Conceptual design of a pressure hull for an underwater pole inspection robot

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The conceptual design of a pressure hull design for an underwater pole inspection system is presented in this study. One of the critical elements in designing the pole-climbing robot is an enclosure for the electronic components that can sustain high pressure and with good protection for underwater application. Four main steps are required in designing the system: selection of the shape of the hull, material selection, hull thickness requirement based on depth limit and safety factor, and end closures design compatible with the hull and design requirements. This study suggests that the preferable material for the proposed structural construction of an underwater pressure hull is aluminum alloy 6061-T6. The methodological approaches applied in the design criteria include analytical calculation and a numerical method for results verification based on the circular cylindrical shape. This paper also proposes that these underwater structures should be able to operate up to 200 m depth.

Introduction

During the year 2008, the USM Robotic Research Group proposed a design and development of a pole inspection system for underwater maintenance and surveillance applications. One of the main features of the pole inspection system is the capability of the robot system to undertake visual inspection of the targeted pole. In addition, the robot is also required to have three bodies with two wheels attached to each body segment. Therefore, the visual inspection system is attached to each of the body segment. Figs 1 and 2 show the conceptual design of the pole inspection system. One of the critical parts in designing a pole climbing robot is an enclosure for the electronic components that can sustain high pressure and with good protection for underwater application. Subsequently, the need for a modular pressure hull has been established.

The literature review established that most of the designs and analysis of a pressure vessel are based on the thin-walled vessel principle. Tsybenko *et al.*[1] studied the state of stress and strain of pressure vessels during pressurization. On the other hand, Liang[2] considered the non-linear responses of a submersible pressure hull in the analysis and proposed an optimum design for filament-wound multilayer sandwich submersible pressure hulls. Ross[3,4] proposed conceptual designs of an underwater vehicle and an underwater missile launcher, respectively. In earlier studies, Ross *et al.*[5-8] also reported the effect of buckling due to variable shapes and materials on pressure hull structures. Blachut and Smith[9] discussed the result of a numerical and experimental study with regard to the buckling performance of a multi-segment pressure hull subjected to a uniform hydrostatic pressure. In addition, Graham[10] predicted the collapse of ring-stiffened cylinders due to high external pressure using the finite element method.

The design of an underwater pressure hull has been studied in the past, and one of the prime design constraints is its resistance to buckling. Some relevant information about these studies can be found in Moss[11], Davis[12], Yousefpour[13], and Ross[3]. In this current pressure vessel design, the aim is not only to have an economical and safe design but also to undertake stress and buckling analyses to determine the thickness and size that suit the other parts within the pole inspection system.

There are four prime steps in designing a pressure hull, (i) selection of shape of the hull,

(ii) material selection, (iii) hull thickness requirement based on the depth limit and safety factor, and (iv) end closures design compatible with the hull and design requirement. The methodological framework used by this study is ascribed in Fig 3. There are several shapes that need to be considering in design a pressure hull. There is a number of material types commonly employed for underwater vehicle applications. It should be noted that the pressure hull must also be able to sustain conditions for shallow underwater application under an external hydrostatic pressure of 4 MPa. During the design process, stress and buckling analysis were performed using analytical and numerical methods.
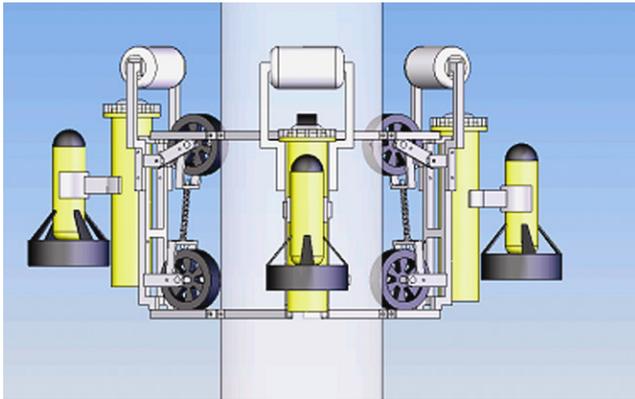

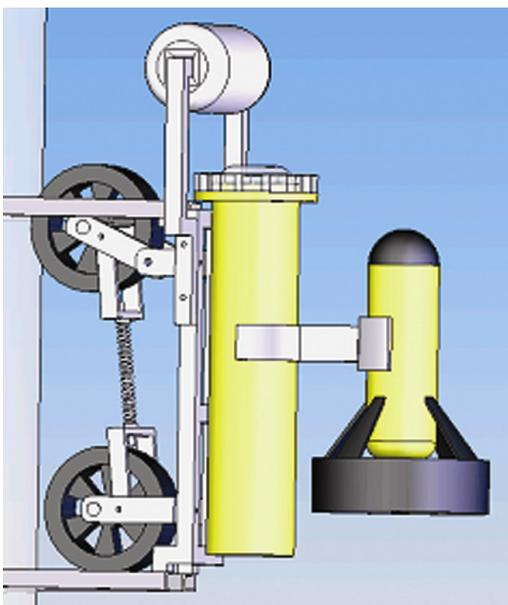Figure 1—An underwater pole inspection system

Figure 2—Close-up of the underwater pole inspection system

Materials and Methods

Shape of the Hull

The typical shapes of a pressure hull structure for underwater application are in the form of spheres, cylinders, cones, ellipsoids, or a composite of all these. All the shapes are considered to be ideal shapes in responding to external pressure. If the wall thickness to diameter ratio is small, they show a nearly uniform distribution of the strains through the thickness. In this paper, a circular cylindrical shape with end caps closure was considered due to its properties and uncomplicated fabrication process.

It was found that a circular cylindrical structure possessed a number of advantages[4].

It was anticipated that for the pressure hull proposed here, the movement of the structure was only in a vertical direction, and for the inspection purposes, it would be operated in a low speed condition, i.e. less than one meter per second. Hence, a very good hydrodynamic form is not a pre-requisite. Thus it can be concluded that a circular cylindrical shape is appropriate for a pressure hull design. After the overall shape was determined, the next methodology was to estimate the enclosure space required to locate all the equipment inside the pressure hull and to fit with the other parts of the pole climbing robot. It was established that the required size was 0.245 m in length with an inner diameter of 0.125 m.

A number of various wall architectures are available for a cylindrical pressure hull such as pure monocoque, ring-stiffened, circumferentially tube-stiffened, and corrugated construction. The ring-stiffened construction is considered better in resisting buckling, but it introduces some difficulties in full-scale manufacturing. Ross[8] presented an alternative design to the traditional ring-stiffened cylinder pressure hull in the form of a corrugated hull wall construction. The study revealed that the corrugated pressure hull is structurally more efficient than the traditional ring-stiffened. There is a research study on a multi-segment of underwater pressure hull[9]. However, in this study, the authors considered the architecture of the wall to have a pure monocoque design for the reasons of ease of manufacturing and cost efficiency. Fig. 4 shows an example of a (a) pure monocoque construction, (b) a ring-stiffened construction, and (c) a corrugated construction.
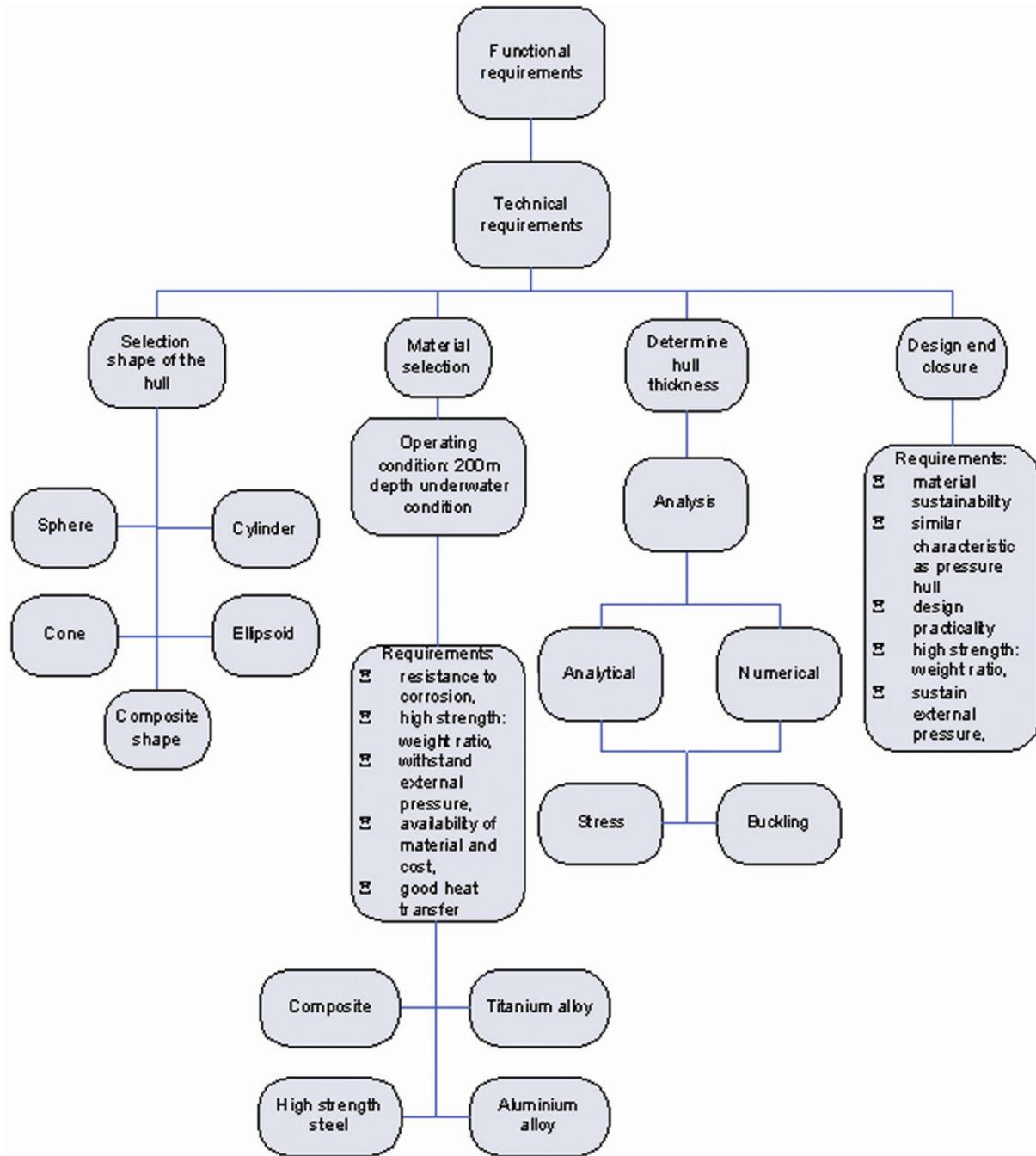
Figure 3—Methodology framework

Material Selection

The structure is designed for conditions down to a depth of 200 m. The successful development of such system would depend on the availability of a suitable material for construction. An advanced material with diverse properties will certainly be required. The material can be composites or complex alloys. In other words, material selection should not only have the capability to withstand a very high external pressure but should also have other suitable properties to withstand hazardous environmental conditions.

Among the requirements for material properties are good resistance to corrosion, better strength-to-weight ratio, low weight capacity, capability to withstand high external pressure, material availability and cost efficiency, fabrication properties (in this case, can the
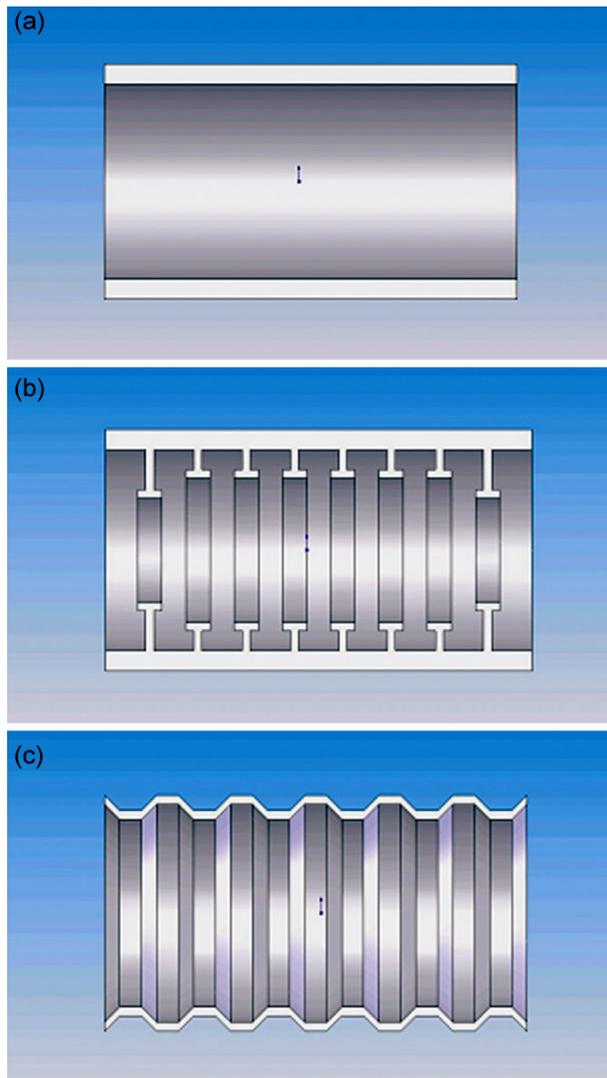
Figure 4—(a) Pure monocoque wall construction; (b) Ring-stiffened wall construction; and (c) Corrugated wall construction

vessel be easily manufactured and follow the pressure hull design?), susceptibility to temperature, and good heat transfer and operating life spans of the material. Material selection based on these properties is important to eliminate or reduce failures in the structure. Such failures include corrosion due to the diffusion of oxygen through the layer of rust, marine organism, and the addition of stress action. This localized failure can occur due to alloy composition, tensile stress (internal and external), and corrosion from the environment, temperature, and time. In addition, other failures that can possibly happen are fatigue failure, brittle failure, and failures induced through fabrication. Based on the characteristic stated above, four materials were considered for possible use

in the design of this underwater vehicle pressure hull: Composite, High strength steels, Aluminum alloys, and Titanium alloys.

The most commonly used composite for marine structures such as ships is glass-fiber reinforced plastic (GRP), but carbon-fiber reinforced plastic (CFRP) is preferred for a structure that is likely to suffer from structure buckling. However, CFRP is expensive to be considered in the current application. Similarly, high strength steel is commonly used for submarine application. Aluminum alloys have a better strength-to-weight ratio compared with high strength steel, better availability among other materials, low cost, and fabricability. However, aluminum can be anodic for most structural alloys and therefore vulnerable to corrosion when utilized in combined structures. Also aluminum alloy is the difficulty in obtaining equivalent strength in weld and base metals. Alternatively, titanium alloys are an ideal material to be used in a pressure vessel design due to its greater strength-to-weight ratio and properties compared with aluminum alloy, but the disadvantage is that the material is very expensive and hard for machining. Table 1 present the general mechanical properties for composite, steels, aluminum and titanium.

Considering all the criteria mentioned above, aluminum 6061-T6 was selected. This is due to its properties which are relatively strong but light, easy to fabricate, available, and with good corrosion resistance. Table 2 presents the mechanical properties of aluminum 6061-T6[14].

Pressure Hull Design

The general shape of the pressure hull design is a circular section of a cylinder with a length of 0.245 m, and an inner diameter of 0.125 m. The circular cylinder shell is made of aluminum alloy 6061-T6 with a Young's modulus of 69 GPa and Poisson's ratio of 0.33. As the structure is intended for deep dive application, it is necessary to construct a wall thickness that can sustain high external pressure to prevent the vessel from structural buckling or collapse. It should be noted that the introduction of the external pressure may cause instability and thus, failures due to the stress that is lower than the elastic limit will occur. Subsequently, a vessel with moderate thickness may collapse under external pressure at a stress just below the yield point.

The wall thickness calculation for a minimum thickness to satisfy the design criteria was accomplished by concluding the analysis using the

Table 1—Mechanical properties for material selection

Material	Tensile Strength (MPa)	Yield Strength (MPa)	Young's Modulus (GPa)	Density (kg/m^3)
Composite	164-1378	-	45.5	1550
Steels	400-620	250-415	200	7860
Aluminum	110-570	95-500	69	2710
Titanium	900	830	115	4730

Table 2—Mechanical properties of aluminum 6061-T6

Property	Value
Modulus (GPa)	69
Y.S (MPa)	275
Shear strength (MPa)	205
Poisson's ratio	0.33

material characteristic. The authors predicted that buckling would be a predominant failure mode, and Roark's equations[15] were utilized to determine the minimum wall thickness based on the design depth of 400 m (4 MPa), although the maximum operating depth of the vehicle was specified to be 200m. For the calculations, material type, wall thickness, inside radius, overall length, Poisson's ratio, and Young's modulus were used as the input parameters for the equations. The first four parameters were generated based on the design specifications, and the latter was from the material properties. The output of the distribution was the pressure at which buckling occurred for several modes, which was successfully predicted using a full scale equation and a simplified model as shown below.

The equations that were employed are shown as follows in equation (1) and (2)[15].

$$q' = \frac{E\frac{t}{r}}{1 + \frac{1}{2}\left(\frac{\pi r}{nl}\right)^2} \left\{ \frac{1}{n^2\left[1 + \left(\frac{nl}{\pi r}\right)^2\right]^2} + \frac{n^2 t^2}{12 r^2 \left(1 - v^2\right)}\left[1 + \left(\frac{\pi r}{nl}\right)^2\right]^2 \right\} \quad \dots (1)$$

where q' is the critical pressure; E is the modulus of elasticity; t is the wall thickness; r is the inside radius; n is the number of lobes formed by the tube in the buckling; l is the hull length; and v is the Poisson's ratio.

If, $60 < \left(\frac{l}{r}\right)^2\left(\frac{r}{t}\right) < 2.5\left(\frac{r}{t}\right)^2$ the critical pressure can be approximated using the following:

$$q' = \frac{0.9E}{\left(\frac{l}{r}\right)\left(\frac{r}{t}\right)^{2.5}} \quad \dots (2)$$

From the calculation, the required thickness is 2.5 mm but for the availability of material, the thickness should be increased. In the final design, the required thickness of the pressure hull was found to be 3 mm; this thickness was extended for the calculations of the axial and hoop (circumferential) stresses, as well as the radial displacement of the circumference. Again, Roark's equations[15] were used and are shown in equations (3), (4), and (5).

$$\sigma_z = \frac{qR}{t} \quad \dots (3)$$

$$\sigma_\theta = \frac{qR}{2t} \quad \dots (4)$$

$$\Delta R = \frac{qR^2}{Et}\left(1 - \frac{v}{2}\right) \quad \dots (5)$$

where q is the unit pressure; R is the radius of curvature of the circumference; σ_z is the axial stress; σ_θ is the hoop stress; and ΔR is the radial displacement of the circumference.

A finite element solution was carried out using CATIA[®16] for the same given data to verify the calculation. The finite element method was employed for several reasons. The main objective was to provide some verification of the results obtained from the theoretical method. Furthermore, it was also intended to investigate the overall buckling behavior of the structure. The pressure hull structure was mainly analyzed for stress and buckling mode of failure. For the analysis, the structure was clamped at both end caps with uniform pressure applied on all three faces of the pressure hull to simulate the effect of hydrostatic pressure.

In this study, parabolic elements were employed, as they presented more accurate results and suited the curved surface better.

CATIA® offers solid elements in the form of a TE10 structural solid of a parabolic tetrahedron, which is a 10-node iso-parametric solid element. The finite element module is geometrically based, in which the boundary conditions cannot be applied to the nodes and elements but only at the part level. In CATIA®, the concept of element size is self-exploratory, wherein the smaller element size provides more accurate results. Another terminology is "sag," which is unique to CATIA®. In the finite element (FE) analysis, the geometry of the part is approximated by the elements. Furthermore, the surface of the part and the approximation of the part should not be coincided. Therefore, the "sag" parameter controls the deviation between the two, with a smaller "sag" value leading to better results[17].

Results and Discussion

During the analytical analysis, the values for σ_x, σ_θ, and ΔR were predicted as 87 MPa, 43 MPa, and 0.067 mm, respectively, for the 3 mm wall thickness of the pressure hull. All values of the calculated stress were well within the allowable values, signifying elastic buckling was indeed the deformation mode as predicted. On the other hand, Figs 5 and 6 show the translational displacement vector and Von Mises stress distribution as integrated by the numerical method, respectively. A typical buckling mode shape of the pressure vessel is shown in Fig 7. The result of the stress values indicated the same trend as the verification model for the metallic pressure vessel. Both axial and hoop stress were compressive, with the results revealing maximum deflection at mid-length. Table 3 present the results of the stress and buckling analysis using numerical method.

The result at the mid-length was considered where the end-cap effects were minimized or discarded. The end-cap effects in the analysis were neglected, as they depended on the type of the end-caps that were used to seal the enclosure. It was found that the result from the displacement due to the static and elastic buckling case shown indicated that the maximum value was lower than that of the allowable stress. Table 4 highlights the results of the analytical solution and corresponding FE analysis results were presented in Table 4.
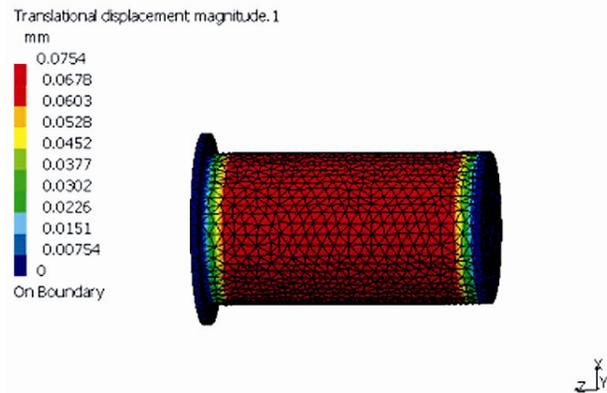

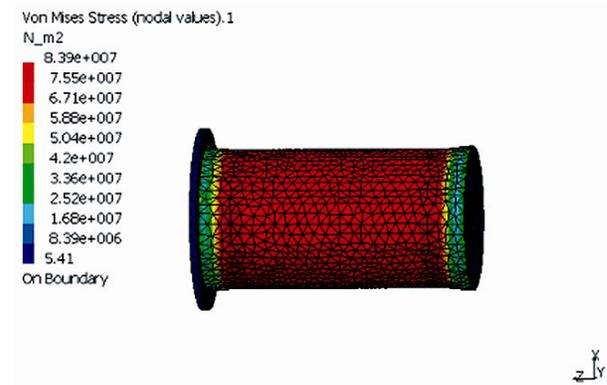Figure 5—Displacement vector on the pressure vessel

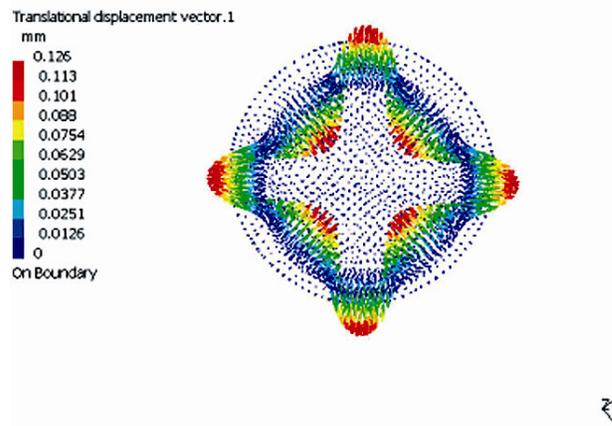Figure 6—Von Mises stress on the pressure vessel

Figure 7—Elastic buckling mode of the pressure vessel

From the simulation, the stress results were analyzed for the modification of the design to reinforce the weak regions of the structures. The results showed the weak regions to be along the mid-length for both static and buckling cases, and as a precautionary action, the thickness of the mid-length section should be increased or reinforced.

Table 3— Results of the stress and buckling analysis of the pressure hull from CATIA software analysis

Wall Thickness (mm)	Displacement (Static case) (mm)	Stress (Nm^{-2})	Pressure (MPa)
3	0.0754	8.39e7	4

Table 4— Comparison of the analytical and numerical solutions for the buckling pressure of an aluminum thin wall circular cylinder shell

	Analytical	Numerical	Deviation
Displacement	0.067mm	0.075mm	10%
Stress	8.7e7Nm^{-2}	8.39e7Nm^{-2}	-3.5%

A cylindrical structure which is subjected to high external pressure, the analysis is simpler. When the wall thickness is thick enough; as the danger from the buckling can be ruled out (instability-induced critical pressure is considerably higher than the allowable one). However, for a thin circular cylinder, the structure can be reinforced by the stiffening ring to increase the value of critical pressure relative to buckling. Another way is to improve the structural efficiency by making a corrugated pressure hull[13].

Once the design of a pressure hull was finalized, the next demanding task was to design the end closures that were compatible with the hull material and satisfied the design requirement. For the end closures, the design for the cap was considered as flat end caps with bore seals. The decision to use flat end caps and bore seals was mainly based on the inexpensive design and simplicity in the manufacturing process. After the design outline had been completed, the selection of material was the second step. The material must be able to sustain a high level of pressure, strong, and durable yet lightweight. To ensure the end closures will have the same characteristic as the pressure hull, the selected material for end closures should be the same with the pressure hull.

Conclusion

The final shape of the pressure vessel was a cylindrical shape with a circular cross-section. A material selection and general design methodology was developed for the conceptual design of underwater pressure vessel. Aluminum alloy 6061-T6 was chosen due to its mechanical and environment performance and ease of fabrication among the four materials. It was also chosen based on the FE analysis. This pressure hull was designed for an external hydrodynamic pressure of 4 MPa with a dimension of 0.245 m long, 0.125 m inner diameter, and 0.003 m thickness.

The stresses and displacements of the pressure vessel were found to be lower than the allowable stress. This indicates that the hull was able to sufficiently support high external and hydrostatic pressure in the operating environment. However, much work is still needed to be carried out such as experiments and testing of the pressure hull prototypes, analysis of the effect at the joints and welding spots, and others before the final estimation of the critical pressure prior to failure can be determined.

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