Analysis Report

MCG 4322A

SUB 1A

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1 Project Charter

1.1 Mandate

The mandate is to design the hull of a versatile manned submersible. The design process will reduce costs while maintaining the necessary functionality and safety. We will be collaborating with the ballast team to incorporate their design with our hull design. The design must be able to be parameterized to satisfy various market needs and accommodate various depth ratings. Our responsibilities include the design of the submarine hull, frame, hatch, seals and access mechanisms. We will be designing:

- Control panel mount and seating assembly.
- Release buoy
- Attachment point for lifting and anchor points.
- Electrical component mounts such as thrusters, batteries and lights.
- To accommodate the ventilation system.

1.2 Requirements

- Submarine should be properly sealed and be corrosion resistant.
- Submarine should be capable of withstanding pressures up to 100 atm.
- Submarine should be capable of seating a pilot and two passengers.
- Submarine should be designed to accommodate a ventilation system, thrusters and compressed air tanks.

1.3 Constraints

- Submersible must attain speeds from 2 to 6 knots
- Diving depth between 330m and 1000m.

1.4 Criteria

- Submersible should be as light as possible.
- Lower costs involved with manned submersibles while maintaining high standard of safety (reduce shipping and operating cost).

1.5 Parameterization Outline

- Submarine weight, size, speed and diving depth

2 Detailed Design

2.1 System Layout

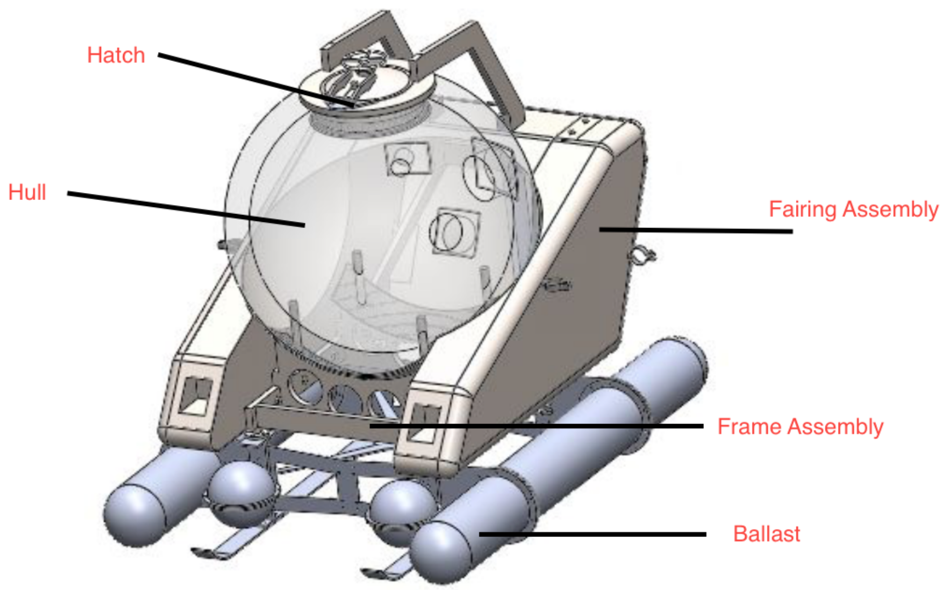

Figure 1: Submarine and Ballasts

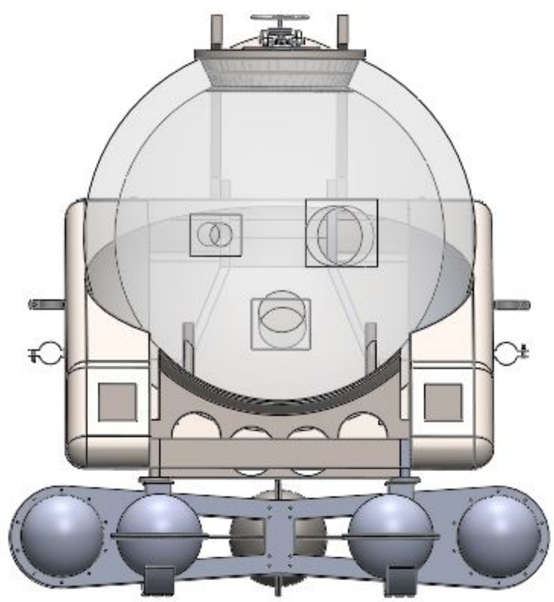

Figure 2 : Submarine and Ballasts front view

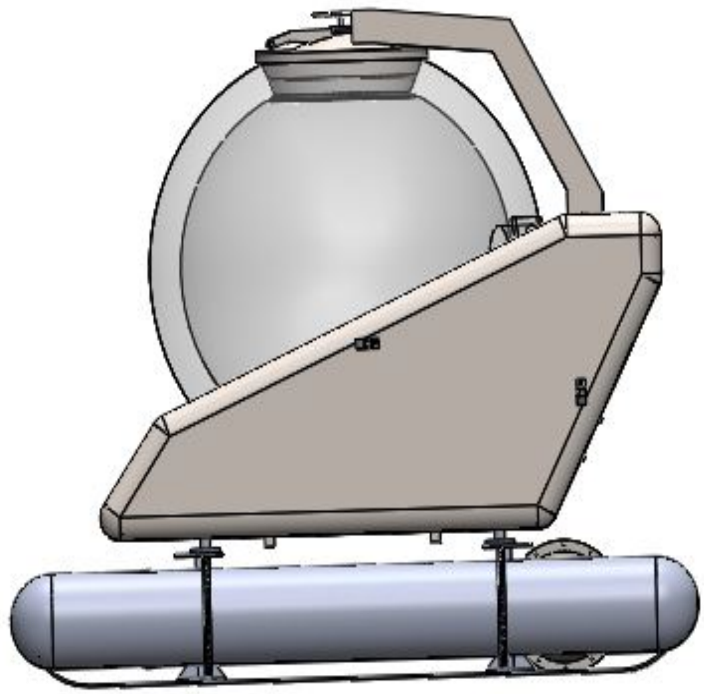

Figure 3: Submarine and ballasts side view

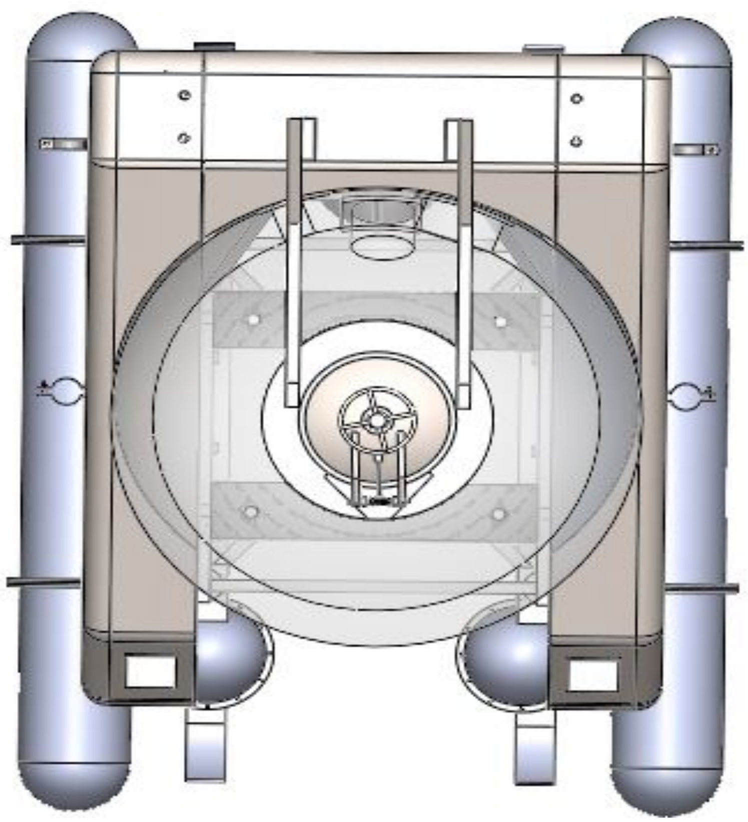

Figure 4: Submarine and ballasts top view

2.2 Detailed Drawings and Description

Frame Subsystem

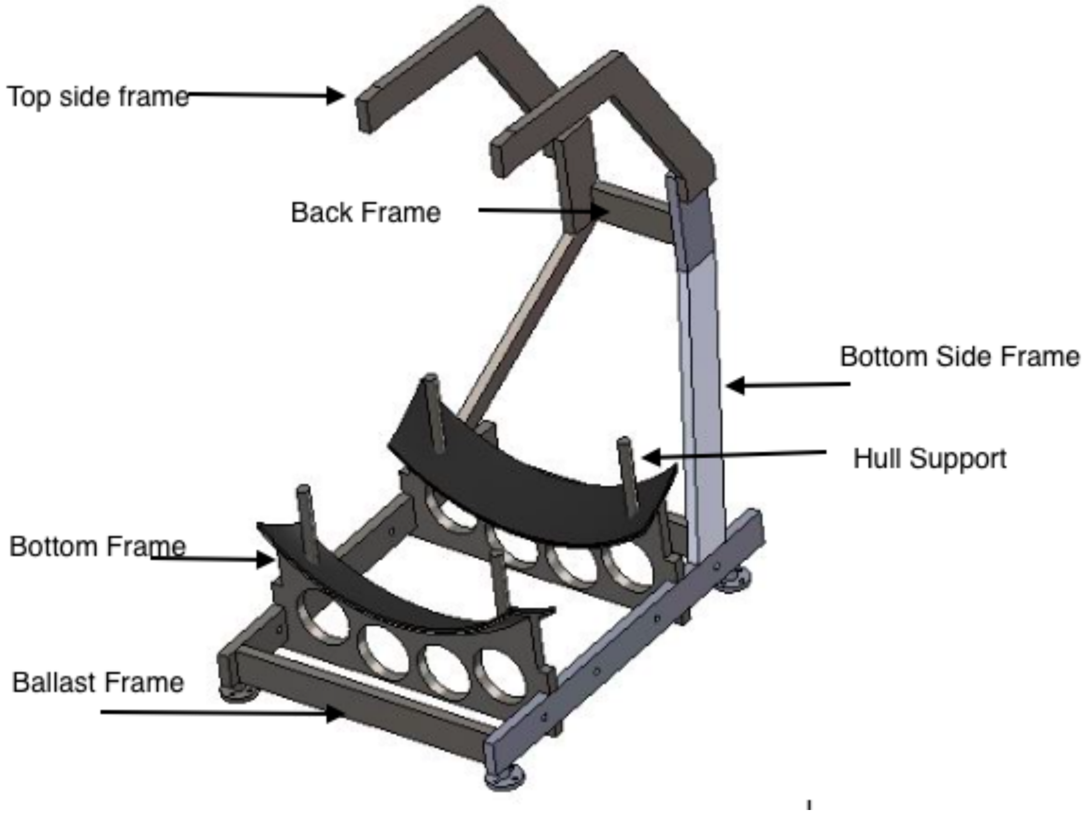

Figure 5: Frame

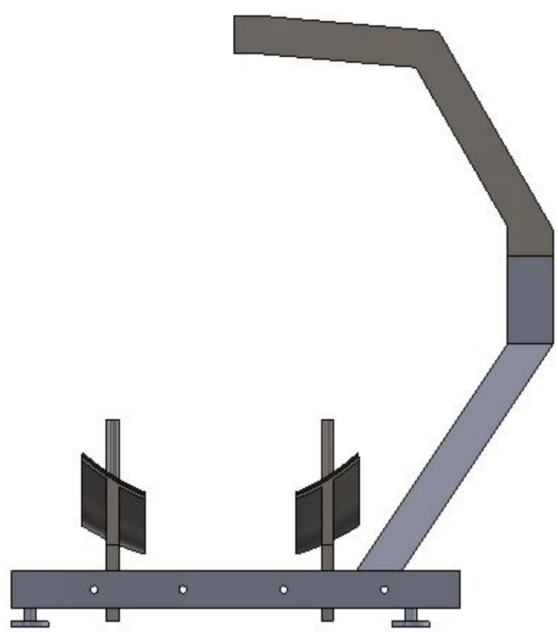

Figure 6: Frame Side View

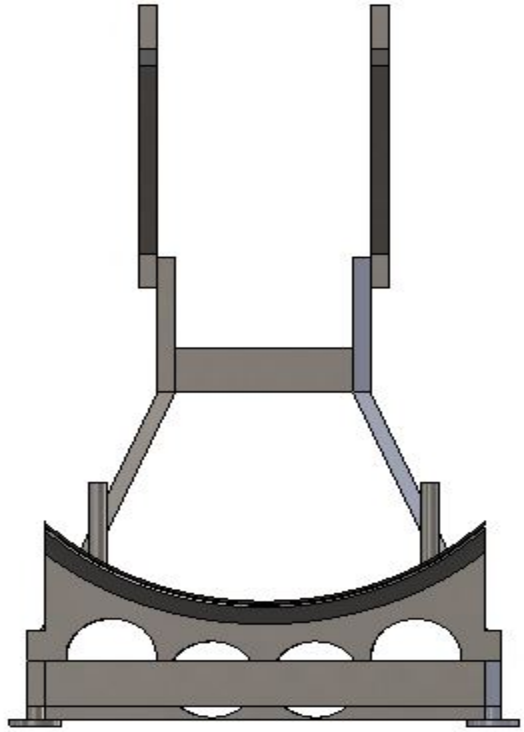

Figure 7: Frame Front View

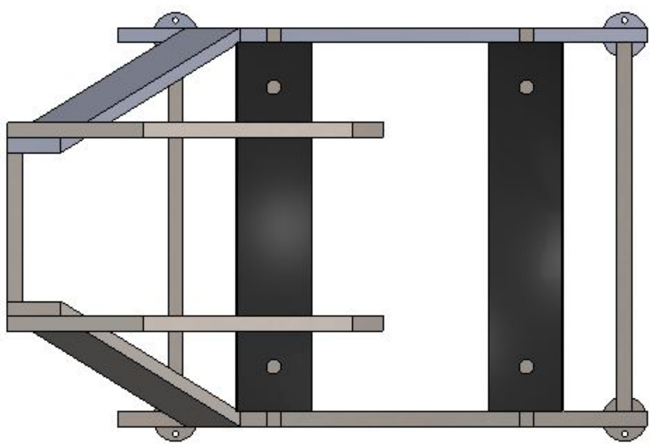

Figure 8: Frame Top View

Fairing Assembly

Figure 9: Fairing

Figure 10: Fairing Side View

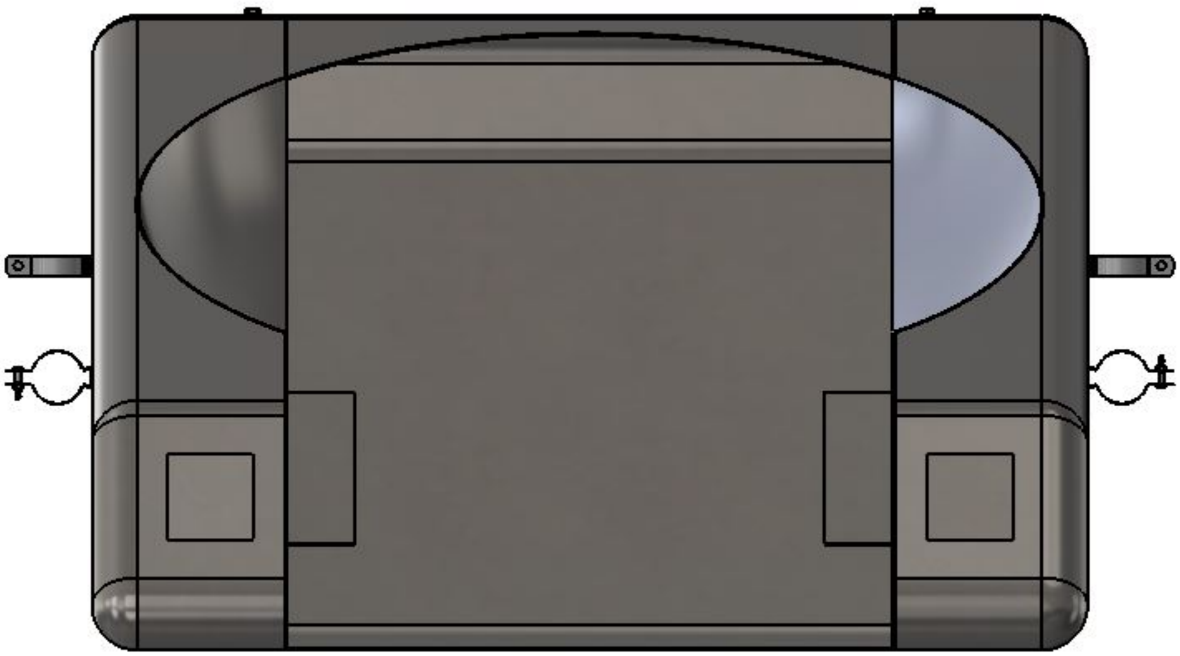

Figure 11:Fairing Front View

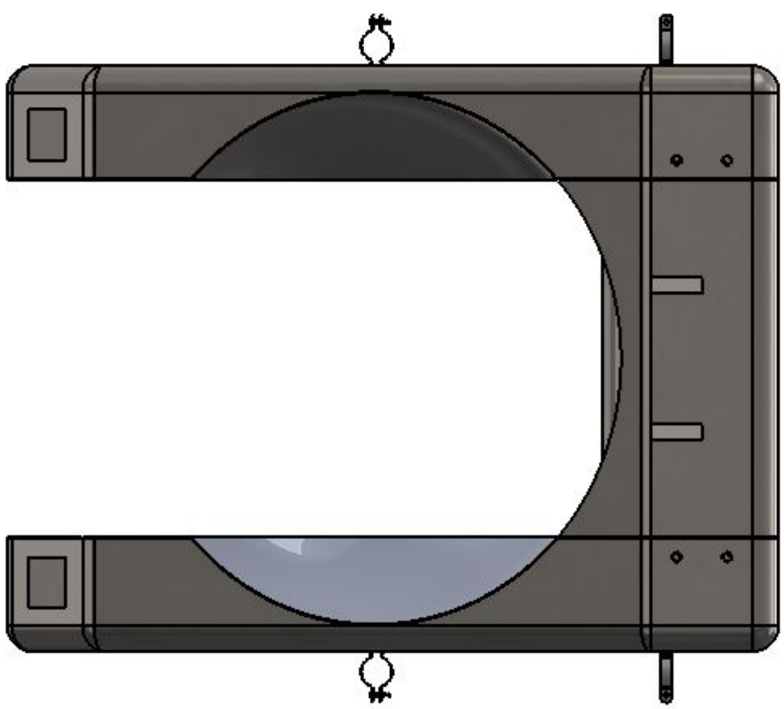

Figure 12: Fairing Top View

Hatch System

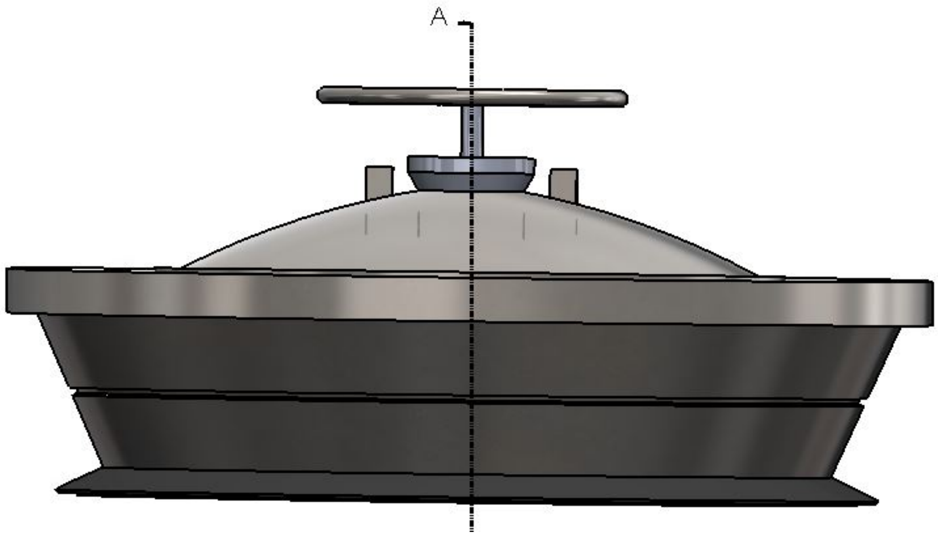

Figure 13: Hatch Assembly

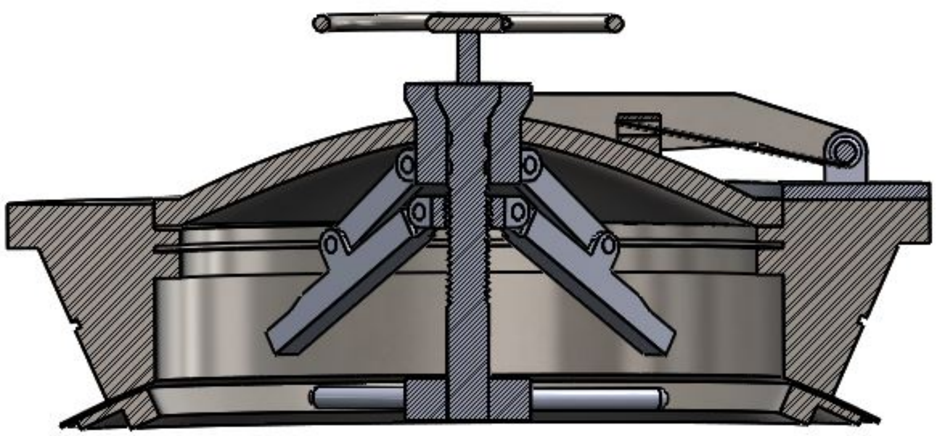

Figure 14: Cross-section of Hatch Assembly (A)

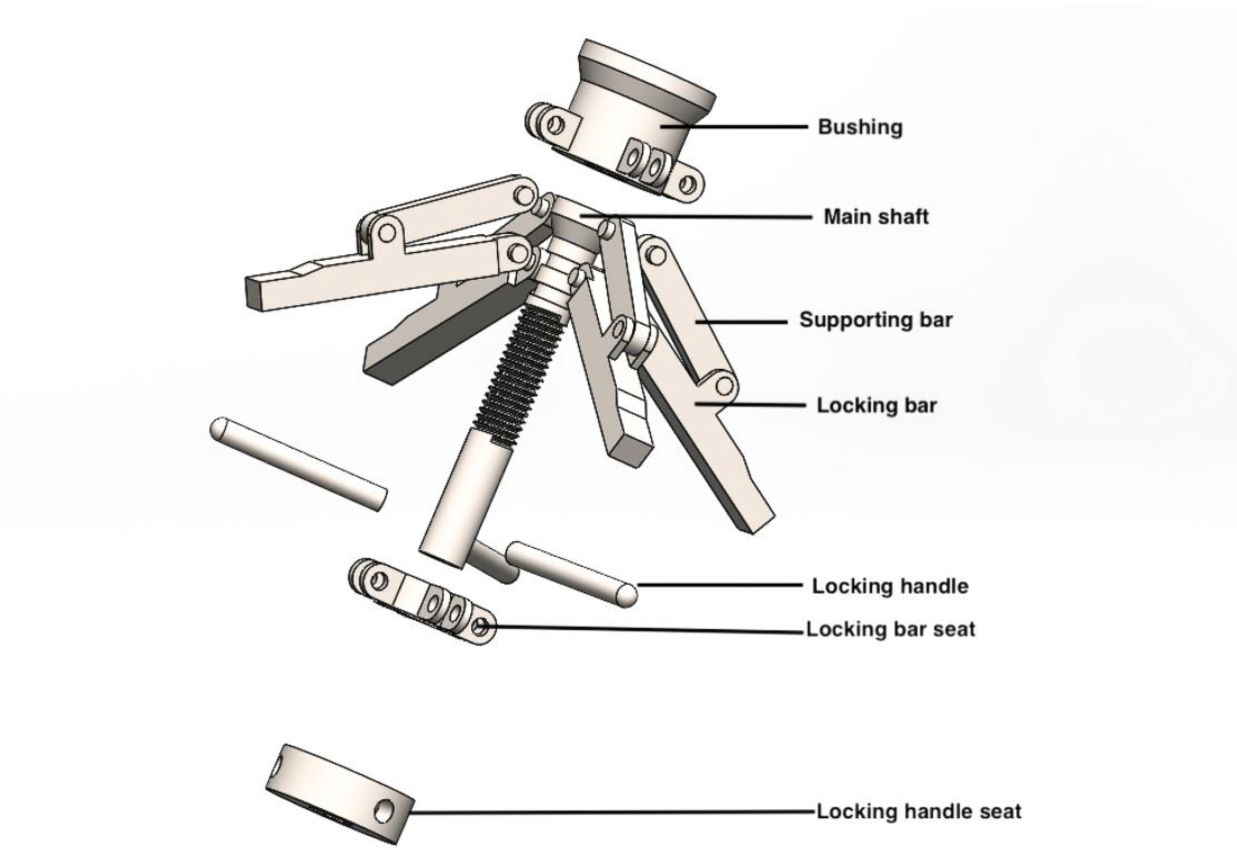

Figure 15: Exploded View of Hatch Locking Mechanism

3 System Modelling

3.1 Assumptions and Approximation

- Conditions evaluated at the maximum diving depth (1000m)
- Submarine speed = 6 knots
- Modelling of the hull and frame without the ballasts, but including ballast weight in force analysis.
- Minimal interference with sea life.

3.2 Geometric Mass and properties (Component Properties)

3.2.1 Geometric relations and restrictions

Hull

Table 1: Hull specifications

Outer Diameter (m)	2.02
Inner Diameter (m)	1.70
Shell Thickness (m)	0.16
Acrylic Density (kg/m^3)[1]	1180
Hull Mass (kg)	2057.06
Hull Displaced Mass (kg)	4440.87

Frame

Table 2: Frame Specifications

Frame cross sectional area (m^2)	0.0075 (0.15m x 0.005m)
Frame mass (kg)	977.77
Frame displaced mass (kg)	133.77

Hatch

Table 3: Hatch Specifications

Hatch opening (m)	0.6
O ring outer diameter (mm)	582
O ring inner diameter (mm)	568
O ring width (mm)	7
Hatch seat outer diameter (m)	0.88

Fairing

Table 4: Fairing Specifications

Fairing thickness (m)	0.005
Fairing Mass (kg)	778.49
Fairing Displaced Volume (m^3)	1.70
Fairing Displaced Mass (kg)	1749.3

Mass and Buoyancy

Table 5: Mass properties

Total Hull Mass (kg)	5292.20
Ballast Mass (kg)	800
Drop-weight Mass (kg)	813.47

Table 6: Buoyancy properties

Total Hull Submerged Mass (kg)	6705.67
Empty Ballast Submerged Mass (kg)	2200
Full Ballast Mass (kg)	144.06

3.2.2 Centre of Gravity

$$x_m = \frac{\sum_{i=1}^{N} m_i x_i}{M}, y_m = \frac{\sum_{i=1}^{N} m_i y_i}{M}$$

Table 7: Centre of Gravity Calculations

Component	Mass (kg)	X (m)	Y (m)	Mass*X (kg*m)	Mass*y (kg*m)
Frame	977.77	1.21	0.69	1183.10	674.66
Fairing	778.49	1.15	0.65	895.26	506.02
Batteries	220.00	2.31	0.28	508.20	60.50
Hatch	103.49	1.61	2.05	166.61	212.15
Inner components	475.00	1.61	1.20	764.75	570.00
Hull	2057.06	1.61	1.20	3311.87	2468.48
Compressed air	544.31	1.61	0.18	876.34	95.25
Oxygen Tank	136.08	0.68	0.73	92.53	99.34
Ballast	800.00	1.61	-0.21	1288.00	-168.00
Drop-weight	813.47	1.61	-0.21	1309.69	-170.83
Σ					
Without ballast and drop-weight	5292.20			7798.68	4686.40
With Ballast (w/ drop-weight)	6905.67			10396.37	4347.57
With Ballast(w/o drop-weight)	6092.20			9086.68	4518.40

Table 8: Centre of Gravity

	x(m)	y(m)	Force of Gravity (N)
Without ballast and drop-weight	1.47	0.89	51916.48
With Ballast (w/ drop-weight)	1.51	0.63	67744.66

With Ballast(w/o drop-weight)	1.49	0.74	59764.48

3.2.3 Centre of Buoyancy

$$x_m = \frac{\sum_{i=1}^{N} m_i x_i}{M}, y_m = \frac{\sum_{i=1}^{N} m_i y_i}{M}$$

Table 9: Centre of Buoyancy

Component	**Mass (kg)**	**X (m)**	**Y (m)**	**Mass*X (kg*m)**	**Mass*y (kg*m)**
Frame	133.77	1.21	0.69	161.86	92.30
Fairing	1749.30	1.23	0.64	2151.64	1119.55
Batteries	0.00	2.31	0.28	0.00	0.00
Hull	4440.87	1.61	1.20	7149.80	5329.04
Compressed air	270.17	1.61	0.18	434.97	48.63
Oxygen Tank	111.56	0.68	0.73	75.86	81.44
ballast (empty)	2200.00	1.61	-0.21	3542.00	-462.00
ballast (full)	144.06	1.61	-0.21	0.00	0.00
Σ					
Without ballast	6705.67			9974.14	6670.97
With empty ballast	8905.67			13516.14	6208.97
With full ballast	6849.73			9974.14	6670.97

Table 10: Centre of Buoyancy with the addition of Ballasts

	x	y	Force of Buoyancy (N)
Without ballast	1.47	0.89	65782.66
With empty ballast	1.51	0.63	87364.66
With full ballast	1.49	0.74	67195.89

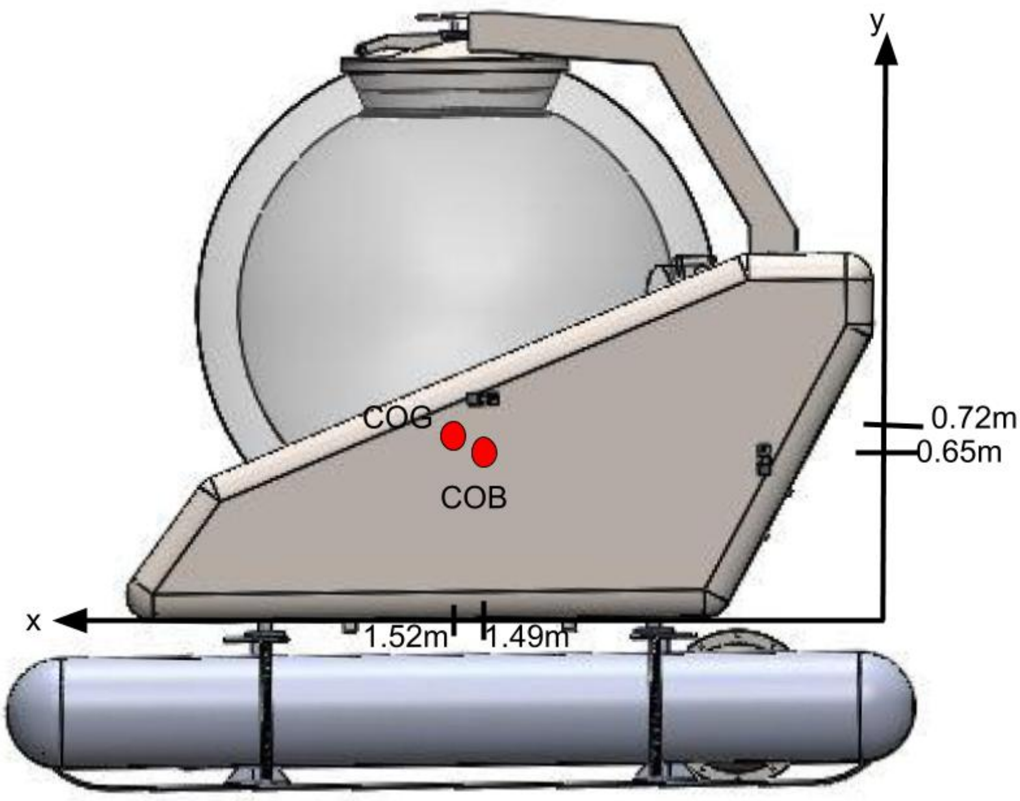

Figure 16: COG and COB with empty ballast and drop weight

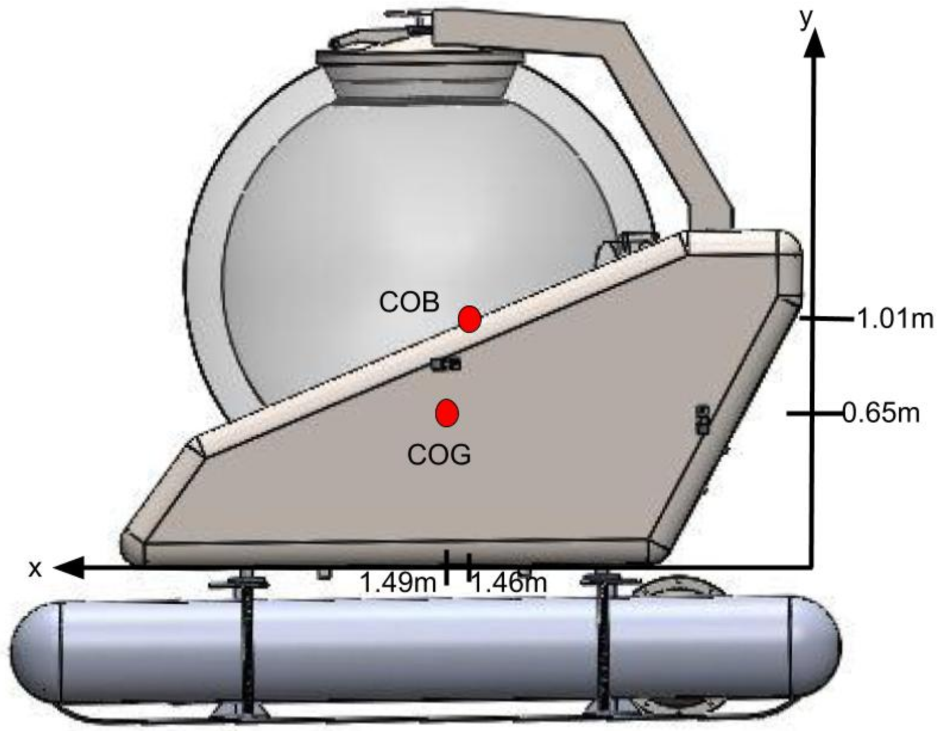

Figure 17: COG and COB with Full ballast and drop weight

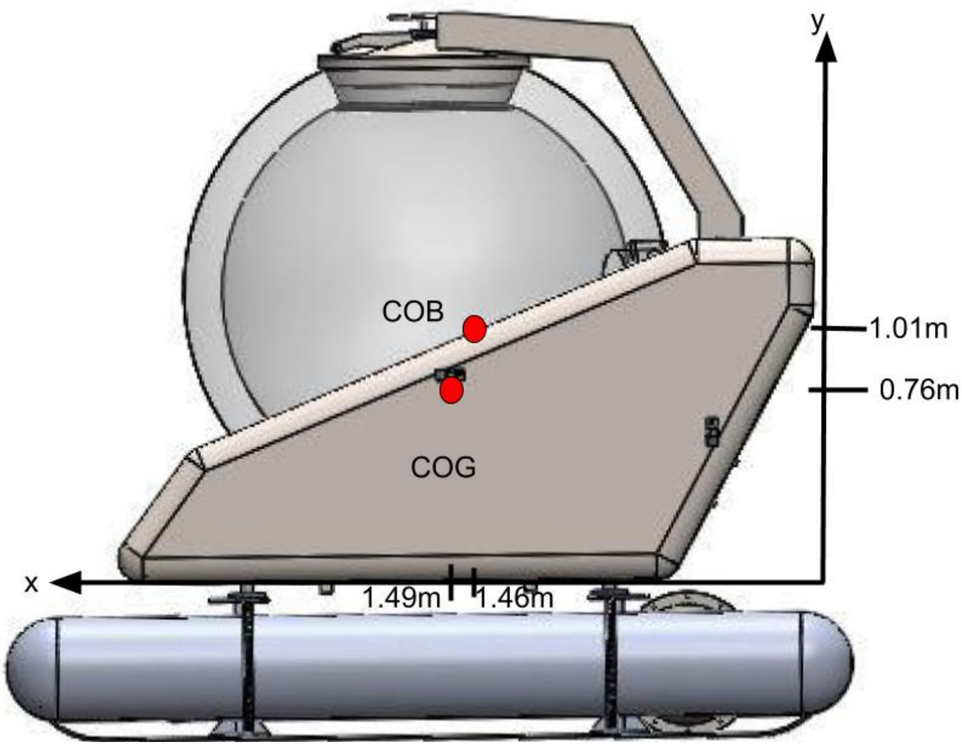

Figure 18: COG and COB with Full ballast and no drop weight

3.2.4 Moment of Inertia

3.3 Kinematics and Applied Forces

3.3.1 Kinematics

Kinematics and Newton's Second Law

Assume:

$v_i=0$, $a_i=0$, $x_i=0$, $v_f=3.09m/s$ and $t=5s$

$$x=\frac{vi+vf}{2}\cdot t=\frac{(3.09m/s)}{2}\cdot(5s) = 7.725m$$

$$a=\frac{\Delta v}{\Delta t}=\frac{vf-vi}{t}=\frac{(3.09m/s)-(0m/s)}{5}=0.618m/s^2$$

3.3.2 Newton's second law

$$F = ma =(6905.67kg)*(0.618m/s^2)= 4267.70N$$

This is the total force that is required to move the submarine forward.

3.3.3 External Applied Forces

Horizontal Drag Force and Hydrodynamics

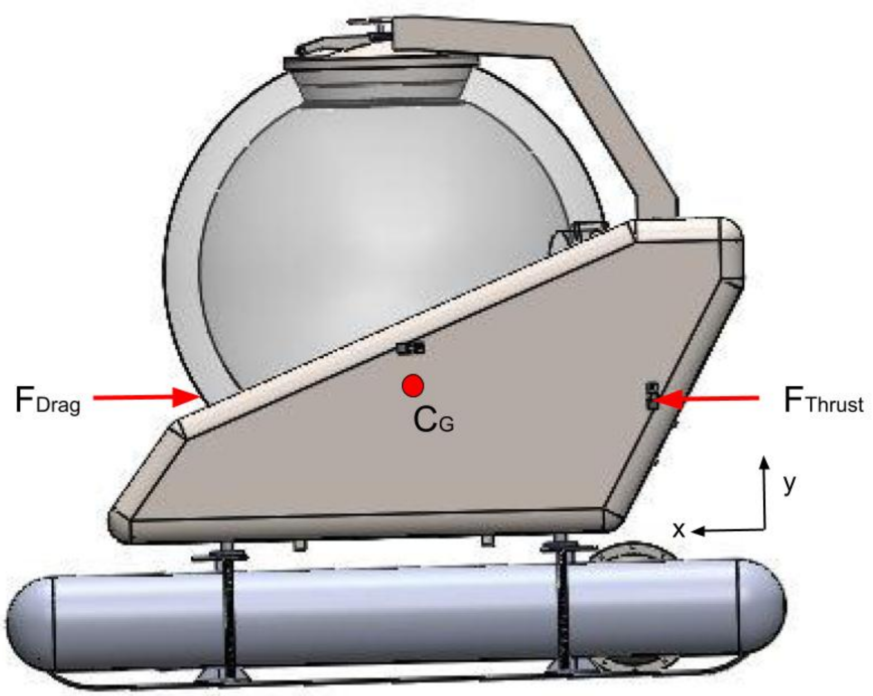

Figure 19: Free Body Diagram of submarine showing external force of drag (Horizontal Motion)

The resultant drag force in the negative x-direction and the force from the thrusters in the positive x-direction are both at a height of 0.99m from the reference

Forward drag:

$$Re = \frac{\rho u D}{\mu} = \frac{uL}{v} = \frac{(1029kg/m^3)(3.09m/s)(2.1m)}{(0.001518Ns/m^2)} = 4.4 \times 10^6$$

$C_D = 0.5$ $(Assuming\ a\ spherical\ frontal\ area)$ and referring to figure 18

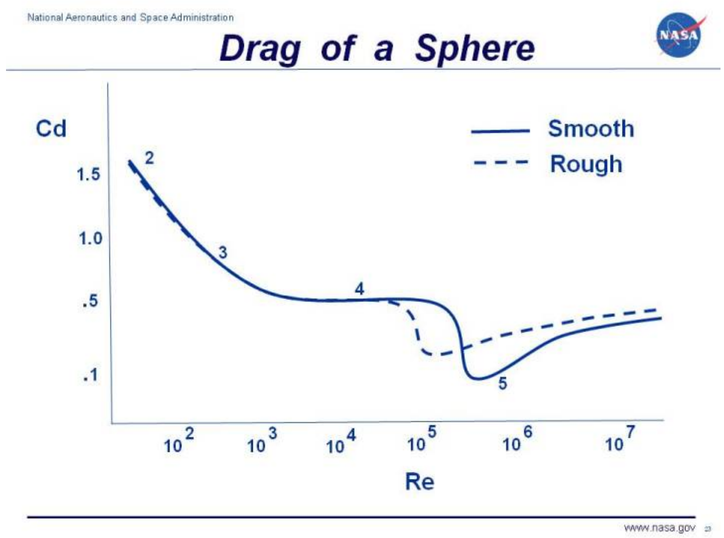

Figure 20 : Coefficient of Drag for spheres against Reynold's number

A = 5.5m^2

Drag Force (F_d)

$F_d = \frac{1}{2}\rho u^2 C_D A = \frac{1}{2}(1029kg/m^3)(3.09m/s)^2(0.5)(5.5m^2)$

$F_d = 13509.37N$

This is the force required by the thrusters that support horizontal motion to operate at 3.09m/s(6knots)

Required Power (P) for 1 of 2 thrusters

P = F*V = $\frac{13509.37N}{2} * (3.09m/s)$= 20871.97Nm/s

P = 20.87KW

Vertical Drag

Area of the top projection= Sub length x sub width

A=8.05m^2

Assuming a spherical body, with different projected area C_d= 0.51

Assuming a rise of 2.06m/s(4 knots)

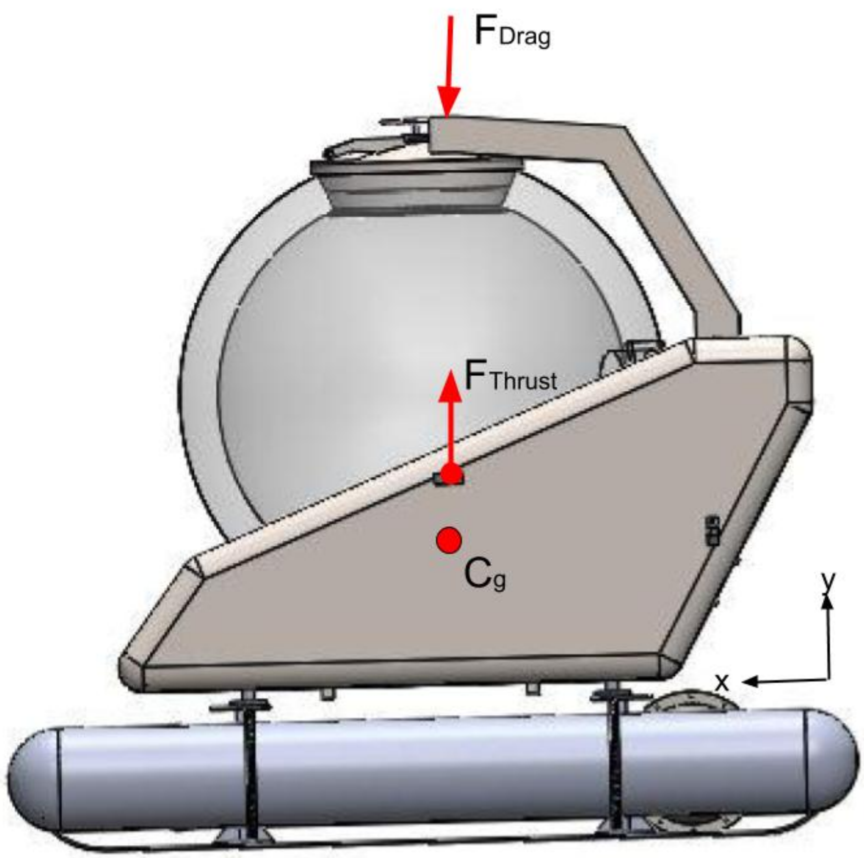

Figure 21 : Free Body Diagram of submarine showing external force of drag (Vertical Motion)

$F_d = \frac{1}{2}\rho u^2 C_D A = \frac{1}{2}(1029kg/m^3)(2.06m/s)^2(0.51)(8.05m^2) = 8963.67N$

F_d = 8963.67N

This is the force required by the thrusters that support vertical motion to operate at 1m/s

Vertical descent drag:

Area of the top projection= Sub length x sub width

A = 8.05m^2

Assuming a spherical body, with a different projected area

C_d = 0.61

Assuming a descent of 2.06m/s (4 knots)

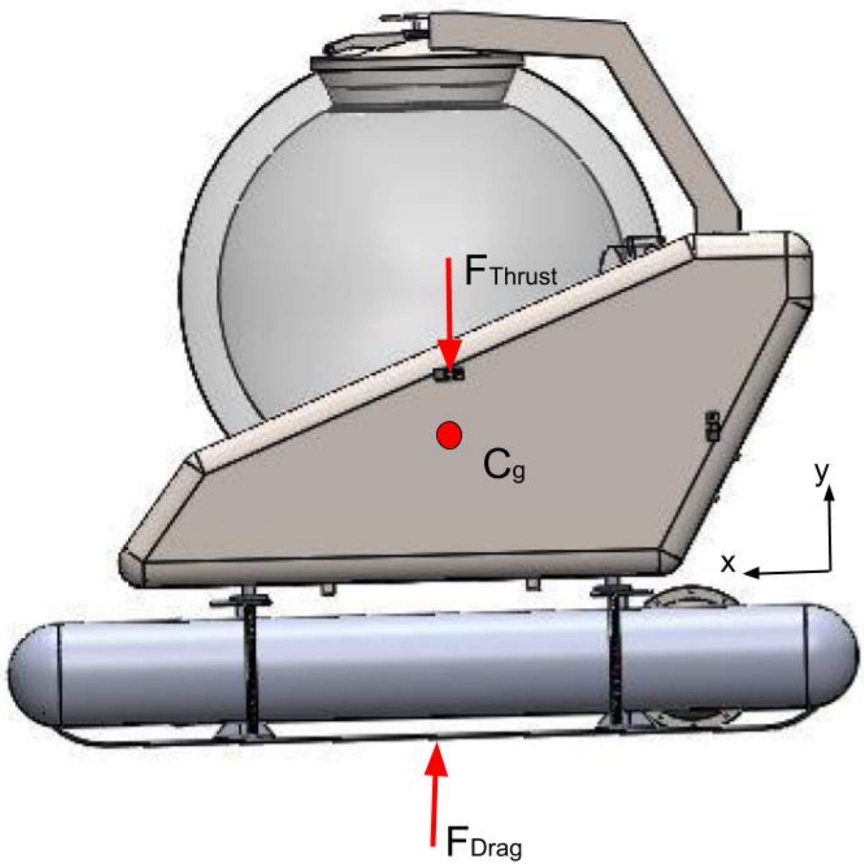

Figure 22: Free Body Diagram of submarine showing external force of drag (Vertical Motion)

$F_d = \frac{1}{2}\rho u^2 C_D A = \frac{1}{2}(1029kg/m^3)(2.06m/s)^2(0.61)(8.05m^2) = 10{,}721.25N$

F_d=*10,721.25*N

This is the force required by the thrusters that support vertical motion to operate at 1m/s

Left/Right turning drag (Yaw)

Need to model for analysis of propulsion (Propellers/Thrusters - 80% efficiency)

Time it takes to get to the surface when drop weight is released with full ballast

Fnet= 67195.89N - 59764.48N=7431.41N (see table 9 and table 10)

To determine the terminal velocity experienced in an emergency situation after the drp weights have been dropped, we apply the equation below:

$$V_{Terminal} = ((2*m*g)/(\rho*A*Cd))^{1/2}$$

m*g is the resultant force due to gravity. We replace this with the resultant of the forces acting on our system as it rises to the surface, after the drop weights have been dropped. Factored in the equation is the weight due to gravity as well as buoyancy and drag.

**Please note that the values for the forces of buoyancy and gravity and how they were determined are stated in the section of Center of gravity and buoyancy. The force due to gravity is without the drop weights in this scenario*

$$m*g = F_{net}$$

$$F_{net} = F_{Buoyancy} - F_{Gravity}$$

Fnet= 67195.89N - 59764.48N=7431.41N (see table 9 and table 10)

F=ma

7431.41N=(6092.20kg)xa

a=1.22 m/s^2

- V represents terminal velocity,
- M is the mass of the falling object,
- g is the difference between acceleration due to gravity and acceleration due to buoyancy

- Cd is the drag coefficient,
- ρ is the density of the fluid through which the object is rising, and
- A is the projected area of the object.

Terminal Velocity

$V_{terminal} = (\frac{2*Fnet}{\rho*A*Cd})^{1/2}$

$V_{terminal} = (\frac{2*7{,}431.41N}{1029kg/m3*8.05m2*0.51})^{1/2}$

$V_{terminal}$ = 1.88m/s

Time to terminal velocity:

$v = v_0 + aT_{terminal}$

$1.88m/s = 0m/s$ +(1.22m/s^2)$t_{terminal}$

$$t_{terminal} = 1.54\, sec$$

Distance covered to get to terminal velocity:

$$v^2 = {v_0}^2 + 2a \cdot \Delta x_{terminal}$$

$$(1.88m/s)^2 = (0m/s)^2 + 2(1.22m/s^{2)} \cdot \Delta x_{terminal}$$

$$\Delta x_{terminal} = 1.45m$$

Time taken for the remaining distance:

$$t_{remainder} = (1000 - \Delta x_{terminal})/Vterminal$$

$$t_{remainder} = 531.14\, sec$$

$$t_{total} = t_{terminal} + t_{remainder} = \mathbf{532.68}\, sec\ =\ \mathbf{8.88}\, min$$

Life Support System Modelling

Life support system needed for the submarine

Values needed for calculation:

Oxygen consumed per person: 26L/hr [2]

Carbon-dioxide produced per person: 22L/hr [2]

Tanks are pressurized and maintained between 0.19 bar and 0.23 bar and aren't affected by hydrostatic pressure

Total submerged time is taken to be 8 hours + 8 hours of emergency support = 16 hours

Oxygen consumed:

26L/hr *16 hours = 416L per person

3 * 416L per person = 1248L for 3 people. Shared among two tanks = 624 tanks to last 8 hours

Mass oxygen is calculated:

PV = nRT

T = 298K (room temperature)

R = 8.314J·K^{-1}·mol^{-1} [3]

V = 624L = 0.64m^3

P = 1.51 *10^7pa (atm)

n = PV/RT

n = (1.51 * 10^7pa *0.64m^3) /(8.314J·K^{-1}·mol^{-1} *298K)

n = 3902.47 mol

Molar Mass of oxygen is 32 G/mol

Mass of oxygen = Molar mass * Number of moles

Mass of Oxygen = 32g/mol * 3902.47 mol

Mass of Oxygen = 124879.37g = 124.88kg.

4 Analysis

4.1 Analysis Outline

- Hull
- Access Mechanism (Hatch)
- Seals
- Frame

4.2 Component Analysis

4.2.1 Hull

Description of Inputs and Outputs

The submarine hull is made of a spherical extruded acrylic, the input is the external force resulting from the hydrostatic force acting on the hull. The output is a hull thickness determined by stress calculations and a safety factor.

Justification for the numerical values of the constants, parameters, safety factors

The outer diameter of the hull will change for the parameterization of a manned submersible between required depths of 300m and 1000m. We are assuming our hull to be a perfect sphere, neglecting the irregularities in geometry occurring at the penetration points.The inner diameter for all submarine models will be 1.7m, while the outer diameter increases based on the thickness required for specific depth ratings.The hull is the most important region of the submarine as it houses the submariners. The inner diameter was selected as the appropriate space for two people with adequate space for storage. The thicknesses of the sphere for this analysis are based on thick shell theory, taking into account the stresses within the shell. The thickness of the hull shell will vary to counter pressure at various depths. The density of seawater is 1029kg/m^3 [4]. We have decided to design our hull with a safety factor of 2 to ensure adequate functionality and safety.

Assumptions, simplifications and Material Selection

- The hull is made of extruded clear acrylic with density of 1180kg/m^3.
- The yield stress of the acrylic is 70 Mpa [5].

- The inner pressure is assumed to be the same as atmospheric pressure (101325 Pa).
- The outer pressure is at a depth of 1000m, P=gh=10,094,490Pa=10MPa.
- The hull is classified as a thick shell.
- All penetrations are filled with a feedthrough and sealed with static and/or dynamic seals.
- The hydrostatic forces acting on the hull cancel out.

Sketch of component and FBD

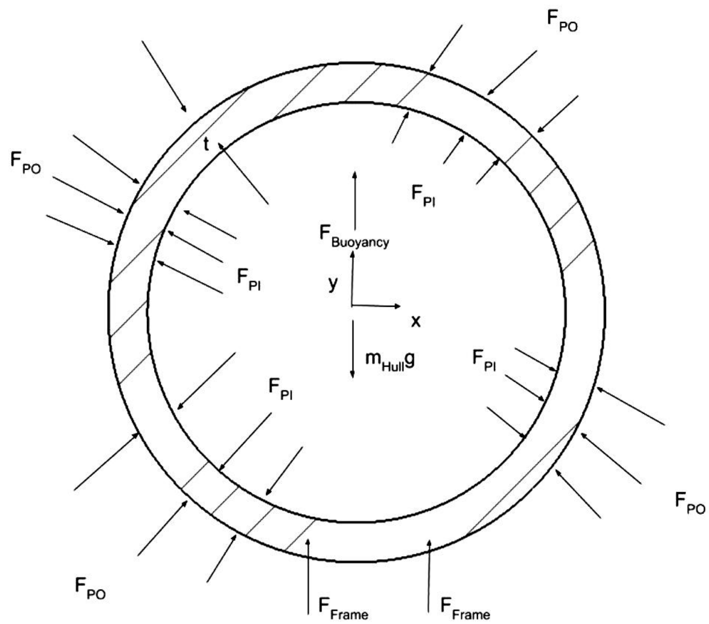

Figure 23: FBD of the hull

Stress Analysis

The hull will fail due to buckling of the shell by pressure at 1000m. The stress acts radially and circumferentially. The stress equations are presented by Jawwad et al [6]

$$\sigma_r = \frac{P_i R_i^3 - P_o R_o^3 - (P_i - P_o)(R_i^3 R_o^3 / r^3)}{R_o^3 - R_i^3}$$

$$\sigma_\theta = \frac{P_i R_i^3 - P_o R_o^3 + (P_i - P_o)(R_i^3 R_o^3 / 2r^3)}{R_o^3 - R_i^3}$$

Where,

$\boldsymbol{\sigma}_r$=radial stress

$\boldsymbol{\sigma}_\theta$=circumferential stress

P_i=Internal pressure

P_o=External pressure

R_i=Inner shell radius

R_o=Outer shell radius

r=radius at any point (we analysed at the shell midpoint)

$$\eta = \frac{S_y}{\sigma'}$$

Where,

$$\eta = safety\ factor$$
$$S_y = yield\ stress$$
$$\sigma' = applied\ stress$$

$$\boldsymbol{\sigma}_r = \frac{(101{,}325 \cdot 0.85^3) - (10{,}094{,}490 \cdot 0.95^3) - (101{,}325 - 10{,}094{,}490)(0.85^3 \cdot 0.95^3 / 0.9^3)}{(0.95^3 - 0.85^3)} = -5{,}651{,}657.76\text{Pa} = -5.65\text{Mpa}$$

$$\sigma_\theta = \frac{(101325 \cdot 0.85^3) - (10{,}094{,}490 \cdot 0.95^3) + (101{,}325 - 10{,}094{,}490)(0.85^3 \cdot 0.95^3 / 2 \cdot 0.9^3)}{(0.95^3 - 0.85^3)} = -50{,}160011.70\text{ Pa} = -50.16\text{Mpa}$$

Above is an analysis of aspherical acrylic shell with a thickness of 10cm, operating at a depth of 1000m. The longitudinal stress is greater than the radial stress. For this analysis, we would be focusing on the longitudinal stresses on the hull. The stresses are negative, signifying a compressive force acting on the hull cross section.

$$\eta = \frac{70MPa}{50.16MPa} = 1.395 \approx 1.4$$

The calculation yields a safety factor of 1.4, hence proving the hull is not of adequate thickness. For a safety factor of 2, the allowable or applied stress acting on the hull cannot be more than 35Mpa.

Table 11: Required Thickness For Different Operating depths

Depth (m)	**Pressure (MPa)**	**Thickness (cm)**	**Midpoint r (m)**	**Longitudinal Stress (MPa)**	**Safety factor**
300	3	4	0.87	-33.08	2.12
500	5	7	0.885	-33.55	2.09
750	7.5	11	0.905	-34.31	2.04
1000	10	16	0.93	-33.96	2.06

Deflection of Hull Shell [6]

The compressive forces acting on the hull cause a shrinkage in the spherical hull volume. This shrinkage is referred to as a deflection.

Deflection, w= $K_1 r + \frac{K_2}{r^2}$ where K_1 and K_2 are constants

K_1=$\frac{1-2\varphi}{E(R_o^3-R_i^3)}(P_i R_i^3 - P_o R_o^3)$

K_2=$\frac{1+\mu}{2E(R_o^3-R_i^3)}(P_i - P_o)(R_i^3 R_o^3)$

In the case of a 1000m modelled submarine,

K_1=$\frac{1-2(0.35)}{3200MPa(1.01^3m-0.85^3m)}(0.101MPa \cdot 0.85^3m - 10MPa \cdot 1.01^3m)$= -0.0023

K_2= $\frac{1+0.35}{2 \cdot 3200MPa(1.01^3m-0.85^3m)}(0.101MPa - 10MPa)(0.85^3m \cdot 1.01^3m) = -0.00317$

w=$(-0.0023 \cdot 0.93m) + (\frac{-0.00317}{0.93^2m}) = -0.00580m = 0.58cm$

Table 12: Deflection Experienced For Various models at rated Depths

Depth	**K_1**	**K_2**	**Midpoint r (m)**	**Deflection w (mm)**
300	-0.00211	-0.00291	0.87	5.68
500	-0.00218	-0.00300	0.885	5.75
750	-0.00227	-0.00313	0.905	5.87
1000	-0.0023	-0.00317	0.93	5.8

Critical Review

For a spherical acrylic hull, the thickness ranges from 4cm to 16cm for a depth range between 300m and 1000m. Increasing the thickness increases the safety factor, while increasing the weight of the hull. The increase in the hull dimensions will increase the frame, hatch and penetrator dimensions. The deflection experienced is similar amongst the analysed hull models, with a mean deflection of 5.8mm. The deflection is inward due to the compression caused by the hydrostatic forces. The deflection causes the shrinking effect experienced by the submarine.

Design Optimization Objectives

The increase in thickness leads to an increase in mass. The hull shell is the area that will be parameterized to select the most appropriate dimensions that provide the lowest weight.

4.2.2 Hatch

Hatch thickness analysis

$$\sigma = \frac{F}{A}$$

Shell: $F = P\pi r^2, A = 2\pi rt$

We can have, $\sigma = \frac{Pr}{2t}, \sigma = \frac{(10.2\times10^6 pa)(0.275m)}{2(0.0255m)} = 5.50 \times 10^7 Pa$

$$\sigma = 55MPa$$

Stainless steel, $\sigma_y = 275MPa$[7]

For safety factor: $N = \frac{\sigma_y}{\sigma}, N = \frac{275Mpa}{55Mpa} = 5$

Hatch Locking and Sealing Analysis

To seal the hatch of our submersible, we will implement the use of o-ring seals mounted to the hatch seat as well as the hatch door by a machined groove. Calculations of the sealing pressure and force generated by the locking mechanism will help identify which specifications an ideal o-ring seal should have.

Locking Mechanism:

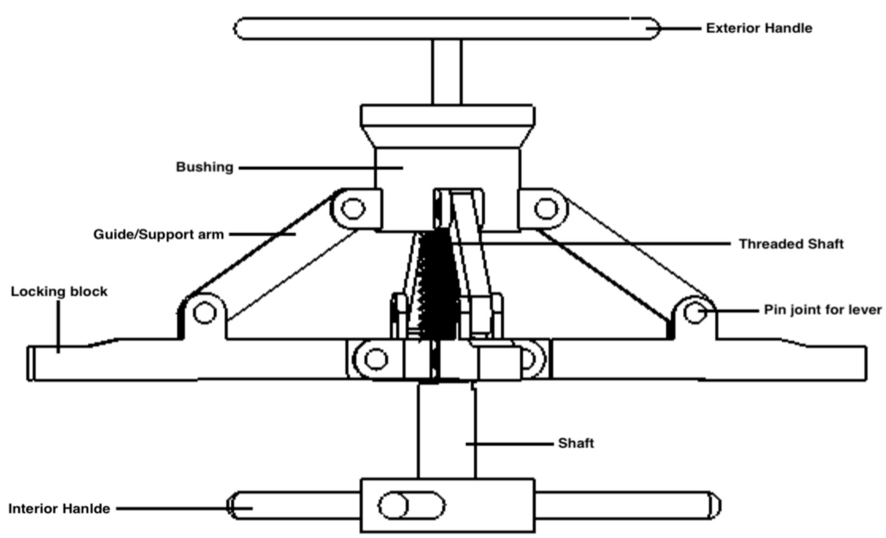

Figure 24: Hatch Locking Mechanism Assembly

Shaft diameter (d) = 40mm = 1.57in

Thread type: UNC 6 Threads per inch

Free Body Diagram:

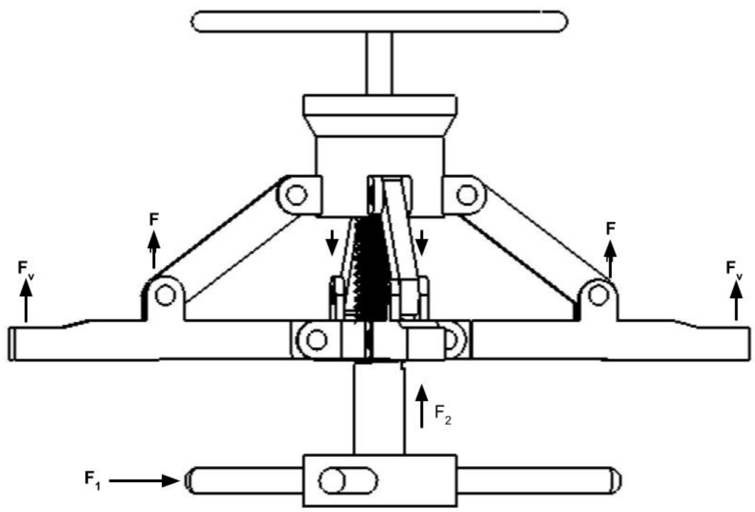

Figure 25: Free Body Diagram of Hatch Locking Mechanism Assembly

T[8] = $(\frac{W*dm}{2})(\frac{f*\pi*dm + L*Cos(\alpha n)}{\pi*dm*cos(\alpha n)-f*L}) + \frac{W*fc*dc}{2}$

F_c = F = 0.16

Pitch = $\frac{1}{N}$= $\frac{1}{6}$= 0.17in

D_m = $d - \frac{p}{2}$= 1.5 - 0.17 = 1.33in

Tan(λ) = $\frac{L}{\pi*dm}$

L = 1*p = 1*(0.17) = 0.17in

Tan(λ) = $\frac{0.17}{\pi*(1.33)}$= 0.03987

λ = 2.28°

α = 30°

Tan(α_n) = Tan(α)*Cos(λ)

α_n = Tan^{-1}(Tan(α)*Cos(λ)) = 29.98°

We need a sealing pressure > 10Mpa in order to properly seal the hatch by compression.

Assumption: Our locking mechanism generates 12Mpa sealing pressure.

Plan: Find the torque/force on the handle in order to generate a 12Mpa sealing pressure.

$P = \frac{Fv}{A}$, $F_v = P*A$

$F_v = (12*10^6 N/m^2)(0.00509 m^2) = 61080N$

$F_v = 61080N$ on all 4 locks

We need the locking system to generate this force in order to have a 12Mpa sealing pressure.

$F_v = \frac{61080N}{4} = 15270N$ on each lock

$F_{v\ real} = 15270N - ((82.17kg)(9.81m/s)) + ((15kg)(9.81m/s)) = 14316.76N$

Using a lever mechanism to translate this force:

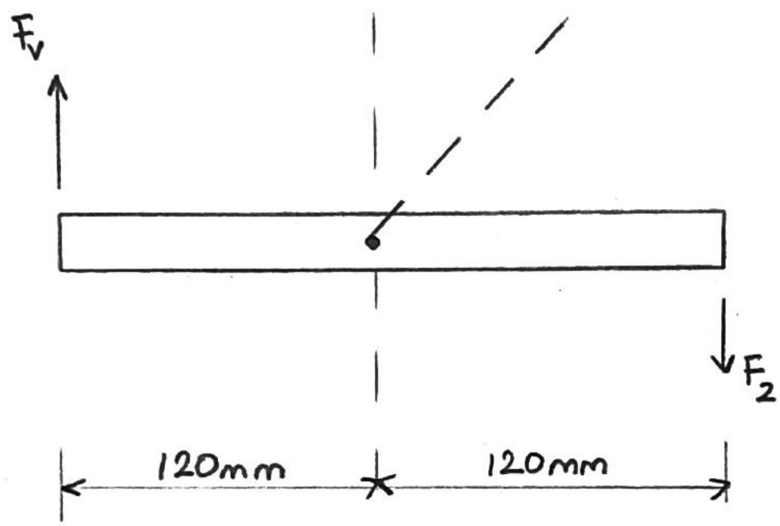

Figure 26: Free Body Diagram of locking arm

$F_{v\ real} * d_1 = F_2 * d_2$

$d_1 = d_2 = 120mm$

$F_2 = \frac{Fvreal * d1}{d2} = \frac{(Fvreal)\ (120mm)}{120mm} = 14316.76N$

$F_2 = W = 3218.53lbs$

$$T[8] = \left(\frac{W*dm}{2}\right)\left(\frac{f*\pi*dm + L*Cos(\alpha n)}{\pi*dm*cos(\alpha n) - f*L}\right) + \frac{W*fc*dc}{2}$$

T = Torque on handle

T = (2145.15)(0.22625) + 451.09 = 936.43 Lb*in

Therefore,

$F_1 = \frac{T}{l}$, *l* = length of handle = 12in

F_1 = 78lb = 346N

A force of 346N on the handle is required in order to generate a sealing pressure of 12Mpa, which is feasible. Hence, a suitable seal will be the silicone rubber O-ring with 30Mpa maximum compressive strength [9].

Hinge spring analysis:

Free body diagram

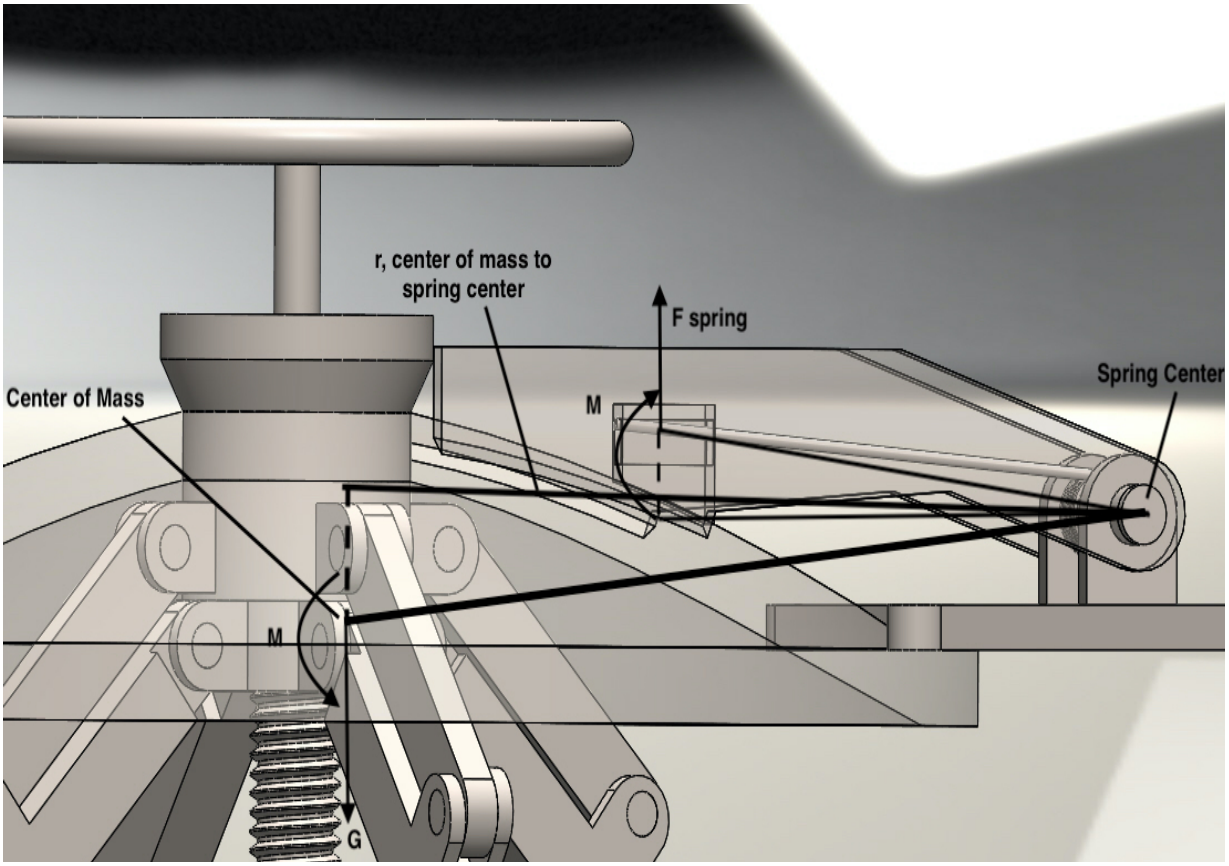

Figure 27:Hatch Reaction forces

Weight of the hatch: $W = (85kg)(9.81m/s^2) = 833.85N$

From hatch mass center to spring center: $r = 0.365m$

We can have the moment: $M = Wr, M = (833.85N)(0.365m) = 304.35Nm = 304350Nmm$

Meaning diameter of the spring: $D = 68mm$

Spring wire diameter:$d = 13mm$

According to the chart (figure 28) :$\frac{D}{d} = \frac{68mm}{13mm}$= $5.23, K = 1.12$

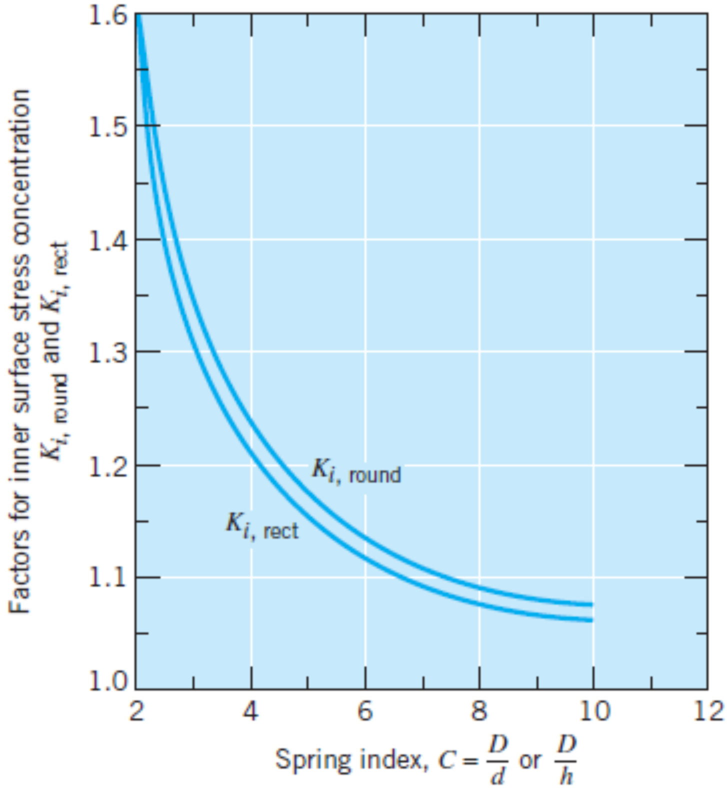

Figure 28: Spring stress concentration vs spring Index [8]

We can have the maximum compressive stress in the coil at the inner surface:

$$\sigma = K\frac{32M}{\pi d^3}, \sigma = (1.12)\frac{32(304350Nmm)}{\pi(13mm)^{3}} = 1580.38MPa$$

Since we have 2 springs, stress on each spring: $\sigma = \frac{1580.38MPa}{2} = 790.19MPa$

We are using ASTM A232 metal, because ASTM A232 contains chromium, which have a good resistance to corrosion, according to the chart:$\sigma = 1350MPa$ at $d = 13mm$

Now safety factor: $N = \frac{\sigma_{ultimate}}{\sigma}, N = \frac{790.19MPa}{1350Mpa} = 1.71$

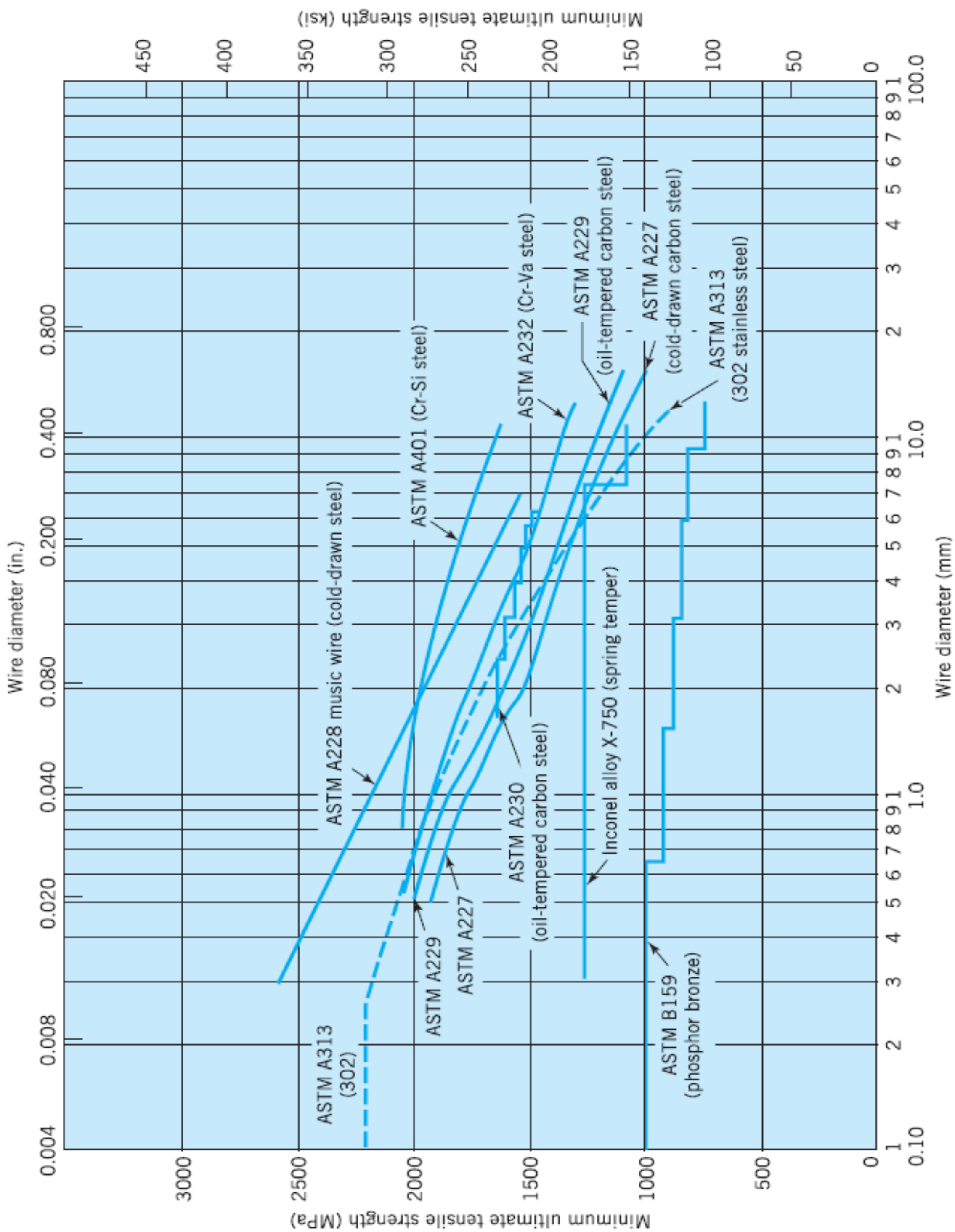

Figure 29: Spring wire DIameter vs Minimum Tensile Strength [8]

4.2.3 Penetration Sealing

Description of Inputs and Outputs

The sealing involves the placement of a seal at regions of openings to prevent the flow of water into the submarine. The inputs for this analysis will be the seal material compression yield factor, the opening diameters, the penetrator dimensions, the seal diameter and flange diameter. The output will be the necessary compressive force as well as the bolt size specification.

Assumptions, Simplifications and Material Selection

- The dynamic o-ring seals are made of encapsulated teflon.
- The penetrator seals are made of silicone rubber with a compressive yield strength of 30MPa[9].
- The o-rings are mounted along the shaft groove.
- Silicone rubber face is compressed against acrylic hull.

Sketch of Components and FBDs

Release Buoy Penetration Sealing

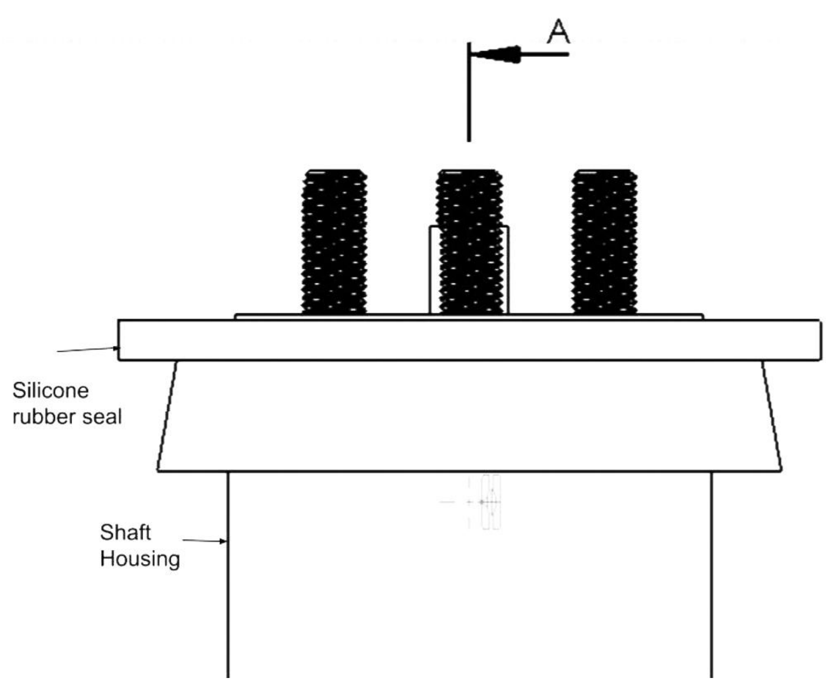

Figure 30: Release Buoy Static Seal

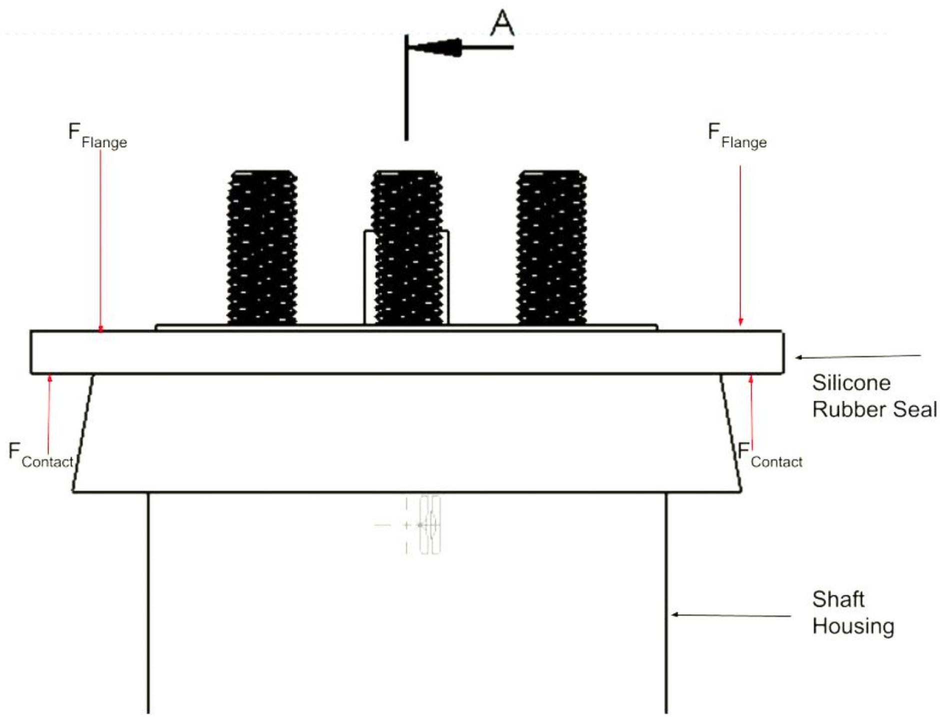

Figure 31: Release Buoy Static Seal reaction forces

<u>Release Buoy Dynamic Seal</u>

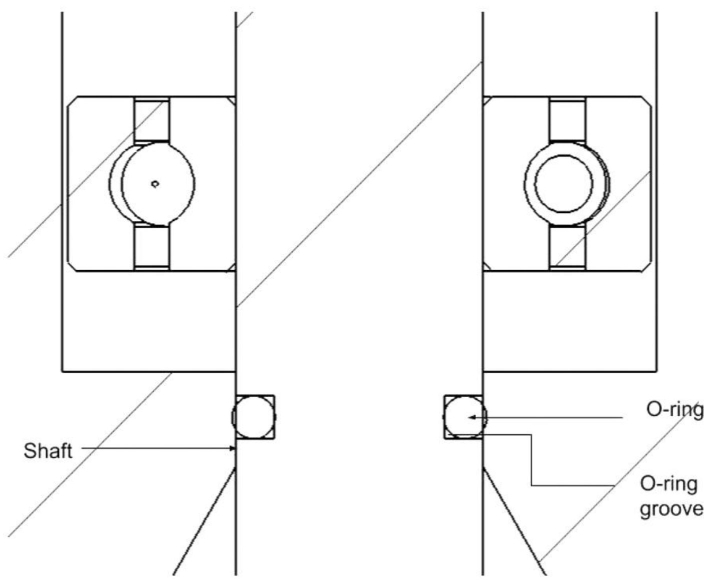

Figure 32: Dynamic sealing of the Release Buoy penetrator shaft

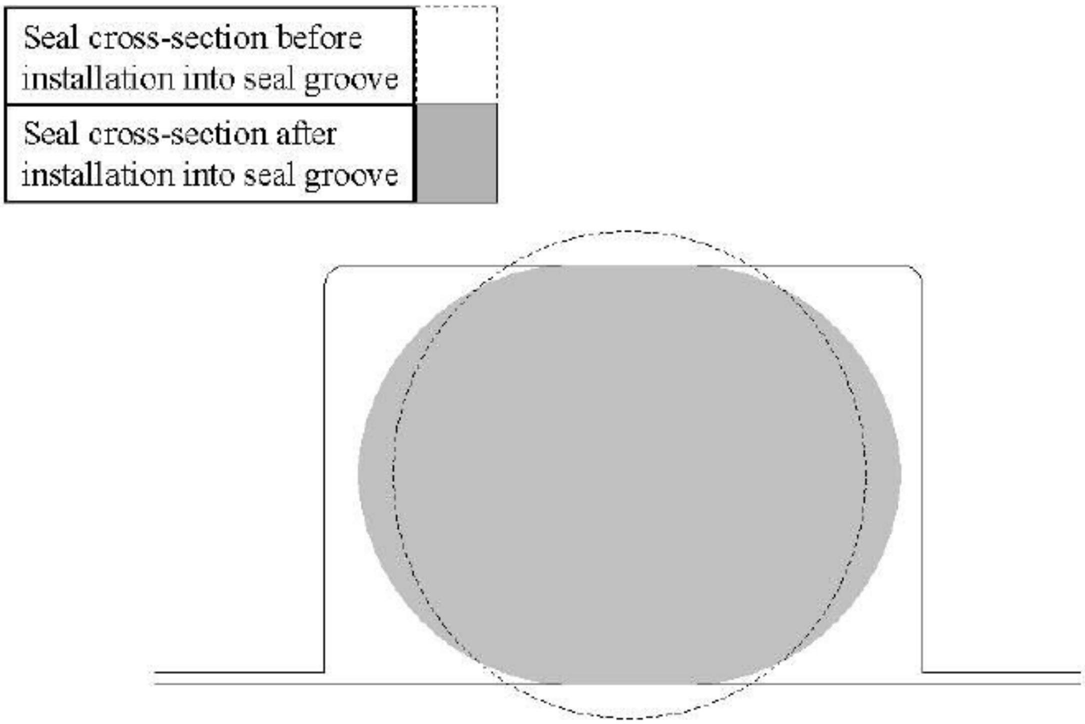

Figure 33: O ring shape deformation due to Compression [10]

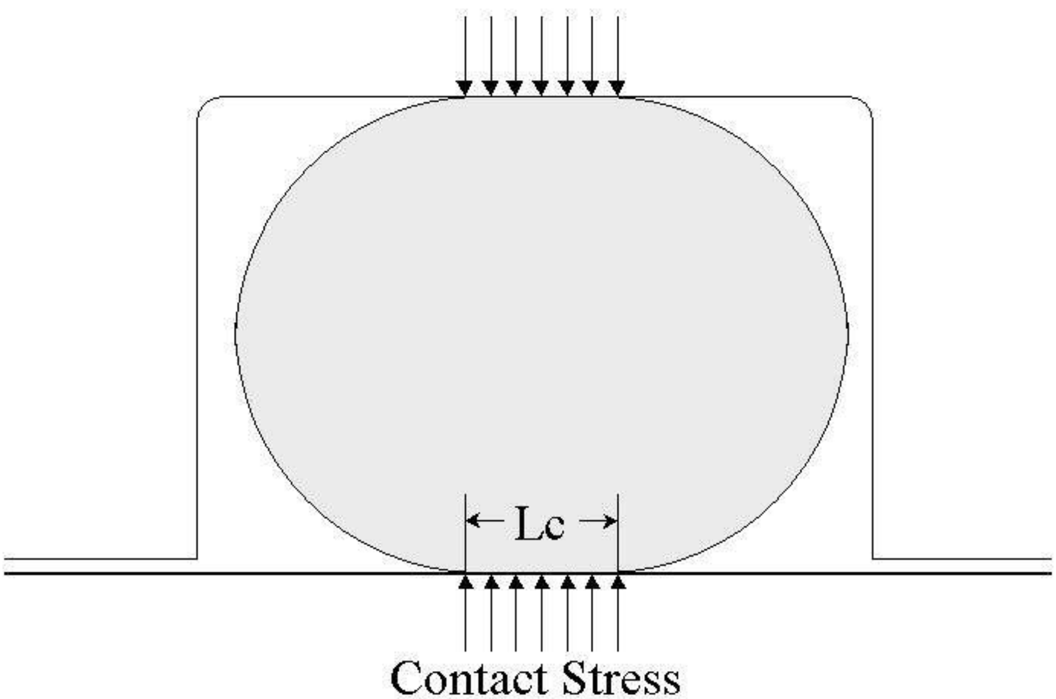

Figure 34:Contact stress on o-ring due to compression [10]

Stress and Failure Analysis

Release Buoy Sealing

Table 13:Release Buoy sealing inputs

Variables	Magnitude	Unit
Compressive Yield (σ_y)	30 [6]	MPa
D_{Flange}	18	cm
Bolting circle D_{bolt}	10	cm
D_{seal}	18	cm
Safety Factor SF	2	-
Applied Stress (σ_a)	15	MPa
Proof Strength of bolt (S_P)	247.5	MPa

Applied strength $\sigma_a = \frac{\sigma y}{2} = 15MPa$

Seal Area (A_S) = Flange Area (A_F) - Bolt Area (A_B)

The flange area representing the area of the flange, while the bolting area represents the bolting circle area along the flange.

$$\sigma_a = \frac{F_a}{A_S}$$

$$15MPa = \frac{F_a}{\left(\frac{\pi \cdot d_{flange}{}^2}{4} - \frac{\pi \cdot d_{bolt}{}^2}{4}\right)} = \frac{F_a}{\left(\frac{\pi \cdot 0.18^2 m}{4} - \frac{\pi \cdot 0.10^2 m}{4}\right)}$$

$F_a = 257{,}250N$ which represents the applied force of the flange on the seal.

Assuming the pressure is constant,

$n \cdot F_{bolt} = \sigma_a \cdot A_b$

(where n=6 is number of bolts)

$$6 \cdot F_{bolt} = 15MPa \cdot 0.0079m^2$$

F_{bolt}= 19,750N

The necessary bolting force per bolt is 19,750N

According to Juvinall [8]

Initial bolting force $F_i=K_iA_tS_p$

K_i- a constant usually from 0.75-1.0. In the case of static loading, approximately 0.9

A_t- Tensile Stress Area of the thread

S_p-Proof strength of the material. Approximately 90% of the material yield strength.

For $F_{i,}$=19,750 N,

19,750N=0.9(A_t)(247.5MPa)

A_t=88.66mm^2 . For a stress area of 88.66mm^{2}as illustrated in table 14, we go to the bolt with the next higher stress area, and that gives us M14 as the required bolt size.

Table 14: Bolt Size as a factor of Thread Stress Area [8]

	Coarse Threads			Fine Threads		
Nominal Diameter d (mm)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)	Pitch p (mm)	Minor Diameter d_r (mm)	Stress Area A_t (mm^2)
3	0.5	2.39	5.03			
3.5	0.6	2.76	6.78			
4	0.7	3.14	8.78			
5	0.8	4.02	14.2			
6	1	4.77	20.1			
7	1	5.77	28.9			
8	1.25	6.47	36.6	1	6.77	39.2
10	1.5	8.16	58.0	1.25	8.47	61.2
12	1.75	9.85	84.3	1.25	10.5	92.1
14	2	11.6	115	1.5	12.2	125
16	2	13.6	157	1.5	14.2	167
18	2.5	14.9	192	1.5	16.2	216
20	2.5	16.9	245	1.5	18.2	272
22	2.5	18.9	303	1.5	20.2	333
24	3	20.3	353	2	21.6	384
27	3	23.3	459	2	24.6	496
30	3.5	25.7	561	2	27.6	621
33	3.5	28.7	694	2	30.6	761
36	4	31.1	817	3	32.3	865
39	4	34.1	976	3	35.3	1030

The analysis for the electrical penetrator and ventilation sealing are similar and can be referenced in the appendix

Dynamic Seal Stress Analysis

Table 15: Dynamic Shaft sealing inputs

Variables	Magnitude	Unit
Shaft Diameter D_{shaft}	1.38	cm
Gland Inner Diameter ID_{Gland}	2	cm

O-ring Inner Diameter ID_{seal}	1.36	cm
O-ring Outer Diameter OD_{seal}	2.06	cm
O-ring width	3.5	mm

$$\%Squeeze\ [10] = \frac{OD_{seal} - \frac{ID_{gland} - OD_{shaft}}{2}}{OD_{seal}} \times 100$$

$$\%Squeeze = \frac{2.06cm - \frac{2cm - 1.38cm}{2}}{2.06cm} \times 100 = 84.95\%$$

O-ring strain due to elongation around shaft

$$\Delta = Shaft\ Diameter - ID_{seal} = 1.38cm - 1.36cm = 0.02cm$$

$$\varepsilon = \frac{\Delta}{ID_{seal}} = \frac{0.02cm}{1.36cm} = 0.014$$

$$\sigma = E \times \varepsilon = 6MPa \times 0.014 = 0.0.84MPa$$

The compressive stresses depend on the contact width and the pressure affecting the cylinder. For an external pressure of 10MPa, the seal is going to be pressed against the gland walls.

Critical Review

The silicone rubber seal is mounted onto the housing penetrating the hull. The flange presses against the seal, causing the seal to rest against the hull, causing a compression. The seal has a compressive yield strength of 30MPa, keeping a safety factor of 2 because of the relationship with the hull gives an applied force of 15MPa that we use to model the bolt size needed. The initial bolting force is a factor of the bolt size and material of the bolt. For all the bolts in the analysis, we have decided to use stainless steel bolts because of the corrosive environment. The dynamic seal is mounted on the shaft and the compression of the seal is a result of the axial force of the gland wall and the pressure in the cylinder.

Design Optimization Objectives

The optimization objective for the sealing of the hull, would be to simplify the design.

Simplifying the design could involve riveting the seal onto the flange plate.

4.2.4 Frame

Bending Stress on Side Frames (When lifting out of water):

For Complete list of FBD See appendix B

Given: [Fs] and [A]; Find: σ_b and η_b

Assumptions, simplifications and Material Selection

- The skeletal frame skeletal is made of Stainless Steel 416
- The yield stress of the Stainless Steel 416 is 275MPa [7].
- The outer pressure is at a depth of 1000m, P=gh=10,094,490Pa=10MPa.
- The frame is classified as a uniform Cross sectional member with no defects.
- The hydrostatic forces acting on the frame cancel out.
-

We are assuming that the resultant of bending stress on the x axis, σ_x, and on the z axis, σ_z, is

$\sigma_b = \sqrt{\sigma_x{}^2 + \sigma_z{}^2}$

The Frame is made out of Stainless Steel 416

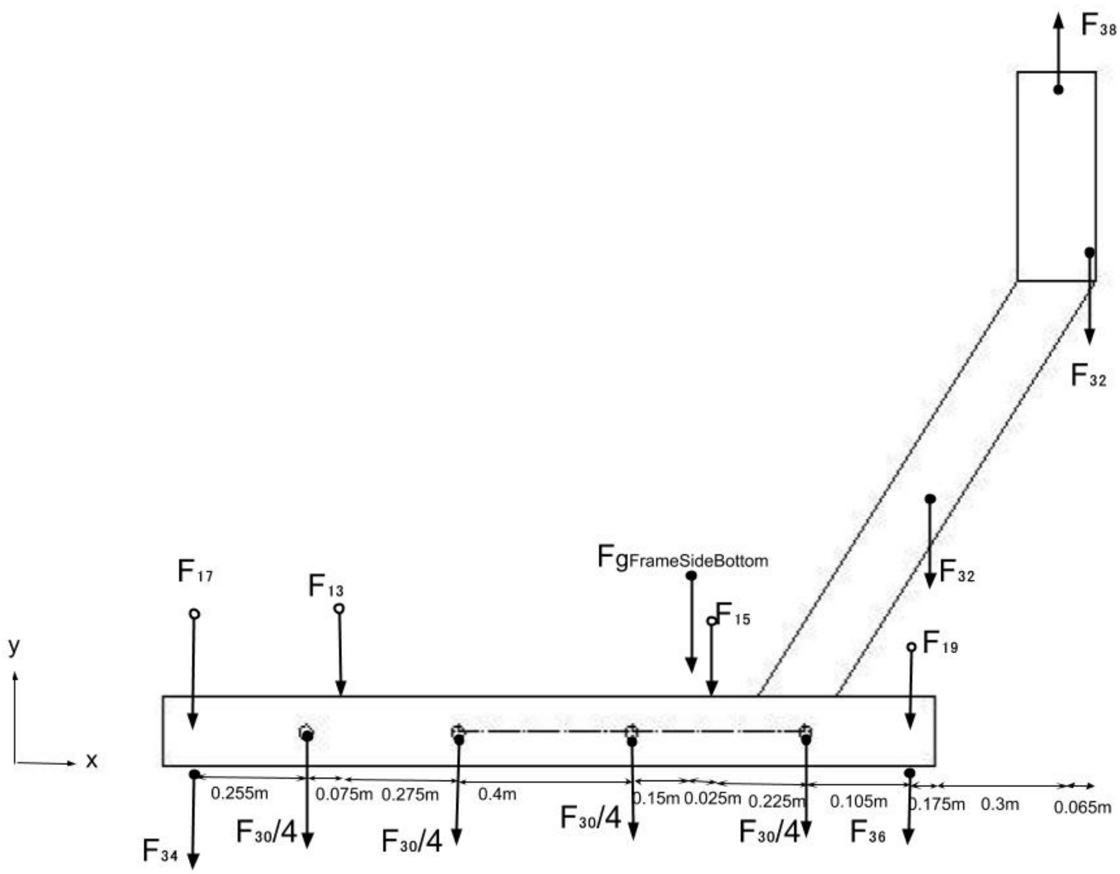

Figure 35: Forces acting Bottom Side Frame (Side View)

Shear force, V=F (Cumulative)

Ex, $V_{0.255->0.33} = (F_{17} + F_{34}) + (F_{30}) = (-355.809 + (-3932.44)) + (-1222.6) = 5510.85\ N$

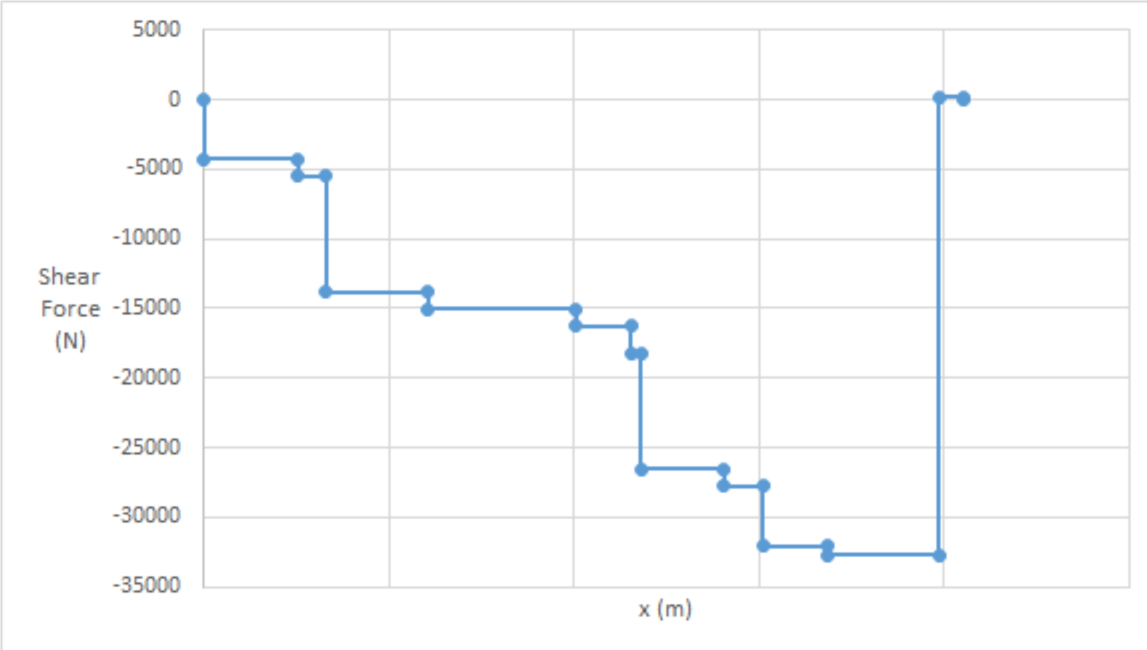

Figure 36: Bottom Side Frame (Side View) Shear Force Diagram

Bending Moment, $M = F \cdot distance$(Cumulative)

Ex, $M_{0.33} = (V_{0->0.255}\ \ ) \cdot x_1 + (V_{0.255-.0.33}) \cdot x_2 = (-4288.25) \cdot 0.255 + (-5510.85) \cdot 0.075 = 1506.82\ Nm$

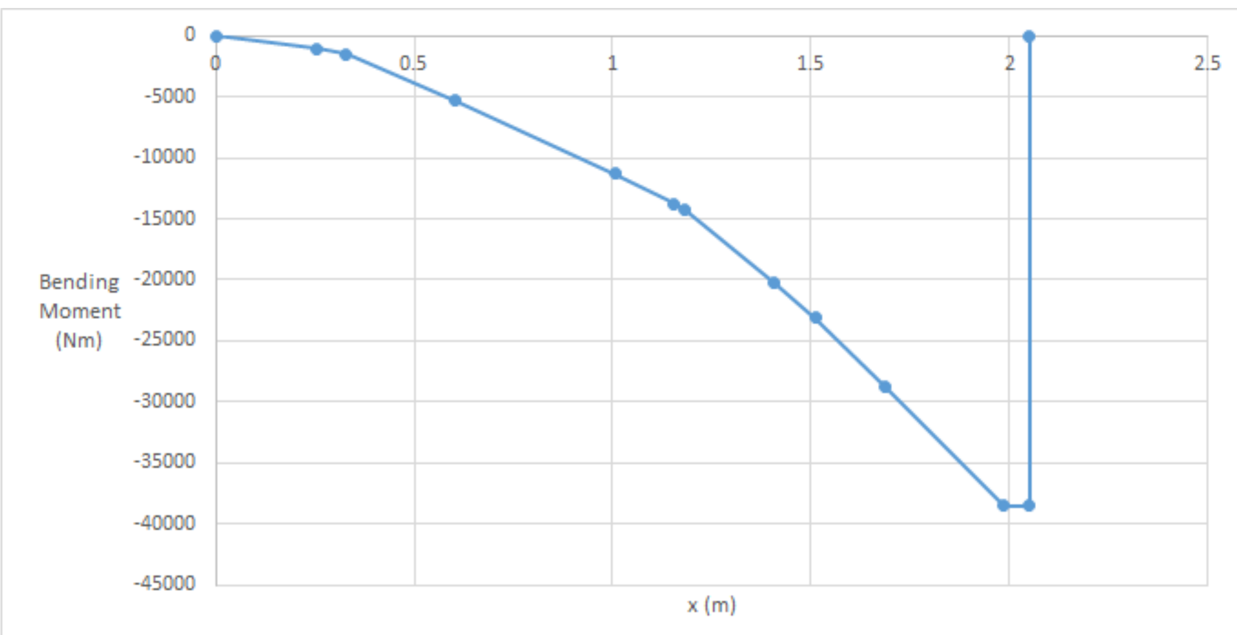

Figure 37: Bottom Side Frame (Side View) Bending Moment Diagram

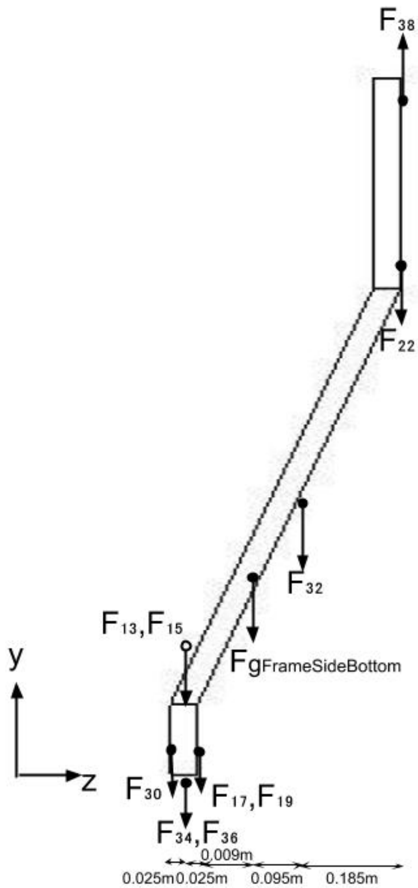

Figure 38: Forces acting Bottom Side Frame (Front View)

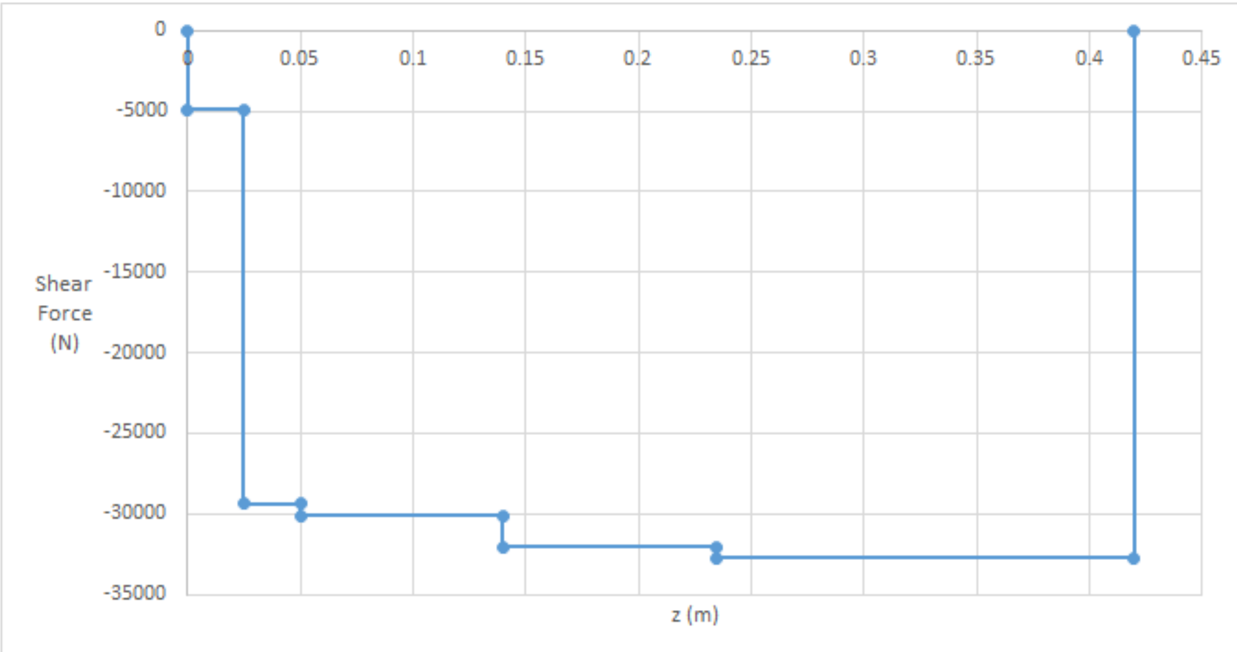

Figure 39: Bottom Side Frame (Front View) Shear Force Diagram

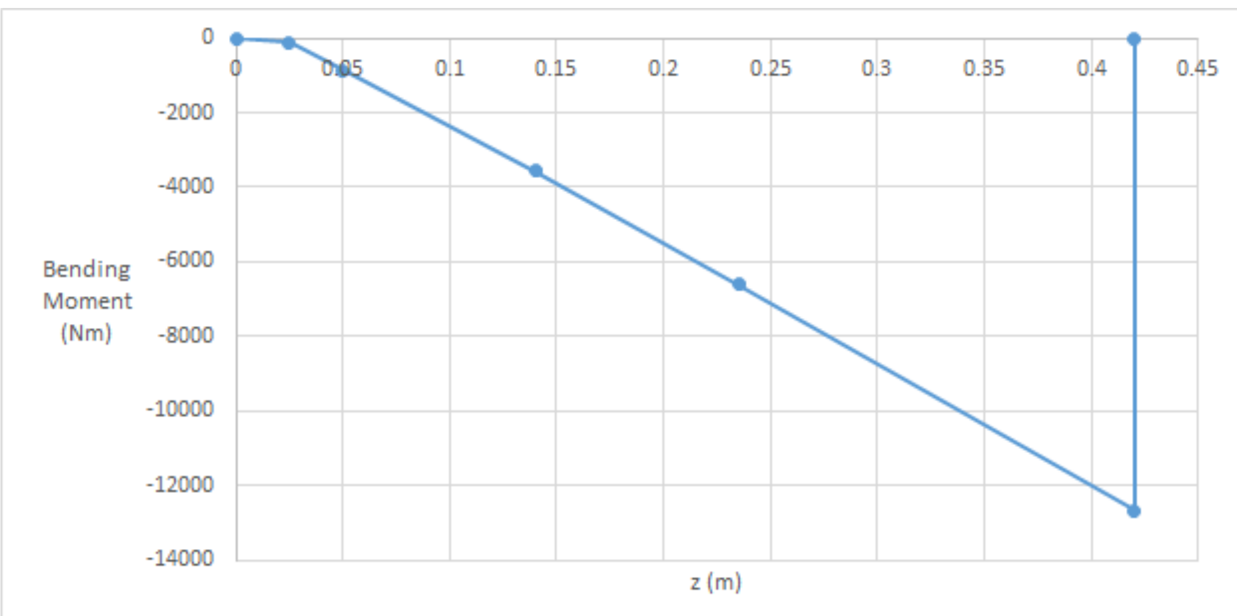

Figure 40: Bottom Side Frame (Front View) Bending Moment Diagram

Safety factor for bottom side frame:

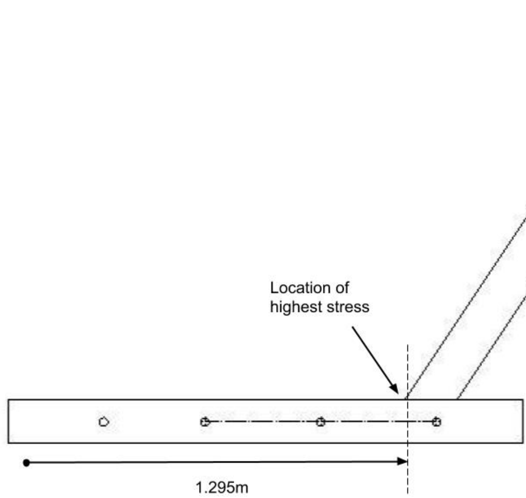

Figure 41: Maximum Stress Location

Largest bending stress occurs at x=1.295m. At this location the bending moment, $M_x = 17{,}284.07\ Nm$

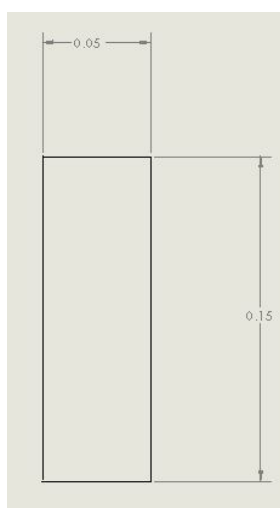

Figure 42: Frame Cross-section

$$y = \frac{h}{2} = \frac{0.15}{2} = 0.075\ m$$

$$I = \frac{bh^3}{12} = \frac{0.05 \cdot 0.15^3}{12} = 1.41x10^{-5}\ m^4$$

$$\sigma_x = \frac{My}{I} = \frac{(17{,}284.07)(0.075)}{1.41x10^{-5}} = 92.18x10^6\ Pa = \quad 92.18\ MPa$$

$\sigma_y$is very small and is negligible. $\sigma_b = \sigma x$.

$$SF(safety\ factor) = \frac{\sigma_y}{\sigma_b} = \frac{275}{92.18} = 2.98$$

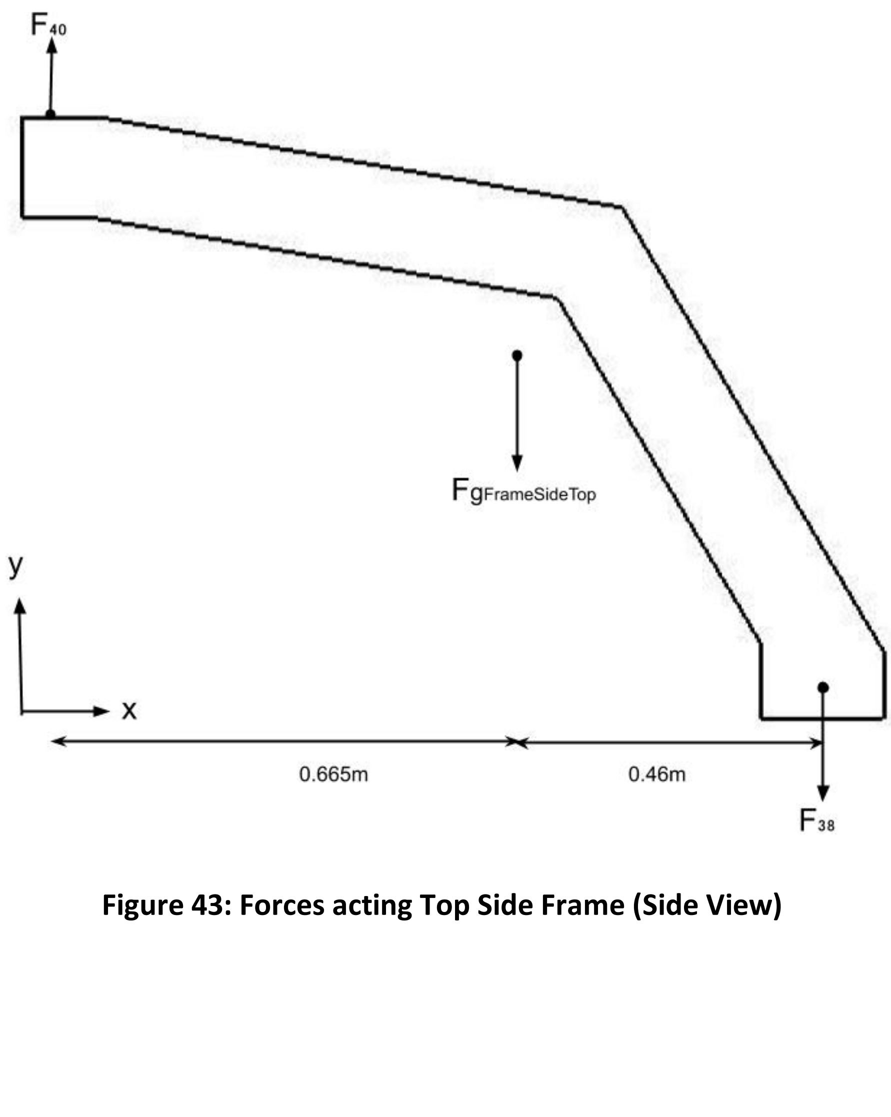

Figure 43: Forces acting Top Side Frame (Side View)

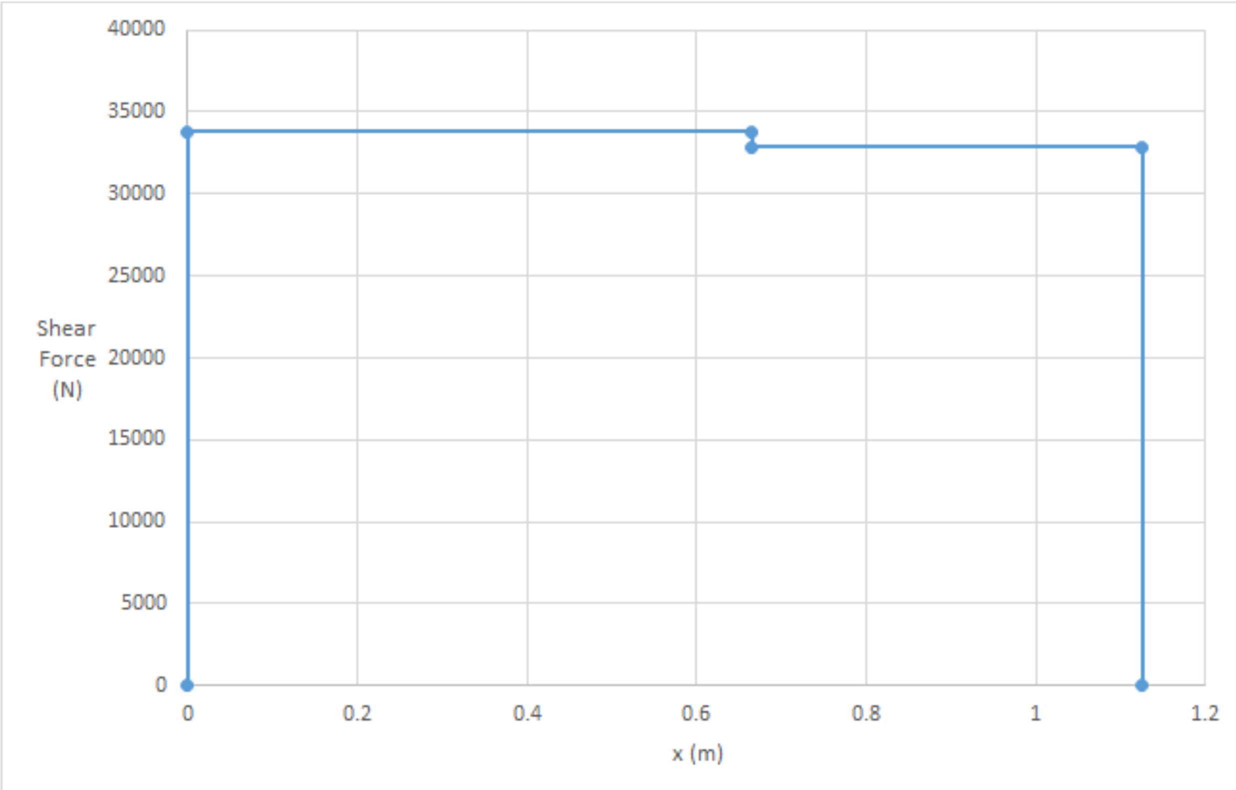

Figure 44: Top Side Frame (Side View) Shear Force Diagram

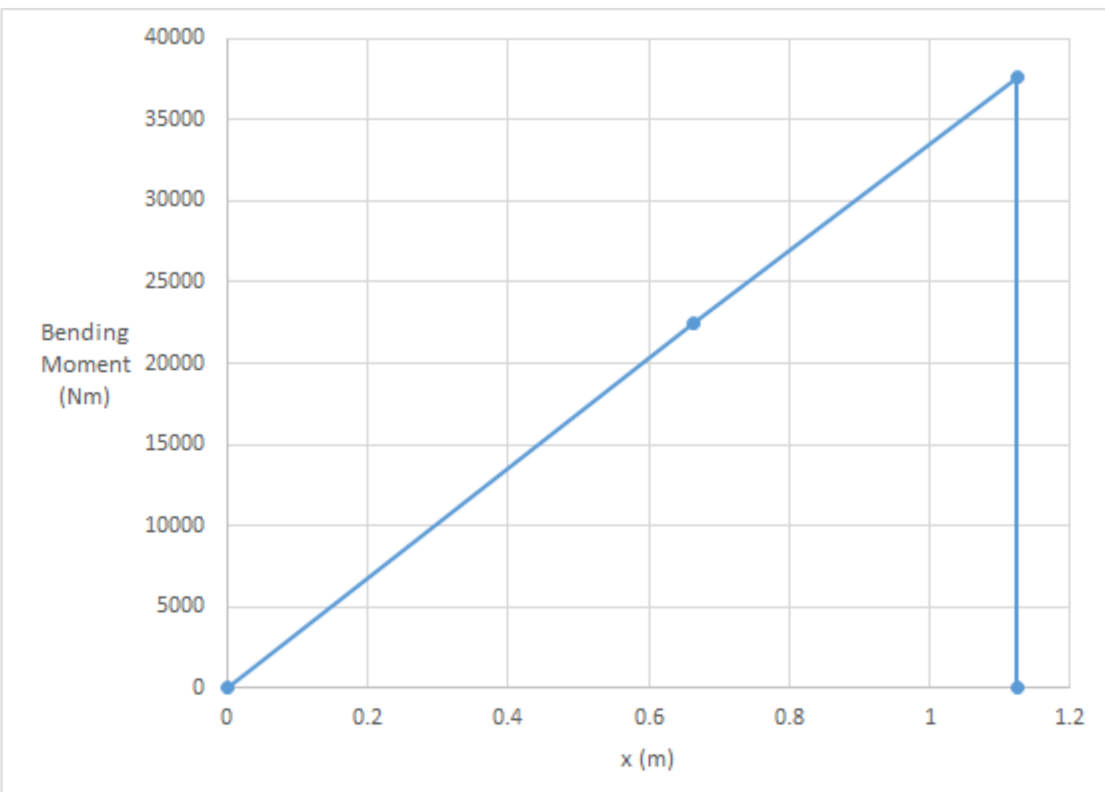

Figure 45: Top Side Frame (Side View) Bending Diagram

Safety factor for top side frame:

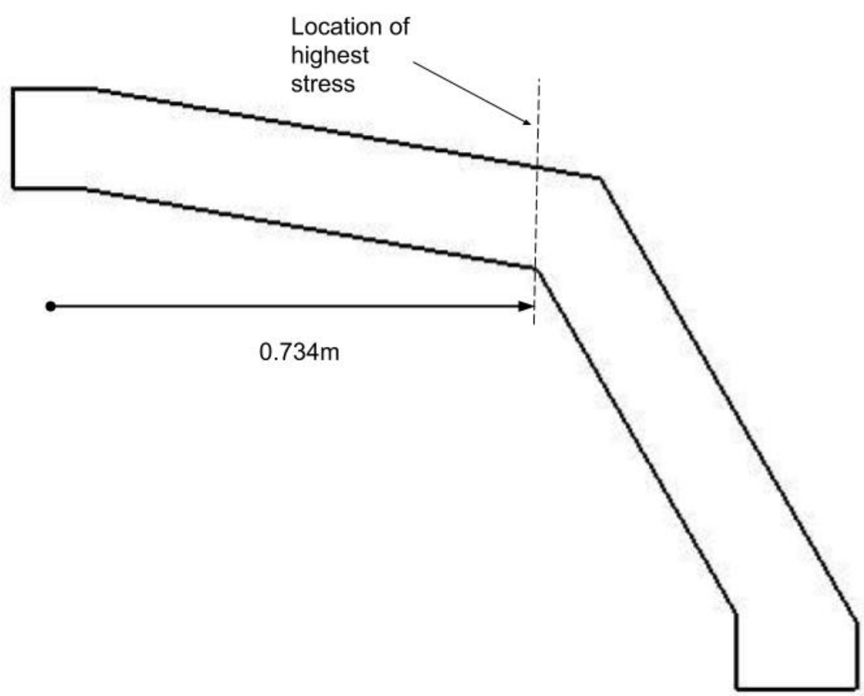

Figure 46: Maximum Stress Location on Side Frame (Top Region)

Largest bending stress occurs at x=0.734m. At this location the bending moment, $M_x = 24{,}755.28\ Nm$ The Frame cross section is constant (figure 42)

$$y = \frac{h}{2} = \frac{0.15}{2} = 0.075\ m$$

$$I = \frac{bh^3}{12} = \frac{0.05 \cdot 0.15^3}{12} = 1.41x10^{-5}\ m^4$$

$$\sigma_x = \frac{My}{I} = \frac{(24{,}755.28)(0.075)}{1.41x10^{-5}} = 132.02x10^6\ Pa = \quad 132.02\ MPa$$

$\sigma_y$is very small and is negligible. $\sigma_b = \sigma x$.

$$SF(safety\ factor) = \frac{\sigma_y}{\sigma_b} = \frac{275}{132.02} = 2.08$$

<u>Critical Review</u>

The results of the analysis make sense. The safety factor is sufficient enough for the frame.

<u>Design Optimization Objectives</u>

A change in the diameter and weight of the hull will change the dimensions and cross section of the frame.

Attachment points for lifting

<u>Given:</u> [F]; <u>Find</u>:L

There are two attachment points for lifting and they are welded to the metal ladder frame structure to properly distribute the lifting force to the bottom of the submarine for lifting
The length and dimensions of the ears needed to avoid shear-out failure has to be evaluated

$$F = S_{Sy}A/SF$$

Where, F is the static load capacity

According to distortion energy theory,

S_{Sy}= 0.58S_y [8]

S_y = 275MPa[7]

S_{Sy} =159.5MPa

S_y = Material yield stress

S_{Sy} = Shear yield strength

F= 67744.66 N (See Table 8, With Ballast (w/ drop-weight))

A = t*L

Where, A is the weld throat area

t = throat length

t= 0.707h

L= length of weld

h= weld height

h=3mm

For a safety factor of 2,

The necessary lifting point area A = $\frac{F*SF}{Ssy}$

A =$\frac{67744.62N * 2}{159.5MPa}$

A = $\frac{44668.56N * 2}{159.5MPa}$

A = 849.46mm^2

A = $0.707 * 3mm * L$

L= $849.46mm^2 \div (2.121mm) = 400.5mm$

There are two lifting points, length of weld will be 200.25mm. We then arrive at a rectangular lifting point from the dimensions.

Frame to Fairing Cover Bolting Analysis

Given: [F] and [d,t,n]; Find: η

Assumptions, simplifications, and materials

- Equal distribution of weight between all the bolts
- Steel frame and Fairing, both being Stainless steel 416 with yield stress of 275Mpa, M15 Bolts

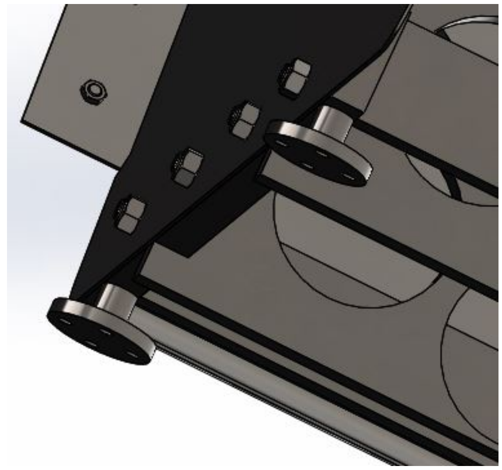

Figure 47: Ballast Chassis Connection Point

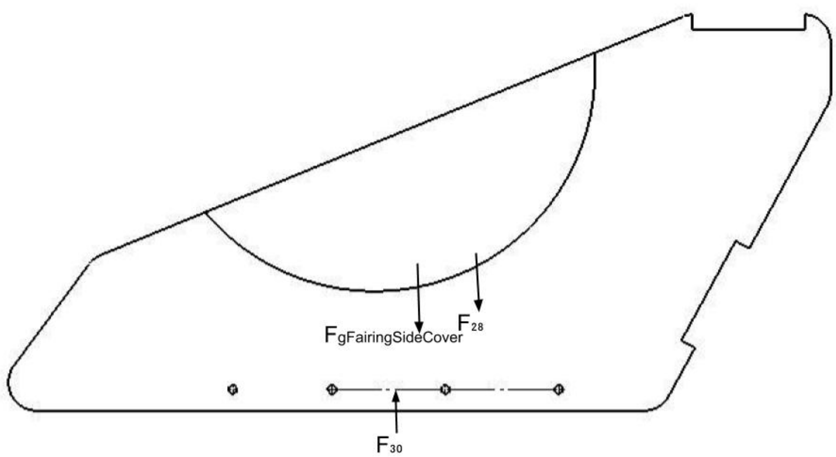

Figure 48: Fairing Reaction Forces

Most likely failure is the yielding of the bolts

M = $\frac{(F/n)t}{2}$ (n=# of bolts=4)

M =($\frac{4890.38}{4}$*0.05) /2

M = 30.56N.m

$$\sigma = \frac{Mc}{I}$$

$\sigma = (M \cdot \frac{d}{2})/(\frac{\pi}{64} d^4))$

$\sigma = (30.56Nm \ \cdot \frac{0.015m}{2})/(\frac{\pi}{64} (0.015m)^4))$

σ = 92.25 x 10^6 Pa = 92.25Mpa

S_y = 275MPa

$\eta = \frac{Sy}{\sigma} = \frac{275Mpa}{92.25Mpa}$ = 2.98

Bottom Side Frame to Top Side Frame Bolting Analysis

Given: [F] and [d,t,n]; Find: η

Assumptions, simplifications, and materials

- Material for frame is Stainless steel 416 with yield stress 275Mpa. 2 Bolts per connection
- Equal distribution of weight in the bolts

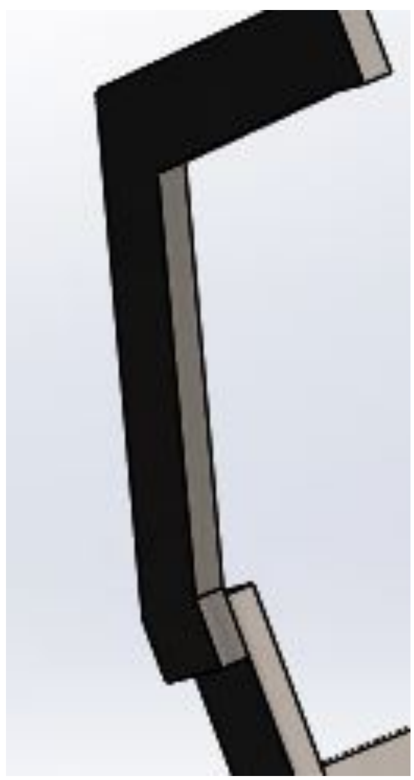

Figure 49: Frame to Frame connection

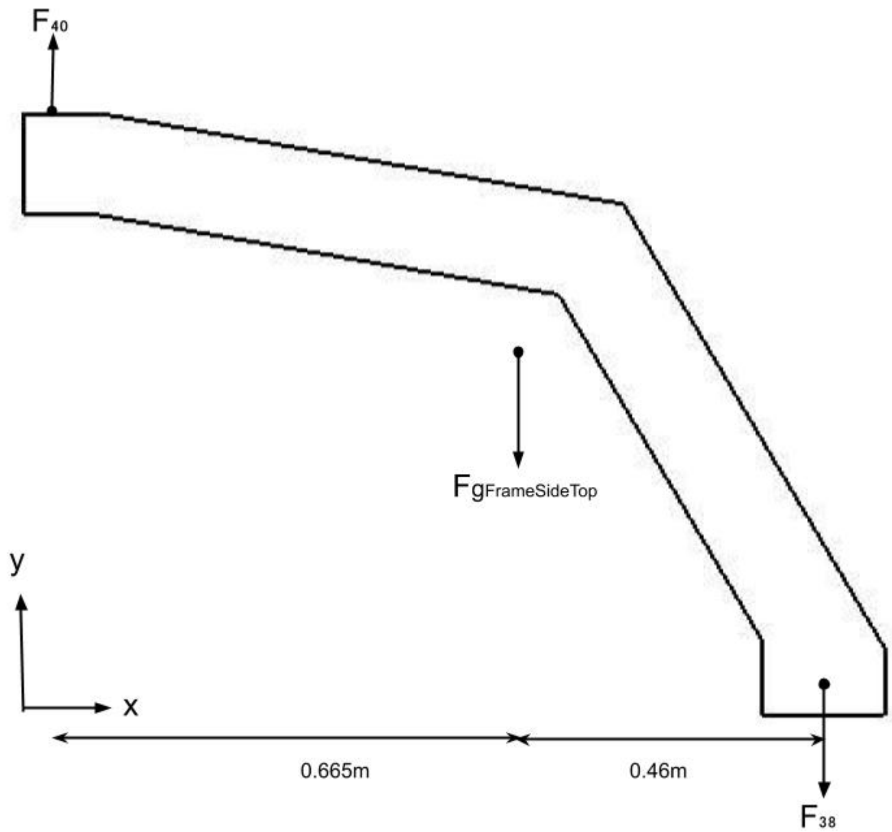

Figure 50: Frame Reaction Forces

M = $\frac{(F/n)t}{2}$ (n=# of bolts=2)

M =($\frac{32,866.42}{2}$*0.05) /2

M = 410.83N.m

$$\sigma = \frac{Mc}{I}$$

$\sigma = (M \cdot \frac{d}{2})/(\frac{\pi}{64}d^4))$

$\sigma = (410.83Nm \ \cdot \frac{0.033m}{2})/(\frac{\pi}{64}(0.033m)^4))$

σ = 116.449 x 10^6 Pa = 116.449 Mpa

Stress = 116.449 * 10^6 Pa = 116.45Mpa

S_y = 275MPa

η= $\frac{Sy}{stress}$= $\frac{275Mpa}{116.45Mpa}$= 2.36

Side Fairings to back Fairning Bolting Analysis

Given: [F] and [d,n]; Find: η

We then analyze the bolts that join the fairings on the side to the one on the back. The fairing on the back is supported solely by the two fairings on the side as it's weight pulls down on the two fairings through the bolt. We also analyze for people standing on the fairing to ensure it doesn't fail.

Assumptions, simplifications and materials

- Weight of the back fairing acts down and doesn't deflect
- Stainless sheet metal steel
- M30 bolt with yield stress 210Mpa

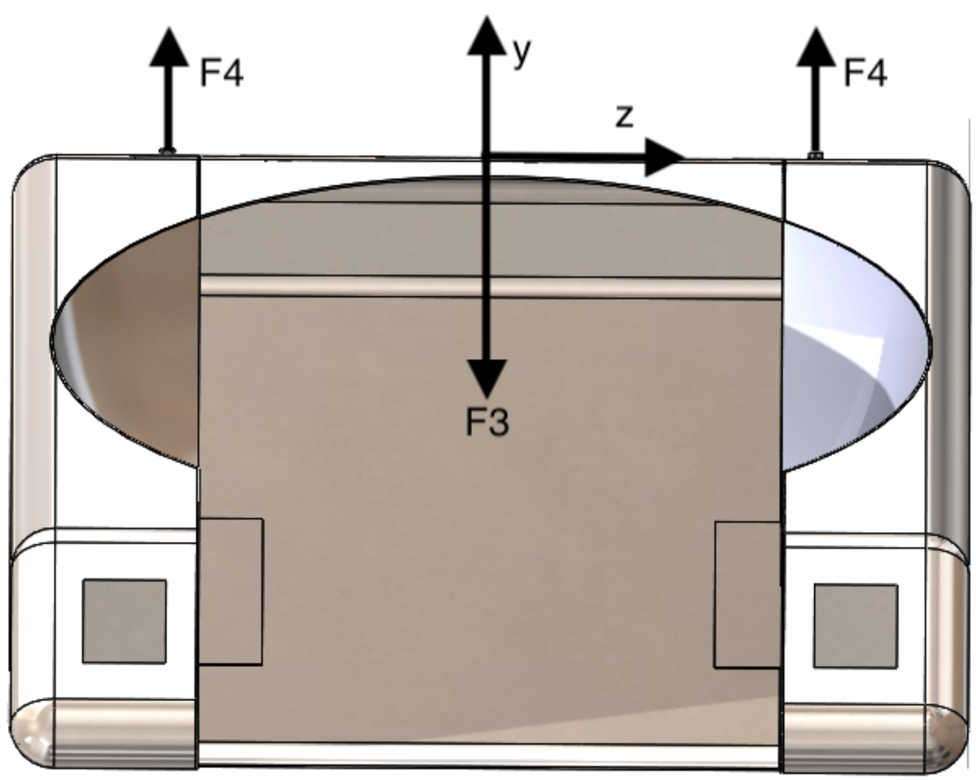

Figure 51: Front view of the fairing assembly showing the different force interactions

F3 = weight of the back fairing + 3 people on the fairing = (136.50kg + 3people*80kg)*9.81m/s

F3 = 3,693.47N

F3 = 2*F4

F4 = F3/2

F4 = 3,693.47N/2

F4 = 1,846.74N

We then proceed to calculate the tensile strength on the bolts.

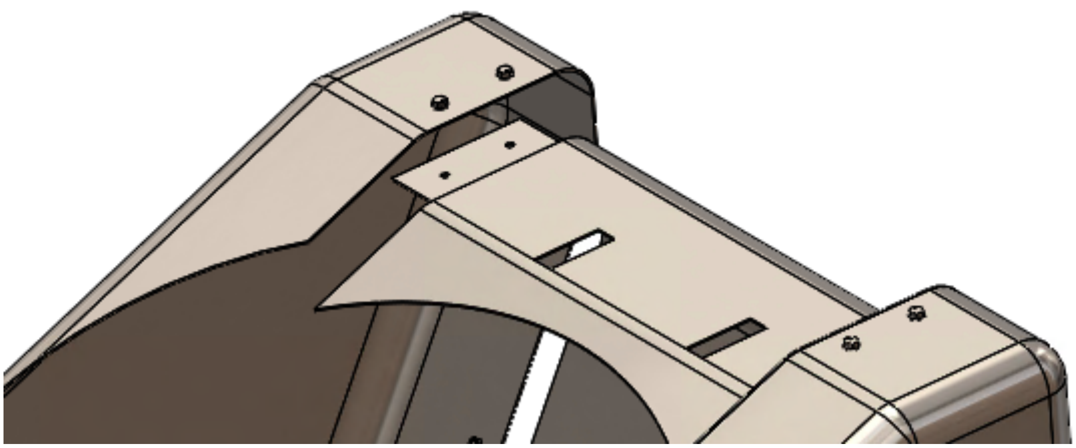

Figure 52: Back fairing fitting into the back fairing

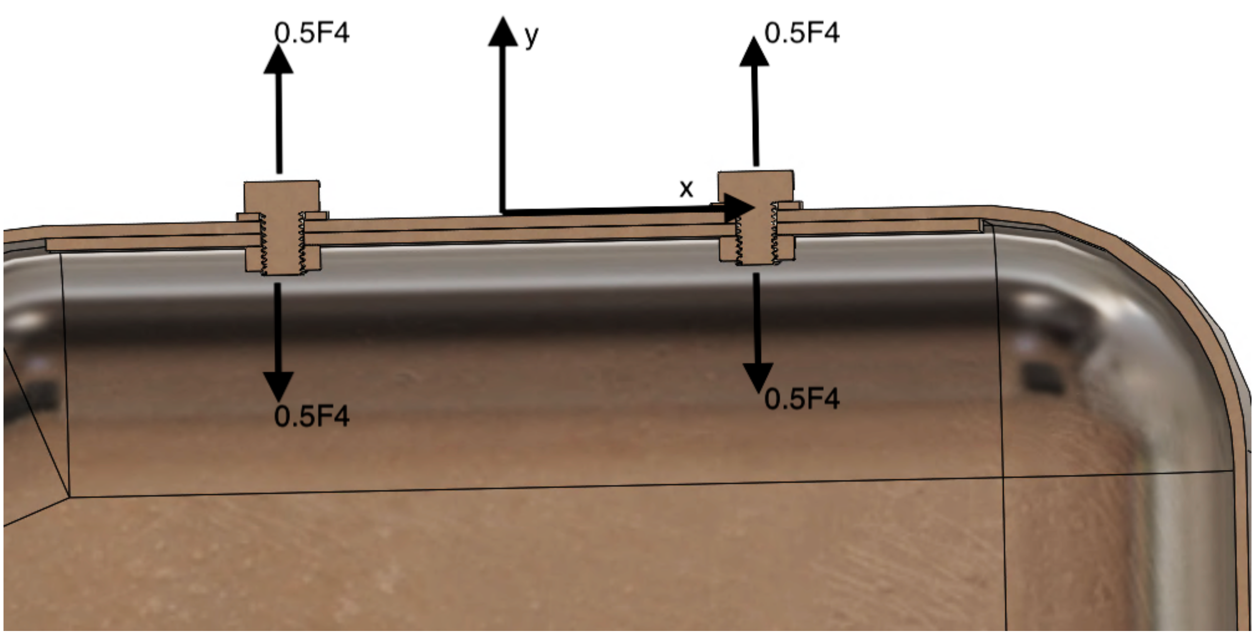

Figure 53: Free-body-Diagram of the forces of the fairing acting on the bolt

F4 = 1,846.74N

Tension force in each bolt is 0.5F4

0.5F4 = 0.5*1,846.74N = 923.37N

Stress = F/A

WIth the chosen M30 bolt, we determine stress using

A = $\boldsymbol{\pi}D^2/4 = \boldsymbol{\pi}*30^2/4$

A = 706.86mm^2

Stress = F/A = 923.37N / 706.86mm^2

Stress = 1.31Mpa.

When a moment is considered i.e. moment for force acting at the center of the fairing

M = $(Force\ *\ Distance)\ /n$ (n=# of bolts=4)

F3 = 3,693.47N

Length in the z direction between the two sets of bolt across the back fairing is 1.36m, therefore perpendicular distance of force to action point is 1.36m/2 = 0.68m

M =($\frac{3693.47}{4}$*0.68)

M = 627.89N.m

$$\sigma = \frac{Mc}{I}$$

$\sigma = (M \cdot \frac{d}{2})/(\frac{\pi}{64}d^4)$)

$\sigma = (627.89Nm\ \cdot \frac{0.030m}{2})/(\frac{\pi}{64}(0.030m)^4)$)

σ = 236.88 x 10^6 Pa = 236.88Mpa

S_y = 275MPa

n= $\frac{Sy}{\sigma}$= $\frac{275Mpa}{236.88Mpa}$= 1.16

Critical Review: The analysis on this bolt results in a factor of safety of n = $\frac{275Mpa}{236.88Mpa}$= 1.16

With this being too small, we will need to get more bolts with to achieve more realistic values and to increase overall safety. This is a worst case scenario, if three people were to be standing directly at the center of the back fairing.

Shrinkage

Given: [F] and [L_o,T1,T2]; Find: ΔL

$\Delta L = \boldsymbol{\alpha} L_o \Delta T$

α[7] = Linear thermal expansion coefficient = 9.9 * 10^6 /^{0}C

An initial room temperature of 24 degrees celsius is used as well as 6 degrees celsius [11] of deep sea salt water at 1000m.

$\Delta T = T1\text{-}T2 = 24\ ^0C - 6\ ^0C = 18\ ^0C$

We then analyze the different shrinkages at different sections of the frame.

* Please note that the frame support beams have uniform cross section, allowing for some assumptions such as uniform to be made

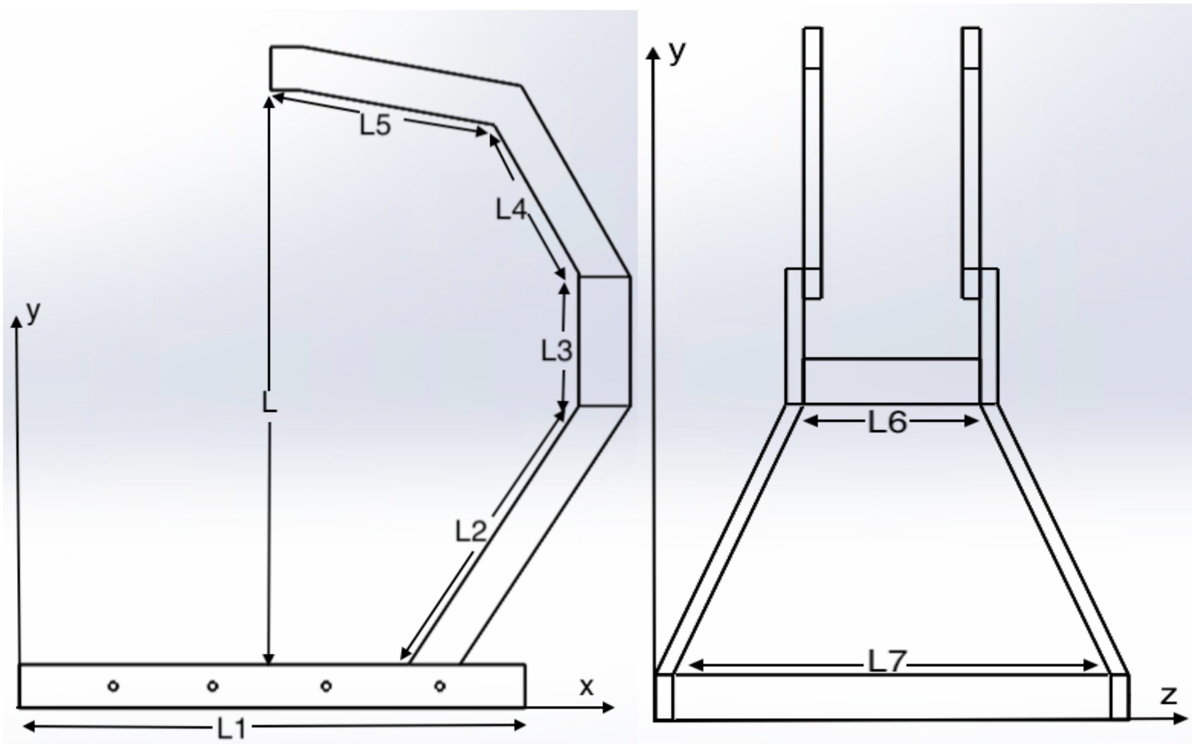

Figure 54: Side and Back view of the frame assembly, indicating the different members

For L1

$\Delta L = \boldsymbol{\alpha} L_1 \Delta T$

$\Delta L = 9.9 * 10^6 /^0 C * 1.78m * 18\,^0 C$

$\Delta L = 0.00032m = 0.32mm$

Likewise

Table 16: Frame Component Lengths

Part	Original Length (m)	Change in length (mm)
L1	1.78	0.312
L2	1.41	0.25
L3	0.45	0.1
L4	0.87	0.2

L5	0.89	0.16
L6	1.24	0.22
L7	0.5	0.09

At an initial temperature of 24 ^{0}C we have an initial L of 2.04m, and after shrinkage, we arrive Have a total vertical change in length of 0.38mm.

The hull support on the frame

Given: [F] and $[L, d, w, \angle]$; Find: η

Assumptions, simplifications and materials

- The weight of the hull acts perpendicular to hull support as intended and is distributed equally along the facing area
- Stainless steel and acrylic hull

Analysis to test the stress on the hull from the metal support

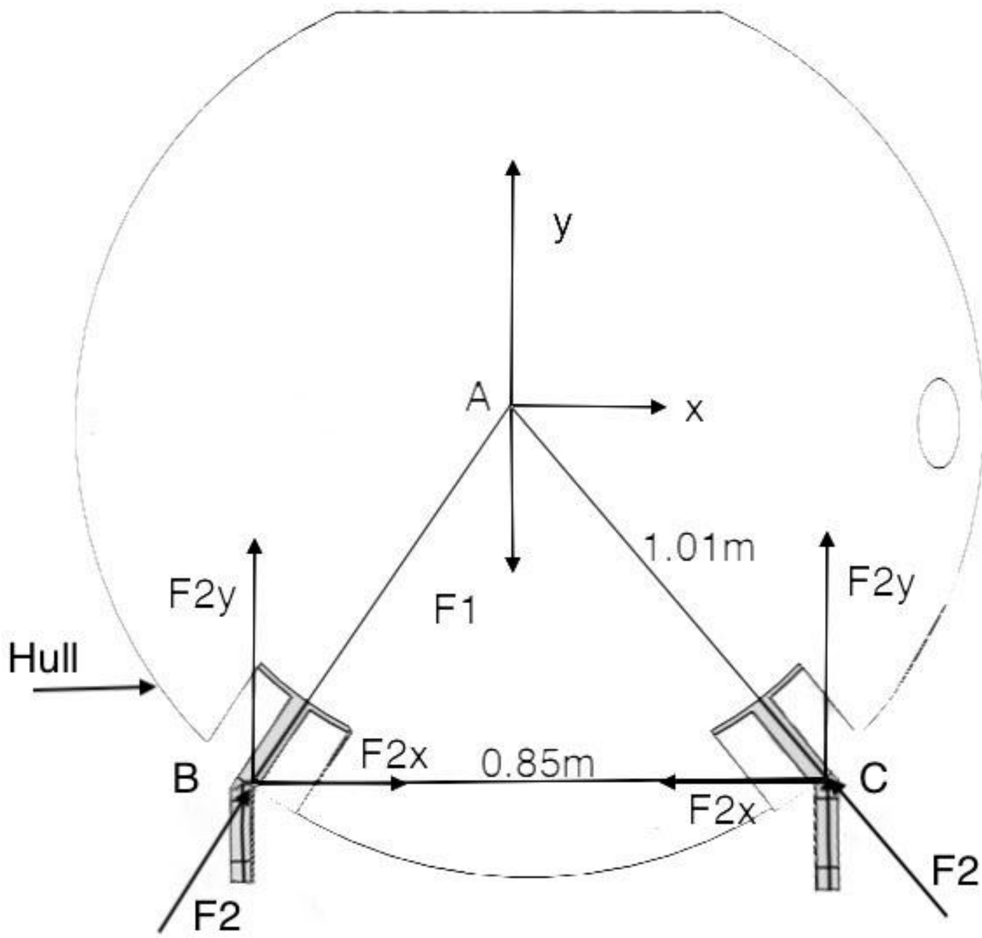

Figure 55: Frame and Hull Reaction Forces

The total force due to gravity of F1 is 25,854.8N. This force is a result of mainly the Hull having a mass of 4440.87kg and the rest of the additional weight is as a result of the hatch, two passengers, a pilot and the control panel mount supports.

Inputs : The angles ∠BAC = 49.7 degrees, ∠ABC = ∠ACB = 65.12 degrees and F1 is 25,854.8N

Due to symmetry, there is equal distribution of weight between the two frames at the bottom of the hull.

To determine the force exerted on one frame member we use trigonometric ratios.

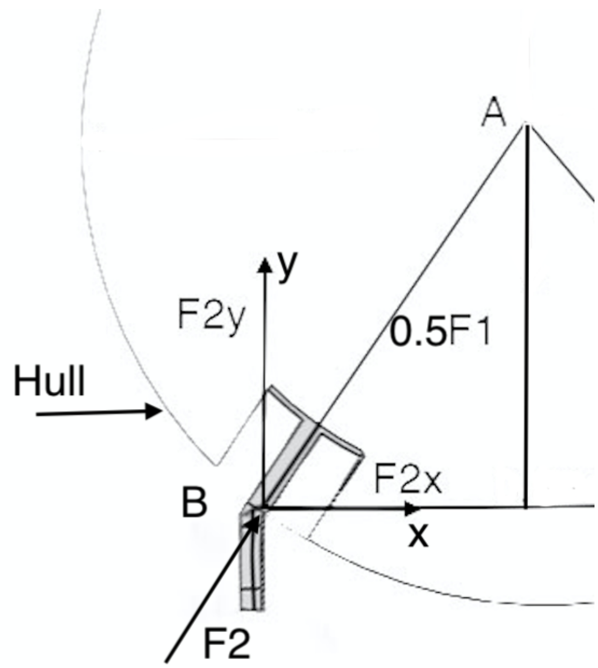

Figure 56: Close up view on Frame and Hull Reaction Forces

To support the weight, $F2_y = 0.5F1 = 12{,}927.4N$

Therefore, the total force (F2) perpendicular the sphere is $\frac{F2y}{sin(\angle ABC)} = \frac{12{,}927.4N}{sin(65.12)}$

F2= 14,249.92N

With this, we then analyse stress/pressure generated on the material by that force to ensure safety and to prevent cracking.

In order to do so, an area must be evaluate the in contact with the force.

First thing we solve for is the angle ∠ theta and with that, we can find the length of the arc L, given the following design parameters.

r = 1.01m, d= 1.24m, width = 0.25m

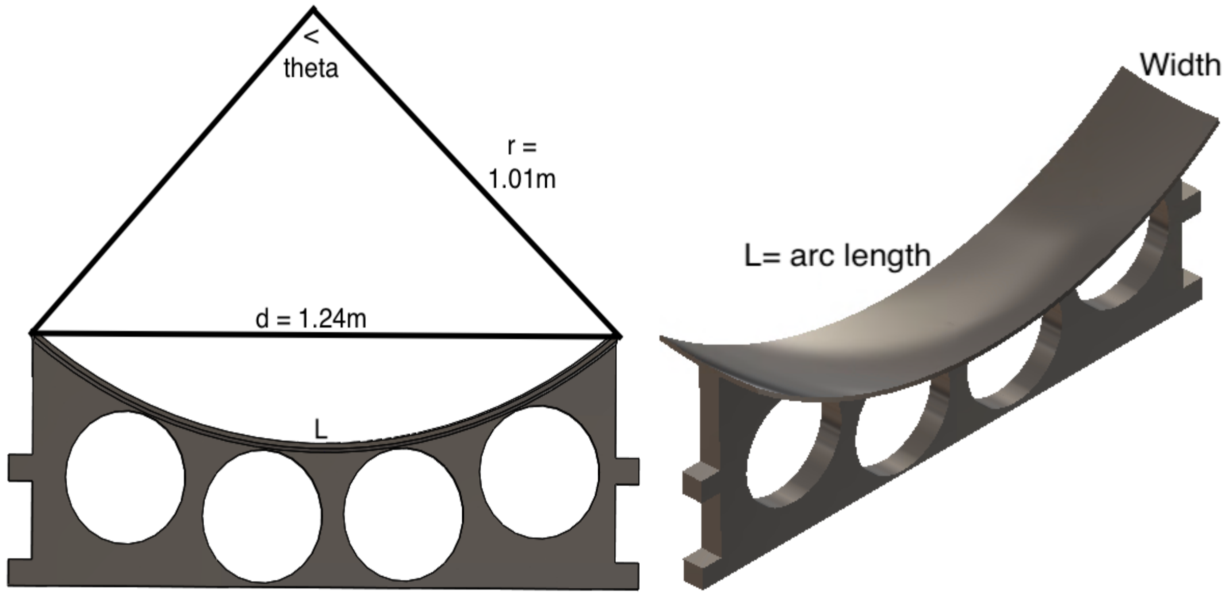

Figure 57: Bottom frame member with the different variables indicated

Knowing that

$$d = 2r\sin(\angle theta/2)$$

$$1.24m = 2(1.01m)\sin(\angle theta/2)$$

$$\angle theta = (\sin(\frac{1.24m}{2(1.01m)})^{-1}) * 2$$

$$\angle theta = 1.322rads$$

Therefore;

L= rt

L= 1.01m*1.322rads = 1.335m

With that length, we finally evaluate the area subjected to force,

$$A = L*Width$$

Area = 1.335m * 0.25m = 0.33m^2

Pressure/Stress **=** $\frac{F2}{A} = \frac{14,249.92N}{0.33m2}$ = 43,181.58Pa = 0.043MPA.

Critical Review :

The yield strength of Acrylic is 70Mpa[5], and is less than the yield strength of the steel chosen so it therefore controls the design. We are then certain the hull will not fail due to stress from reaction forces from the frame at 0.043Mpa, bringing the safety factor to

n = $\frac{70Mpa}{0.043Mpa}$= 1627.9. This result shows that we overestimated how much stress would be applied on the hull. We will be updating the dimensions of the bottom frame in order to minimize material used and to get a more realistic factor of safety.

Release Buoy Analysis

Description of Inputs and Outputs

The release buoy mechanism is a scotch yoke mechanism which replaces the piston model with a pin model. The inputs for the analysis will be the weights of the marker buoy, while the output will be the needed torque.

<u>Justification for the numerical values of the constants, parameters, safety factors</u>

The release buoy mechanism was designed to be compact while the marker buoy would occupy most of the space.

Assumptions, simplifications and Material Selection

- The marker buoy is made of HDPE with a density of 924g/m^3 [12].
- The hole the pin passes through is covered in teflon.
- Friction forces are neglected.
- Marker Buoy mass is 5.3kg.
- Marker Buoy Volume is 0.01m^3

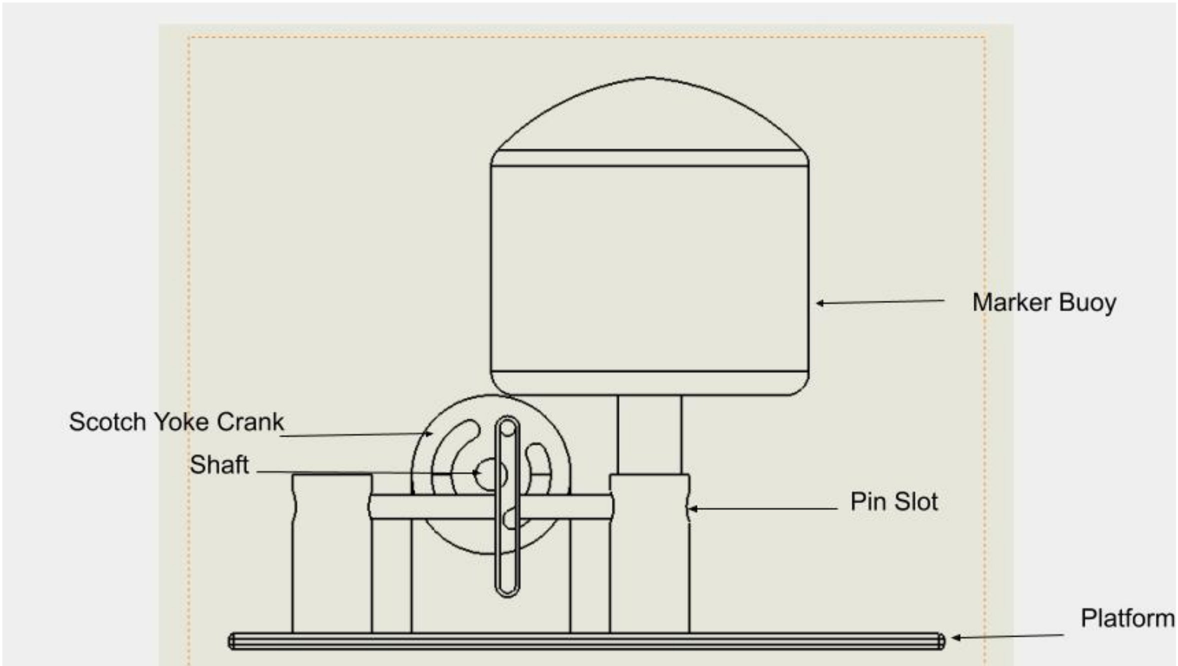

Figure 58: Release Buoy Assembly

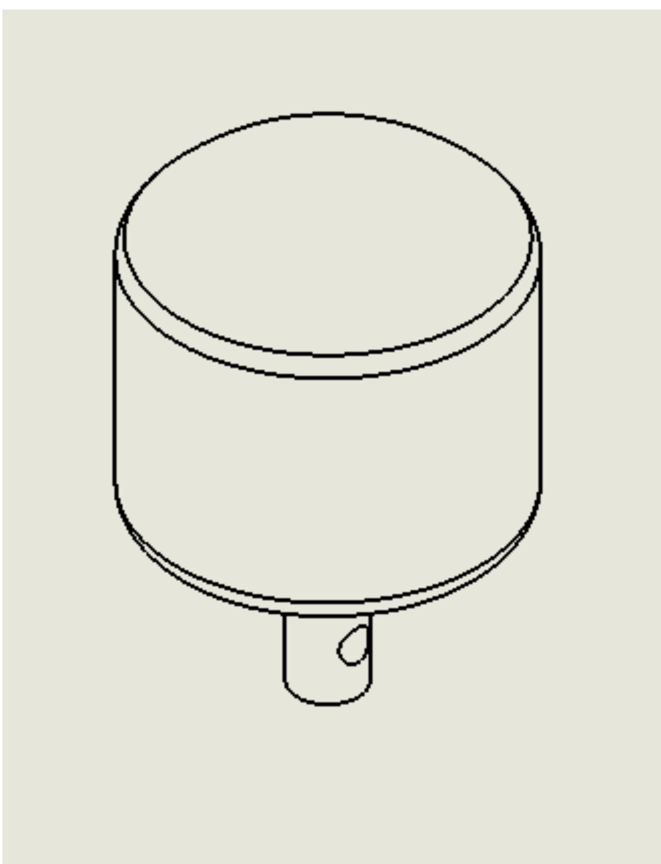

Figure 59: Marker Buoy

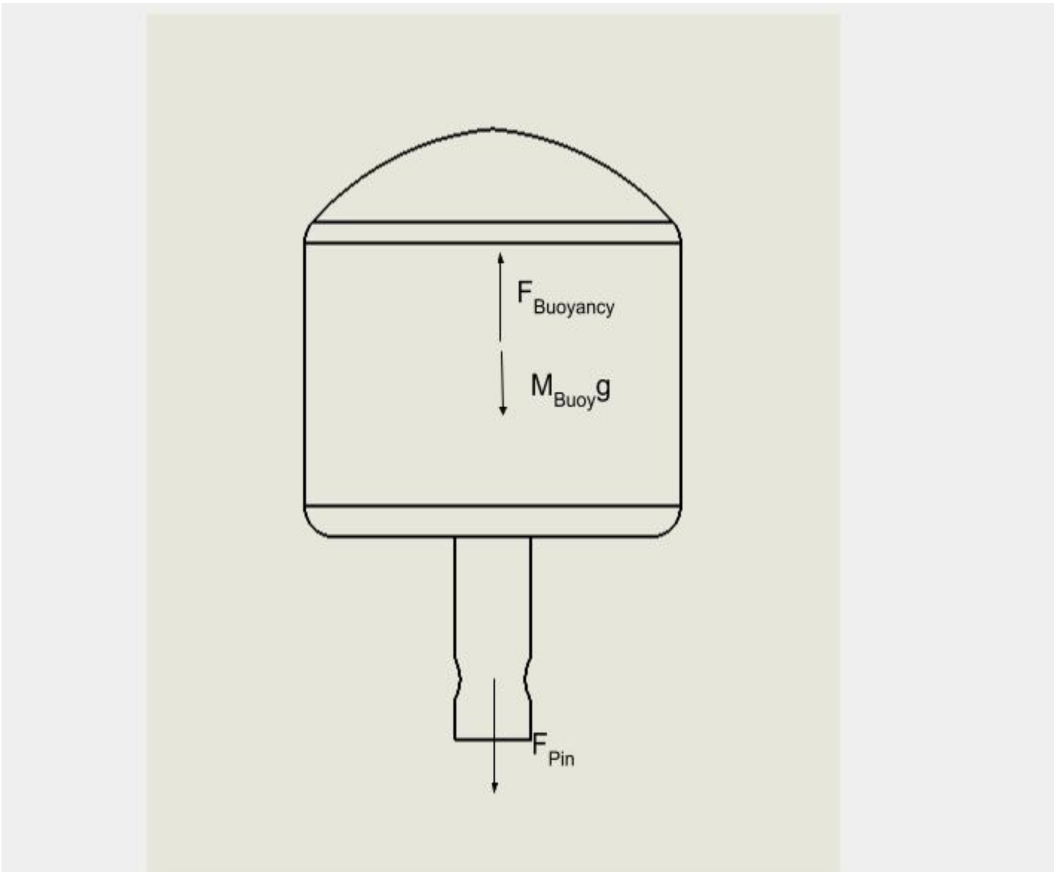

Figure 60: Forces acting on marker buoy

$$\Sigma F_Y = 0$$

$$F_{Pin} = F_{Buo} - m_{Buoy} \cdot g = \rho_{Seawater} \cdot V \cdot g - m_{Buoy} \cdot g$$

$$F_{Pin} = (1029kg/m^3 \cdot 0.01m^3 \cdot 9.81m/s^3) - (5.3kg \cdot 9.81m/s^2) = 48.95N$$

$$T = F_{Pin} \cdot r_{crank} = 48.95N \cdot 0.04m = 1.96Nm$$

5 Discussion and Future Work

The acrylic hull of the submersible assumes a perfect spherical shape for the analysis of the compression stresses, but in reality there are discontinuities that occur at regions of hull penetration. It has been established during the analysis that the major factor for submerging deeper is the thickness of the acrylic hull shell. The increase is dependent on the desired factor of safety, and in our analysis, we evaluated with a safety factor of 2. Our major concern going forward to the CAD is the sealing of the hull penetrations. To adequately seal the hull, we need to extrude flat surfaces at the region of penetrations to ensure effective contact for sealing.

The hatch of our submersible can be split into 4 major parts including: the hatch door, hatch seat, locking mechanism and hinge/spring. As a result of the very high weight of this assembly, it's safety factor was determined to be around 5. Our aim for the future is to redesign with a goal of achieving a safety factor around 2. The thickness of the hatch, as well as the hatch door can be reduced to bring down the overall weight and ensure the spring is strong enough to lift the hatch without failure.

The connection between the bushing and main shaft is sealed using a dynamic sealing method. However, a combination of a static sealing and a ball bearing method will be employed in order to achieve a more reliable seal.

The spring in the hatch-hinge assembly is always exposed to seawater, as a result, the ideal material should have good resistance to corrosion. Adding chromium can significantly improve the resistance to corrosion, hence, ASTM A232 [8] was chosen as the spring wire material. Another viable material will be stainless steel wire ASTM A313 [8], but as seen on Figure 29, an increase in diameter of this material yields a lower tensile strength.

Currently, we have established that our penetrations will be sealed by means of a silicone rubber mounted on the penetrator in the form of a cylinder (release buoy, electricals and ventilation) or flange in the case of the frame penetration. We are considering binding the silicone rubber to the flange by means or rivets to simplify the design and guarantee the force distribution for the sealing is uniform. The o-ring acting as a dynamic seal in the case of our release buoy, is directly mounted on the shaft in the figures. According to Parker's o-ring handbook [ ] that is not recommended because of the friction that can play a part. In future submissions, we intend to redesign the shaft and create grooves for the o-ring to be situated in. Doing this creates an issue in maintenance, because if the cylinder is divided into two halves such that they can be joined together by bolts, whenever the shaft is removed, the o-ring is removed and vice versa.

We have been able to do most of the analysis for the frame. It is necessary to have a very strong and secure frame when lifting the sub out of the water. The Analyses that are yet to be completed are the connection type of the fairing side to fairing cover weldment. We also need to do some analysis on the bottom frame, and the forces exerted on the bottom frame, and the side frame. For the bottom frame structure, that will need to be parametrized and re-analyzed because the current Factor of safety to too large, unreasonable and unnecessary. The facing area as discussed will be reduced so that we don't have excess material in the frame, this inturn will lower costs of materials and make the submarine lighter.

The future work involves the design and analysis of the seating platform, the seats, control panel mount and the anchor points for two passengers. The seating platform will be attached to the frame which penetrates the hull from the bottom, while the seats and control panel mounts will be connected to the platform.

6. References

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Appendices

A. Additional Analysis

Electrical penetrations

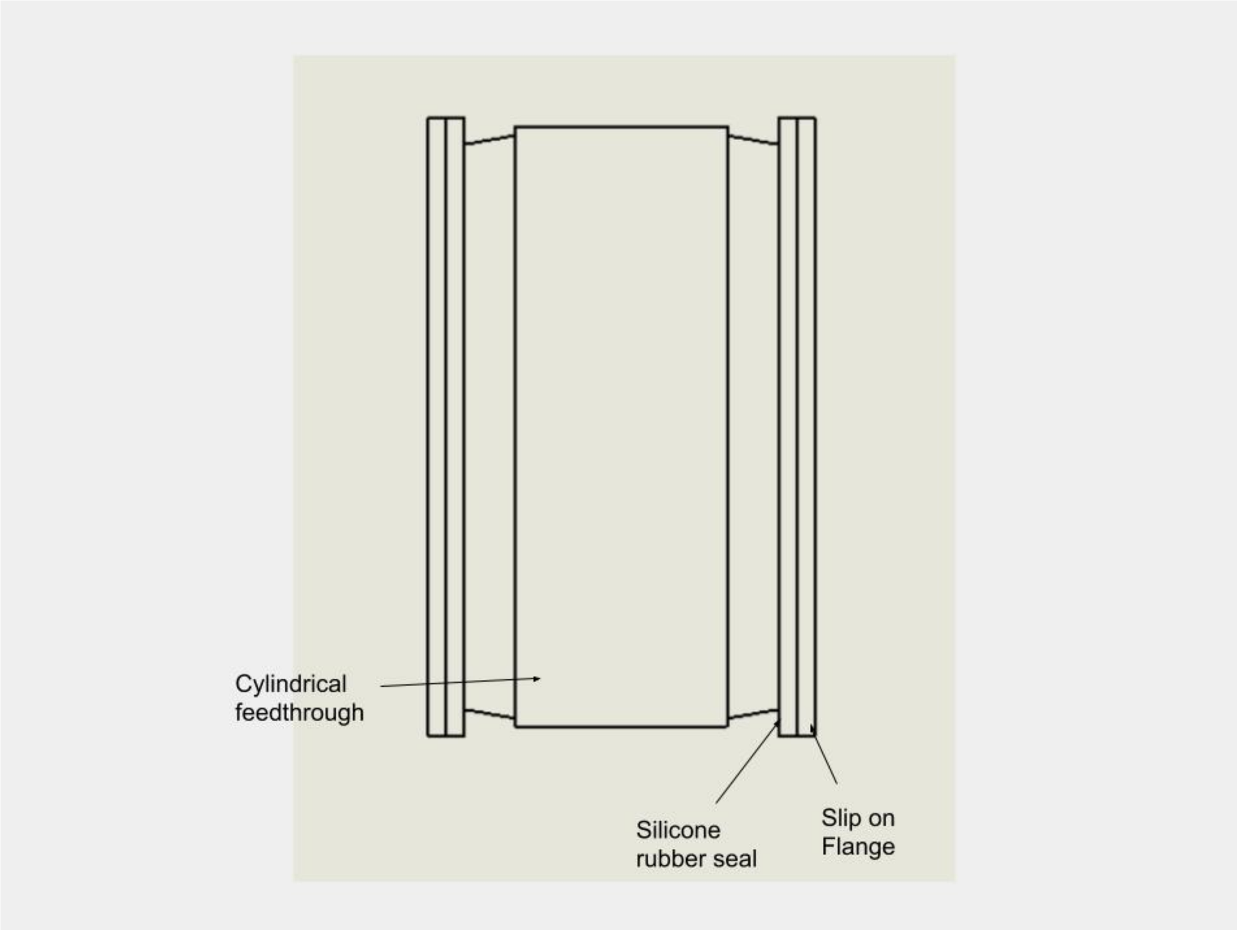


Figure 61: Electrical penetration Sealing

Table 17:Electrical Penetrator sealing inputs

Variables	Magnitude	Unit
Compressive Yield σ_y	30	MPa
D_{Flange}	34	cm
Bolting circle D_{bolt}	24	cm
D_{seal}	34	cm
Safety Factor SF	2	-
Applied Stress σ_a	15	MPa

Proof Strength of bolt S_P	247.5	MPa

$$15MPa = \frac{F_a}{\left(\frac{\pi \cdot d_{flange}^2}{4} - \frac{\pi \cdot d_{bolt}^2}{4}\right)} = \frac{F_a}{\left(\frac{\pi \cdot 0.34^2 m}{4} - \frac{\pi \cdot 0.24^2 m}{4}\right)}$$

F_a= 690,000N

$n \cdot F_{bolt} = \sigma_a \cdot A_b$ where n=12

$$12 \cdot F_{bolt} = 15MPa \cdot 0.045m^2$$

F_{bolt}= 56,250N

For $F_{i,}$=56,250 N,

56,250N=0.9(A_t)(247.5MPa)

A_t=252.53mm^2, From the table, required bolts are M22.

Ventilation System Penetration Sealing

Table 18: Ventilation Sealing Inputs

Variables	Magnitude	Unit
Compressive Yield σ_y	30	MPa
D_{Flange}	26	cm
Bolting circle D_{bolt}	22	cm
D_{seal}	26	cm
Safety Factor SF	2	-
Applied Stress σ_a	15	MPa
Proof Strength of bolt S_P	247.5	MPa

$$15MPa = \frac{F_a}{\left(\frac{\pi \cdot d_{flange}{}^2}{4} - \frac{\pi \cdot d_{bolt}{}^2}{4}\right)} = \frac{F_a}{\left(\frac{\pi \cdot 0.26^2 m}{4} - \frac{\pi \cdot 0.22^2 m}{4}\right)}$$

F_a= 225,000N

$n \cdot F_{bolt} = \sigma_a \cdot A_b$ where n=8

$$8 \cdot F_{bolt} = 15MPa \cdot 0.038m^2$$

F_{bolt}= 28,125N

For $F_{i,}$=28.,125 N,

28,125N=0.9(A_t)(247.5MPa)

A_t=126.26mm^2, the necessary bolt size is M16

Frame Sealing

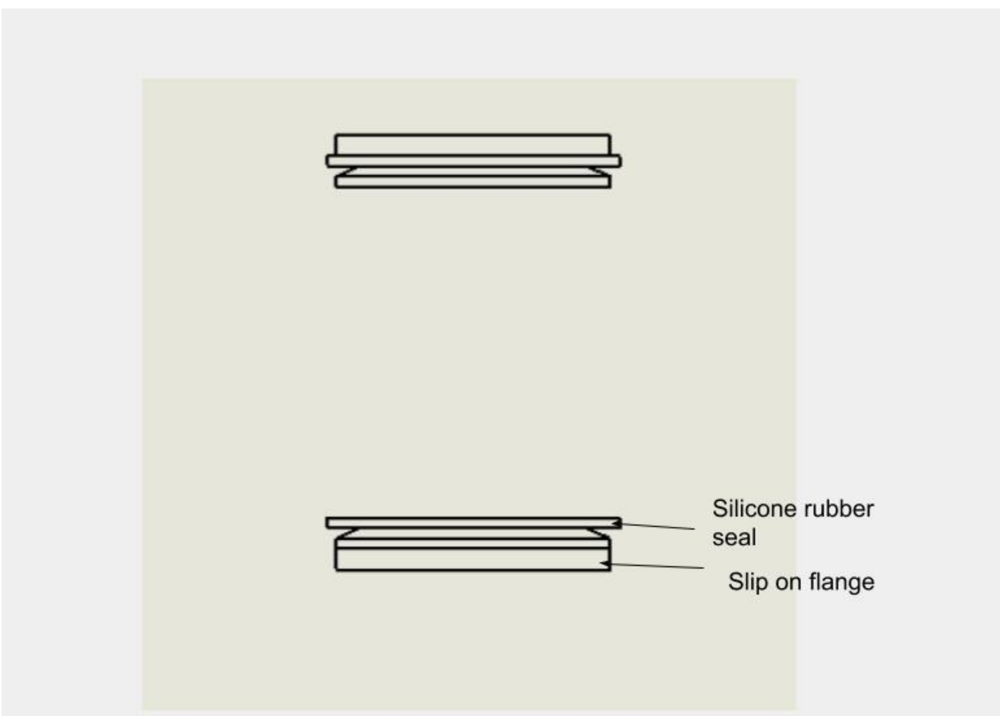


Figure 62: Sealing of hull-frame penetration

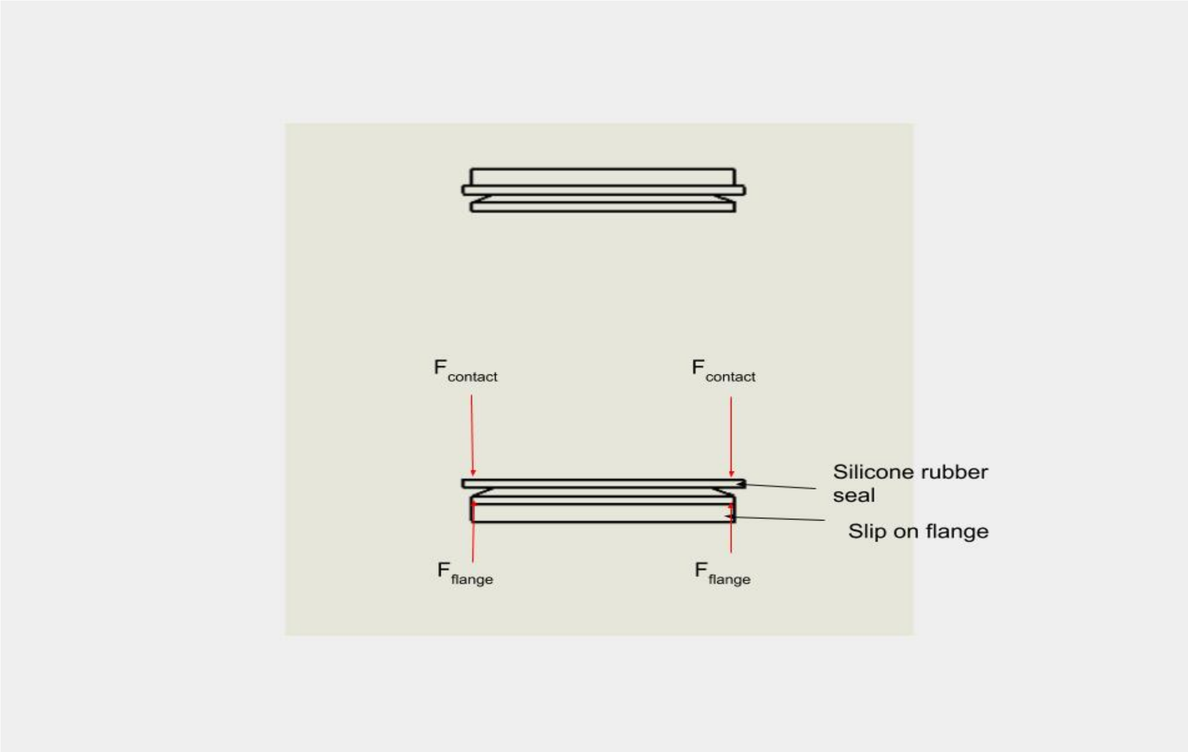


Figure 63: Reaction forces on the Seal

Table 19: Frame Sealing Inputs

Variables	Magnitude	Unit
Compressive Yield σ_y	30	MPa
D_{Flange}	12	cm
Bolting circle D_{bolt}	8.8	cm
D_{seal}	12	cm
Safety Factor SF	2	-
Applied Stress σ_a	15	MPa
Proof Strength of bolt S_P	247.5	MPa

$$15MPa = \frac{F_a}{(\frac{\pi \cdot d_{flange}{}^2}{4} - \frac{\pi \cdot d_{bolt}{}^2}{4})} = \frac{F_a}{(\frac{\pi \cdot 0.0.12^2 m}{4} - \frac{\pi \cdot 0.088^2 m}{4})}$$

F_a= 78,000N

$n \cdot F_{bolt} = \sigma_a \cdot A_b$ where n=6

$$6 \cdot F_{bolt} = 15MPa \cdot 0.006m^2$$

F_{bolt}= 15,000N

For $F_{i,}$=15,000 N,

15,000N=0.9(A_t)(247.5MPa)

A_t=67.34mm^2, the necessary bolt size is M12

Additional FBD (for forces when lifting sub out of water):

Force	Magnitude (N)
F1, Fghatch	1015.24
F2, Fginner	4659.75
F3, F4	10597.52
F5, F6, F7, F8, F9, F10, F11, F12	667.46
F13, F14, F15, F16	8323.41
F17, F18, F19, F20	355.81
F21, Fgscrubber	(Not currently available)
F22, F23	286.94
F24, F25, Fgbattery	1079.10
F26, F27	669.53

F28, F29	3945.09
F30, F31	4890.38
F32, F33, Fgo2tank	667.46
F34, F35, F36, F37	3932.44
F38, F39	32866.42
F40, F41	33815.83
Fghull	20179.80
Fgcomp	1334.93
Fgframebottom	1049.57
Fgframeballast	711.62
Fgframeback	286.94
Fgfairingback	1339.07
Fgfairingside	2196.46
Fgfairingsidecover	1000.23
(Fgballast + Fgdropweight)	15729.75
Fgframesidebottom	1941.79
Fgframesidetop	949.41

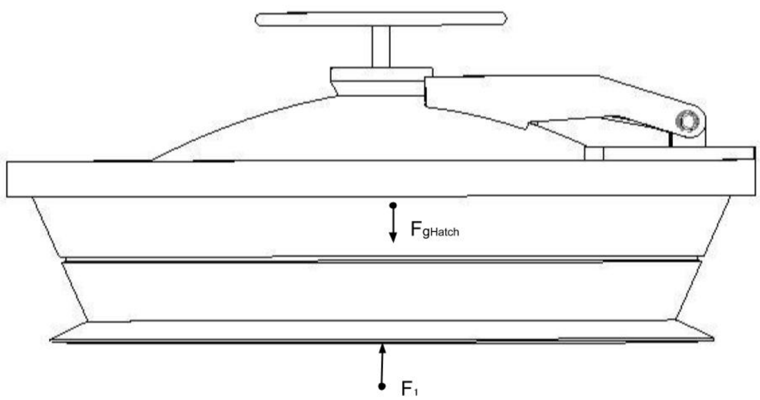


Figure 64:FBD Hatch

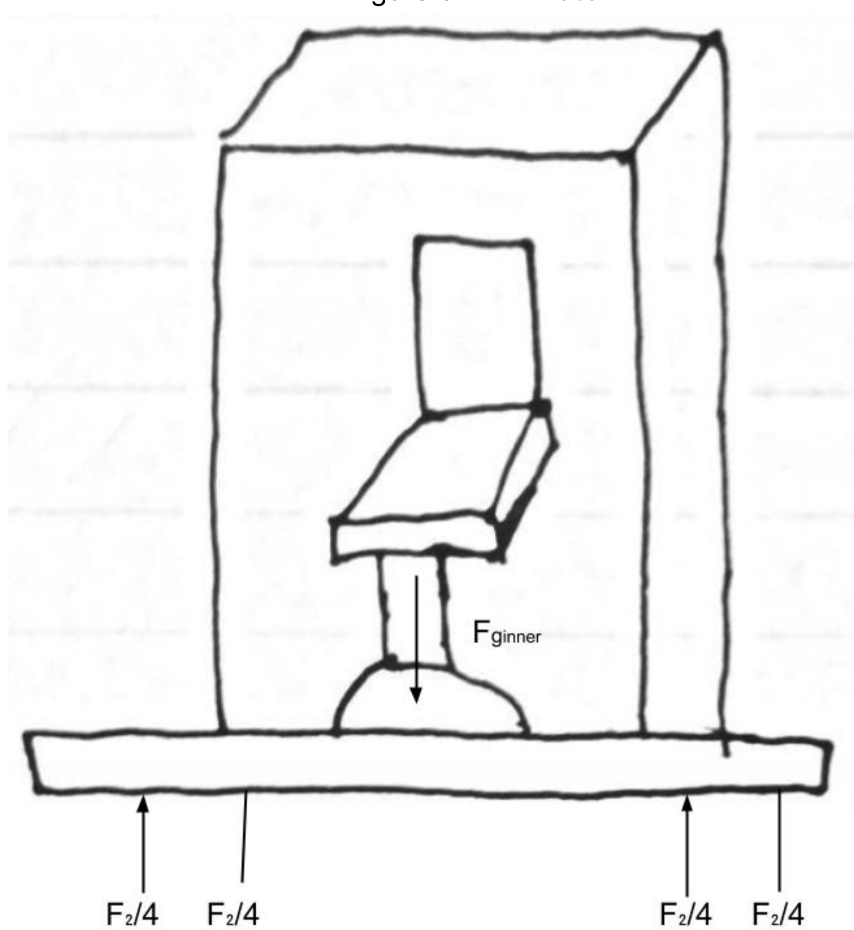


Figure 65: FBD Seating Assembly

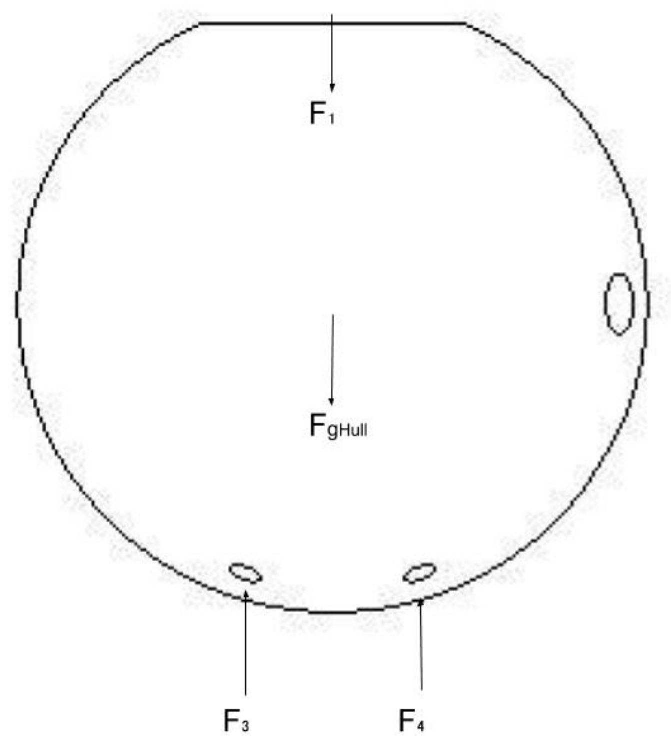


Figure 66: FBD Hull

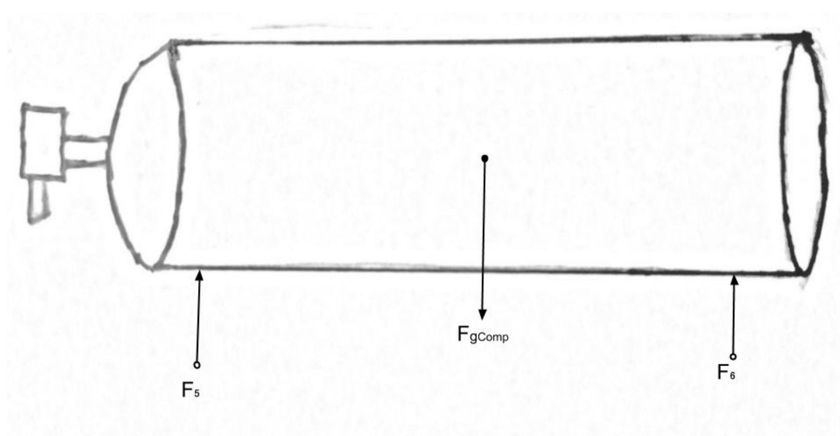


Figure 67 : FBD compressed air. ×4 (F5, F6; F8,F9;F9,F10;F11,F12)

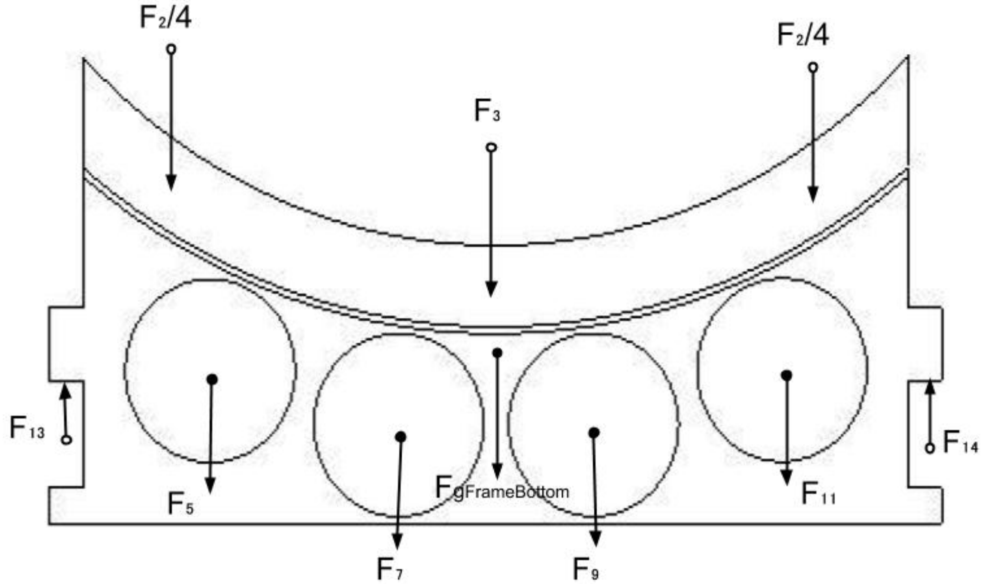


Figure 68 : FBD Bottom Frame. $\times 2$ (F13, F14; F15,F16)

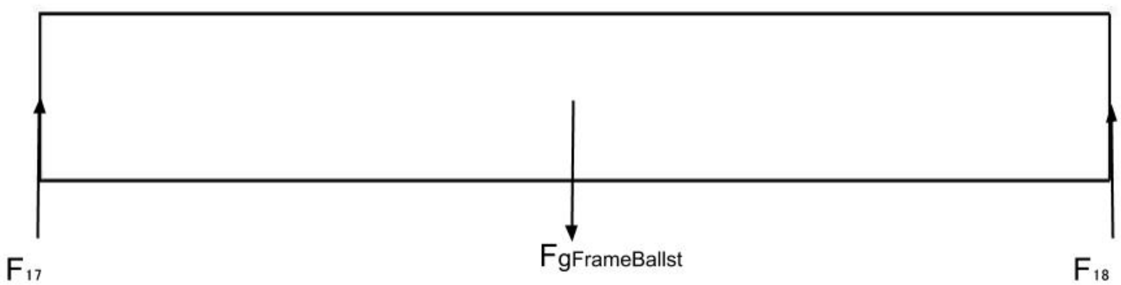


Figure 69: FBD Frame Ballast. ×2 (F17, F18; F19,F20)

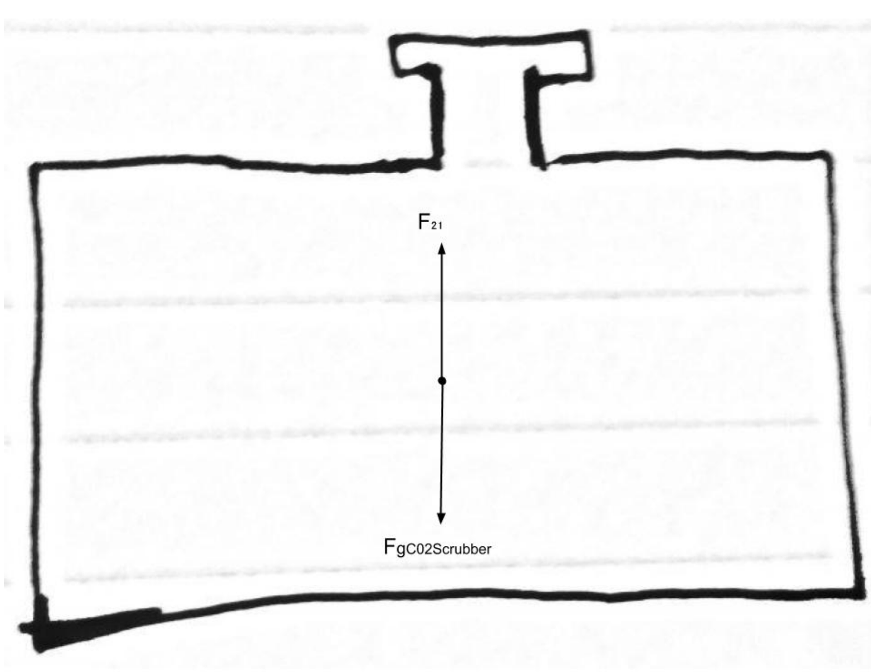


Figure 70: FBD CO_2 Scrubber

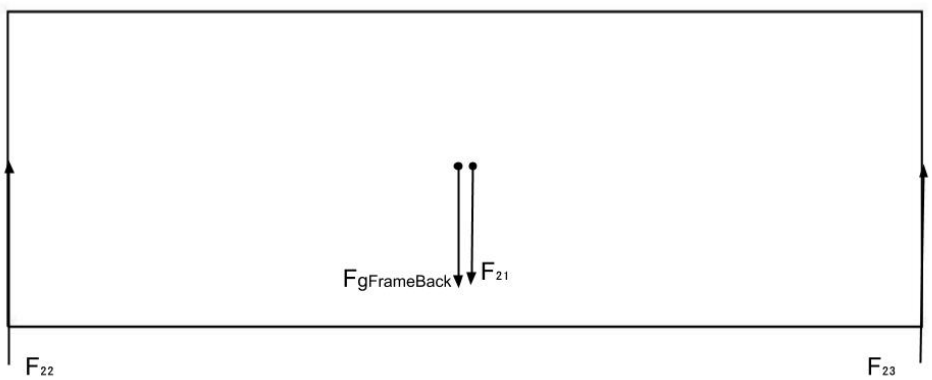


Figure 71: FBD Frame Back

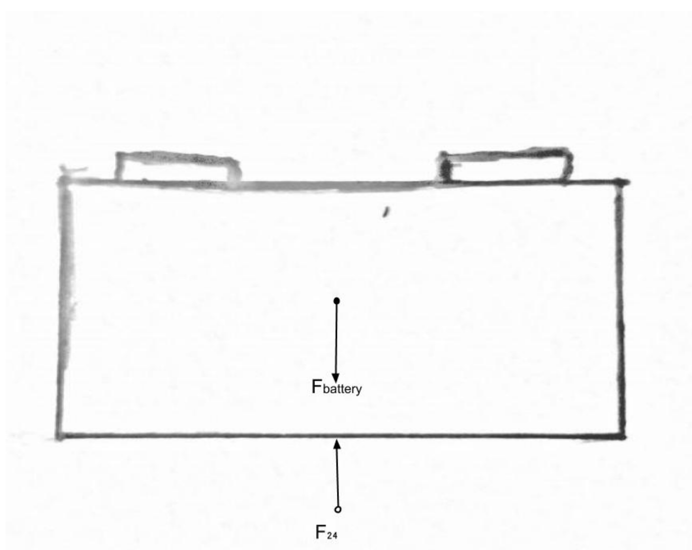


Figure 72: FBD Battery. $\times$2 (F24; F25)

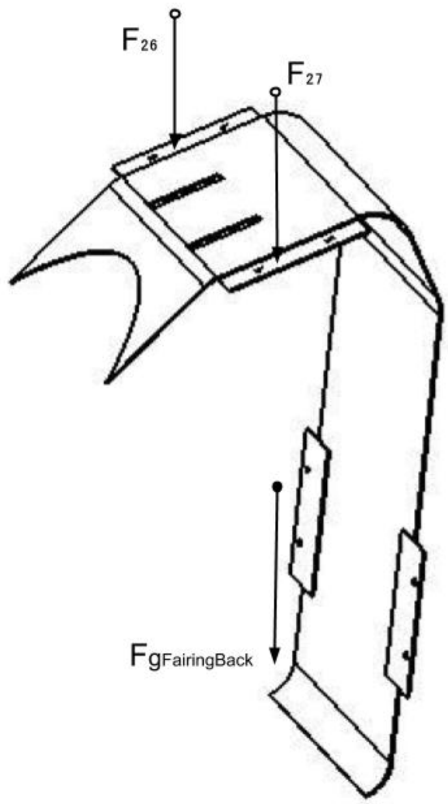


Figure 73: FBD Frame Back

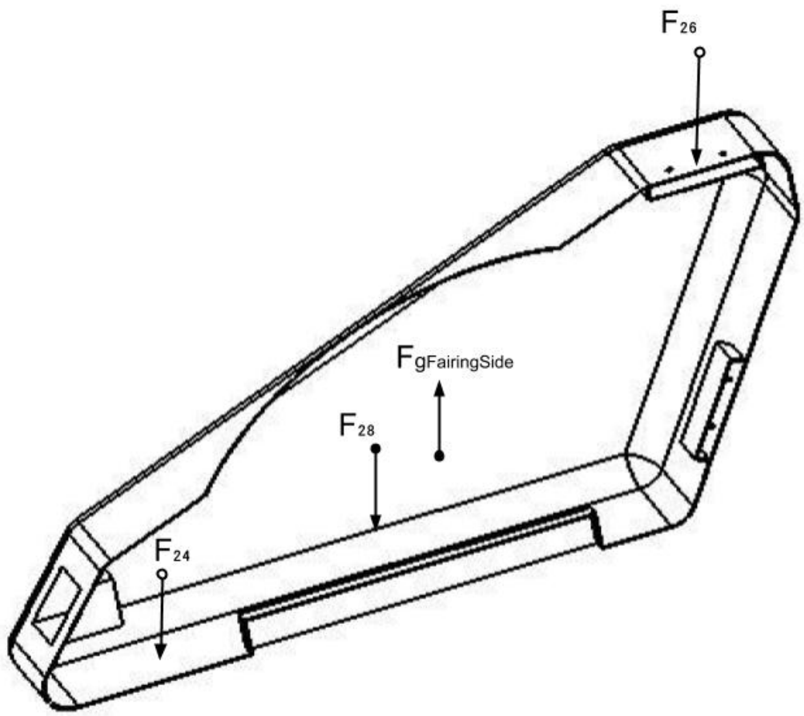


Figure 74: FBD Fairing Side. ×2 (F28; F29)

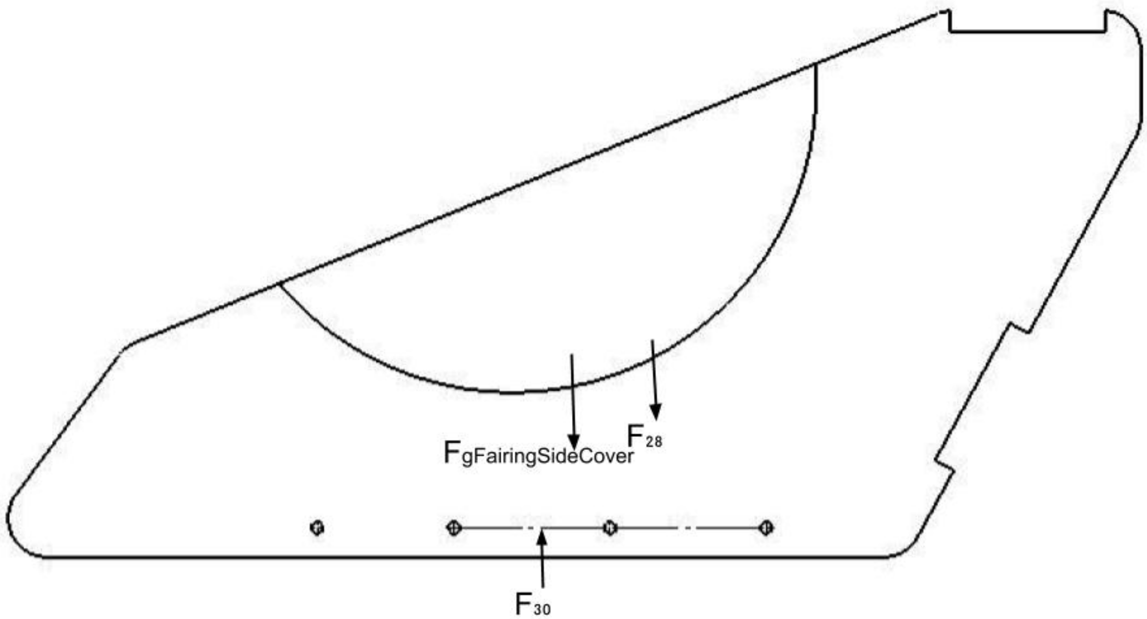


Figure 75: FBD Fairing Side cover $\times$2 (F30; F31)

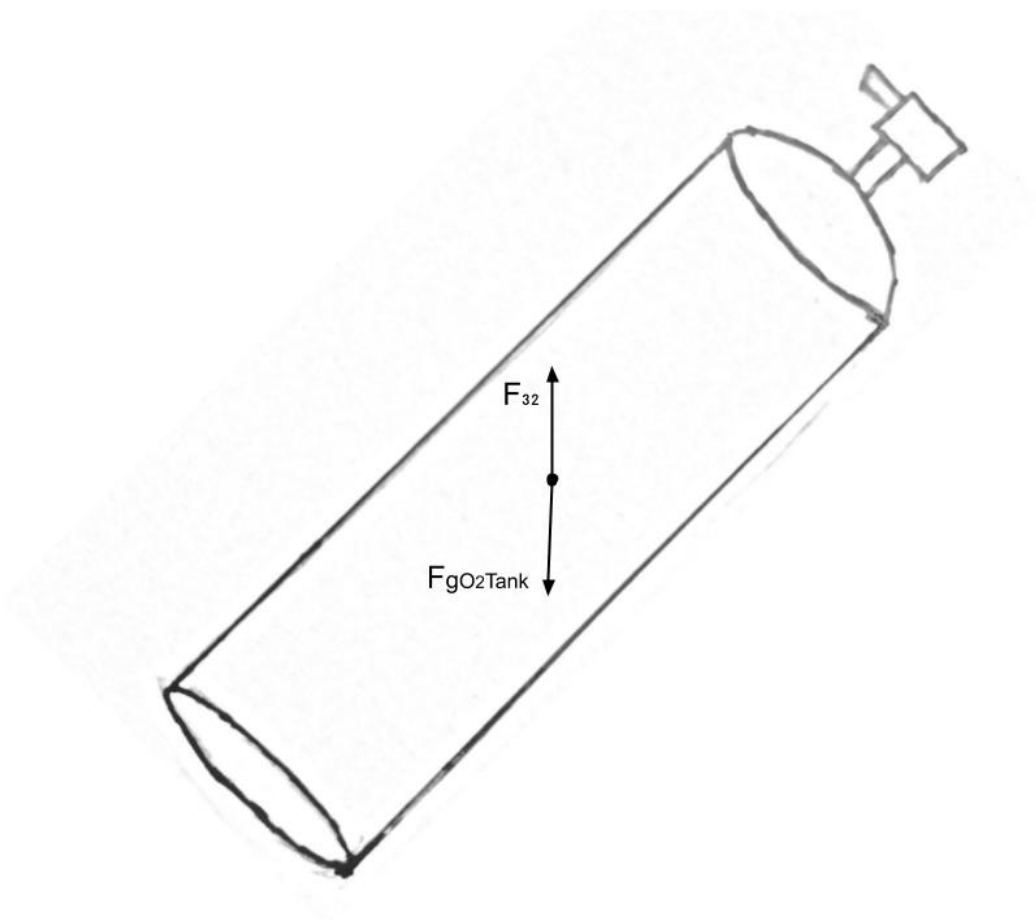


Figure 76: FBD O2 Tank ×2 (F32; F33)

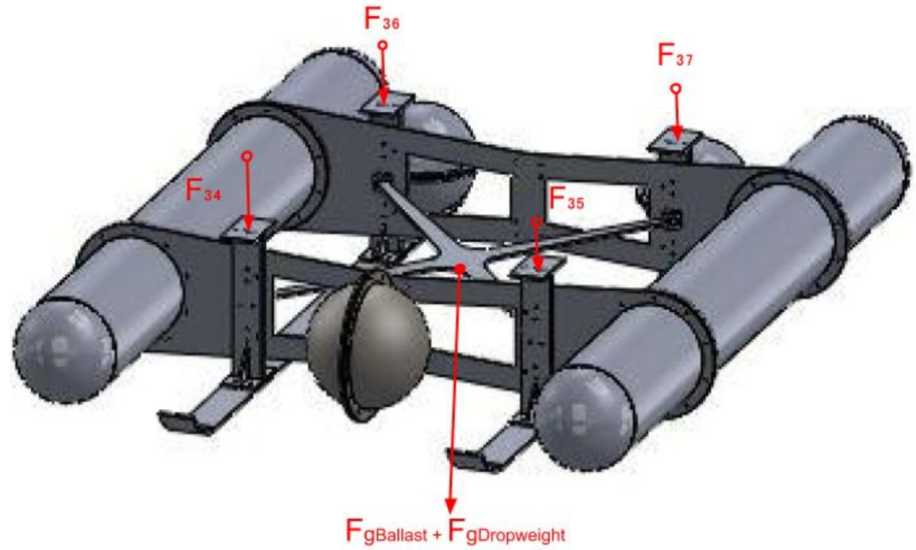


Figure 77: FBD Ballast Assembly

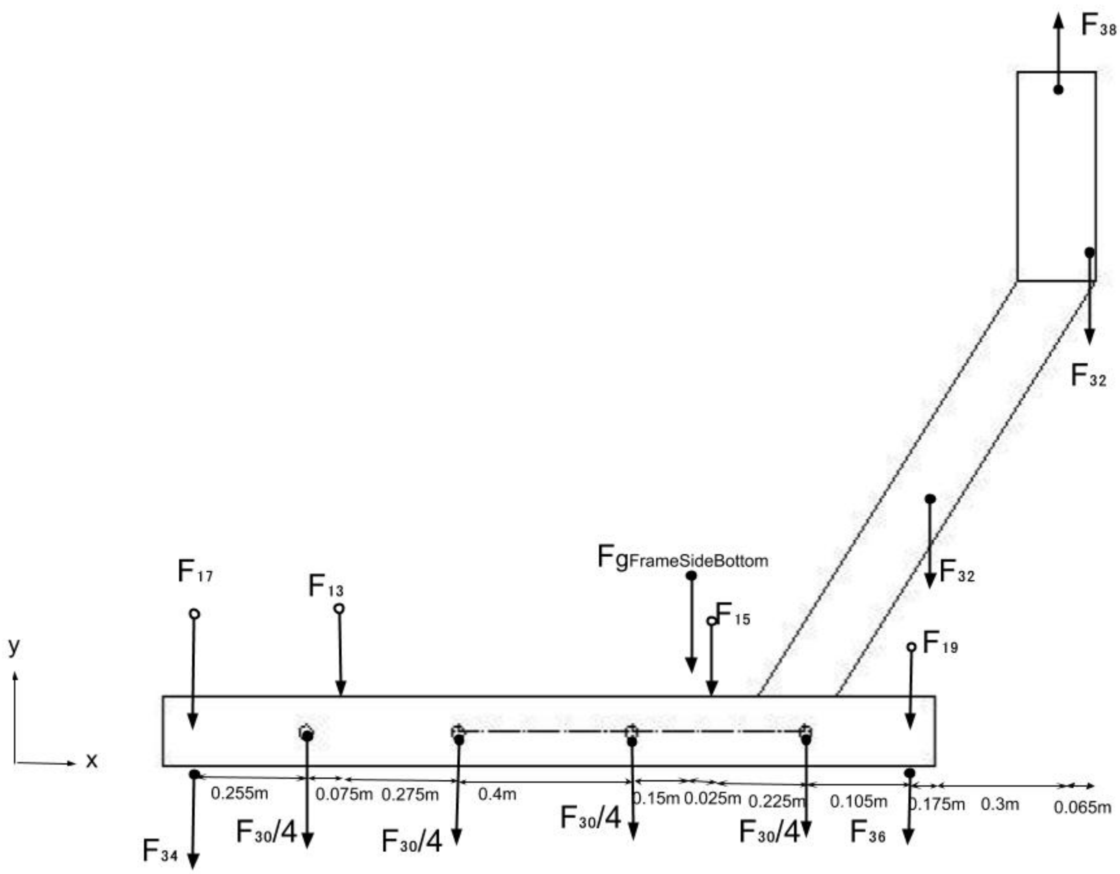
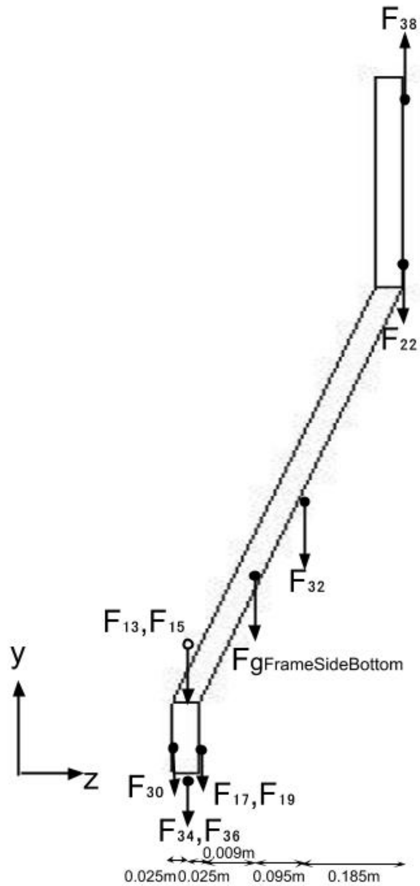


Figure 78: FBD Bottom Side Frame (Side View). ×2 (F38; F39)



Figue 79: FBD Bottom Side Frame (Front View). ×2 (F38; F39)

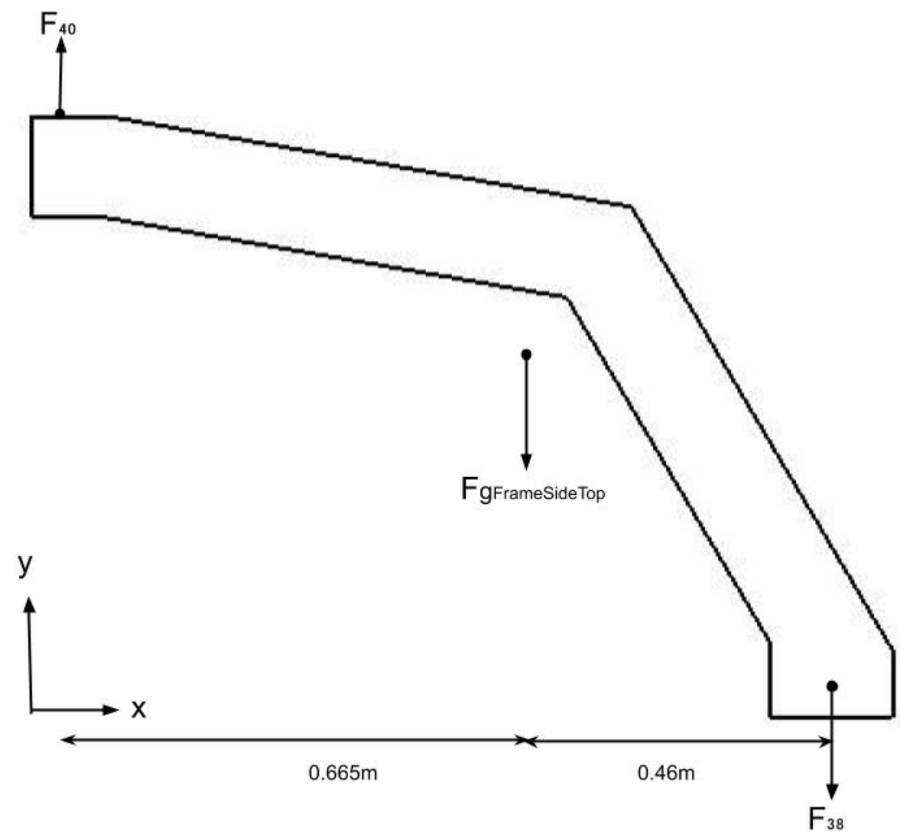


Figure 80: FBD Top Side Frame (Side View). ×2 (F40; F41)

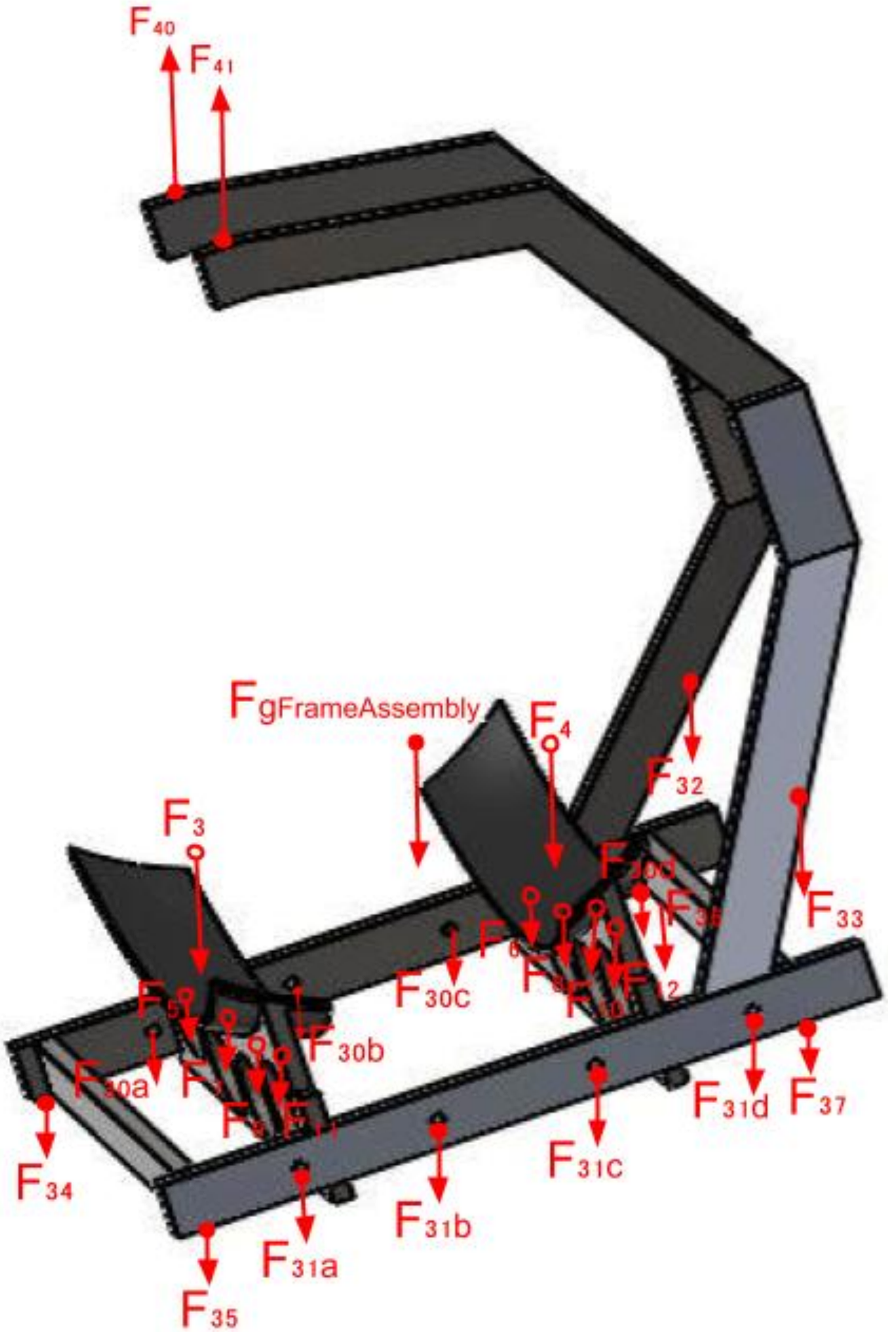


Figure 81: All the forces exerted on the Frame sub-assembly