margin for buoyancy reserve is allowed in the presence of design errors

-The resistance to corrosion of pressure hull made of composite is larger than for steel in Marine environment.

-The fabrication process of these elements has a precision much larger than the similar task for the steel, in which the distortions at the welding process are very difficult to control.

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