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Numerical Study and Finite Element Analysis of Submerged Cylindrical Pressure Hull

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ABSTRACT

Weight and volume are a direct impact on diving which is an important indicator of the overall performance of the deep-sea submersible structure. In order to increase the payload, improve endurance, reduce energy consumption, improve work efficiency, and therefore must accordingly reduce the total weight of the submersible. The design of underwater vehicles with minimum buoyancy factor is a major requirement from the underwater vehicles design point of view. The composite material pressure hull is a new concept due to the excellent structural performance. Therefore, in the present study, a comparison between three different pressure hulls with the same volume and various materials subjected to the same hydrostatic pressure will be investigated. The first, model was constructed from carbon fibre, while the second one constructed from steel (HY100) and the last one constructed from Titanium alloy. The finite element analysis (FEA) was executed using ANSYS. The results illustrated that the best buoyancy factor and the minimum mass occurred in the case of the pressure hull constructed from carbon fibre.

Keywords: Composites, Pressure hulls, Buoyancy Factor, Hydrostatic Pressure, Finite element, ANSYS

I. INTRODUCTION

The pressure hulls are the main watertight structural component of a submerged submarine that houses people, thrust machinery, sensor systems, weapons, besides the other sensitive equipment [1]. Composites materials have useful properties such as low weight, better corrosion resistance, low magnetic and acoustic signatures [2]. Submarines are weight-sensitive structures and widely used in the exploration of ocean resources [3]. The pressure hulls are the most key structures of the underwater vehicles that provide high load capacity for the buoyancy and the electric systems and [4,5]. Taetragool et al. [6], investigated the optimal angel ply in a laminated plate to maximize the failure load. Additionally, many researchers studied the laminated first lay failure as in [7, 8]. One of the major problems being faced with the submarine designer is to minimize the pressure hull leading weight to raise payload and propulsion for the submarine hull at a designed depth [9]. Fathallah et al.[10,11], explored the composite pressure hulls optimization for maximizing and minimizing the buckling load capacity and the buoyancy factor (B.F), respectively Messager et al. [12], presented the design optimization of submerged vehicles furthermore, presented the optimum lamination of the un-stiffened thin composite hull. Ca et al. [13], examined the composite hull with the external hydrostatic pressure and demonstrated that, the buckling and subsequent burst behaviours will occur. Besides, Lee et al. [9], investigated the optimization of a composite hull including both the buckling and material failure. Smith et al.[14], illustrated an advanced approach to design and model analysis of the submerged vehicles [15]. Weight/displacement (W/Δ) ratio is frequently used for comparing the various construction materials [16,17]. Fathallah et al. [18,19], used ANSYS to optimize the elliptical composite pressure hull to minimize the buoyancy factor. Both Tsai-Wu and maximum stress failure criteria were used as constraints in the design optimization process. Helal et al.[8], presented the sandwich core multi-objective design to maximize both buckling load factor and deck area and minimize the buoyancy factor using the ANSYS software.

II. Shapes of Submarine Pressure Hull

Figure 1 shows the different shapes of a submarine pressure hulls. These structures can be failing either by buckling or through axisymmetric yield as shown in

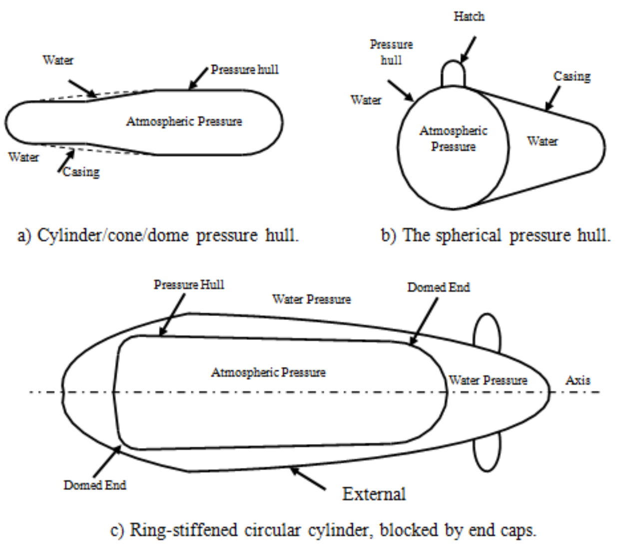


Figure 1: Shape of a submarine pressure hull

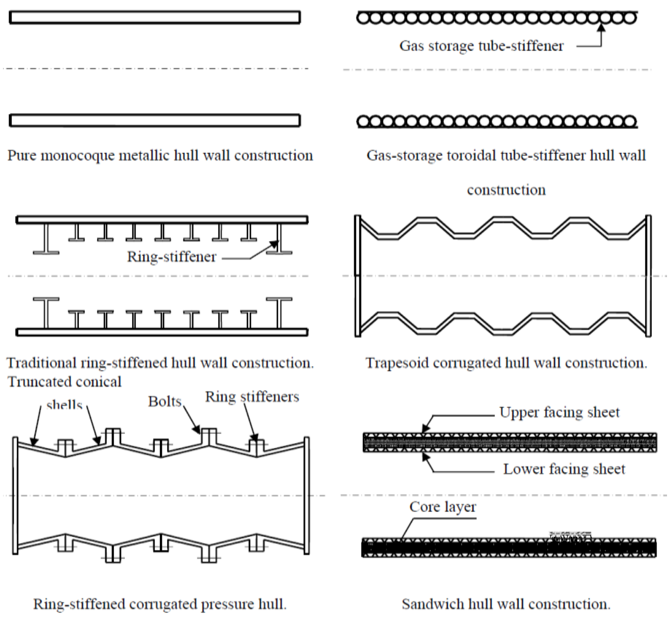


Figure 2: Shape of various wall architectures used for pressure hulls.

Figure 1a. Figure 1b shows the spherical pressure hull which is usually fabricated in the form of a thin-walled spherical shell with a pressure-tight hatch to allow access. In order to improve the hydrodynamic streamlining, the pressure hulls were usually covered with casing for mini-submarines. Figure 1c shows the ring-stiffened circular cylinder which blocked by end caps. It was the usual form of a submarine pressure hull. This can make the cylinder form longer than a spherical form with the same volume. Numerous wall architectures used for pressure hulls were presented in Figure 2 [20,21]. Usually, in the same displacement pressure shell of various shapes, the spherical pressure hull its weight is small, but the space utilization ratio is small. Also, the accuracy is not easy to control in the case of the process of making the roundness and the production cost is relatively expensive [22]. In this research, a submarine pressure hull is in a cylinder form as presented in Figure 3. It has a better space utilization, more efficient than a spherical one for housing large numbers of personnel, good hydrodynamic form with the same volume, facilitates various instruments layout of the electronic equipment and easy to manufacture. The ratio of weight-to-displacement is the best way for measuring pressure hull efficiency [23].

In the present study, a finite element analysis of the submarine pressure hull was presented. A comparison between three different pressure hulls with various materials subjected to the same hydrostatic pressure will be investigated. The finite element analyses were completely performed using ANSYS workbench. The comparative studies were carried out to analyze the buoyancy factors of the cylindrical pressure hull made of metal and composites to find out the decrease in the buoyancy factor of the composite pressure hull with the use of composite materials instead of steel (HY100) and Titanium alloy.

A. Selection of the geometry and Modelling

The structure of the pressure hull has several shapes as in [24]. In our study, the pressure hull is cylindrical amidships and both ends have conical sections closed by end bulkheads as in Figure 3. A comparative study was achieved to analyze the cylindrical pressure hull shell made of steel Titanium alloy and composites. The model consists of: Long stiffener T-section (200×15) mm, Ring stiffener T-section (flange 250×25, web 300×25, diameter 2960) mm, as in Figure 4. Both stiffeners were arranged to give a cylindrical geometry of length 13500 mm covered with a shell of thickness 20 mm calculated with two hemispheres end cap as in Figure 3.

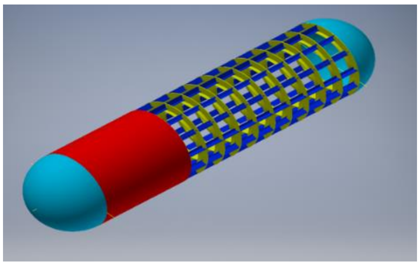


Figure 3: cylindrical pressure hull

The asymmetry boundary condition was applied in all nodes at (Y = 0 (X-Z plane) and X = 0 (Y-Z plane)) furthermore and due to the fluid flow in the longitudinal direction (Z-direction). In the Z-direction, only one single node was constrained as in [8, 25]. No constraints were imposed on the vertical direction as the submarine was balanced by the buoyancy and the gravity. These boundary conditions allowed capturing a heave and pitching motion of submarine in the simulation. The pressure hull was loaded by external hydrostatic pressure as a uniform external pressure load. The mesh density is greatly influenced by the results; therefore the mesh convergence check was conducted to choose the optimum mesh for the model. The loads on a submarine can be calculated as atm. Where: ρ was the density of seawater 1025 kg/m³, g was the acceleration due to gravity 9.81 m/s² and h was the maximum operating depth. All these loads were simulated in ANSYS as applied pressure on the outer surface of the pressure hull, as illustrated in Figure 5.

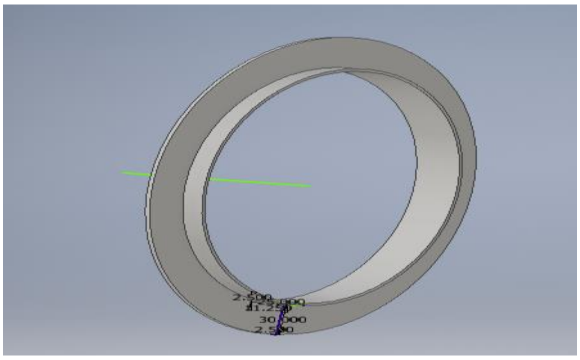


Figure 4: Ring stiffener

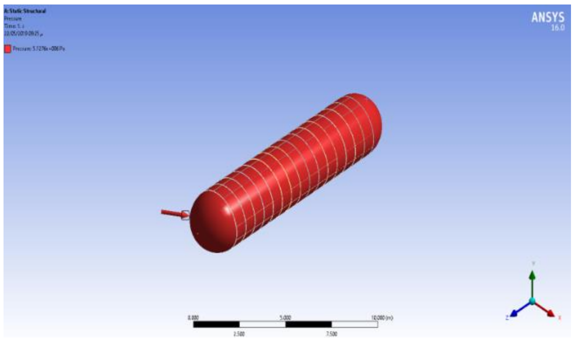
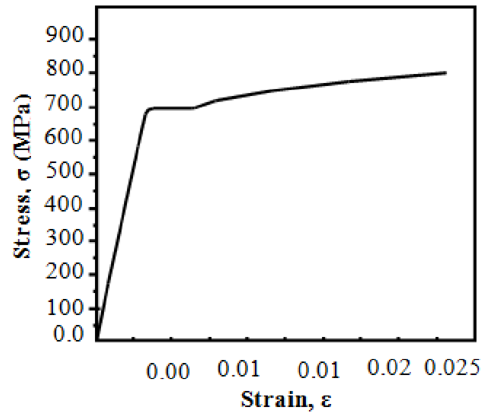


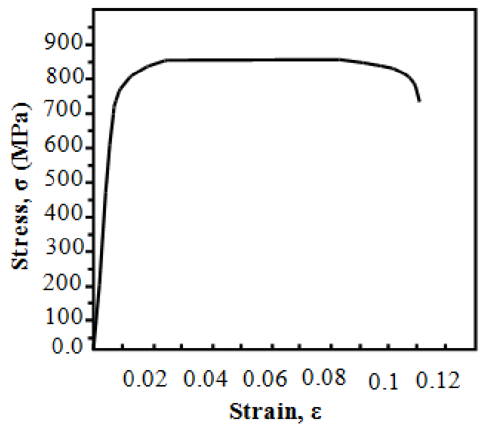
Figure 5: Applied pressure

B. Submarines pressure hulls materials

The materials for the underwater pressure hull must be capable of withstanding very high external pressures and also, have suitable properties that can withstand the environment. Substituting composites for the metallic structures has many advantages due to the higher specific stiffness and strength of the composite materials. Table I and Table IIIII show the properties of the selected materials [22, 25-27]. Figure 6 illustrated the stress-strain curve for both HY100 and Titanium alloy. The comparison between HY100, Titanium alloy, and carbon fibre was made to investigate the deformation and stresses.



a) HY100



b) Ti-4-V-6

Figure 6: The stress-strain curve [27, 28]

TABLE IV

MATERIAL PROPERTIES FOR STEEL AND TITANIUM ALLOY

Material	Titanium Alloy	HY100
Specific Density [ton/m³]	7.8	4.5
Ultimate Strength [MPa]	794	890
Compressive Yield Strength [MPa]	690	827
Young's Modulus [MPa]	210	120
Poisson Ratio	0.3	0.3

In this study, the total deformation, directional deformation, and equivalent stress were investigated and taken into consideration both the mass of each model and the buoyancy factor. All the results were illustrated in table (3) and (4). The buoyancy factor B.F was calculated as [29, 30]:

$$B.F = \frac{\text{Total hull weight}}{\text{The fluid displaced by the body volume.}} \quad (1)$$

TABLE VVI

Properties of Carbon Fiber

Density (kg/m³)	X and Y	1480
Tensile Yield Stress (MPa)	X and Y	829
	Z	50
Compressive Yield Stress (MPa)	X and Y	439
	Z	140
Young's Modulus (MPa)	X and Y	91820
	Z	9000
Poisson Ratio	X Y	0.05
	Y Z	0.3
	X Z	0.3

III.RESULTS AND DISCUSSION

The HY100, Titanium alloy, and composite pressure hull were analyzed. The achieved results were analyzed as follows: Figure 7 shows the distribution of the deformation in the X direction in the case of pressure hull constructed from HY100 the figure illustrates as the maximum value equals to 2.6 mm. Figure 8 shows the distribution of the equivalent stresses in the case of pressure hull constructed from HY100. The figure illustrates as the maximum value equals to 9.6x108 Pa. Figure 9 shows the distribution of the total deformation in the case of pressure hull constructed from HY100 the figure illustrates as the maximum value equals to 3.2mm.

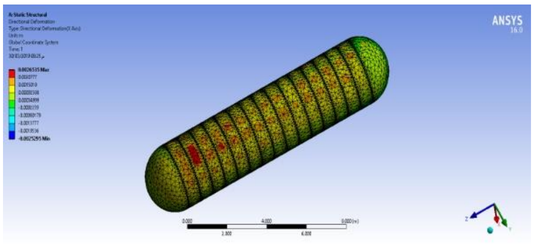


Figure 7: The deformation in X direction in case of pressure hull constructed from HY100

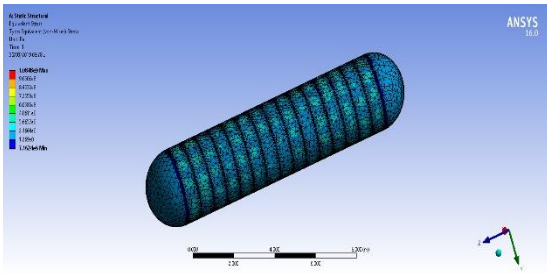


Figure 8: Equivalent stress in case of pressure hull constructed from HY100

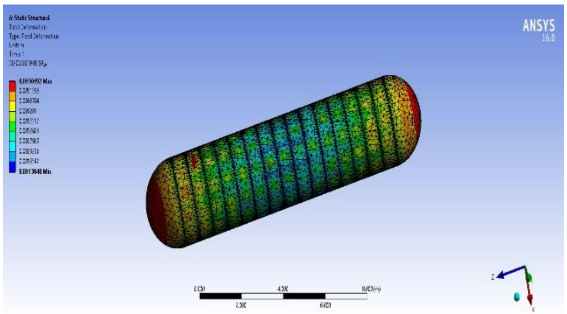


Figure 2 Total deformation in case of pressure hull constructed from HY100

Figure 10 shows the distribution of the deformation in the X direction in the case of a pressure hull constructed from Titanium alloy the figure illustrates as the maximum value equals to 4.6 mm. Figure 11 shows the distribution of the equivalent stresses in the case of pressure hull constructed from Titanium alloy the figure illustrates as the maximum value equals to 8.2x108 Pa. In the composite model, the shell was made from carbon fibre (woven) while the ring stiffener and the long stiffener were made of Titanium alloys. Figure 12 shows the distribution of the total deformation in case of pressure hull constructed from composite. The figure illustrates the maximum value equals to 7.3 mm. Figure 13 shows the distribution of the equivalent stresses in case of pressure hull constructed from composite the figure illustrates that the maximum value equals to 9.9x108 Pa.

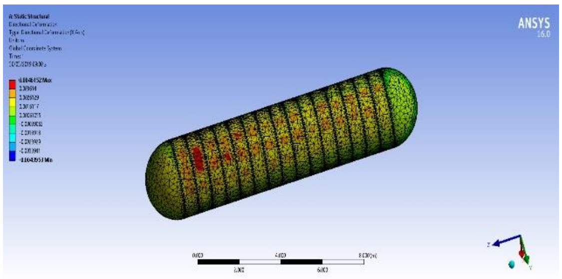


Figure 10:3 Maximum deformation in the X direction in case of pressure hull constructed from Titanium alloy

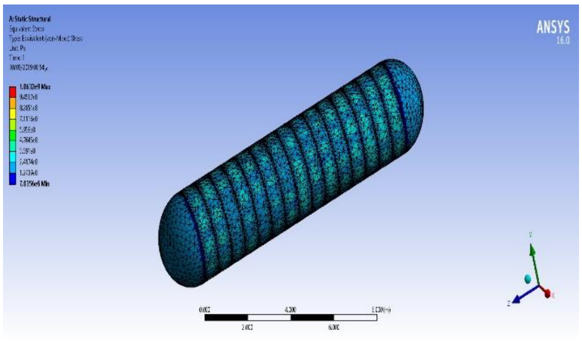


Figure 41 Equivalent stress in case of pressure hull constructed from Titanium alloy

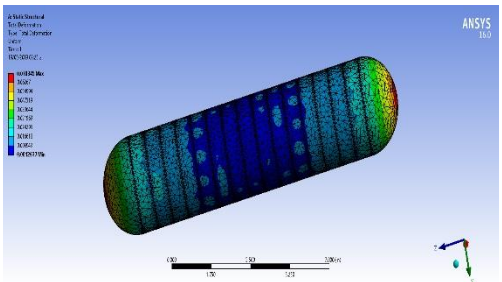


Figure 12: Maximum total deformation in case of pressure hull constructed from composite

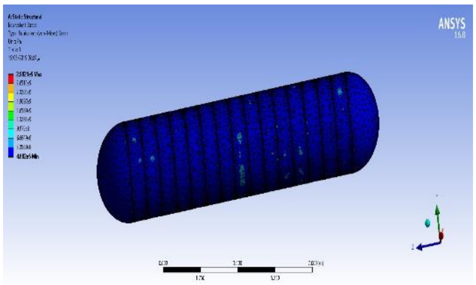


Figure 13: Equivalent stress in case of pressure hull constructed from composite

Table VIIVIIIIX presents the failure index and the deformation for all models. The table illustrates that the best failure index occurs in the case of the composite model and equals to 0.38.

TABLE XXIXII

THE FAILURE INDEX AND THE DEFORMATION FOR ALL MODELS

Material	Failure index	Total deformation (mm)	Directional deformation X-axis (mm)
HY100	0.46	3.2	2.6
Titanium Alloy	0.4	5.6	4.6
Carbon Fiber	0.38	7.3	13.6

Table XIIIXIVXVXVI presents the mass and the buoyancy factor for all models. The table illustrates that the minimum mass and the best buoyancy factor occur in the case of the composite model.

TABLE XVIIXVIIIXIXXX

The mass and the buoyancy factor for all models

Material	Mass (KG)	Buoyancy factor
HY100	38190	0.419
Titanium Alloy	21954	0.242
Carbon Fiber	14444	0.159

IV.CONCLUSION

Finite element modelling and simulation of submarine pressure hull were improved using ANSYS in the study. The results illustrated that the best buoyancy factor and the minimum mass occurred in the case of the pressure hull constructed from carbon fibre. Additionally, the best buoyancy factor occurs in the case of the pressure hull constructed from carbon fibre. The model constructed from HY100 had the least deformation but has a higher failure index than the others. The Titanium alloy came in second place in all the calculated parameters. This research could be used as a helpful tool in the pressure hull designing of underwater vehicles.

V. Conflict of Interest

The authors declare no conflict of interest.

VI.Acknowledgment

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