Finite element modelling and multi-objective optimization of composite submarine pressure hull subjected to hydrostatic pressure

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Abstract. The design of submarine pressure hull made of laminated composites depends on number of layers and fibre orientation of ply angle. In the present study an overview and comprehensive study about the multi-objective optimization of composite pressure hull under hydrostatic pressure to minimize the weight of the pressure hull and increase the buckling load capacity according to the design requirements. Three models were constructed, two models constructed from Carbon/Epoxy composite (USN-150) with and without core layer the third model is metallic submarine hull constructed from HY100. The low-density PVC foam material is used as a core material. The optimization process is carried out in ANSYS Parametric Design Language (APDL). The constraints on the optimization process are Tsai-Wu and maximum stress failure criteria. The results obtained emphasize that, the submarine constructed from Carbon/Epoxy composite (USN-150) is better than the submarine constructed from HY100. Furthermore, the submarine constructed from carbon fiber-epoxy composite (USN-150) with core layer is better than the submarine constructed from carbon fiber-epoxy composite (USN-150) without core layer. Finally, an optimized model with an optimum pattern of fiber orientations was presented. Hopefully, the results may provide a valuable insight for the future of designing composite underwater vehicles.

Keywords: Multi-objective optimization; buckling load; composite pressure hull; failure criteria

1. Introduction
The excellent mechanical properties and behaviour of composite material make composite as a suitable material for underwater pressure hull and helps to reduce the structure’s weight [1]. Design optimization of composite submerged pressure hull and buckling behaviour has been attracted some recent attention [2-13]. Mian et al. [14], presented the design optimization procedure for composite

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pressure vessel, considering both maximum stress and Tsai-Wu failure criteria for the first-ply failure. Zhu et al. [15], studied the buckling behaviours of a spherical shells with an opening. The critical load with an opening was 4.4% - 8% lower than that of the complete spherical shell. Ross and Little [16], predicted the buckling pressure of a carbon fiber vessel subjected to an external pressure. It was higher than the experimental by 20%, because the FE results assume perfect circular geometry. Lopatin and Morozov [17], presented an analytical solution for the buckling problem of a cylindrical composite shell subjected to hydrostatic pressure. The maximum critical pressure was achieved for the helical angles close to 65°. Sekulski [18], presented the multi-objective optimization of high speed catamaran structure incorporating the strength criteria. John et al. [19], investigated the corrosion effect on stability and strength of a pressure hull. The results illustrated that, there is a reductions in the overall collapse and the yield pressures with values 20% and 40%, respectively. Metin and Tolga [20], studied the buckling of the composite plates considering the linearly varying in plane loads and different plate theories. Zhang et al. [21], investigated the egg-shaped pressure hulls to improve the poor hydrodynamics of the spherical pressure hull, the low buckling resistance and the difficult interior arrangement. Han et al. [22], studied the buckling and post-buckling behaviour of composite cylindrical shells under external pressure. Song et al. [23], investigated the optimization of a composite hull to minimize the weight. Ren et al. [24] used the finite element analysis to calculate the buckling behavior of a composite pressure hull. In this study, the multi-objective optimization of a composite intersecting cross elliptical pressure hull shell under external hydrostatic pressure with and without core layer are presented in order to obtain minimum mass (the weight / displacement) ratio and maximum buckling pressure. The modelling and the multi-objective optimization are performed completely using ANSYS Parametric Design Language (APDL).

2. Buckling in composite pressure hull
To ensure the stability, the critical buckling strength \( N_{cr} \) of the composite pressure hull must exceed the actual load \( N_{act} \). The buckling load strength factor \( \lambda \), is introduced to identify the buckling of the pressure hulls and defined as:

\[
\lambda = \frac{N_{cr}}{N_{act}}
\]

Where: \( N_{act} \) and \( N_{cr} \) denote the actual and the critical buckling load, respectively. Buckling will be occurring when \( N_{cr} \) less than \( N_{act} (\lambda < 1) \), which \( N_{cr} \) is defined as [25]:

\[
N_{cr} = \left( R \left( n^2 + 0.5 \left( \frac{m \pi R}{L} \right)^2 \right) \right) \left[ C_{11} C_{12} C_{13} C_{21} C_{22} C_{23} C_{31} C_{32} C_{33} \right]
\]

Where: \( L \) is the length, \( R \) is the radius, \( m \) and \( n \) are the number of buckle half waves in the axial and circumferential direction, respectively. The \( C, A, B \) and \( D \) are expressed as in [26]. The structural stability requirement is assumed to be met when the critical buckling pressure \( N_{cr} \) is greater than design pressure \( N_{d} \) [27].

3. Composite failure criteria
The successful design requires an efficient and safe use of materials. Therefore, theories are needed to develop and compare the state of the stresses and the strains in the material [28].

3.1. Maximum stress failure theory
The failure occurs if any one of the stresses in the principal material coordinates is higher than the respective strength. The failure index is defined as [29]:

\[
I_f = \max \left\{ \frac{\sigma_{11}}{X}, \text{ if } \sigma_{11} > 0 \text{ or } \frac{\sigma_{11}}{X}, \text{ if } \sigma_{11} < 0 \right. \\
\left. \frac{\sigma_{22}}{Y}, \text{ if } \sigma_{22} > 0 \text{ or } \frac{\sigma_{22}}{Y}, \text{ if } \sigma_{22} < 0 \right. \\
\left. \frac{\tau_{12}}{S} \right\}
\]

(3)

Where: \(\sigma_{11}, \sigma_{22}\) and \(\tau_{12}\) denote the applied longitudinal, transversal and shear stress components, respectively and calculated using the following relationship:

\[
\begin{bmatrix}
\sigma_{11} \\
\sigma_{22} \\
\tau_{12}
\end{bmatrix} =
\begin{bmatrix}
Q_{11} & Q_{12} & 0 \\
Q_{12} & Q_{22} & 0 \\
0 & 0 & Q_{66}
\end{bmatrix}
\begin{bmatrix}
\varepsilon_{11} \\
\varepsilon_{22} \\
\gamma_{12}
\end{bmatrix}
\]

(4)

Where: \(Q_{11} = (1 + u_{12} v_{21})^{-1}E_{11}, Q_{12} = (1 - u_{12} v_{21})^{-1}E_{11} v_{21}, Q_{22} = (1 - u_{12} v_{21})^{-1}E_{22}, Q_{66} = G_{12}, X, Y, E_{11}, E_{22}, \) and \(S\) denote the ultimate longitudinal, transversal and shear strength constants, respectively. Since the sandwich core is considered as an isotropic material, the maximum stress criterion can be used for its failure analysis.

3.2. Tsai-Wu Failure Criteria

Tsai-Wu failure criterion is the most generalized criterion, it distinguishes between the compressive and tensile strength and used for the failure prediction in a ply. The criterion can be expressed as [30, 31]:

\[
FI = \sigma_{11} \left( \frac{1}{X} - \frac{1}{X} \right) + \sigma_{22} \left( \frac{1}{Y} - \frac{1}{Y} \right) - \frac{\sigma_{11}^2}{X \times X} - \frac{\sigma_{22}^2}{Y \times Y} - \frac{\tau_{12}^2}{S^2}
\]

(5)

Where: \(X, Y, X, Y, E_{11}, \sigma_{11}, \sigma_{22}\) and \(\tau_{21}\) are aforementioned. The failure will be occurring when the calculated stresses are reaching the ultimate stresses and the \(I_f\) reaches or exceeds the value 1[32].

3.3. Von Mises yielding criteria

The von Mises yielding criteria is employed here to assess the capability of both the metal and the core materials to withstand the yielding failure. The failure index \(I_f\) is defined as follows [33].

\[
I_f = \frac{\sqrt{\sigma_{11}^2 - \sigma_{11}, \sigma_{22}, + \sigma_{22}^2}}{\sigma_0}
\]

(6)

Where: \(\sigma_0\) denote the material allowable yielding strength, \(\sigma_{11}\) and \(\sigma_{22}\) are the in-plane principal stresses.

4. Finite element modelling and simulation

Preferably pressure hull structure should have minimal bouncy factor through achieving the hull density to be close to the sea water as possible [34]. Figure 1 shows the different shapes of a submarine pressure hull. Strong lightweight materials for the pressure hulls are the most urgent requirement in the development of a technology for deep diving vehicles. Strength and lightweight are not the only requirements. Also, toughness, formability, weldability, resistance to corrosion, fracture toughness, low cycle fatigue strength, reliability and cost are importance [1, 35]. The material properties and the strength parameters are given in Table 1 [36-38]. The multiple intersecting cross-elliptical submersible pressure hulls used in this study presented in Figure 2, with total length \((L_{tot})\), major diameter \((D_{major})\), minor diameter \((D_{minor})\), intersecting angel \((\theta)\) and radius \((R)\). The pressure hull modelled using (Shell 93) in case of metallic hull (HY100). For composite pressure hull without
core the structure modelled using (Shell 99) with stacking sequence \([(-\theta/\theta_{4})^s_s]_s\). For composite pressure hull with core layer the structure modelled using element (Shell 281) with stacking sequence \([(-\theta/\theta_{4})^s/[C]/_s]_s\). The ring beams and stringers are modelled using BEAM189 [39, 40]. The boundary conditions applied here as in [41]. The pressure hull is loaded by external hydrostatic pressure \(P = \rho g H\) as a uniform external pressure. Where \(\rho\) is the density of sea water with value 1025 kg/m\(^3\), \(g\) is the gravity acceleration and equals to 9.81 m/s\(^2\) and \(H\) is the depth below the water surface. To predict the failure index, both Tsai-Wu and maximum stress failure criteria are used. Figure 3 indicates the mesh for the overall model.

![Diagram](image1)

Figure 1 Possible forms of submerged vehicles.
5. Optimization concept

The most design problems are often involve constrained minimization. One example of such constrained minimization problem is minimizing the weight of a structure under constraints such as stress and deflection. In general optimization statement, the objective is to minimize a single function which known as objective function [42, 43]. In this study, the first order optimization method is used. It is presented in the optimization tool in ANSYS. The lower and upper bound for the update parameters were applied. Random analysis in ANSYS is used to find the constraints for the updating parameter. Figure 4 shows the multi-objective optimization procedure flow chart. The optimization can be described as the follows:

![Diagram](image)

Figure 2 Multiple intersecting cross elliptical submersible pressure hulls.

![Diagram](image)

Figure 3 Finite element modeling of the cross-elliptical submersible pressure hulls.

| Material                        | Material and strength properties                               |
|---------------------------------|----------------------------------------------------------------|
| Carbon/epoxy composite (USN-150)| $E_{11} = 131 \text{GPa}$, $E_{22} = 10.8 \text{GPa}$, $E_{33} = 10.8 \text{GPa}$, $G_{12} = 5.65 \text{GPa}$, $G_{23} = 5.65 \text{GPa}$, $\nu_{12} = 0.28$, $\nu_{23} = 0.059$, $X_t = 2000 \text{MPa}$, $X_c = 1400 \text{MPa}$, $Y_t = 61 \text{MPa}$, $Y_c = 130 \text{MPa}$, $S = 70 \text{MPa}$, $\rho = 1540 \text{kg/m}^3$ |
| HY100                           | $E = 210 \text{GPa}$, $\nu = 0.3$, $\rho = 7890 \text{ kg/m}^3$ Compressive yield strength ($F_y = 690 \text{MPa}$), Ultimate strength ($F_u = 739.5 \text{MPa}$) |
| H200                            | $E = 250 \text{MPa}$, $G = 73 \text{MPa}$, $\nu = 0.3$, $X_t = 7.1 \text{MPa}$, $X_c = 5.4 \text{MPa}$, $Y_t = 7.1 \text{MPa}$, $Y_c = 5.4 \text{MPa}$, $S = 3.5 \text{MPa}$, $\rho = 200 \text{kg/m}^3$ |
Define the objective constraints and levels for the design parameters of
Initial values for design variables.

Finite Element Analysis
Using ANSYS to calculate

Renew
Design
Variables

Critical Buckling Load
Failure Criterion Coefficients
Stiffens Matrix
Stress Components

Check the conversion of the Multi-objective function

Converged?

No

Optimal results

End

Figure 4 Multi-objective optimization procedure flow chart.
5.1. Optimization Statement and the objective of the optimization

The optimization model of the design including the objective function, design constraints and design variables. The multi-objective optimization problem can be stated mathematically as:

\[
F(X): \text{Minimize } B.F = \frac{\text{Total hull weight}}{\text{The fluid displaced by the body volume}}. 
\]

And

\[
\text{Maximize buckling capacity: } \lambda = \frac{N_{cr}}{N_{act}}. 
\]

5.2. Material strength Constraints:

The factors of safety for both Tsai-Wu (FS\textsubscript{TW}) and maximum stress (FS\textsubscript{MS}) failure criteria are used as material failure constraints. In order to avoid material failure, both (FS\textsubscript{TW}) and (FS\textsubscript{MS}) must be greater than 1. For the core layer, and steel von-Mises stress (\(\sigma_v\)) must be lesser than the yield stress (\(\sigma_y\)) of the core layer. These constraints are represented as:

\[
g_1 : FS_{TW} (i) \geq 1, \ i = 1,2,3,\ldots,n \\
g_2 : FS_{MS} (i) \geq 1, \ i = 1,2,3,\ldots,n
\]

Where: \(FS_{TW}\) and \(FS_{MS}\) are the factors of safety for both Tsai-Wu and maximum stress failure criteria respectively, for each \(i^{th}\) layer and \(n\) is the number of layers.

\[
g_3 : \sigma_y \geq \sigma_v 
\]

Where: \(\sigma_i\) and \(\sigma_y\) are the actual stress and yielding strength in the core layer, respectively.

5.3. Side Constraints:

The radii of the cross-elliptical submersible pressure hull ((\(D_{\text{min}}\) and \(D_{\text{max}}\)), the thickness (\(t\)) of individual ply thicknesses, the thickness (\(t_{\text{core}}\)) of the core layer, the fiber orientation angle (\(\alpha\)) of the individual ply, the operating depth (\(H\)), the intersecting angle (\(\theta\)).

\[
D_L^i \leq D_i \leq D_U^i, \ i = \text{max, min}; \\
t_L^i \leq t_i \leq t_U^i, \ i = 1,2,3,\ldots,n; \\
T_L^i \leq T_{\text{core}} \leq T_U^i; \\
\alpha_L^i \leq \alpha_i \leq \alpha_U^i, \ i = 1,2,3,\ldots,n; \\
H_L \leq H \leq H_U; \\
\theta_L \leq \theta \leq \theta_U
\]

Where: \(D_L^i, D_U^i\) and \(D_U^i\) denote the \(i^{th}\) cross pressure hull diameters and the upper and lower limits, respectively; \(t_L^i, t_U^i\) and \(t_U^i\) denote the \(i^{th}\) ply thickness and the lower and upper bounds, respectively; \(T_L^i, T_U^i\) and \(T_U^i\) denote the core thickness of cross-elliptical deep pressure hull and the upper and lower limits, respectively; \(\alpha_L^i, \alpha_U^i\) and \(\alpha_U^i\) denote the \(i^{th}\) orientation angle of each ply and the upper and lower bounds, respectively; \(H, H_U^i\) and \(H_U^i\) denote the operating depth for the cross-elliptical pressure hull and the upper and lower limits, respectively; \(\theta, \theta_L^i\) and \(\theta_U^i\) denote the \(i^{th}\) intersecting angle and the lower and upper bounds, respectively.

6. Results and discussions
The optimization results for the metallic multiple intersecting cross elliptical pressure hull are summarized in Table 2. These results will be used as a reference for the comparison with the composite hulls with and without core layer. From Table 2, the Multi-objective function is 0.07373 with buoyancy factor \( B.F \) equals to 0.39 and buckling strength factor \( \lambda \) equals to 5.2899. The maximum von Mises stress \( \sigma_{\text{MAX}} \) equals to 765 MPa and the maximum deflection value \( \delta_{\text{MAX}} \) equals to 2.3297mm. The total hull weight equals to 778.589kg. For the same loadings, the optimization procedures were performed on the composite multiple intersecting cross elliptical pressure hull. Using (Carbon/Epoxy composite (USN-150)) with and without core layer have been used. The optimization results are presented in Tables (3 and 4).

| Maximum deflection \( \delta_{\text{MAX}} \) | Maximum Von mises stress \( \sigma_{\text{MAX}} \) |
|------------------------------------------|------------------------------------------|
| 2.3297 mm | 0.76557×10^9 Pa |

Table 2 The results of the optimal design for metallic pressure hull. (Steel HY100).

6.1. Optimization results analysis for Carbon/Epoxy composite (USN-150) with and without core layer

The optimization results obtained in case of (Carbon/Epoxy composite (USN-150)) without core is presented in Table 3. Which indicates that the MOF is 0.0487 with \( \lambda = 33.9 \% \). The B.F equals to 0.1976 with improvement ratio \( I_B = 49.33\% \). The total hull weight equals to 394.574 kg with improvement ratio \( I_B = 49.322 \% \). While, the buckling load capacity \( \lambda \) in this case is (4.05) which is smaller than the metallic one with percent = 23.39058 \%. The optimization results in case of (Carbon/Epoxy composite (USN-150)) with core layer are shown in Table 4 indicating that, the MOF is 0.0769 which is higher than the metallic pressure hull with percent = 4.3 \%. While, the B.F equals to 0.139 with improvement ratio \( I_B = 64.324 \% \). The total hull weight equals to 277.843kg with improvement ratio \( I_B = 64.314 \% \). The buckling load capacity \( \lambda \) in this case is (1.81) which is smaller than the metallic one with percent = 65.79 \%. From the above results, the pressure hull constructed from (Carbon/Epoxy composite (USN-150)) without core has better MOF than the (Carbon/Epoxy composite (USN-150)) with core layer. Moreover, (Carbon/Epoxy composite (USN-150)) with core layer has minimum weight and minimum B.F while, (Carbon/Epoxy composite (USN-150)) without core has maximum buckling load capacity \( \lambda \) and minimum deflection value. The maximum Tsai-Wu and stress failure occur at upper face (upper most layer) in case of (Carbon/Epoxy composite (USN-150)) with core layer due to its strength properties. The maximum Tsai-Wu and stress failure occur at lower face (lower most layer) in case of (Carbon/Epoxy composite (USN-150)).

| Ply-1 | Ply-12 | Ply-13 | Ply-14 | Ply-15 | Ply-16 | Ply-17 | Ply-18 |
|-------|-------|-------|-------|-------|-------|-------|-------|
| \( F_{\text{SW}} \) | 1.3490 | 1.4743 | 1.2178 | 1.3284 | 1.2713 | 1.2079 | 1.1653 |
| \( F_{\text{SAE}} \) | 1.5068 | 1.5631 | 1.4512 | 1.4176 | 1.4895 | 1.4036 | 1.5836 |
| \( D_{\text{major}} \) | 2.0 m | 1.6836 m | | | | | |
| \( D_{\text{minor}} \) | 2.0 \( h_1 \) | 1.6836 m \( b_1 \) | | | | | |
| \( h_2 \) | 56 mm | 50mm | | | | | |
| \( a \) | 49° | 40° | | | | | |

Table 3 The results of the optimal design for pressure hull without core. (Carbon/ePoxy composite (USN)).
In order to explore the relationship between Tsai-Wu failure and the fibre orientation, models with different fibre orientation were developed. In order to explore the relationship between buckling strength factor and fibre orientation, models with different fibre orientation were also, developed as illustrates in Figure 5. The figure illustrates that the maximum value of buckling strength factor equals (2.1) and occurred when $\alpha$ equals to 43°.

Table 4 The results of the optimal design for sandwich pressure hull. (Carbon/epoxy composite (USN)).

| Ply  | $FS_{TW}$ | $FS_{MS}$ | $FS_{TW}$ | $FS_{MS}$ | Buckling strength factor ($\lambda$) | $Layer \ thickness \ (t)$ | Total weight | $D_{major}$ | $D_{minor}$ | $t_{core}$ | $\theta$ | $\alpha$ | $H$ | $\delta_{MAX}$ | $\delta_{MIN}$ | $\text{VONMISES}$ | $b_1$ | $b_2$ | $b_3$ |
|------|------------|------------|------------|------------|-----------------------------------|--------------------------|--------------|-------------|-------------|----------|--------|--------|-----|---------------|--------------|----------------|-------|-------|-------|
| Ply-1 | 1.5301     | 1.4659     | Ply-12     | 1.7084     | 1.6620   | Layer thickness (t)  | 0.69229(mm) | 277.84kg     | 2.0 m       | 1.6836m    | 35mm     | 40°    | 48°    | 500 m | 6.003 (mm) | 0.15895×10^{7} Pa | 0.139153986 | 56 mm | 75.4 mm | 50 mm |
| Ply-2 | 1.5212     | 1.4843     | Ply-13     | 1.7204     | 1.6506   | MOF                      | 7.6904×10^{2}    |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-3 | 1.5328     | 1.4689     | Ply-14     | 1.7131     | 1.6657   | Total weight             | 277.84kg        |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-4 | 1.5239     | 1.4873     | Ply-15     | 1.7249     | 1.6544   | $\theta$                 | 40°           |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-5 | 1.5354     | 1.4719     | Ply-16     | 1.7175     | 1.6694   | $\alpha$                | 48°           |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-6 | 1.5263     | 1.4902     | Ply-17     | 1.7310     | 1.6600   | $t_{core}$               | 35mm          |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-7 | 1.5378     | 1.4749     | D$_{major}$| 2.0 m      |          | Ply-9._SMAX              | 99856 Pa       |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-8 | 1.5286     | 1.4932     | D$_{minor}$| 1.6836m    |          | Ply-9._SMIN              | -0.18650×10^{7} Pa |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-9 | 1.7035     | 1.6583     | Operating depth ($H$) | 500 m     |          | Ply-9._SYMAX             | -74346 Pa      |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| Ply-10| 1.7156     | 1.6468     | Maximum deflection ($\delta_{MAX}$) | 6.003 (mm) |          | Ply-9._SYMIN             | 0.15895×10^{7} Pa |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| B.F  | 0.139153986 | R          | 0.50 m     |           |          | Ply-9._VONMISES          | 0.68149×10^{7} Pa |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| $h_1$ | 75.4 mm    | $h_2$      | 50 mm      | $h_3$     | 50 mm    | $b_1$          | 56 mm          |              |             |             |         |        |        |       |               |                      |                |       |       |       |
| $b_1$ | 56 mm      | $b_2$      | 5mm        | $b_3$     | 5mm      |             |               |              |             |             |         |        |        |       |               |                      |                |       |       |       |
6.2. Conclusions
In this paper the design optimization of composite pressure hull is carried out. The methodology for the multi-objective optimization is presented and performed using APDL. The Comparison is carried out between the optimized designs of the composite pressure hull based on the improvement ratio ($I_R$) in the total hull weight and the B.F over an equivalent steel hull. From the results it is observed that, the pressure hull constructed from (Carbon/Epoxy composite (USN-150)) without core layer has minimum MOF than the pressure hull constructed from the same materials with core layer. (Carbon/Epoxy composite (USN-150)) with core layer has minimum weight, minimum B.F. and minimum buckling load (Carbon/Epoxy composite (USN-150)) without core layer. A higher strength material provides a better buoyancy to increase the permitted payload as in (Carbon/Epoxy composite (USN-150)) with core layer has minimum weight and minimum B.F. The fiber orientation is the most influential design parameter. Moreover, it has a great effect upon the Tsai-Wu failure and the buckling strength factor. When the objective is to obtain minimum weight, carbon fiber-epoxy composite (USN-150) with core layer is preferred. All of these results from this study will be helpful in the future of development and designing composite underwater vehicles.

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