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# STRUCTURAL ANALYSIS OF A SUBMERGED COMPOSITE HULL, INTENDED FOR OIL RECOVERY FROM SHIPWRECKS

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## ABSTRACT

In the present work, the survivability of a submerged structure against hydrostatic loads, imposed by the operational environment, such as hydrostatic pressure due to depth, is presented. The structure under consideration is a component (Buffer Bell) of a larger system called DIFIS that recovers fuel from shipwrecks in order to prevent oil spills. This system was developed within the frame of an FP 6 EU project under the acronym DIFIS and relies on gravity forces to channel the spilled fuel flux towards a Buffer Bell-reservoir, 30m below the sea surface, by means of a light, easy to deploy and flexible structure that would stay in place until the pollution threat is eliminated. The Buffer Bell is a main element of the DIFIS system as it is the component that receives and stores the recovered oil and allows its transfer to surface vessels. It consisted of two parts; the Reservoir and the Floater.

The Reservoir is a large tank used for the temporary storage of the recovered oil during the operational phase. The Floater is the part that produces the buoyancy force that holds the whole system in place and therefore its structural integrity is crucial for the operation of the system and the safety of the structure.

During the final concept, the Floater is designed as a double hull structure made of steel plates and stiffeners. In the present analysis, an optimized structure made of GFRP and foam is proposed in order to reduce the steel's hull weight. The structural analysis of the composite hull is carried out for the worst case scenario.

**KEYWORDS**: Finite Element; Composite hull; Sub Sea; GFRP; DIFIS Project; Foam.

## INTRODUCTION

The analysis of underwater structures using the finite element approach is a methodology applied in ship design, offshore and submarine engineering during the last decades. The structural analysis of ship, naval and submarine structures is investigated by researchers [1],[2],[3],[4],[5],[6].The hydrostatic pressure is the main load that the hull of a submerged structure must withstand. In addition, the sea environment is corrosive (salinity) and in the case of the present study, the underwater tank contains oil (leakage from the shipwreck), which is corrosive. All these factors could affect the structural integrity of the hull.

The main challenge of the present work was to develop a submerged structure, that its component have to meet many contradicting requirements such as predefined general dimensions, limited maximum weight in water, produce adequate buoyancy forces to hold the whole DIFIS system in place and tensioned, corrosive operational environment due to salt water and oil, need of high safety level during operations together with the low risk of any possible failure. [7]

Thus the methodology followed for the analysis includes preliminary simplified analytical calculations for the foam structure, Finite Element (FE) analysis of GFRP plates and finally the FE analysis of the complete composite hull.

During the preliminary analysis phase, materials and material behavior models were selected, and the main dimensions were calculated based on stiffness and strength requirements. Then, FE analysis of the system components was performed and the basic structural characteristics were concluded. These results were input to the full scale FE model in order to validate the stress distribution over the parts of the structure and the optimization with respect to the thickness of the GFRP plates. During the worst case scenario, the hull structure has to withstand the extreme hydrostatic loads keeping the appropriate safety margin against failure.

## DESCRIPTION OF THE DIFIS SYSTEM

The DIFIS system consists of seven parts and is anchored on the sea bed for oil recovery from shipwrecks [7],[8],[9]. A schematic representation of the system is given in Figure 1. These parts are: 1) **The buffer bell (BB):** An underwater tank for the temporary storing of recovered oil. It also keeps the system fully pre-tensioned, producing buoyancy forces, 2) **The dome (DM):** A conical shaped structure made of a fabric material which covers the shipwreck and drives the collected oil through the riser tube 3) **The riser tube (RT):** An almost vertical pipe made of HDPE for the connection of the buffer bell to the dome that covers the shipwreck, 4) **The dome interface unit (DIU):** A conical steel substructure serving as a connection interphase between the riser tube and the dome structure, 5) **The mooring lines (ML):** Vectran cables which support the riser tube and the dome, and are anchored to the seabed 6) **The stiffening rings (SR):** Aluminum Disks that connect each part of the riser tube with the mooring lines, 7) **The anchoring system (AS):** Deadweight cement anchors, holding the overall structure to the seabed.

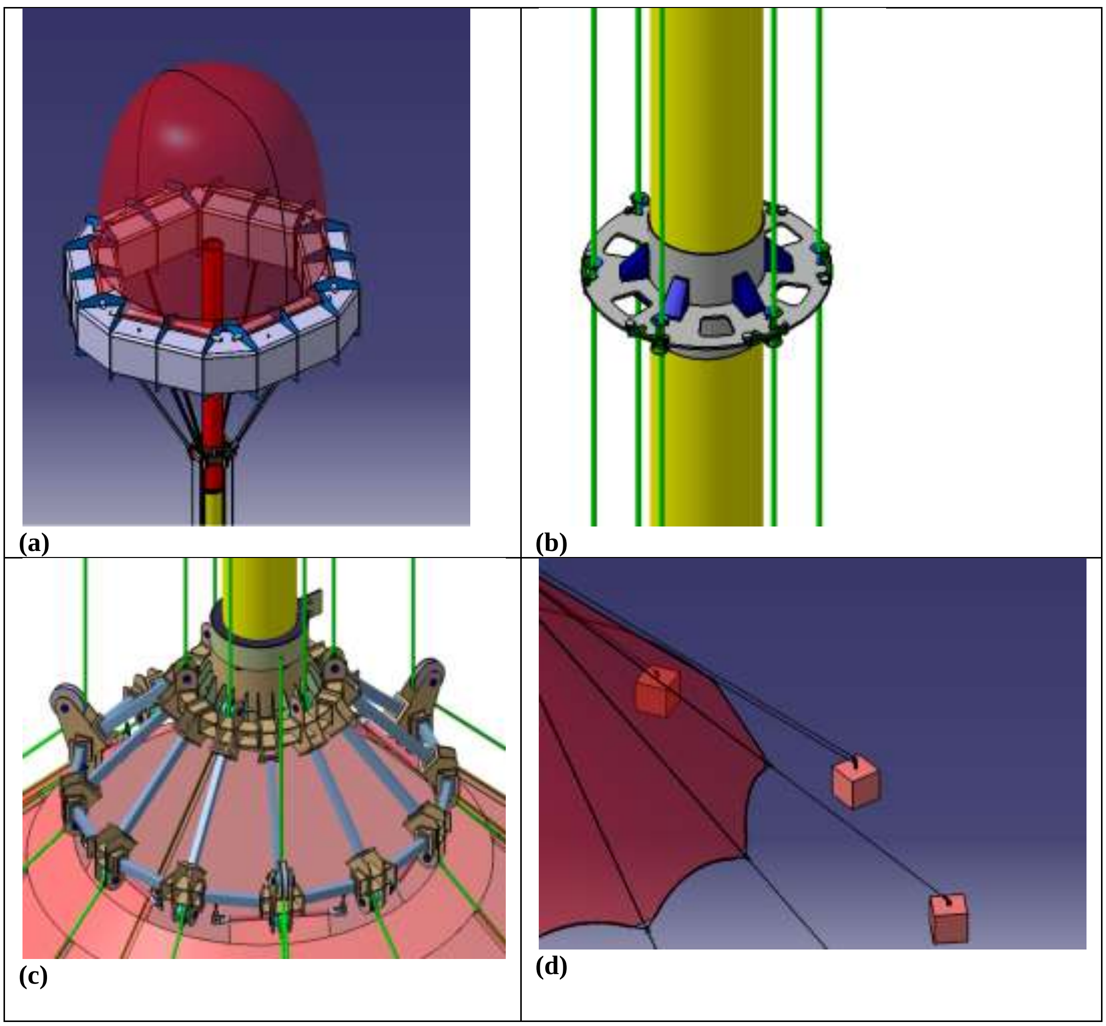

**Figure 1: The Underwater Structure (DIFIS System), a) Buffer Bell, b) Riser Tube, Stiffening Rings, and Mooring lines c) Dome Interface Unit, d) Dome and Anchoring System**

In contrast to common offshore structures, this new design for oil recovery is not affected by weather conditions at the sea surface such as waves, storm conditions etc, because it is fully submerged. As a result the structure needs to withstand only the hydrodynamic loads from sea currents and the high hydrostatic pressure due to the operational depth. This is an advantage as the system may need to remain submerged for long periods of time until oil recovery is completed [7].

## DESCRIPTION OF BUFFER BELL

### Buffer Bell Dimensions

The buffer bell (BB) is the main element of this underwater structure (DIFIS, 2007), composed of two main parts: the upper and the lower. A schematic configuration of the BB is shown in Figure 2. The upper part is a cylindrical structure with a spherical cup and it is called capacitor (A). The capacitor is actually a tank for the temporary storage of the recovered oil. The lower part, the floater, is a steel structure that consists of parallelepiped (B) and pyramid (C) substructures, of a double hull configuration that held together with connector plates (D) [6]. The floater is connected with the capacitor by means of steel tubes (E). The floater transfers the tensioning (upward) force through the steel rods (F) to the first stiffening ring where the mooring lines are anchored (G).

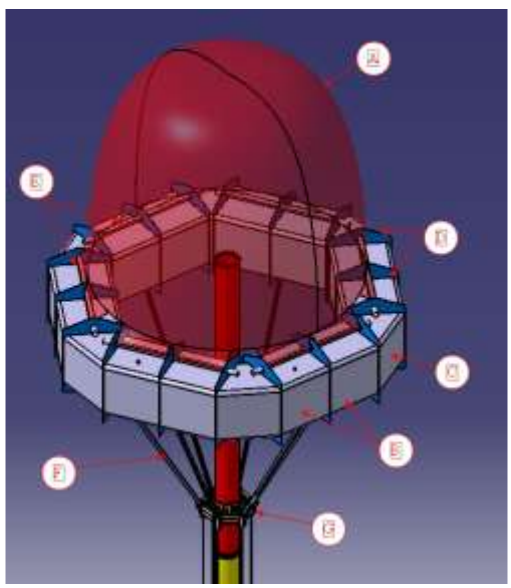

***Figure 2: The Buffer bell parts***

The production of net upward force is the difference between the buoyancy and the weight of the structure. The preliminary calculations and weight estimations [8], [9] determined that the buffer bell should produce 2000 tons buoyancy in order to retain the whole structure under tension [8]. Since the heaviest part of the system is the Buffer Bell and especially the floater, the design challenge was to develop a structure able to withstand the operation loads while keeping the weight limited under a maximum value. The basic external dimensions of the Buffer were dictated by its oil storage capacity (based on the functional specifications of the system.

### Floater's Double Hull structure

The floater part is essentially an assembly of a number of parallelepiped and pyramid double hull sub-structures. During the final concept, the double hull design was selected because it can be more efficient in terms of weight than the single hull, as far as the stiffness concerns [7]. Each hull must be able to withstand the hydrostatic pressure and its weight should not exceed a nominal value. In this work the structural analysis of an optimized Composite hull is presented for the case of the maximum hydrostatic pressure, in order to investigate the survivability of the structure. The functional specifications for the double hulls of the floater are presented in Table 1:

| **Functional specification** | **Value** |
|---|---|
| Hydrostatic pressure | • Maximum depth<br>• 70 m, 0.7MPa |
| Corrosion environment | • Sea salinity 35 ppt<br>• Withstand the Crude oil corrosion |
| Seawater temperature | 15ºC |
| Extreme loading requirements | • Withstand the maximum hydrostatic pressure without rupture |
| Maximum hull mass on air (tons) | • 45 |

***Table 1: Double hull's functional specifications***

The pyramid hull is formed of three parts a rectangular and two triangular. In contrast, the parallelepiped hull is consisted of rectangular parts only. Each of these parts is a composite structure which is connected with the other two. The Composite hull design is based on a structure, which consists of the external GFRP plates and a foam core. The foam core is reinforced externally with the GFRP plates in order to withstand the hydrostatic pressure. The combination of these two materials can lead to a high stiffness to weight ratio.

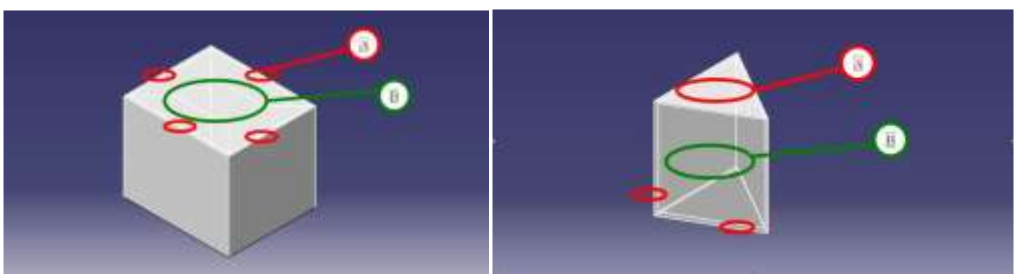

***Figure 3: Composite double hull basic elements, (A) GFRP plates, (B) Foam core***

The basic dimensions of the Composite hull are presented in Table 3:

| **Dimension** | **Value** | |
|---|---|---|
| | **Rectangular part** | **Triangular part** |
| Overall Length (L) (mm) | 5000 | 5000 |
| Overall Width (W) (mm) | 5000 | 5000 |
| Overall Height (H) (mm) | 6000 | 6000 |

***Table 3: Double hull's main dimensions***

The manufacturing of the Composite hull structure can be made with typical processes used in composite materials industry. The foam core is manufactured as continuous structure with molding. The GFRP plates are manufactured from GFRP fabric and mated on foam with handle up, vacuum bagging or infusion methods.

## MATERIAL SELECTION

The Glass Fiber Composites with epoxy or vinyl ester resin are very common in marine applications. The Glass fibers are produced in fabric materials and reinforced with resin in order to create a continuous material that can withstand structural loads. For the current analysis, the Glass Fabric 280 gr/$m^2$ (silane) twill weave with epoxy resin was selected. It is a low-priced glass fabric for standard applications in the design of boats and hulls. It has good drapability, good impregnation and satisfactory transparency. The mechanical properties for the fabric and epoxy resin are presents in Table 4:

| **Property** | **Value** |
|---|---|
| Glass Fabric 280 gr/m$^2$ | |
| Density (kg/m^3) | 2000 |
| Elasticity Modulus (GPa) | 19/19 |
| Compressive strength (warp/weft) (MPa) | 310/310 |
| Tensile strain (%) | 1.5 |
| Epoxy Resin | |
| Material | Isotropic |
| Density (kg/m^3) | 1300 |
| Elasticity Modulus (GPa) | 2.9 |
| Tensile Strength (MPa) | 55 |
| Tensile strain (%) | 3.0 |

***Table 2: Glass Fabric and Epoxy Resin mechanical properties***

Foams are used on many aerospace and marine applications as core materials. Especially, when compressive loads are applied, foams are very useful due to lightweight characteristics. At this work, AIREX R63 foam is used for the floater's hull core material. It is a damaged tolerant material, made of a closed-cell, linear, thermoplastic polymer foam. It has extraordinary impact strength and excellent fatigue resistance. This one of a kind formula combines very high elongation and excellent bond strength. It is cold formable to simple shapes and thermoformable to complex three-dimensional curves, and is non-friable. It is also exceptional material for dynamically loaded and shock absorbing structures. Many products such as: canoes, kayaks, hull bottoms, containers and helmets are manufactured by AIREX R63. The AIREX's mechanical properties are summarized in Table 5:

| **Property** | **Value** |
|---|---|
| Material | Isotropic |
| Density (kg/m^3) | 90 |
| Elasticity Modulus (MPa) | 56 |
| Compressive strength (MPa) | 0,6 |
| Shear Strength (MPa) | 1.0 |
| Shear Modulus (MPa) | 18 |
| Impact Strength (kJ/m^2) | 5.0 |

***Table 3:AIREX R63 mechanical properties***

The GFRP and foam properties were used for the structural design and FE models.

## STRUCTURAL DESIGN AND FINITE ELEMENT SUB-MODELS

The structural design was assessed on the interaction between the GFRP plate and the core foam. The hull had to withstand a high hydrostatic pressure due to the operational sea depth. At the preliminary phase of design, an extensive amount of sub models was proposed in order to select the thickness of the GFRP plate, investigating the behavior of the GFRP plate in rupture for parallelepiped configuration. The larger part of the plates was selected and the appropriate boundary conditions were applied in order to perform a parametric study based on the GFRP plate's thickness.

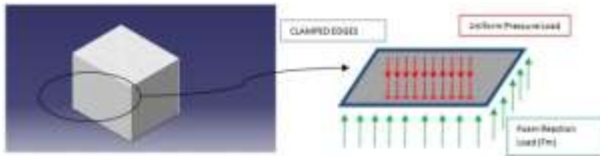


***Figure 4: Parallelepiped hull's equivalent sub model***

As it can be seen in the sub model, the GFRP plate is clamped at four edges. The hydrostatic pressure is applied at the top surface of the GFRP. The foam acts as a simple support, reducing the displacements of the GFRP plate. The foam can withstand a total force ($F_m$) equal to his maximum compressive strength (Figure 4).

The Foam Force (Fm) is modeled as an inertia load acting on the GFRP Plate at opposite direction, relative to the hydrostatic pressure. This force simulates the reaction force from the foam core which reduces the global displacements of the GFRP plate. The thickness of the GFRP plate and the weight were investigated, for the parallelepiped shape.

At the thickness direction, the GFRP plate is consisted of a GFRP/ epoxy layers. The plate can be manufactured with the Infusion or Vacuum bag method, succeeding higher volume fraction (Vf>0,4) and leading to a more efficient structure. The foam can be used as a mold in order to build the GFRP around the core. The combination of these materials can lead to a lighter structure [10] with the same or higher stiffness to weight ratio. For the different plates, the mechanical properties were estimated according to the Classical Laminate Theory for thin plates and the Rules of Mixtures (Vf=40%) [11]. The mechanical properties were lowered, including the effect of the seawater on the GFRP structure. Due to the dimensions (plate thickness to length (t/L<1/20), shell (2D) elements were used for the sub models. The $F_m$ reaction load was estimated as inertial load, based on the rupture criteria for the foam. These expressions are presented in (1), (2), (3):

The maximum foam's rupture load is: $F_f = \sigma_{cf} * A_f / n$ (1),
where n is the safety factor, $\sigma_{cf}$ is the foam's compressive strength $A_f$ and is the area of the applied load (the foam's surface).

The inertial $F_m$ reaction load is expressed by: $F_m = \rho * a * V$ (2),
where ρ is the GFRP plate's density, V is the volume of the GFRP plate and a is the calculated inertia acceleration.

As the GFRP plate is attached to the foam core, the plate can be deformed, reaching the maximum strength limits of the foam. A safety factor is used for the foam structure, minimizing the risk of failure.

$F_f = F_m$, thus $\sigma_{cf} * A_f / n = \rho * a * V$ and $a = \sigma_{cf} * A_f / n * \rho * V$ (3)

The hydrostatic pressure is applied on the top surface of the GFRP plate, the inertia acceleration is applied on GFRP volume, at the opposite direction of the hydrostatic pressure, reducing the stresses and the displacements on the plate. The GFRP plate's stress and the displacement field are investigated in order to select the most appropriate thickness for the GFRP plate. The composite's hull mass was also included in order to be compared with the steel hull mass. The FE analyses were performed for five sub models with GFRP plate thicknesses equal to 50mm, 100mm, 150 mm and 170mm. For the same thickness, the triangular plate was proved to be stiffer than the parallelogram plate (Figure 5), so further analyses were concentrated on the parallelogram GFRP plates and parallelepiped hull only. The mechanical properties and the input data that were used for the analysis are summarized in Table 4:

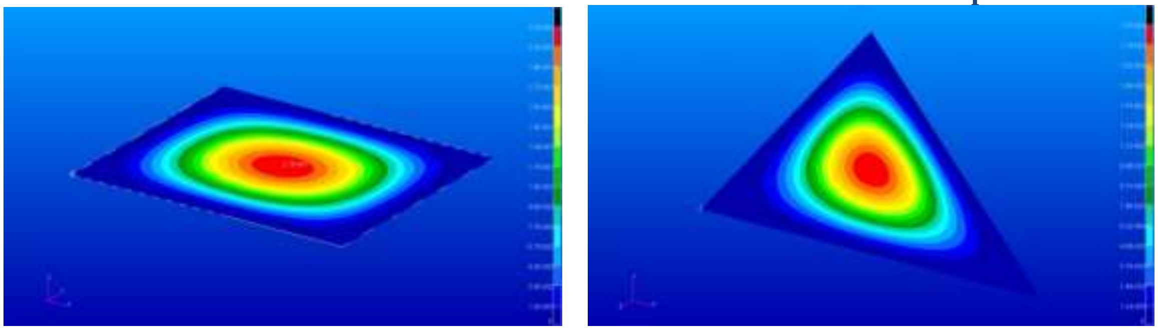

***Figure 5: Rectangular & Triangular plate comparison. Triangular plate (Disp 18,7 mm) is stiffer than rectangular (Disp 215 mm)***

| Input data for the Sub models' analyses | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| GFRP plate thickness (mm) | GFRP Plies (number x ply thickness) | GFRP plate mechanical properties (CLT Theory & Rules of Mixtures) | GFRP plate Volume ($m^3$) | Safety factor n | Total Area ($m^2$) | Inertia acceleration ($m/s^2$) | Reaction Force Fm (kN) | Composite Parallelepiped Hull mass (ton) |
| 50 | 186x 0,236 | E1= 9.34 GPa<br>E2= 9.34 GPa<br>ρ=1630 kg/$m^3$<br>v12=0,36<br>σ1=134 MPa<br>σ2=134 MPa | 5.45 | 1.5 | 29.5 | 7239.64 | 17700 | 21.9 |
| 100 | 424x 0,236 | | 10.8 | 1.5 | 29.0 | 3558.2 | 17400 | 30.2 |
| 150 | 636 x 0,236 | | 16.1 | 1.5 | 28.5 | 2331.2 | 17100 | 38.3 |
| 170 | 720x 0,236 | | 18.12 | 1.5 | 28.3 | 2042.6 | 16980 | 41.4 |

***Table 4: Input data considerations for the sub models' FEA***

The chemical degradation of the GFRP due to the immersion in the seawater was studied from many researchers [12]. The seawater temperature has also reduces the mechanical properties. According to the researchers, the GFRP aging can lead to a total reduction of 65% for the tensile strength and 32% for elasticity modulus. These parameters had also included in the FE analyses. A total amount of 3111 2D elements were used for each sub model analysis. Pseudo isotropic material properties were used in this analysis due to the glass fabric orientation and epoxy resin properties. The plates were clamped at the four edges. The results for the stress/strain and displacement field are presented in Figures 6-13:

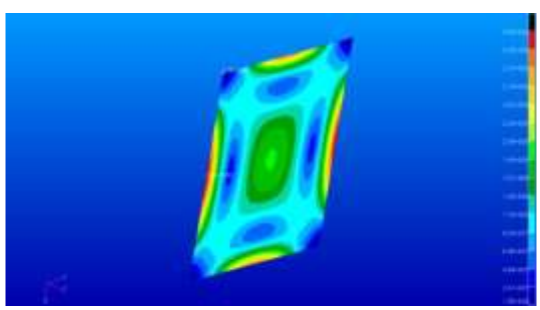

***Figure 6: Maximum Von- Misses stresses (Pa) for thickness 50 mm***

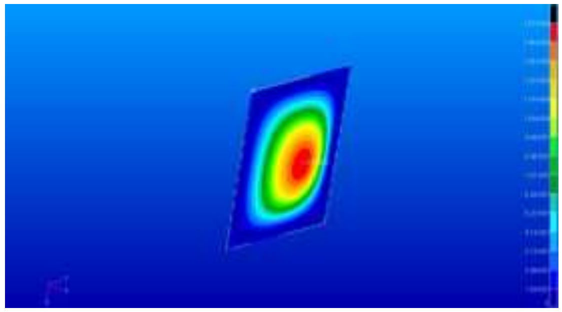

*Figure 7: Plate Displacements (m) for thickness 50 mm*

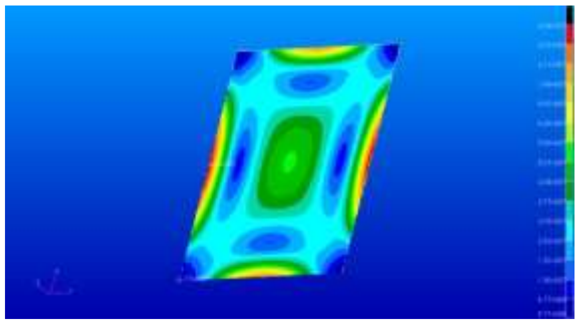

*Figure 8: Maximum Von- Misses stresses (Pa) for thickness 100 mm*

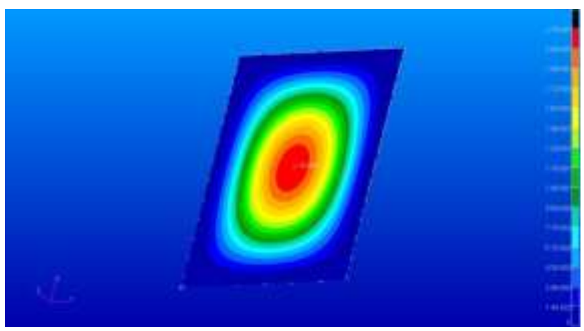

*Figure 9: Plate Displacements (m) for thickness 100mm*

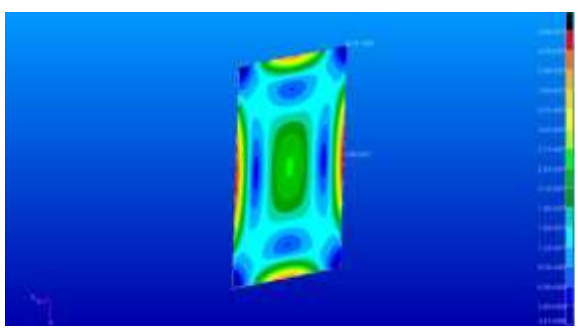

*Figure 10: Maximum Von- Misses stresses (Pa) for thickness 150 mm*

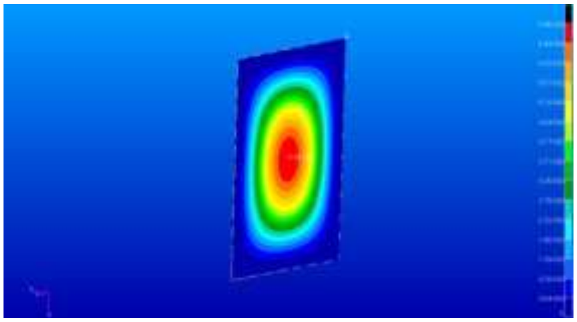

***Figure 11: Plate Displacements (m) for thickness 150 mm***

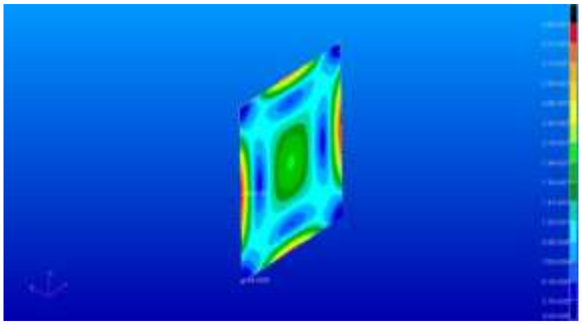

***Figure 12: Maximum Von- Misses stresses (Pa) for thickness 170 mm***

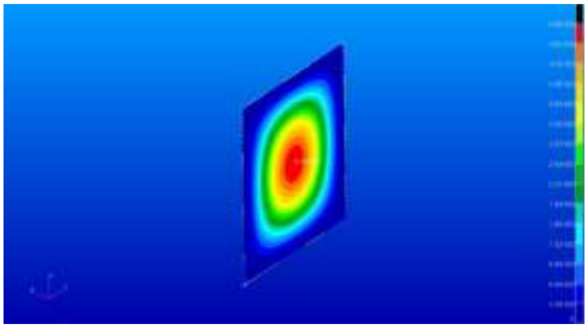

***Figure 13 : Plate Displacements (m) for thickness 170 mm***

FEA results are summarized in Table 5:

| FEA Comparisons for the GFRP plates | | | |
|---|---|---|---|
| GFRP plates thickness (mm) | $\sigma_{max}$ (MPa) | $\varepsilon_{max}$ (%) | Displacement at the center (m) |
| 50 | 343.0 | 5.400 | 1.570 |
| 100 | 93.4 | 1.470 | 0.2150 |
| 150 | 44.9 | 0.707 | 0.0690 |
| 170 | 36.0 | 0.567 | 0.0495 |

***Table 5: Sub models' FEA results for various GFRP thicknesses***

Regarding the structural results and the hull's weight considerations, the GFRP plate with thickness equal to 170 mm was selected as the most appropriate solution for the composite hull design. As the applied pressure of 0.7 MPa is the worst case scenario, the GFRP has to withstand this load without rupture. For this GFRP thickness, the hull can carry the maximum load while its weight is kept on 42 tons. The strains are maintained close to 0.567%, which is valid for GFRP composite structures. A further full scale 3D FEM analysis was performed for this applied pressure and further investigation was necessary during the detail design. The shear strength between the Foam Core and GFRP plate had to be testified in order to avoid the shear failure.

## FULL SCALE FINITE ELEMENT MODEL

At the latter stages of design, the full scale Composite Hull was analyzed in order to extract very specific results for its response. The double hull floater of the Buffer Bell structure was modeled and analyzed in Patran/Nastran commercially available FEM code [13]. At the full scale, the interfacial shear stresses should be investigated in order to avoid the shear failure between the GFRP plate and foam. For this reason, 3D solid elements were selected for this analysis, both modeling GFRP plates and Foam. Due to the huge dimensions, the overall structure was modeled as a continuum structure. The stress, strain and displacement field was investigated. In current analysis the survivability at the maximum depth was main scope, so the maximum yield stress/strain must not be exceeded. The shear strength between Foam Core and GFRP was also another topic for analysis. The dimensions of each component which were used in the FE model are presented in Table 7:

| **Parameter** | **Value** |
|---|---|
| Thickness of GFRP plate (mm) | 170 |
| Foam core volume ($m^3$) | 139.2 |

***Table 6: Components dimensions for FEM analysis***

The finite element model is presented in Figure 13:

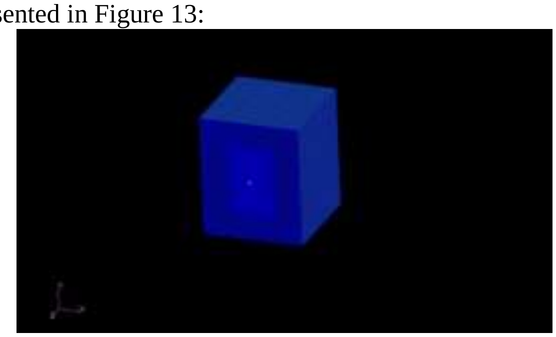

***Figure 14: Floater's parallelepiped composite hull mesh***

***Figure 15: Floater's parallelepiped composite hull detailed mesh***

A total number of 500.000 8-node (hex) elements were used for the parallelepiped structure. The hull is clamped at the eight surfaces (where one hull is attached to the other using the connector plate) (Figures 14-15).

The material properties that were selected for the analysis were presented in Tables 4 and 5. The maximum hydrostatic pressure of 0.7MPa is applied as a uniform load on the element faces. The material is modeled as linear elastic and the analysis is linear static. The Nastran solver (SOL 600) was used for the calculations.

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## RESULTS

The results presented in the next are based on the analysis of the worst case scenario.

**Worst case Scenario (Applied pressure 0.7 MPa)**

The stress and displacement distribution for both basic structural components of the Floater's parallelepiped hull made out of GFRP and Foam, in the worst case scenario are presented in Figures 16-17:

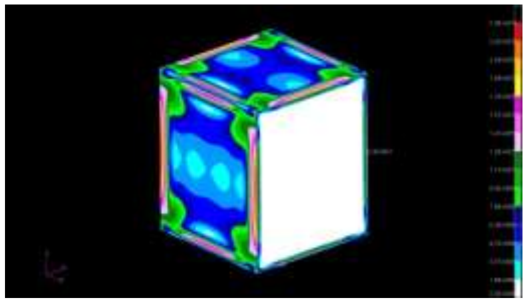

*Figure 16: Worst case scenario: results for the parallelepiped hull's von Mises stresses (Pa)*

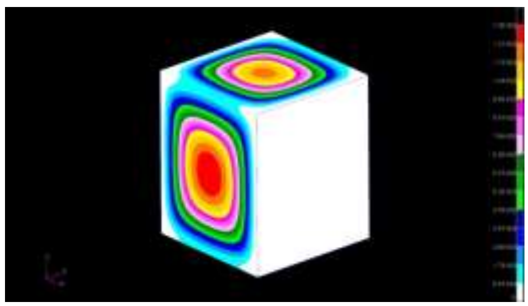

*Figure 17: Worst case scenario: results for the parallelepiped hull's displacements (m)*

Again, in order to better visualize the developed maximum stresses and displacements on Foam core, the above results have been plotted again for the middle plane of the analyzed structural components (excluding the GFRP). These results are presented in Figures 18-21:

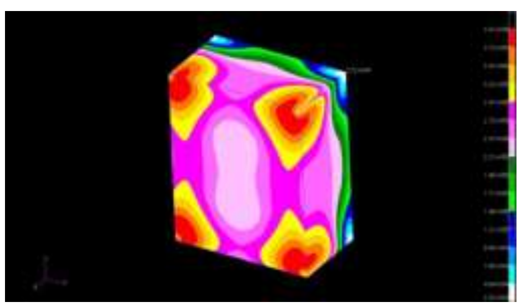

*Figure 18: Worst case scenario: results for the parallelepiped hull's core foam von Mises stresses (Pa)*

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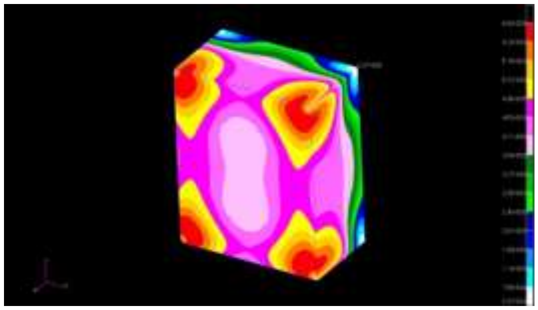

***Figure 19: Worst case scenario: results for the parallelepiped hull's core foam von Mises strains***

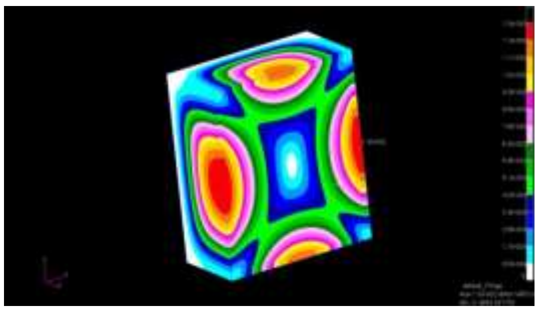

***Figure 20: Worst case scenario: results for the parallelepiped hull's core foam displacements (m)***

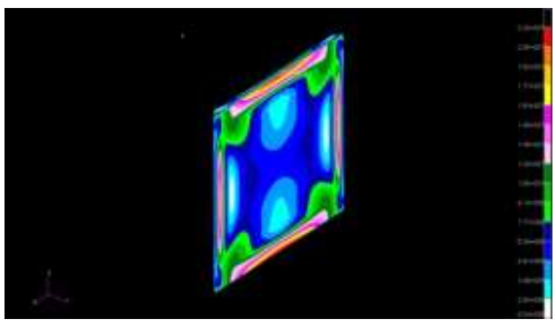

***Figure 21: Worst case scenario: results for the parallelepiped hull's GFRP plate Von Mises stresses (Pa)***

The shear stresses and strains had also been investigated close to the interface between the foam and the GFRP plate.

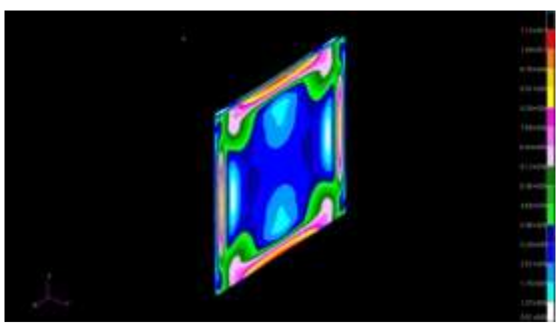

***Figure 22: Worst case scenario: results for the parallelepiped hull's GFRP plate max shear stresses (Pa)***

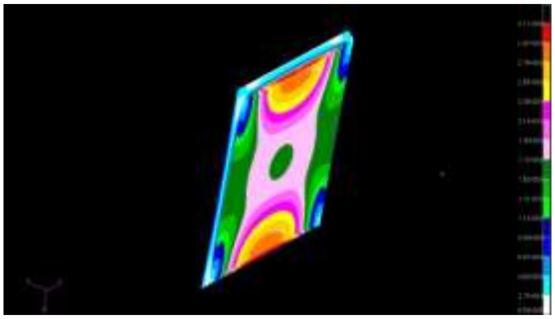

***Figure 23: Worst case scenario: results for the parallelepiped hull's GFRP plate max shear strains***

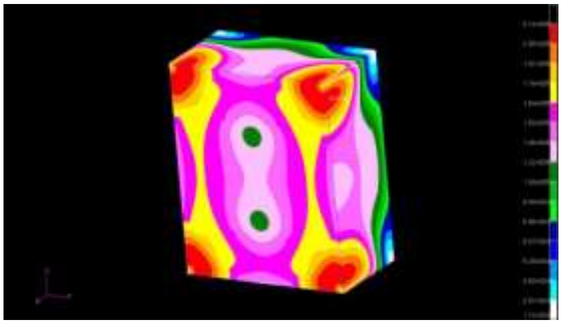

***Figure 24: Worst case scenario: results for the parallelepiped hull's foam core max shear stresses (Pa)***

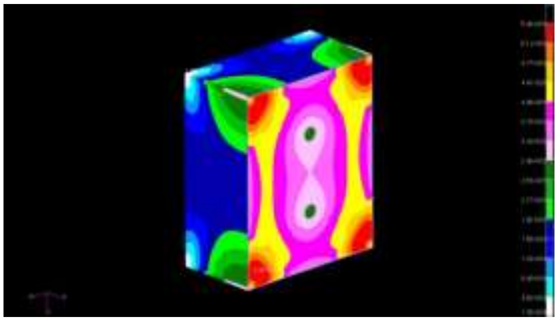

***Figure 25: Worst case scenario: results for the parallelepiped hull's foam core max shear strains***

As it can be seen by the results presented above, in the case of worst case scenario the maximum Von-Misses stress is close to 0.4 MPa for the Foam Core and 24 MPa for the GFRP plate. The shear stresses were not so large to cause interfacial or interlaminar failures (0,22 MPa for Core Foam and 11 MPa for GFRP plate) . Finally, the total weight of the parallelepiped hull is 41 tons, succeeding the weight specification and it is 8% lighter than the steel hull.

## CONCLUSIONS

In this work, finite element sub models were built up to predict the stress distribution over the critical zones of a composite BB structure. The results of this analysis were used for the full scale hull model. The results showed that for the worst case, the composite hull can withstand the hydrostatic pressure. The maximum weight of the hull does not exceed the maximum limit according to the functional specifications, so the required tensioning force is obtained. Future work includes the redesign of the hull with different shape (cylindrical, spherical) in order to reduce the volume and the weight.

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