

Design of a modular submersible platform for monitoring marine
ecosystem (*Benthic Lander*)

Hugo João Miranda Alves

Dissertação de Mestrado

Orientador na FEUP: Professor Doutor Mário Augusto Pires Vaz

Orientador no INEGI: Engenheiro Tiago António Nunes da Silva Morais



Mestrado Integrado em Engenharia Mecânica

Junho 2017

À minha mãe

Conceção de uma plataforma modular submersível para a monitorização do ecossistema marinho (*Benthic Lander*)

Resumo

A presente dissertação insere-se no âmbito do Projeto AMALIA – ‘Algae-to-Market Lab IdeAs’. AMALIA é um projeto europeu que pretende transformar uma atual ameaça dos oceanos, as algas invasoras, numa oportunidade. Valorizar as algas do Noroeste da Península Ibérica e criar produtos alimentares inovadores, rações com potencial para estimular o sistema imunitário de peixes e camarões em aquacultura, extratos para a indústria cosmética e medicamentos, são alguns dos objetivos do projeto AMALIA. Para a monitorização destas algas invasoras serão utilizados avançados sistemas e soluções de engenharia e recolha de imagem, integrados num sistema subaquático que dará informações em tempo real sobre o aparecimento e as quantidades de algas. O trabalho desenvolvido nesta dissertação apresenta a fase inicial da conceção e projeto de uma plataforma oceânica submersível modular e reconfigurável para suporte de sensores que permitam monitorizar e detetar o aparecimento de algas invasoras.

Numa fase inicial procedeu-se à recolha de informação referente ao estado-da-arte das tecnologias utilizadas para a observação do espaço marinho, dos diferentes materiais utilizados e de distintos equipamentos de monitorização. Após análise e seleção da tecnologia que mais se adequa à operação pretendida, foram gerados diversos conceitos da plataforma e sistemas propostos pelo autor.

Após ter sido selecionado um conceito procedeu-se ao dimensionamento dos principais componentes da estrutura e a um estudo dos diferentes materiais que poderiam maximizar a sua *performance*, tendo sido dada especial atenção à redução do peso e preço do produto final. Para isso, foram realizadas várias análises de esforços e tensões na estrutura tendo em conta os diversos casos de carga previstos e, posteriormente, foi feito um estudo do comportamento estrutural previsível.

Concluiu-se que a inserção de materiais de origem polimérica neste tipo de tecnologias, embora não convencional, surge como uma alternativa viável para a obtenção do produto com as características descritas, pelas suas propriedades mecânicas e comportamento para aplicação em ambiente marinho.

Design of a modular submersible platform for monitoring marine ecosystem (Benthic Lander)

Abstract

This dissertation integrates the project AMALIA – ‘Algae-to-Market Lab IdeAs’. AMALIA is an European project that aims to transform a current ocean threat, invasive seaweeds, into an opportunity. Give value to seaweeds of the northwest of the Iberian Peninsula and create innovative food products, animal feed with potential to stimulate the immunity system of fish and shrimps in aquaculture, extracts for cosmetic industry and medicines, are some of the objectives of the AMALIA project. To monitor the appearance of these macroalgae, advanced engineering and imaging systems and solution will be deployed into the seafloor to provide real-time information regarding the appearance and quantities of algae. The worked developed on this dissertation presents initial steps of the design of a ocean modular submersible platform adjustable to support sensors that allow to monitor and detect the surge of invasive seaweeds.

Initially, it was made an information collection regarding the state-of-the-art of current technologies used for marine observation, different used materials, and distinct monitoring equipment. Analysed and selected the most suitable technology to the operation, it were generated several concepts for the platform and systems, proposed by the author.

Selected on single concept, it was designed the main structural components, and studied different materials in order to maximize its performance, always giving special attention to weight and cost reduction of the final product. For that, there were made several stress analysis on the structure, considering different predicted load cases and, posteriorly, it was made a study regarding the predicted structural behaviour.

It was concluded that the insertion of polymeric materials on this type of technologies, although not conventional, are a viable alternative to obtain a final product as described, given their good mechanical properties and behaviour under marine environment applications.

Agradecimentos

Mais do que a meta de um percurso académico prestes a findar, vejo esta dissertação como o resultado de um esforço conjunto, um pináculo de uma viagem composta por mil e um obstáculos ultrapassados, na sua maioria, pelo apoio dos muitos que me acompanharam.

Como tal, começo por agradecer ao Professor Doutor Mário Vaz e ao Engenheiro Tiago Morais, cruciais na orientação deste projeto.

Uma palavra de apreço aos Professores José Luís Esteves, Paulo Tavares de Castro e António Torres Marques, e aos Engenheiros Nuno Viriato, Diogo Vale, Carlos Machado e José Cunha pela disponibilidade e pelo conhecimento transmitido em diferentes matérias, fundamentais para a execução do projeto.

De cariz mais pessoal, inicio, agora, por agradecer aos colaboradores do INEGI que estiveram presentes aquando deste projeto: à Elisabete Barros, aos meus colegas do Grupo TECMAR, Guilherme, Pedro, Ashank e Vasco, e aos restantes companheiros das efémeras e esporádicas pausas!

Ao Grupo de amigos do Fafes, que me acompanharam desde o meu primeiro dia nesta casa, com quem partilhei momentos inesquecíveis e que, indubitavelmente, contribuíram para que este momento fosse possível. Aos meus *póneis* e àqueles com quem muitas horas passei na sala de estudo do Departamento. À Tuna de Engenharia, com quem passei imensos momentos de diversão, a fazer o que mais gosto, com quem mais gosto.

Ao Clube das Estrelas e restantes amigos flavienses a quem devo muito do que sou hoje. Ao *staff* da Ilha do Cavaleiro que, mais do que incluir-me como apenas um colaborador, acolheram-me como um deles. À família Durão pelas oportunidades que me deram, quer a mim, quer à minha família. A vocês um eterno obrigado.

À minha família, nomeadamente o meu pai, Alberto, o meu irmão João e a minha avó Helena, que por muitas vezes foram obrigados a aturar os meus desnorteios ao longo deste curso. À minha tia Sandra, padrinho Jano e prima Beatriz por terem sempre sido mais do que seria espetável. Aos meus avôs, Maria e Guilhermino, e à Nana, as minhas estrelas, por quem farei sempre o máximo por honrar o que por mim fizeram.

Um agradecimento especial à minha mãe, a minha guerreira, lutadora, que fez o impossível para que o hoje fosse possível. A ti, dedicar-te-ei todo e qualquer sucesso que a mim esteja destinado.

Por fim, àqueles que, nem que apenas por uma breve e tardia passagem, marcaram e que, certamente, o vosso contributo se revê neste projeto. E ao Justin Vernon, que muito me acompanhou nas horas de escrita deste documento.

A todos vós, o meu mais sincero e profundo obrigado.

Este projeto teve apoio da União Europeia através do Projeto AMALIA - Algae-to-Market Lab IdeAs da EASME Blue Labs (EASME/EMFF/2016/1.2.1.4/03/SI2.750419).

Hugo João Miranda Alves

Contents

1	Introduction.....	1
1.1	Ocean Exploration.....	1
1.2	Algae-to-Market-Lab IdeAs – TECMAR - INEGI.....	2
1.3	Project objectives.....	3
2	State-of-the-art	4
2.1	Deep-sea Technologies	4
2.1.1	Manned Submersibles.....	4
2.1.2	Remotely Operated Vehicle.....	6
2.1.3	Unmanned Submersibles	7
2.2	Benthic Landers	9
2.2.1	The Role of Benthic Landers in the science fleet.....	9
2.2.2	Types of Landers.....	10
2.2.3	Mechanical and Material Issues	11
2.2.4	Sea keeping and Mooring System.....	17
3	Lander Structural Concepts	20
3.1	Project Requirements.....	20
3.2	Lander Geometry	21
3.2.1	Central Module	23
3.2.2	Supports	33
3.2.3	Buoyancy.....	34
3.2.4	Modularity.....	36
4	Structure design	38
4.1	Central Module Housing design	38
4.1.1	Coordinate System.....	38
4.1.2	Theoretical Approach	38
4.1.3	Open Cylinder Dimensioning.....	40
4.2	Frame dimensioning.....	49
4.2.1	Buckling Analysis.....	53
4.2.2	Steel Structure.....	56
4.2.3	Structure Material Alternative – POM	62
5	Drag Force	69
6	Finite Element Analysis - Simulation.....	73
6.1	Load Case no 1.....	74
6.2	Load Case no 2.....	77
6.3	Load Case no 3.....	80
6.4	Load Case no 4.....	83
6.5	Load Case no 5.....	86
6.6	Modal Analysis	88
	Steel	88
	POM	91
7	Analysis Discussion.....	95
8	Conclusions and Future works.....	97
	Bibliography.....	98
	Appendix A – Different Materials Behaviour under External Pressure	103
	Appendix B – Different components stress representation for distinct Load Cases	120

List of figures

Fig. 1 (a) AMALIA logo; (b) INEGI logo	3
Fig. 2 Remotely Operated Vehicle (a) ROV VICTOR 6000; (b) ROV KIEL 6000.....	7
Fig. 3 Autonomous Underwater Vehicles (a) AUV SENTRY; (b) AUV BLUEFIN-21; (c) AUV DEPTHX; (d) Hybrid Remotely Operated Vehicle Nereus.....	8
Fig. 4 Hybrid Remotely Operated Vehicle 11k from <i>Promare</i>	9
Fig. 5 Benthic Landers - (a) HADAL Lander A; (b) HADAL Lander B	12
Fig. 6 Benthic Landers - (a) ROBIO Lander; (b) Medusa Lander; (c) Bait system <i>e-jelly</i>	13
Fig. 7 Benthic Landers - (a) K/MT100 Lander; (b) DOBO Lander	14
Fig. 8 Benthic Lander DELOS.....	14
Fig. 9 Benthic Landers - (a) OBSEA Lander; (b) K-Lander	15
Fig. 10 Lander Design/Geometry sketches.....	21
Fig. 11 Modelled concepts of lander studied in this project.....	22
Fig. 12 Rendered model of the selected concept - SOLIDWORKS;.....	22
Fig. 13 (a) Central Module representation; (b) Handle detail.....	23
Fig. 14 Technical draw of the housing cylinder	23
Fig. 15 (a) Undistorted pattern; (b) Pincushion distortion; (c) Barrel Distortion (<i>adapted</i> [55])	29
Fig. 16 (a) Angle of coverage representation; (b) Flat port angle of coverage [57].....	30
Fig. 17 Virtual image formation when a dome cover is used [58]	32
Fig. 18 Definition of a plane based on three non-collinear points.....	33
Fig. 19 (a) Concept for the Lunar Excursion Module (May of 1962) [60]; (b) Main support variable length, AMALIA Lander concept.....	34
Fig. 20 (a) TELEDYNE Glass Sphere; (b) Hard Hats Shapes	35
Fig. 21 Pumped Water Variable Buoyancy System;	36
Fig. 22 AMALIA Lander modularity feature	37
Fig. 23 Stress Representation	38
Fig. 24 Thin-walled cylinder cross section representation.....	39
Fig. 25 (a) Cross section of a thick-walled cylinder loaded by both internal and external pressure; (b) Elementary ring with thickness dr	39
Fig. 26 Circumferential and Axial Stresses distribution through the cylinder thick wall.	40
Fig. 27 POM cylindrical tube.....	41
Fig. 28 Impact of External Pressure on Equivalent Stress	42
Fig. 29 Equivalent (von-Mises) Stress (Pa) for 2.23 MPa external pressure – POM	43
Fig. 30 Equivalent Elastic Strain (m/m) for 2.23 MPa external pressure - POM	44
Fig. 31 Total Deformation (m) for 2.23 MPa external pressure - POM.....	44

Fig. 32 Stress/thickness relation on Syntactic Foams.....	47
Fig. 33 Representation of rigidity rings	48
Fig. 34 First model Assumed for the main structure	50
Fig. 35 Relative coordinates of the main joints	51
Fig. 36 Buckling factor function of boundary conditions	54
Fig. 37 - Representation of thickness deformation	60
Fig. 38 Steel Structure	61
Fig. 39 POM structure.....	68
Fig. 40 Drag coefficient as a function of the Reynold's number and geometry.....	70
Fig. 41 Structure dimensions - Steel	70
Fig. 42 Structure dimensions - POM.....	72
Fig. 43 Total Deformation (displacements in m) on Load Case no 1 - Steel Structure	74
Fig. 44 Equivalent Elastic Strain on Load Case no 1- Steel Structure	74
Fig. 45 Equivalent von-Mises Stress (Pa) on Load Case no 1- Steel Structure	75
Fig. 46 Total Deformation (displacements in m) on Load Case no 1 - POM Structure.	75
Fig. 47 Equivalent Elastic Strain on Load Case no 1- POM Structure.	75
Fig. 48 Equivalent von-Mises Stress (Pa) on Load Case no 1- POM Structure.	76
Fig. 49 Total Deformation (displacements in m) on Load Case no 2 - Steel Structure.	77
Fig. 50 Equivalent von-Mises Stress (Pa) on Load Case no 2- Steel Structure.	77
Fig. 51 Equivalent von-Mises Stress (Pa) on Load Case no 2- Steel Structure.	78
Fig. 52 Total Deformation (displacements in m) on Load Case no 2 - POM Structure.	78
Fig. 53 Equivalent Elastic Strain on Load Case no 2- POM Structure.	78
Fig. 54 Equivalent von-Mises Stress (Pa) on Load Case no 2- POM Structure.	79
Fig. 55 Total Deformation (displacements in m) on Load Case no 3 - Steel Structure.	80
Fig. 56 Equivalent Elastic Strain on Load Case no 3- Steel Structure.	80
Fig. 57 Equivalent von-Mises Stress (Pa) on Load Case no 3- Steel Structure.	80
Fig. 58 Total Deformation (displacements in m) on Load Case no 3 - POM Structure.	81
Fig. 59 Equivalent Elastic Strain on Load Case no 3- POM Structure.	81
Fig. 60 Equivalent von-Mises Stress (Pa) on Load Case no 3- POM Structure.	81
Fig. 61 Total Deformation (displacements in m) on Load Case no 4 - Steel Structure.	83
Fig. 62 Equivalent Elastic Strain on Load Case no 3- Steel Structure.	83
Fig. 63 Equivalent von-Mises Stress (Pa) on Load Case no 4- Steel Structure.	83
Fig. 64 Total Deformation (m) on Load Case no 4 - POM Structure.....	84
Fig. 65 Equivalent Elastic Strain on Load Case no 4- POM Structure.	84
Fig. 66 Equivalent von-Mises Stress (Pa) on Load Case no 4 - POM Structure.	84

Fig. 67 Total Deformation (displacements in m) with Drag force- Steel Structure.....	86
Fig. 68 Equivalent Elastic Strain with Drag force - Steel Structure.....	86
Fig. 69 Equivalent von-Mises Stress (Pa) with Drag force - Steel Structure.....	87
Fig. 70 Total Deformation (displacements in m) with Drag force - POM Structure.....	87
Fig. 71 Equivalent Elastic Strain with Drag force - POM Structure.....	87
Fig. 72 Equivalent von-Mises Stress (pa) with Drag force - POM Structure.....	88
Fig. 73 The first 6 eigen-frequencies for the proposed Steel Structure.....	89
Fig. 74 Mode 1 Shape – 48.283 Hz	89
Fig. 75 Mode 2 Shape - 48.397 Hz	90
Fig. 76 Mode 3 Shape – 79.789 Hz	90
Fig. 77 Mode 4 Shape – 79.789 Hz	90
Fig. 78 Mode 5 Shape – 108.87 Hz	91
Fig. 79 Mode 6 Shape – 108.96 Hz	91
Fig. 80 The first 6 eigen-frequencies for the proposed POM Structure.....	92
Fig. 81 Mode 1 Shape – 23.301 Hz	92
Fig. 82 Mode 2 Shape – 23.4 Hz	92
Fig. 83 Mode 3 Shape – 32.279 Hz	93
Fig. 84 Mode 4 Shape – 32.327 Hz	93
Fig. 85 Mode 5 Shape - 42.927 Hz	93
Fig. 86 Mode 6 Shape - 43.037 Hz	94
Fig. 87 Stress in function of External Pressure - Structural Steel	103
Fig. 88 Stress in function of External Pressure - Stainless Steel	104
Fig. 89 Stress in function of External Pressure - Titanium.....	105
Fig. 90 Stress in function of External Pressure - Aluminium	106
Fig. 91 Comparison of metal material stress.....	107
Fig. 92 Stress in function of External Pressure – PTFE	108
Fig. 93 Stress in function of External Pressure – FEP	109
Fig. 94 Stress in function of External Pressure - FPA.....	110
Fig. 95 Stress in function of External Pressure – LDPE	111
Fig. 96 Stress in function of External Pressure HDPE	112
Fig. 97 Stress in function of External Pressure - UHMW-PE.....	113
Fig. 98 Stress in function of External Pressure - PBI.....	114
Fig. 99 Comparison of polymeric material stress	115
Fig. 100 Stress in function of External Pressure - 94Al ₂ O ₃	116
Fig. 101 Stress in function of External Pressure - 96 Al ₂ O ₃	117

Fig. 102 Stress in function of External Pressure - Si ₃ N ₄	118
Fig. 103 Comparison of ceramic material stress	119
Fig. 104 Main Support (Upper Part) - Load Case 1 - Steel	120
Fig. 105 Main Support (Upper Part) - Load Case 2 - Steel	120
Fig. 106 Main Support (Upper Part) - Load Case 3 - Steel	121
Fig. 107 Main Support (Upper Part) - Load Case 4 - Steel	121
Fig. 108 Main Support (Upper Part) - Load Case 1 - POM	122
Fig. 109 Main Support (Upper Part) - Load Case 2 - POM	122
Fig. 110 Main Support (Upper Part) - Load Case 3 - POM	123
Fig. 111 Main Support (Upper Part) - Load Case 4 – POM	123
Fig. 112 Main Support (Inferior Part) - Load Case 1 - Steel	124
Fig. 113 Main Support (Inferior Part) - Load Case 2 - Steel	124
Fig. 114 Main Support (Inferior Part) - Load Case 3 - Steel	125
Fig. 115 Main Support (Inferior Part) - Load Case 4 - Steel	125
Fig. 116 Main Support (Inferior Part) - Load Case 1 - POM	126
Fig. 117 Main Support (Inferior Part) - Load Case 2 - POM	126
Fig. 118 Main Support (Inferior Part) - Load Case 3 - POM	127
Fig. 119 Main Support (Inferior Part) - Load Case 3 – POM	127
Fig. 120 Secondary Support - Load Case 1 - Steel	128
Fig. 121 Secondary Support - Load Case 2 - Steel	128
Fig. 122 Secondary Support - Load Case 3 - Steel	129
Fig. 123 Secondary Support - Load Case4 - Steel	129
Fig. 124 Secondary Support - Load Case 1 - POM	130
Fig. 125 Secondary Support - Load Case 2 - POM	130
Fig. 126 Secondary Support - Load Case 3 - POM	131
Fig. 127 Secondary Support - Load Case 4 – POM	131
Fig. 128 Columns - Load Case 1 - Steel	132
Fig. 129 Columns - Load Case 2 - Steel	132
Fig. 130 Columns - Load Case 3 - Steel	133
Fig. 131 Columns - Load Case 4 - Steel	133
Fig. 132 Columns - Load Case 1 - POM	134
Fig. 133 Columns - Load Case 2 - POM	134
Fig. 134 Columns - Load Case 3 - POM	135
Fig. 135 Columns - Load Case 4 - POM	135
Fig. 136 Superior Hoop - Load Case 1 - Steel	136

Fig. 137 Superior Hoop - Load Case 2 - Steel.....	136
Fig. 138 Superior Hoop - Load Case 3 - Steel.....	137
Fig. 139 Superior Hoop - Load Case 4 - Steel.....	137
Fig. 140 Superior Hoop - Load Case 1 - POM.....	138
Fig. 141 Superior Hoop - Load Case 2 - POM.....	138
Fig. 142 Superior Hoop - Load Case 3 - POM.....	139
Fig. 143 Superior Hoop - Load Case 4 - POM.....	139

List of tables

Table 1 Active Manned Submersibles	5
Table 2 ROV Groups [15]	6
Table 3 Mechanical Properties and Ratio Price/Weight for different materials [5][35][36][37].	16
Table 4 Mechanical properties and Price/kg ratio of different metallic materials [37].	24
Table 5 Mechanical properties and Price/kg ratio of different composite materials [37].	25
Table 6 Mechanical properties and Price/kg ratio of different ceramic materials.	26
Table 7 Mechanical properties and Price/kg ratio of fluoropolymers [49].....	27
Table 8 Mechanical properties and Price/kg ratio of polyethylene polymers [49].	27
Table 9 Mechanical properties and Price/kg ratio of POM and PBI [49].	28
Table 10 Mechanical properties of MZ-24	28
Table 11 Comparison of angles of coverage underwater with a flat port and air	31
Table 12 Mechanical properties and Price/kg ratio of Glass and Acrylic [1]	32
Table 13 Pumped Water Variable Buoyancy System Valves functioning	36
Table 14 Tube thickness for each material.....	41
Table 15 Equivalent Stress resultant from different loads.....	42
Table 16 Maximum ANSYS values for POM.....	44
Table 17 Depth range for each material.....	45
Table 18 Ratio €/m for each material for maximum depth	46
Table 19 Syntactic Foam ANSYS analysis.....	46
Table 20 ANSYS and theoretical values comparison for POM cylinder.....	49
Table 21 Imperfection Factors.....	55
Table 22 different classes of profiles.....	56
Table 23 Buckling analysis for steel frame main supports	56
Table 24 Buckling analysis for steel frame main supports - chosen profile.....	57
Table 25 Buckling analysis for steel frame secondary supports	58
Table 26 Buckling analysis for steel frame secondary supports - chosen profile	59
Table 27 Bending radius for steel profiles.....	60
Table 28 Steel Structure submersed volume	61
Table 29 Buckling analysis for POM frame main supports.....	63
Table 30 Safety factor for polymeric buckling analysis	64
Table 31 Buckling analysis for POM frame main supports - chosen profile.....	65
Table 32 Buckling analysis for POM frame secondary supports - chosen profile	66
Table 33 Structural Components Stress - Load Case 1.....	76

Table 34 Structural Components Stress - Load Case 2	79
Table 35 Structural Components Stress - Load Case 3	82
Table 36 Structural Components Stress - Load Case 4	85
Table 37 Steel Structure Vibration modes' frequencies.....	89
Table 38 POM Structure Vibration modes' frequencies	91
Table 39 Maximum Stress Values - Steel Structure	95
Table 40 Maximum Stress Values - POM Structure	96
Table 41 Structural Steel Analysis Values.....	103
Table 42 Stainless Steel Analysis Values.....	104
Table 43 Titanium Analysis Values	105
Table 44 Aluminium Analysis Values	106
Table 45 PTFE Analysis Values	108
Table 46 FEP Analysis Values	109
Table 47 FPA Analysis Values.....	110
Table 48 LDPE Analysis Values.....	111
Table 49 HDPE Analysis Values.....	112
Table 50 UHMW-PE Analysis Values	113
Table 51 PBI Analysis Values.....	114
Table 52 94Al ₂ O ₃ Analysis Values.....	116
Table 53 96Al ₂ O ₃ Analysis Values.....	117
Table 54 Si ₃ N ₄ Analysis Values	118

1 Introduction

1.1 Ocean Exploration

The ocean carries a fundamental role in human life, from the climate pattern, to the air we all breathe [1]. However, despite its value, the ocean remains highly unknown, inhibiting us from exploiting full advantages of all its potentialities.

Citing The Ocean Portal Team, *“Deep below the ocean’s surface is a mysterious world that takes up 95% of Earth’s living space.”* [2]. The deep-sea homes a large diversity of geological features and living beings, whose adaptation to the harsh medium characterized by cold temperatures, high pressure and low light, created unique and fascinating creatures with special features, that can be exploited in many areas of industry.

The Ocean Exploration, mainly laying on deep-sea, will give us new information on marine geology and biology, which may allow to discover new potentialities and resources that so far are not in use. Water quality, mineral resources, biological stock evaluation, biodiversity protection against invasive species, determination of highly valued species, are some examples of potentialities ocean exploration offers.

One of the main fields that deserves an increased focus are benthic macroalgae, given their industrial potential and their constitution that can be highly valuable in food, feed, pharmaceutical and cosmetic industries. The main advantages of macroalgae as a biological resource are their fast grown, the ability of growth in all climatic zones, their high content of valuable carbohydrates, proteins and lipids [3]. Regarding the ecological functioning of marine ecosystems, these seaweeds have an essential part since they are a habitat for other leaving creatures [4] and they are one of the main producer of nutrient in the benthic food chain, therefore, a rigorous control of seaweeds may have direct effect in the upper food chain levels.

From detecting a specific specie to mere curiosity, the study of the deep ocean is something that have always intrigued scientists and engineers, however the water barrier that split us from the seafloor has been a major challenge [5].

Cornelis Drebbler built the first submarine ever made, in 1623: a machine comprised by an outer hull of greased leather over a wooden frame that could operate to depths of 3.5 to 5 *m*. One of the first successful technologies developed to know the fauna and flora living on the sea bottom, and turning point for deep-sea exploration paradigm, was the *marine biology dredge*, by naturalist Sir Charles Wyville Thomson in 1830, used to collect sample

organisms from seafloor. With this technology, simply composed by a net and a digging mechanism, samples were collected at 548.64 meters deep in the ocean (300 *fathoms*). Up to date, it was believed that there were no life under 200 m [6], since there was no light. With this discovery, there was a bloom in deep-sea exploration technologies development.

The first advanced engineered technology came up in the 1950's with the first untethered manned submersibles, like ALVIN. The first unmanned vehicles appeared one decade after, and the last 25 years of the XX century was the main period for development of Autonomous Unmanned Vehicles [5]

1.2 Algae-to-MARket-Lab IdeAs - TECMAR - INEGI

This project can be seen as a part of a project entitled AMALIA. This project counts on the participation of higher education institutions, research units, companies and local development associations, in particular INEGI - Institute Of Science And Innovation In Mechanical And Industrial Engineering, which is a research and technology organization, acting as an interface between the university and the industry, and it is mainly focused on applied research and development, innovation and technology transfer activities [7]. The Sea Technologies group (TECMAR), an INEGI sector, where this dissertation project was developed, aims to promote and create technologic solutions that can fit industrial needs, particularly economy of Sea, through innovation of traditional marine activities or through the emergence of new high value economic activities (on the domain of biotechnology, energy, robotics). This working group skills comprise the hydrodynamic evaluations of any offshore and marine structures (such as SPARs, FPSOs, TLP, energy converters), design and mooring systems global performance analysis, dynamic stability analysis, among others.⁴

The surge of invasive macroalgae species has becoming a major concern both in an ecological and economic level. These species displace natives, causing the loss of their genotype, affecting marine habitats, ecosystem processes, and food-web properties, having impact in human health. In addition, all of these lead to an economic loss [8].

Instead of looking at this event has a threat, is possible to see it as a promising opportunity.

The AMALIA (Algae-to-Market Lab IdeAs) project aims to exploit these seaweed species, in the northwest of the Iberian Peninsula, considering their industrial potential and their compounds that can be used in the food, feed, pharmaceutical and cosmetic industries, thus, increasing their value and contributing to the economy. Besides, the effective control of these macroalgae will lead to an improvement in the quality of the oceans.

To monitor the appearance of these macroalgae, advanced engineering and imaging systems and solution will be deployed into the seafloor to provide real-time information regarding the appearance and quantities of algae.



(a)



(b)

Fig. 1 (a) AMALIA logo; (b) INEGI logo

1.3 Project objectives

Underwater operations in deep-sea have a major importance in the knowledge of both chemical and physical properties of water, and also in the study of its biodiversity. The seabed is the habitat for a great variety of organisms, therefore constitutes a distinct stratum for benthic life [9].

The analysis of this environment can be conducted either *in-situ* or *ex-situ*, regarding where this analysis is taken: directly from deep-sea or from a samples extracted from their natural habitat, respectively.

In *ex-situ*, when brought up to the surface, samples are subjected to hydrostatic pressure and temperature variations; it consequently becomes difficult to collect accurate data for further analysis. It is therefore desirable to carry out deep-sea experiments and measurements *in-situ*, avoiding artefacts induced by disturbance. Such operations are carried out using manned or unmanned vehicles that explore the sea floor whose main goal is to collect all data needed while minimizing the possible perturbation it may cause to seabed and samples during both landing and operation [9].

The main objective of this project is to design a submersible modular system for seashore monitoring, capable of conducting *in-situ* analysis. During this report, the procedure to select the system geometry and construction materials for different components it will be describer, as different load cases. Other parameters were considered as vertical movements', hydrodynamic design, methods for launching, landing and recovering, sampling, observation and measurement techniques, choice of electronic components, as sensors, and, for last, energy requirements, are specific points to be considered when designing these vehicles in order to obtain successful results, regarding data collect [9].

2 State-of-the-art

2.1 Deep-sea Technologies

Explore seafloor is an appealing topic since it is the largest geographic feature in our planet. It will therefore boost the need for new and more developed marine technologies for ocean exploration and discovery [10].

These technologies can be distinguished either if they are manned or not, or based on its autonomy.

2.1.1 Manned Submersibles

Manned vehicles are vehicles that drive with, at least, one operator into deep-sea. For *Manned Underwater Vehicles Committee*, these vehicles can be divided in two different groups: GROUP 1 – HADAL Depth, for a working depth over 1000m; and GROUP 2 – Deep Ocean, for working depth between 250 and 1000 m [11]. With these deep diving capabilities, manned submersible vehicles are able to explore approximately 98% of the ocean floor [10].

The main advantages of manned submersibles lies '*in placing the human eye, hand and brain at the point of observation*', as it is said by R.A. Geyer in *Submersibles and Their Use in Oceanography and Ocean Engineering* [12]. This means that a closer and more direct look can be took, allowing one to pick out more details, thus improving accuracy when compared to less sophisticated remote system. The most durable virtue for manned submarines is '*(...) the ability to react, to pursue the unexpected, to alter plans quickly and continuously in response to a changing situation (...)*'. Despite all this, it requires complex, expensive, and non-optional life-support and safety systems, thus less preferable for deep-sea exploration. An additional disadvantage, when compared to other technologies, is the limitation exploration period, once it does not allow long-term operations [12].

One of the most known manned submersible project is de *Deepsea Challenger*, where a partnership between *Woods Hole Oceanographic Institution (WHOI)* and James Cameron an explorer and filmmaker, culminated in a one single person expedition to the deepest point of the world's Ocean, the Mariana Trench, at the depth of nearly 11 000 m. The Deepsea Challenger construction was secretly made under the leadership of the Australian engineer Ron Allum, in a partnership with the National Geographic Society and with support from Rolex.

The vehicle descend speed was about 2.6 m/s, taking 2 hours to reach the ocean bottom. There, due to the sub battery life, the expeditor had 6 hours to '*(...) explore the deep-ocean frontier for clues new life-forms and the forces that shape our planet (...)*', as it is said in *NATIONAL GEOGRAPHIC* magazine [13], which reported the project. On ascent, as a result of the sub technology, the velocity of the vehicle achieved approximately 3.6 m/s, reaching the surface one hour after leaving the seafloor.

Manned submersible vehicles may present some risks and dangers to its operator(s). Some of those where mentioned by James Cameron. The possibility of implosion due to miscalculated sphere project; penetration failure; freezing; fire; adrift, when only some of the ballast weight, whose liberation allows the sub to ascend, drops, causing the hazard of getting

lost in the water column; hypothermia and hyperthermia are some examples. All of these present a major life risk [13].

Alongside this manned submersible, there are many other currently active vehicles, Table 1 Active Manned Submersibles presents some of the models.

Table 1 Active Manned Submersibles

			
	<i>Deepsea Challenge</i>	<i>JIAOLONG</i>	<i>SHINKAI 6500</i>
Depth	11 000 m	7000 m	6500 m
Capacity	1	3	
Operator	Woods Hole Oceanographic Institution (WHOI)	China National Deep Sea Centre	Japan Agency for Marine-Earth & Tech Science (JAMSTEC)
Country	USA	China	Japan
			
	<i>MIR 1</i>	<i>NAUTILE</i>	<i>ALVIN</i>
Depth	6000 m	6000 m	4450 m
Capacity	3	3	3
Operator	PP Shirshov Inst. of Oceanology (RAS)	IFREMER	Woods Hole Oceanographic Institution (WHOI)
Country	Russia	France	USA

2.1.2 Remotely Operated Vehicle

According to Remotely Operated Vehicle Committee of the Marine Technology Society, a Remotely Operated Vehicle, further referred to as ROV, is ‘(...) *a tethered underwater robot that allows the vehicle's operator to remain in a comfortable environment while the ROV works in the hazardous environment below.*’

A ROV system is composed by a vehicle connected to the control van via a tether, which comprises a group of cables responsible to transmit video and data signal back and forth between the vehicle and the operator, as well as carry electrical power, a handling system for cable dynamics control, a launch system and power supplies. Additional equipment needed for data collecting may be assembled to the vehicle, fitting its requirements. Such equipment can be video cameras, sonar systems, lights, or an articulating arm [14].

Due to high drag on the vehicle and tether cable, ROVs are slow in speed, however, they can move with full control along all axes.

Concerning size, depth capability, on-board horsepower, and whether it has or not hydraulic components, ROV systems can be grouped as it is shown in the following table.

Table 2 ROV Groups [15]

Class	Depth	Type	Power	Observations
Micro Observation	<100 m	• Low Cost • Small • Electric	<5 hp	• Less than 3 kg • An alternative to a diver for places he may not be able to reach
Mini Observation	<300 m	• Small • Electric	<10hp	• Around 15 kg
Light/Medium Work Class	<2000 m	• Medium • Electric/Hybrid	<100 hp	• Can be made from polymers (e.g., polyethylene)
Observation/Light Work Class	<3000 m	• High Capacity Electric	<20 hp	
Heavy Work Class /Large Payload	<3000 m	• High Capacity • Electric/Hybrid	<300 hp	• Ability to carry at least two manipulators
Observation/Data Collection	>3000 m	• Ultra-Deep • Electric	<25 hp	
Heavy Work Class /Large Payload	>3000 m	• Ultra-Deep • Electric/Hybrid	<120 hp	• Ability to carry at least two manipulators

Some ROVs are capable of achieving 6000m depth, for example KIEL 6000 from GEOMAR [16], presented on Fig. 2 (b) used in a ‘live-boating-mode’, by means of a deep-sea glass fibre cable; and VICTOR 6000 [17], show on Fig. 2(a), from IFREMER. With this working depth, both of these ROVs can reach more than 90% of the seafloor.

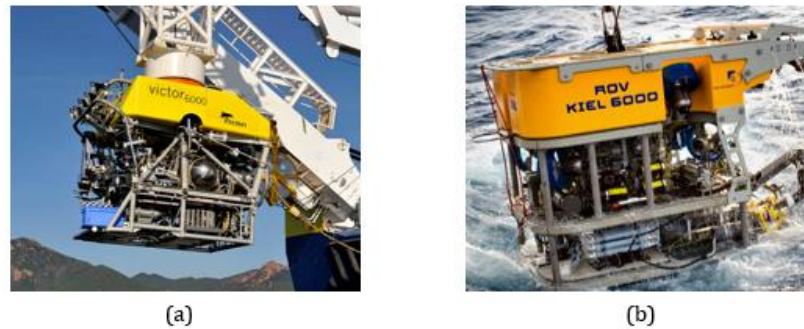


Fig. 2 Remotely Operated Vehicle (a) ROV VICTOR 6000; (b) ROV KIEL 6000

2.1.3 Unmanned Submersibles

Autonomous Underwater Vehicles

Autonomous Underwater Vehicles, further referred to AUVs, are unmanned and self-propelled vehicles, usually deployed into the sea from a surface vessel. AUVs can operate independently for periods of a few hours to several days [18].

These vehicles differ from ROVs in the way that AUVs are un-tethered to the host vessel; therefore, their speed, mobility and spatial range are less constrained. Nonetheless, ROVs present advantages in real-time communication and power transmission.

AUVs are able of maintaining a linear trajectory through the seawater, just unlike submarines gliders, which due to its variable buoyancy system, profile the water column in a sawtooth pattern by shifting small amounts of ballast to dive and climb. The glides moves both horizontally and vertically. These vehicles can follow a pre-programmed course and they are able to navigate using a combination of Ultra Short Base Line acoustic communication, GPS positioning, and inertial navigation; or using arrays of acoustic beacons on the seafloor.

Operations with these vehicles cannot be done everywhere; therefore, there is a need to consider some particular aspects. Usually, the maximum speed of AUVs used for sea exploring is 1.5-2.0 *m/s*, this value can be influenced by currents, namely tidal, approaching or exceeding these velocities. This may affect data quality collect due to navigational drift [18].

The ability to operate close to the seabed, less than 5 *m* altitude in low relief, allows these vehicles to collect seafloor mapping, profiling and imaging data of far higher spatial resolution [18].

AUVs may have a torpedo-shape, the most usual, or a more complex configuration, allowing them to move in slower motion and across complex terrain. These are commonly related with hovering navigation and hybrid AUV/ROV capabilities [10].

Some AUVs examples are Sentry from WHOI, capable of exploring down to 6000 meter, presents a complex shape, allowing it to carry a range of devices. Sentry AUV (Fig. 3 (a)) produces '(...) *bathymetric, sidescan, subbottom, and magnetic maps of the seafloor and is capable of taking digital bottom photographs in a variety of deep-sea terrains such as mid-*

ocean ridges, deep-sea vents, and cold seeps at ocean margins(...)' as it is said in WHOI website. It also can locate and quantify hydrothermal fluxes [19].

Another example is the Bluefin-21 (Fig. 3(b)), from Bluefin Robotics, a modular vehicle able to carry different sensors and payloads. It presents high energy capacity that grant long-term operations at big depths, up to 4500 meters [20]. DEPTHX AUV - *Deep Phreatic Thermal eXplorer* (Fig. 3 (c)) from Stone Aerospace is another AUV example. It was the first mobile robotic system to implement 3D-SLAM (Simultaneous Localization and Mapping) as part of real-time navigation engine, the first to explore and map a subterranean cavern – a hydrothermal spring, and the first robotic system able to decide autonomously the precise moment and place to collect biological data [21].

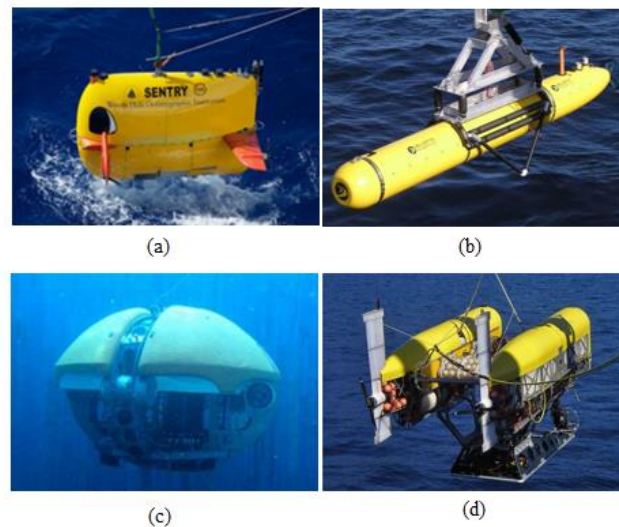


Fig. 3 Autonomous Underwater Vehicles (a) AUV SENTRY; (b) AUV BLUEFIN-21; (c) AUV DEPTHX; (d) Hybrid Remotely Operated Vehicle Nereus

Hybrid Remotely Operated Vehicle

The AUV/ROV hybrid, commonly known as HROV (*Hybrid Remotely Operated Vehicle*), e.g. Nereus (Fig. 3 (d)), from WHOI, which was the highest level of development concerning the deep-sea unmanned vehicles. It was able to operate in two different modes, for deeper or more expansive areas it operates untethered as an AUV, and it could be converted to a ROV to enable close-up images and sampling [22]. Nereus was lost at sea while exploring the Kermadec Trench at approximately 10,000 meters [23].

Another example of AUV/ROV hybrid is the Promare's 11k (Fig. 4). This is a low cost solution, able of achieving 11 000 meters, with a total weight of 60 kilograms. What controls all vehicle functions is a single board computer, which, in AUV mode the system is completely programmed by it. When it comes to ROV mode, the vehicles control is via a fibre-optic connection that provides two channel of communication: one for operator command and sensor measurements, the other aims to provide real-time high definition video signals [24].

The most peculiar feature of this technology is that most of the hardware is embedded in a glass sphere, held in place by a thermoplastic structure [24].



Fig. 4 Hybrid Remotely Operated Vehicle 11k from *Promare*

Benthic Landers

It is known as '*Benthic Lander*', commonly just *lander*, any unmanned, autonomous and instrumented vehicle, which is deployed in the seafloor, unattached to any cable, to gather physical and chemical variables *in situ* over a period of time. The lander operations may be taken in a few days, for biological studies, to several years, for physical oceanography studies [9][10].

The deployment can be in a free fall mode or in a high precision location to measure geomorphological features using a special design launching device connected to a crane from the ship or vessel in the surface. The climb is usually made by the release of ballast weights or using variable buoyancy systems. They are then recovered and put on board, lifted by cranes [9].

2.2 Benthic Landers

2.2.1 The Role of Benthic Landers in the science fleet

Given the concerns regarding manned and tethered submersibles, there was a need to develop a vehicle able to perform underwater without any physical connection to the surface. However, underwater operations present many engineering challenges, for example, electromagnetic waves that hardly spread underwater, creating communications barriers between the surface and the vehicle [5]. Therefore, the emergence of a completely autonomous vehicle or structure would ease the deep-sea exploration. To this end, AUVs and Benthic Landers appear face to ROVs and Manned Submersibles.

AUVs and Benthic Landers can now easily be deployed into the ocean and retrieve the required data for a specific operation, however, when it comes to longer terms, or time lapsed comparison of static images, benthic landers present a clearly advantage. The fact of being completely static down the sea bottom, allows the lander to capture easily the same image over the time, allowing us to monitor the progress of macroalgae growth, for example. An AUV, for instance, could capture the same picture as well, however, it would need a much more power consumption, and a simple disturbance in the current could damage the image.

Another advantage of benthic landers is the fact that they can carry experiments and measurements *in situ*, reducing the possibility of artefacts on collected samples [9].

2.2.2 Types of Landers

As inserted in autonomic and unmanned deep-sea technology group, the depth rate of the site the lander will operate characterizes its category [5]. There are, therefore, the following categories:

- Shallow Water;
- Mid-water;
- Deep-Water.

Shallow Water Landers

Shallow Water Landers usually operate at a maximum depth of 500 m [5], in a near offshore zone. Here, the main loads applied on the structure are dynamic forces caused by current or wind, thus it is very important to minimize the projected area of all equipment in order to reduce drag effect. The hydrostatic pressure they have to bear, compared to other loads, is negligible.

These landers are usually small and consequently the power supply equipment they carry may limit the operation term.

Mid-Water Landers

This category refers to Landers rated up to 2500 m [5]. In order to handle the hydrostatic pressures at the seafloor, these landers are typically voluminous. Since the water current at these depths is negligible, also the drag effect is, therefore the area is not a major facto to take in count.

Deep-water Landers

These landers operate at depths of more than 2500 m [5]. Here, the oceanographic pressures are enormous and thus the whole structure must be robust. To compensate its own weight on the recovery, it must have a complex variable buoyance system, however it my carry some risk of implosion of some of its equipment, e.g. glass spheres [9].

2.2.3 Mechanical and Material Issues

Lander Design - Lander Structural Types

Regarding its applications and measuring instruments they carry, landers come in different shapes and sizes. They can generally be arranged in micro open-frame landers, macro open-frame landers, and flat shape landers [10].

Micro open-frame landers

A modular structure capable of carry instruments for monitoring and measuring *in situ*. These landers have an open-framed structure to minimize the effects of horizontal drag due to sea currents, allowing water to flow through the instruments and sensors payload. This type of lander has ballast weights in its bases that will be left in the bottom of the sea after its operation.

Examples of this kind of structures are the free falling baited landers HADAL – Lander A and HADAL – Lander B from WHOI, ROBIO and Medusa.

HADAL – Lander A and HADAL – Lander B

HADAL-Lander A, Fig. 5 (a), was equipped with a 3CCD hi-resolution colour video camera, mounted vertically at an altitude of 1 m, providing a visible area of 0.62 x 0.465 m, illuminated with two 50 W halogen lamps. I also had a Conductivity, Temperature and Depth (CTD) sensor that recorded temperature, salinity, and pressure every 10 seconds after deployment. Both camera and CTD probe were programmed *a priori* its deployment and all data were download after lifting and recovering [25].

The lander deliver system was comprise with the frame structure, made out of aluminium, with a mooring in which 6 buoyancy modules where couple off-line, three ballast weights and two acoustic release systems to discard the ballast weight. The total weight of the system during the free fall was 135 kg and reached the descent velocity of approximately 0.83 m/s. After the observational period, a unique acoustic command was sent in order to release the ballast, allowing the lander to ascend at the 0.5 m/s by virtue of the positive buoyant mooring [25].

HADAL-Lander A reached depths of 9800 m, prior to its loss in 2009 [25].

The lander deliver system of HADAL – Lander B, Fig. 5 (b), was similar to A, the only exception was that it comprised two acoustic release systems to jettison the ballast weight. These two are used in the event of failure of one of them. This lander reached 8074 m depth [26].

Despite this difference, the monitoring and observation equipment was different as well. It was equipped with a 5 megapixel digital camera and a CTD sensor. The camera was mounted vertically at an altitude of 1 m, providing a visible area of 0.62 x 0.465 m, as well [26].

Both of them were also equipped with small baited funnel traps, located on the extremity of each of the 3 legs or on the mooring line, used to collect samples [25][26].

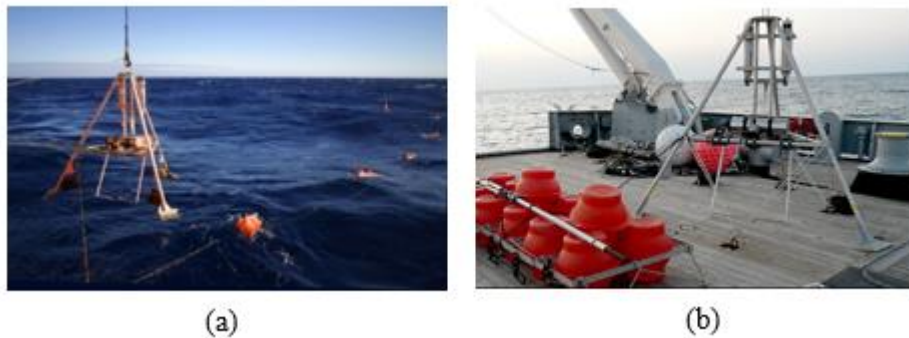


Fig. 5 Benthic Landers - (a) HADAL Lander A; (b) HADAL Lander B

ROBIO

The RObust BIODiversity, *ROBIO* (Fig. 6 (a)), is a baited lander built to monitor the fauna of benthic communities near possible subsea oil and gas exploration sites, autonomously, capable of reaching 3000 *m*. These landers are equipped with 3 Megapixel digital camera, able of taking up to 1400 frames per year, and other sensors to characterize currents properties such as direction, velocity and salinity [10].

It has two different deployment methods: i) tethered two meters above seabed with the camera positioned vertically, pointed downward looking at the bait; ii) landed on the sea-floor, with the camera pointed outward [27].

Medusa

The Medusa lander (Fig. 6 (b)) is deployed in a free fall mode and recover in the surface after the acoustic signal that allow the ballast release. It uses red light illumination, which is not visible to most of deep-sea fauna, and it is equipped with a camera system that is able to amplify both this dim illumination as bioluminescence. This camera will record for 24-hour time intervals for up two days [28]. The data is not transmitted in real time; therefore, the analysis can only be made after its recovery [29].

This lander uses two different bait systems: a bait box on the end of the bar, mounted directly in front of the camera; or a lure simulating a bioluminescent jelly to attract large predators in the area. These 16 blue LEDs system called *e-jelly* (Fig. 6 (c)) simulate the signal that *Atolla* jellyfish produces in need for help [28].

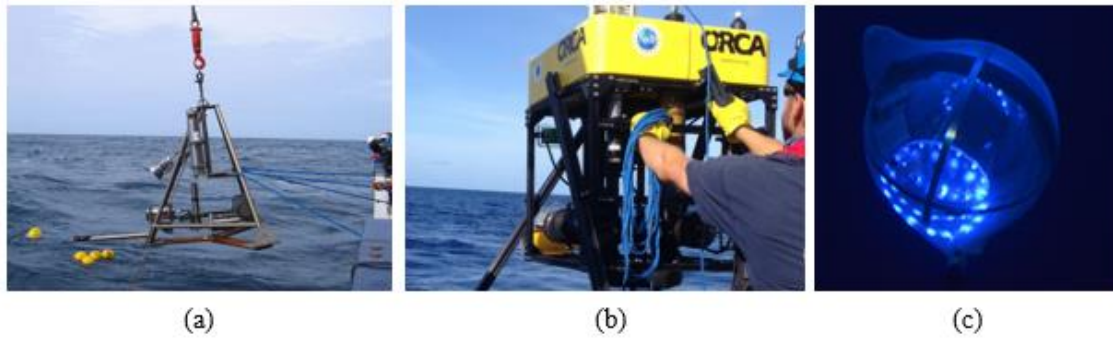


Fig. 6 Benthic Landers - (a) ROBIO Lander; (b) Medusa Lander; (c) Bait system *e-jelly*

Macro open-frame landers

These type of benthic landers are similar to the prior referred. However, these are used for longer term deployments with a major number of equipment, which leads to a considerable size increase.

Some examples of these landers are K/MT 100 from KUM Kiel, DELOS and DOBO, both from Aberdeen Ocean Lab.

K/MT 100

The K/MT 100 Lander Fig. 7 (a) comprises an open framed titanium structure with three legs, float units and measuring and monitoring equipment. The deployment is due the negatively buoyant ballasted lander that descends to the sea floor at a speed of 0.5 to 1.0 m/s where it lands. After the research period, steel weights are dropped through time release or an acoustic signal, switching the buoyancy to positive. After ascending to the surface, the research vessel recovers the lander. Its maximum operation depth is 6000 m [30].

Apart from the camera, the lander is also equipped with a benthic chamber. It is within the enclosed environment of this chamber where *in situ* experiments are carried out [9].

DOBO

Deep Ocean Benthic Observatory Fig. 7 (b) lander, known as DOBO, from Aberdeen Ocean Lab, is a benthic lander, which comprises a titanium frame, able to stay immersed for long term, up to 10 months, at a maximum depth of 2500 m operation. It has bait release systems that releases portions of bait at pre-programmed periods. This bait will attract benthic fauna, such as fish and invertebrates, photographed then by the camera incorporated in the lander [10].

This lander does not allow real-time data access, however, it has six to twelve months of data storage capability.

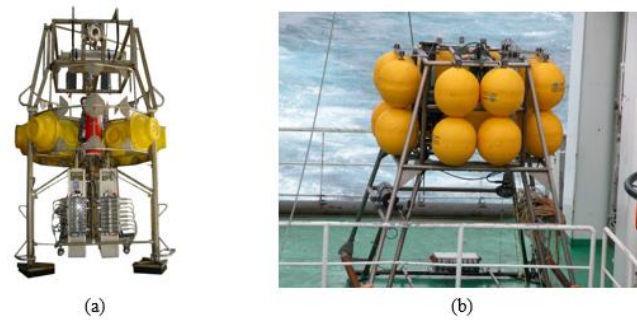


Fig. 7 Benthic Landers - (a) K/MT100 Lander; (b) DOBO Lander

DELOS A and DELOS B

The DELOS (Fig. 8) system comprises two environmental monitoring platforms: DELOS A, located within 50 meters of a seafloor wellhead; and DELOS B, located 16 km from sea floor structure. Each one of them comprises two parts: the seafloor anchor station, a robust triangular shaped structure made out of glass fibre designed to withstand 25 years; and observation modules, with a power storage enough for autonomous operation for 12 months. After this period, each platform requires intervention to recover observatory modules to the surface for calibration, data offload and servicing, for this operation is used a ROV. Therefore, for this system is not required a buoyancy system or other system for further recover [31].

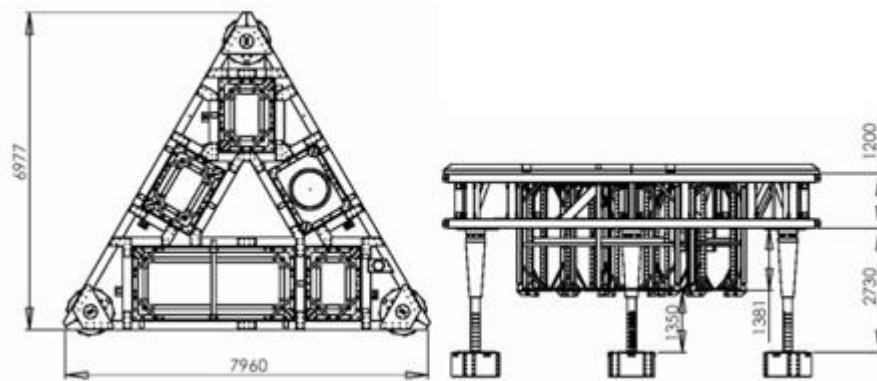


Fig. 8 Benthic Lander DELOS

OBSEA

OBSEA lander (Fig. 9 (a)), a part of the FixO3 project, Fixed point Open Ocean Observatory, is a stainless steel structures designed for not permitting unauthorized manipulation, therefore protecting all the equipment. The cylinder that holds the control electronics is within this metal housing and it was projected to withstand water pressure at 300 m depth [32]. It will provide the interface between the marine cable, made up of six mono mode fibre optics for data transmission and two conductor tubes for power supply, and other instruments connected to the observatory [33].

After deployed, these platforms will be connected to shore by the marine cable, which allows continuous data flow and to supply enough power to operate [33].

Flat Shape Landers

The prior design concepts lies in the minimization of the projected surface area, orthogonal to the current movement direction, in order to reduce horizontal drag effect. This new concept uses a flat shape structure that seats in the seafloor. Thus, due to existence of the boundary layer and the fact that the wave induced effects decrease exponentially with the increase of the depth, this type of lander present low drag exposures. In addition, it uses its hydrodynamic effect to generate downward forces that anchors the system with the increase of seafloor currents [10].

K-Lander

This autonomous seabed sensor carrier, from KONGSBERG Modular Subsea Monitoring-Network (Fig. 9 (b)), has complete flexibility on sensor type installed and electric power from integral battery packs. It is a robust steel framed structure, which allows the implementation of multiple data source sensors allowing a wide area coverage [34].

It can operate for up to 24 months with low intervention at a depth rating of 2000 *m*.

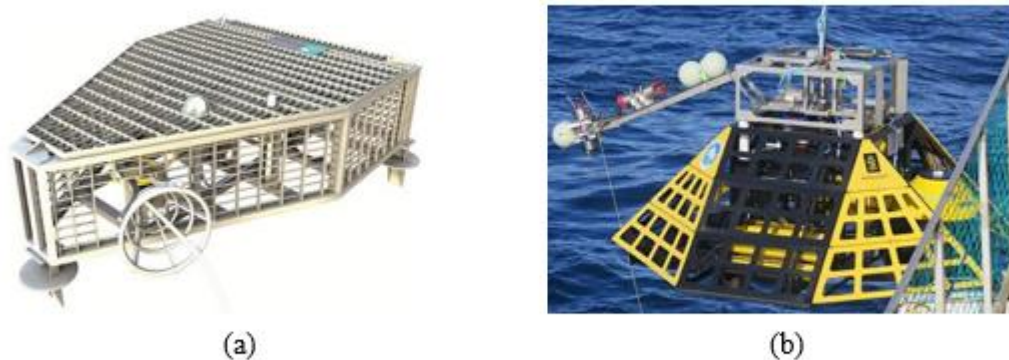


Fig. 9 Benthic Landers - (a) OBSEA Lander; (b) K-Lander

Outer Frame Material

The lander is comprised by a basic support structure on which the equipment, ballast weight, and other mechanisms are mounted. When projecting, and posterior construction, these frames the main goal is to keep them as light as possible without compromising its mechanical properties, such as strength, in order to minimize the need for extra bouncy material. When deployed, the static stress that the lander is exposed is trivial, therefore, the main risks of mechanical damage occur during launch and recovery. For easier transport, most constructors designed outer frames that are readily dismantled [9]. Another crucial factor to

keep in mind when designing these devices is the non-inert sea environment, since it is highly corrosive.

The most common material is aluminium, but, despite of being relatively inexpensive and light, it is not strong as other materials and it needs some special welding equipment that may not be available in many ships [9].

Stainless steel is heavier than aluminium (about three times denser), however it presents better mechanical properties [9]. Its Modulus of Elasticity, E , is bigger than the other common materials module. The values can be compared in table 3.

Titanium is also a common material to build the outer frame; however, its higher price turns it into an unappealing solution for applications that do not require its properties. Titanium presents some disadvantages apart from its cost, such as the difficulty of weld it and polish it. A good quality welding must be undertaken in an oxygen free environment, and polishing titanium must have considerable precaution since its dust can ignite spontaneously [9].

Galvanized steel is a less likely alternative since it presents several disadvantages. Closed sections of the frame, such as cylindrical tubes, are difficult to galvanize internally, leaving the frame vulnerable to corrosion. In order to reduce this problem, constructors recommend the use of sacrificial anodes (made of Zn or Mg) [9].

To give the structural shape and strength to the frame, composite structures have not been used yet, due to their low behaviour under high external pressures [9]. However, in order to minimise corrosion, composites overwrapped profiles are an alternative [31].

The table below presents the average values for mechanicals properties density and Modulus of Elasticity, and the average price per kilogram of most used materials. Fibres and a matrix, a resin, comprise composite materials. Thus, these materials combine both components mechanical properties. For comparison, the resin chase was Epoxy.

Table 3 Mechanical Properties and Ratio Price/Weight for different materials [5][35][36][37].

Material	Density [kg/m³]	E [GPa]	€/kg (14 Feb 17)	Corrosion Resistance
<i>Stainless Steel</i>	7800	198	2.01	Fair
<i>Aluminium</i>	2699	68	1.41	Fair
<i>Titanium</i>	4500	116	3.45	Very Good
<i>Epoxy + Carbon fibre reinforcement*</i>	1575	141.5	36.20	Excellent
<i>Epoxy + Glass fibre (S Class) reinforcement*</i>	1905	47.7	23.20	Excellent
<i>Epoxy + Aramid fibre reinforcement*</i>	1380	70	58.00	Excellent

* Unidirectional prepreg/UD lay-up

Housing Material

Many of the equipment used in landers are designed to operate at one atmosphere. So, most of them, such as non-compensable batteries, computer electronics, etc., must be placed in housings at this pressure condition. Since these housings' walls bear a big differential pressure between the internal one atmosphere pressure and the relatively higher exterior hydrostatic pressure, it is crucial that the housing material grants high strength [5]. This material selection presents a major concern regarding the final weight of the structure, and its shape is a crucial factor as well, since it will be the lander's component with the biggest projected area (orthogonal to current movement), consequently the one that will create a bigger drag.

With the big surface area of the housing, corrosion is another critical problem to be solved. Unless a composite or a ceramic material is used, the harsh environment of the ocean will corrode the metal housing and, as it was said in the previous section, some techniques must be used in order to decrease this risk. One of the most commonly solutions used is to attach a sacrificial anode or a more galvanically active material, e.g. zinc, that would corrode preferentially before the housing [38].

In some cases, the components that will take part in the lander are pressure compensable, thus, they experience a high and uniform pressure. To prevent the equipment from getting wet, these are normally enclosed in an oil-filled housing. Although this set up presents an advantage concerning the water bouncy, since the oil density is lower than the water's, it is not recommended for long-term deployments once it need regular maintenance.

Variable Buoyancy Systems

To lift off the seafloor and following ascending, landers use positive bouncy. Depending on the system they use, after the surface signal is sent, vertical forces will be no longer in static equilibrium, and, therefore, a vertical movement will be generated by impulsion.

Variable buoyancy systems (VBSs) provide the lander the capacity of controlling descendent and ascendant speed; relocating [10]; low operating cost and energy consumption; increased payload capacity; simplified pre-dive maintenance [39]; and better lander control. Some VBSs already used are: *Discharge Variable Buoyancy Systems*; *Pumped Water Variable Buoyance System*; *One-way Tank Flood Variable Buoyancy System* and *Pumped Oil Variable Buoyancy System* [39].

2.2.4 Sea keeping and Mooring System

It is a major concern that the lander stays in the required position to collect the desirable data. In order to obtain that, there are some aspects which must take into account regarding its landing, descent movement and its stationary properties.

Deployment

As was previously said the deployment of the lander can be made in a free fall mode or, if the operation requires high precision positioning, using a special design-lunching device connected to a crane from the vehicle in the surface [10].

Depending on the lander's dimensions and weight, its transportation to the deployment site may vary from a simple rib boat to a ship-of-opportunity equipped with a crane and winch system, having direct effect in the final coast. Therefore, the final structure must be able to fit a standard sized maritime container of 20' x 8' x 8.6' [40].

Descent and Landing

The projected lander's position in the ocean's floor before its deployment is strongly conditioned by its descent movement, which, in turn, is highly affected by the currents within the ocean water column. This, allied to lander's sink speed, can be the cause of a crucial obstacle in keep the lander in its desirable position [9].

To oppose these effects, some techniques are used. In order to obtain greater dive velocities, the structure must not comprise a big projected area in the dive direction. This will reduce the hydrodynamic effects, such as drag effects and *added mass effects*, on the lander not deviating it from its planned dive path [9].

Regarding the lander's landing, the descent speed is also important, once, if it is too fast, it is likely to produce great disturbance among the seafloor and it will be driven deeper into the sediment, increasing the risk of the lander becoming trapped in sticky bottom sediments [9].

However most landers successfully land by simply crash into the bottom, to minimize the impact, the seafloor disturbance, and the risk above mentioned, some techniques have been developed in order to adjust lander's sink speed. When launched, landers have a negative buoyancy, which value, as it sinks into the water column, decreases due to the increase of seawater density. Some landers have a ballast weight suspended beneath it that will firstly reach the bottom. When this happens, the lander buoyancy has already turn into positive and, therefore, it will not hit the floor, keeping suspended. An acoustic command from the surface will then pull the lander onto the bottom at a small speed. This system is used in ROLAI²D Lander [9].

Other identical technique consists in the same principle of using buoyancy variations with suspended ballast bellow the structure; however, in this set the lander stays suspended, not touching the seafloor, not requiring the feet. This method is used in *BANYULS* and *GOTEBORG* [9].

Concerning footpads, these must be big enough to prevent low penetration of the lander into the sediment. To prevent suction when recovery ascent movement starts, these footpads must have holes.

Mooring Systems

When in the seafloor the lander is constantly subjected to dynamic forces that may remove it from its desirable position. In order to prevent it, there are different approaches that engineers adopt. One of these systems is the mooring.

The purpose of a mooring system is to keep a floating structure on its desired position. The most common components of a mooring system are the mooring line and the anchors, connected with specific connectors.

The most used materials for the mooring line are wire, chain, or synthetic fibre rope or even, in some required cases, such as ultra-Deep-water station keeping, a combination of them. For permanent moorings in shallow waters up to 100 m, the most likely option is the chain [41].

Some mechanisms, such as *Single Anchor Leg Mooring (SALM)*, a mooring line is not used. Instead, this mooring system comprises an anchoring structure with built-in buoyancy, anchored to the seabed by an articulated connection [42].

Regarding the anchoring component, this is the element where the system relies for its strength. There are three main types of anchors: *Drag Embedment Anchors*, *Vertical Load Anchors* and *Suction Anchors* [41].

The first referred, Drag Embedment Anchors, are most used. Here, as the anchor is dragged along the seabed until it reaches the desirable depth, it uses soil resistance to get hold. It does not have a good performance when the anchor is under vertical forces; therefore, it is mainly used for centenary moorings [41].

Vertical Load Anchors work in a similar way as the previous described, however they can withstand both horizontal and vertical mooring forces. This kind of anchors is mainly used in Taut Leg Mooring Systems once the mooring line arrives at the seafloor with an angle [41].

At least, Suction Anchors systems performance lays in tubular pipes that are driven into the seabed, where a pump sucks out the water from the top of the tubular, pulling the pipe deeper into the seabed. This kind of anchors cannot operate in porous soils, such as gravel, once the water must not flow through the ground into installation [41].

3 Lander Structural Concepts

3.1 Project Requirements

This project started with the presentation of several concept models. These could be distinguished regarding their geometry and functionality. However, all of them must obey a list of rules and desires, named *Project Requirements*. This is a collection of requirements that should be followed in order to obtain the final desired product. These may be demanded (**D**), which, if not met, the final product is not truly complete, or desirable (**d**), which add value to the lander, making it more attractive in the industry market.

Regarding its geometry, we must consider the following requirements:

- **D** - Modularity – the lander must be projected considering that it must be able to engage more components regarding its operation.
- **D** - The centre of gravity of the lander must be below its centre of buoyancy.
- **D** - The total volume of the different lander modules disassembled must be inferior to 20' x 8' x 8.6' (6.096 m x 2.4384 m x 2.62128 m).
- **d** - The projected area in the perpendicular plane to the currents direction must be minimized in order to reduce drag effects.
- **d** - The horizontal projected area must be minimizing as well to reduce the drag during the landing.

Considering the forces applied to the structures, the requirements are:

- **D** - The weight of the hardware module housing must be inferior to 6 kg;
- **D** - Ability to adjust its ballast weight, and consequently, varying its buoyancy;
- **D** - Landing velocity inferior to 0.5 m/s;
- **D** - Ability to handle hydrostatic pressures up to 2 MPa without damaging structural joints.

The demanded requirement for the lander's modularity aims to adjust the components on the lander regarding its operation, therefore, among others, the lander should:

- **D** - Use an integrated hyper spectral camera to collect the possible source of seaweed in the ocean floor;
- **d** - Use a digital camera for image acquisition through the water column above the lander;
- Collect data regarding:
 - **D** - Water temperature;
 - **D** - Salinity;
 - **D** - Dissolved Oxygen;
 - **D** - Turbidity;
 - **D** - Ph;
 - **d** - Depth;

- d - Velocity of the currents;
- d -Wave amplitude.

For the material choice for the structure, the main requirements are:

- D - Minimum working temperature of 0°C;
- D - Chemically inert;
- D -Ability to handle hydrostatic pressures up to 2GPa without damage;

3.2 Lander Geometry

The geometry of a lander must be chosen regarding its application and requirements. The Lander developed within the scope of AMALIA project first priority is to detect invasive macroalgae species through their spectral reference; hence, the camera allocation is crucial. Aside from this, the lander must carry specific equipment able to characterize the environment of the macroalgae emergence. Once the need for sensors and monitoring equipment varies in accordance to our needs, the lander geometry must be able to adapt. Therefore the lander's modularity is a central factor, as well.

As presented on the *State-of-the-art*, regarding their geometry, landers can be defined as 'Open Frame' or 'Flat Shape'. In this specific application, invasive macroalgae will appear on the seafloor, at approximately 20 m depth; therefore, the camera must point upside down. For this main reason, the lander cannot be flat shaped. Thus, all concepts presented are open framed.

The first step consisted on reviewing existing benthic landers and understand why that was the chosen geometry and how could it be optimized. As the search progressed, a simple pen and a notebook were crucial to imagine the most suitable configuration foreseen for this application, presented on Fig. 10.



Fig. 10 Lander Design/Geometry sketches

After analysing some concepts proposed, some of them were modelled three-dimensionally using the software SOLIDWORKS for a better space perception.

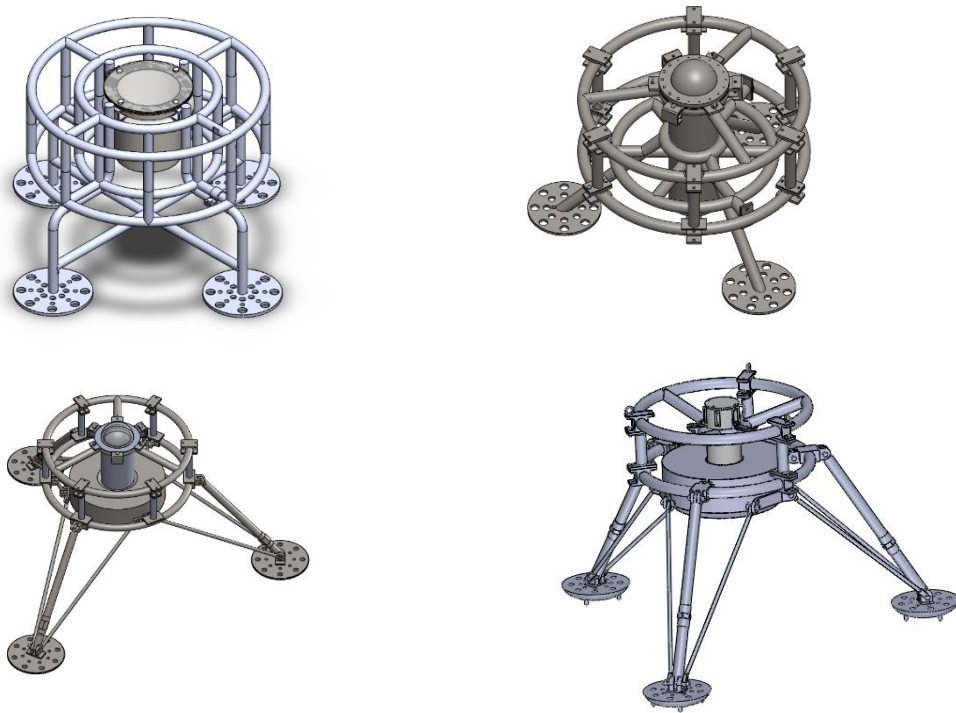


Fig. 11 Modelled concepts of lander studied in this project

The chosen geometry for the lander in an initial approach was the represented on Fig. 12. As mentioned, this selection was based on the state-of-the-art, fitting both requirements and purposes. As a dynamic process, as the project progresses, the geometry may vary according to errors or necessities that may appear.



Fig. 12 Rendered model of the selected concept - SOLIDWORKS;

Next, it will be presented reasons behind this first approach for the lander's geometry. In next chapters, some modules will be analysed in detail due to more likely to happen load cases.

3.2.1 Central Module

As some sketches arise, there is a constant feature: a central module. For monitoring invasive macroalgae, we will need expensive and delicate equipment in a hostile environment; this represents a huge risk of damage and data loss. Said so, this central module would be responsible for housing this equipment. It must be resistant, not corrosive, watertight, capable of bearing a pressure differential, and be easily removable.

Said that, our concept for this main module consists in an open cylinder that will carry up to 30 kg payload, with the possibility of assembly a cover or a port in both tops. It also comprises three handles that allow people to comfortably lift it and relocate it, or to use it with a winch.

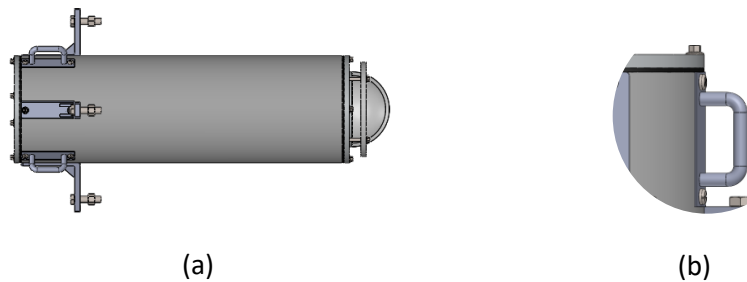


Fig. 13 (a) Central Module representation; (b) Handle detail.

Thinking ahead, for future applications, it would be highly valuable if this central module could be able to work as an Autonomous Unmanned Vehicle. For this, when studying the geometry and material for this module, when possible, it should fit some requirements referring this hypothetical ambition, as example: hydrodynamic shape, low weight, resistant to higher hydrostatic pressure. With this feature, this main part of the lander, which contain all data, could autonomously leave the lander's frame in case of emergency, which could result in a huge save.

Open cylinder

To start the dimensioning, based on the state-of-the-art, it was assumed that the housing geometry consisted in a cylinder, whose weight could not exceed 6 kg, with a length of 600 mm and an outer diameter of 200 mm, as it is shown in the Fig. 14.

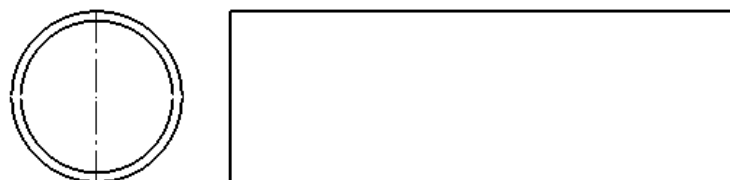


Fig. 14 Technical draw of the housing cylinder

The material choice for this module was based in the ratio *Price/Depth*, i.e. the lowermost amount of money that we are willing to pay for each depth meter it can operate,

considering that it must stand, at least, 2 MPa of hydrostatic pressure, which corresponds to 200 m depth, and meet the geometric requirements listed previously.

With that said, after revising the state-of-the-art and considering of new materials, it will be now presented some alternatives to use in the construction of this module.

Material

Metallic Materials

As already mentioned, due to their mechanical properties, metallic materials are widely used materials for high external pressure applications, such as this specific case, once they provide a good ration between price and stiffness. However, as also said in a previous chapter, the big surface area creates a huge risk of corrosion, what may compromise the structure lifetime.

The mechanical properties of most used metallic materials are listed on the table below.

Table 4 Mechanical properties and Price/kg ratio of different metallic materials [37].

	<i>Steel</i>	<i>Stainless Steel</i>	<i>Titanium</i>	<i>Aluminium</i>
<i>Density (kg/m³)</i>	7850	7750	4620	2770
<i>Yield Strength (MPa)</i>	250	520	940	414
<i>Price (€/kg)*</i>	0.545	5.915	19.6	2.175

*estimated

Composite Material Overwrapped Tube – Filament Winding

A composite material comprises two or more components which properties complement each other. With this blend, it is possible to achieve mechanical properties that, from one isolated component, would be difficult to obtain, being possible to obtain a tailor component in what concerns its load and structural performance. The final mechanical properties are approximately a linear function from properties of each part. In composites we can distinguish two phases that must be chemically inert and immiscible, the matrix, continuous phase, and the filler, dispersed phase [43].

Considering different type of matrix, composites can present very different purposes. The matrix can be polymeric, metallic or ceramic. Among the polymeric matrix, they can be thermosets and thermoplastics. The polymeric are the most used in the industry.

The matrix has to assure the cohesion and alignment of the fibres, distribute the applied loads among them, and protect them whether from the environment or from its handling.

Regarding the filler, its main goal is to optimize the mechanical properties of the final composite material. The most used are glass fibre, carbon fibre and aramid fibre, commercially known as *KEVLAR*®.

The table below means to compare the mechanical properties we can obtain from different type of fillers. The chosen resin used to compare the fillers was *Epoxy*.

Table 5 Mechanical properties and Price/kg ratio of different composite materials [37].

	<i>Epoxy + Carbon Fibre</i>	<i>Epoxy + Glass Fibre(Class S)</i>	<i>Epoxy + KEVLAR®</i>
<i>Density (kg/m³)</i>	1.550-1.580	1.840-1.970	1.380
<i>Yield Strength (MPa)</i>	603-738	457-504	355-392
<i>Price (€/kg)*</i>	34.3-38.1	17.9-28.5	43.9-72.1

Note that assigned prices are merely estimated and referred to the material, it does not consider the application and design.

Given the axisymmetric shape of the tube, the application of the composite system would be through a process named *Filament Winding*; this is an automatized process with great efficiency, where the fibres are previously impregnated in a polymeric matrix and are, then, rolled around a liner or a mandrill. The thickness and fibre orientation are chosen according to the future application of the tube.

Since the main goal of this module is to obtain a low weight, low cost, and watertight housing to use underwater, composite overwrapped tubes present several disadvantages.

- Process cost – Filament Winding at a small scale is a highly cost process, thus only economically viable if we predict a commercial use of the lander. A single overwrap cost may vary from 2000€ to 25000€, according to its finish and its mechanical properties;
- Composite materials are highly porous; this eases the growth of marine life and the impregnation of microorganism inside the material. Thus, this process requires a polymeric sleeve and, therefore, a cost increase;
- A composite overwrap over a cylinder under axisymmetric loads may be advantageous if these refer to internal pressure, since composites present high traction resistance. When subjected to outer pressure, in the presented case, hydrostatic pressure, the composite overwrap may easily collapse, once the majority of failure in composite systems come from its low compression resistance [44];
- Buckling is a highly importance phenomenon to consider, as well, when dimensioning a composite system, however, the analysis of this kind of geometry is extremely complex and time consuming [45]. Other detail that increases the possibility of buckling is that the cylinder has not both tops, thus, precluding the longitudinal overwrap and allowing the axial displacement of the composite.

The main advantage using composite systems, by the other hand, is its high corrosion and impact resistance, greatly desirable for this particular application.

To conclude, the use of composite overwrap is advantageous if the only goal is to create a highly impact resistant structure without any budget limitations. In our case, it is intended to create a low coast and low weight structure capable of withstanding high

hydrostatic pressures, hence, composite systems are not a suitable option for the project requirements.

Ceramic Materials

The properties of this kind of materials make them a proper option for great depth range applications once they provide the required weight to strength ratio, keeping them positively buoyant [46]. The most used ceramic materials are the alumina based ones, due to their highly resistance to compression and corrosion [47]. Glass is usually avoid for underwater applications for its low cracking propagation resistance [46].

Woods Hole Oceanographic Institution (WHOI) used watertight ceramic materials for electronic hardware housing on *NEREUS* project design [47]. Apart from the cylinder, the structure comprised titanium rings that reinforced the cylinder on the circumferential direction, the critical area for the cracking creation and propagation.

The material selected for this study were 94% and 96% Al_2O_3 and Si_3N_3 . The selection was made based on most used ceramics found on the state of the art.

Table 6 Mechanical properties and Price/kg ratio of different ceramic materials.

	$94Al_2O_3$	$96Al_2O_3$	Si_3N_3
<i>Density (kg/m³)</i>	3665	3710	3195
<i>Yield Strength (MPa)</i>	220.5	220.5	500.5
<i>Price (€/kg)*</i>	11.41	17.1	40.7

*estimated

Polymeric Materials

Polymers are other option for the central module. Some of these materials provide comparatively good specific mechanical properties, such as Yield strength or water absorption, that are very desirable for this application.

The exposed data refers to polymers without any additives; however, in order to improve their mechanical properties, thus, their performance, it is possible to add other elements, such as fibres.

The *fluoropolymers* family comprise high performance plastics, such as Tetrafluoroethylene with Perfluoroalkoxy, *PFA*, and Tetrafluoroethylene with Perfluoropropylene, *FEP*, and Polytetrafluoroethylene, *PTFE*, usually used in chemically austere environments and high temperatures. Although these polymers provide good properties, their processing is costly and complex [48].

Table 7 Mechanical properties and Price/kg ratio of fluoropolymers [49].

	<i>PTFE</i>	<i>FEP</i>	<i>PFA</i>
<i>Density (kg/m³)</i>	2140-2190	2120-2170	2120-2170
<i>Water absorption(%/24h)</i>	<0.01	<0.01	0.03
<i>Yield Strength (MPa)</i>	15-25	14.9-17.1	13.8-15.2
<i>Price(€/kg)*</i>	11.5-12.8	19.6-29.7	31.3-50.9

*estimated

Polyethylene's family comprise a wide range of polymeric materials characterized as an economic solution for applications that require low water absorption and high chemical and corrosion resistance. For this application, it should be noted the high-performance Marine Grade HDPE, specifically developed for marine demands, however, it will not be analysed once it is difficult to collect the required data.

Table 8 Mechanical properties and Price/kg ratio of polyethylene polymers [49].

	<i>LDPE</i>	<i>HDPE</i>	<i>UHMW-PE</i>	<i>MG-HDPE</i>
<i>Density (kg/m³)</i>	910-925	941-965	928-941	960
<i>Water absorption (%/24h)</i>	<0.01	<0.01	<0.01	-
<i>Yield Strength (MPa)</i>	8.96-14.5	19.3-26.9	21.4-27.6	21.4-38
<i>Price (€/kg)*</i>	1.82-2.23	1.82-2.23	2.66	6.07

*estimated

Other frequently used polymer is POM, Polyoxymethylene, commercially designed as TECAFORM™ [50]. The main advantages presented by this polymer are its easiness to be machined, highly desirable for prototypes, and its excellent mechanical properties, such as high stiffness, low abrasion, good resilience and low water absorption [51]. Its disadvantages are contraction after moulding (up to 2%), and it is chemically active with acidic and basic reagents.

Polybenzimidazole – PBI, commercially designated as Celazole®, is characterized as a thermoplastic with one of the best performance in engineering. Its drawbacks come from the difficulty of machining and the, comparatively, high cost.

Table 9 Mechanical properties and Price/kg ratio of POM and PBI [49].

	<i>POM</i>	<i>PBI</i>
<i>Density (kg/m³)</i>	1410	1300
<i>Water absorption (%/24h)</i>	0.22	0.4
<i>Yield Strength (MPa)</i>	71.7	130-160
<i>Price (€/kg)*</i>	2.57	193-229

*estimated

Syntactic Foams

This class of material is created using glass, ceramic, polymer or metal spheres bound together with a polymer matrix [52]. This foams present high resistance keeping their low density, what makes them a very appealing material for marine applications.

For this study, we considered MZ – 24 from Engineered Syntactic Systems [53]. Since this is a very specific material, it was not possible to find the same properties as the previously presented; however, it will be presented the needed data for the same analysis.

Table 10 Mechanical properties of MZ-24

	<i>MZ – 24</i>
<i>Density (kg/m³)</i>	380
<i>Weight gain (24h@depth)</i>	2% Max
<i>Compressive Strength (MPa)</i>	22063.23

Hatches, Doors and Access Ports

The main purpose of this project requires a perfect recognition of the macroalgae spectrum, to achieve this goal it will be used a digital camera inside the previous mentioned ‘*central module*’. For this, it will be needed a watertight and transparent interface between the equipment and the environment which can be a flat port or a sphere’s hatch.

When using an underwater camera, the port behaves as an extra optical component, which, if not used properly may disturb the obtained image, causing errors on data collection. For the lander’s purpose, the most relevant are *Chromatic Aberration*, *Radial and Tangential Distortion*, and *Refraction*. This have a major effect on flat ports.

The *Chromatic Aberration* occurs when the lights rays, depending on their wavelength, pass through different points of the lens focus. It can be either axial chromatic aberration or lateral chromatic aberration. The first one refers to a variation in the length of each wavelength, while the last occurs due to a variation of the magnification colours of light [54].

In a collected image, *Radial Distortion* may come in two different forms: or the magnification increases from the centre to the periphery, *pincushion distortion*, or it decreases from the centre, *barrel distortion* [55], shown on Fig. 15.

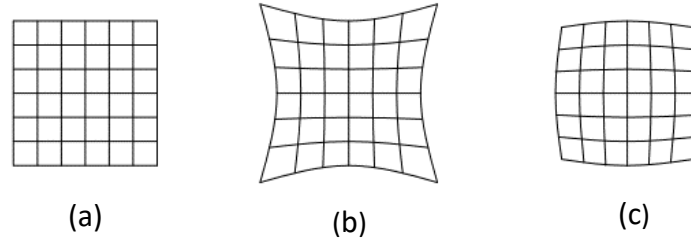


Fig. 15 (a) Undistorted pattern; (b) Pincushion distortion; (c) Barrel Distortion (*adapted* [55])

The non-alignment of the different optical components is the main responsible for *Tangential Deformation*. This shows up in an image magnification that varies perpendicularly to the rays from the image centre. Radial and tangential distortion usually come together in a complex phenomenon. This resultant distortion may be modelled using Conrady-Brown polynomial approximation, where the distorted points are mapped to the distortion-free position [56].

Another problem that can occur in underwater imaging is *Refraction*. This phenomenon is the bending of light rays when passing from a medium to another with different refractive indexes; this makes rays to travel in a non-perpendicular direction to the boundary between the different medium. The *Snell's law* gives us an equation that relates the angle of incidence of the ray lights, the *phase velocity*, and *refractive index*.

$$\frac{\sin\theta_1}{\sin\theta_2} = \frac{v_{p1}}{v_{p2}} = \frac{n_1}{n_2} \quad (1)$$

Where

- θ_1 and θ_2 are the angle of incidence and the angle of refraction, respectively;
- v_{p1} and v_{p2} are the phase velocity in the respective medium;
- n_1 and n_2 are the refractive index, which tells how much a ray will slow for an angle of incidence. The refractive index can be defined as the quotient between the velocity of light, c , and the phase velocity, v_p .

$$n_p = \frac{c}{v_p} \quad (2)$$

The refraction as a major effect on flat port, reducing the angle of coverage (Fig. 16 (a)) of the lens considerably. The angle of coverage for a specific medium can be obtained as follows.

Considering a 35 mm (36x24mm) film as an example, we need to calculate the film diagonal using the Pythagoras's Theorem:

$$d = \sqrt{h^2 + w^2} = \sqrt{24^2 + 36^2} = 43.3 \text{ mm} \quad (3)$$

Using 35 mm as the focal length, merely as an example, we can obtain the angle of coverage.

$$\alpha = \text{Arctan}\left(\frac{d}{2f}\right) \quad (4)$$

After substitution, we know that, for this example, we have an angle of coverage of 46.8°.

For a flat port, as mentioned previously, the geometry of the port and the difference of refractive index will have a major effect on the angle of coverage. For the same values as the previous example, we can obtain the modified angle of coverage as follows.

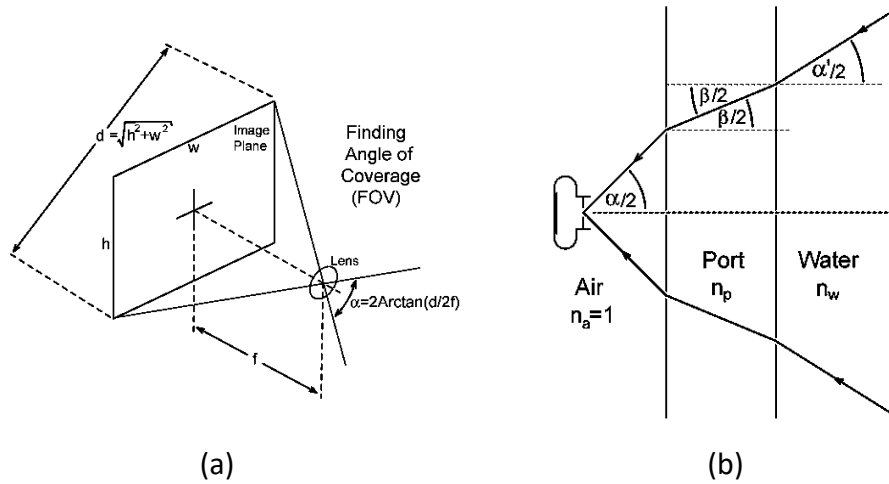


Fig. 16 (a) Angle of coverage representation; (b) Flat port angle of coverage [57]

Considering the diagram represented on Fig. 16 (b), we have that $\frac{\alpha}{2}$ is half of the angle of coverage in air, $\frac{\beta}{2}$ the half angle of coverage in the port material, and $\frac{\alpha'}{2}$ the half angle of coverage in the water. The relation among them is given through Snell's Law as follows [57]:

$$\frac{\sin \frac{\beta}{2}}{\sin \frac{\alpha}{2}} = \frac{n_a}{n_p} \quad (5)$$

And

$$\frac{\sin \frac{\beta}{2}}{\sin \frac{\alpha'}{2}} = \frac{n_p}{n_w} \quad (6)$$

After substitution, we have

$$\frac{\sin \frac{\beta}{2}}{\sin \frac{\alpha'}{2}} = \frac{n_p}{n_w} \quad (7)$$

$$\alpha' = \text{Arcsin} \left[\frac{\sin \frac{\alpha}{2}}{n_w} \right] \quad (8)$$

This means that the effect of the port material on the ray lights is null, i.e., it has no effect on the angle of coverage.

For a refraction index equal to one for air, 1.339 for seawater, and using the 35 mm film as an example, we have the following angles of coverage for different focal lengths:

Table 11 Comparison of angles of coverage underwater with a flat port and air

<i>Focal lenght</i>	<i>Coverage in air</i>	<i>Coverage underwater with flat port</i>	<i>Error</i>
<i>(mm)</i>	<i>(rad)</i>	<i>(rad)</i>	
20	1,649218	1,160854	29.61%
25	1,426656	1,021172	28.42%
30	1,249508	0,904114	27.11%
35	1,107236	0,807039	26.74%
40	0,991555	0,726421	26.74%
45	0,896243	0,65902	26.27%
50	0,816692	0,602174	26.11%
55	0,749491	0,553779	25.99%

As seen on the shown table, the angle of coverage when using a flat port decreases up to 30%. For a dome port, if the entrance pupil of the lens is perfectly located at the centre of

the curvature, the ray lights will behaviour as they would in the air, once each light ray passes through the dome perpendicularly, this means that the angle of coverage remains the same.

The dome port eliminates *Chromatic Aberration*, *Radial and Tangential Distortion*, and *Refraction* effects on the image when aligned with other optical equipment. Although it is more expensive, it represents a major benefit for AMALIA's application. The major con with this port geometry, however, is the virtual image effect.

The point at which the ray lights emitted from an object converge is called *focal point*. Once the dome port used for underwater applications is a divergent lens, the *focal length* is negative, i.e., the image is virtual, creating the illusion that the object is closer to the lens (Fig. 17).

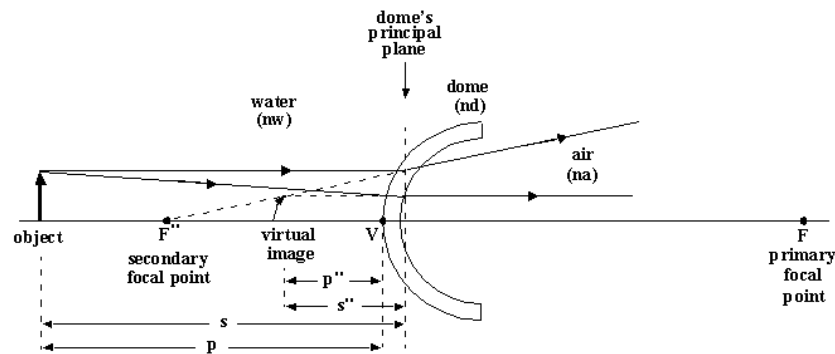


Fig. 17 Virtual image formation when a dome cover is used [58]

Material

For viewports, the most used materials are glass and acrylic. These two materials' mechanical properties are listed below.

Table 12 Mechanical properties and Price/kg ratio of Glass and Acrylic [1]

	Glass	Acrylic - PMMA
Density (kg/m^3)	2650	1220
Yield Strength (MPa)	62.5	63.1
Price (€/kg)*	6.65	3.155

*estimated

These two materials often come in literature compared to each other, is, therefore, easy to match the properties desirable for AMALIA's Lander operation.

As previously mentioned, the main goal of AMALIA project is to detect the spectrum reflected by different macroalgae when exposed to light, is, therefore, demanded a perfect ray transmit through the lens port. While acrylic transmits up to 92% of visible light, mineral glass only transmits from 80 to 90%, depending on the quality of glass and manufacturer.

Regarding the Scratching, for glass, it requires a relatively hard material to scratch it, while acrylic is easy to scratch, which may compromise the data acquisition.

At least, acrylic has a higher thermal conductivity value, which decreases the possibility of condensation on the dome [59].

3.2.2 Supports

A considered aspect for the geometry was the number of supports. Usually, open framed landers have three or four 'legs', this decision, additionally to the lander's purpose, was based on the type of soil of the place where it will be deployed. The location, sited at the Berlengas archipelago, is characterized by its rocky soil with a thin mud layer on top; therefore, it is likely that the lander is deployed at an uneven floor. Considering the following theorem '*If P , Q , and R are three non-collinear points in R^n , there is one and only plane M that contains these three points.*', as it is seen on Fig. 18.

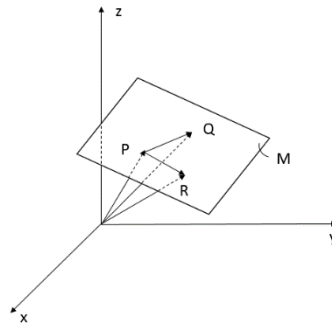


Fig. 18 Definition of a plane based on three non-collinear points

Therefore, a fourth support, in case of a not levelled soil, will be likely to be suspended, making the structure unstable, which is highly undesirable. For these reason, the chosen concept comprises a three 'leg' structure.

As an image acquisition requirement, the camera must be positioned vertically. For that, we considered a lunar lander, as an example. This lander that may operate at a hostile environment that, as also, must be correctly positioned.

For this concept, each set comprises three 'legs', the main one, which length is variable, and two secondary. All of them must be able to rotate over an axis; however, both of secondary supports must rotate over the same axis.

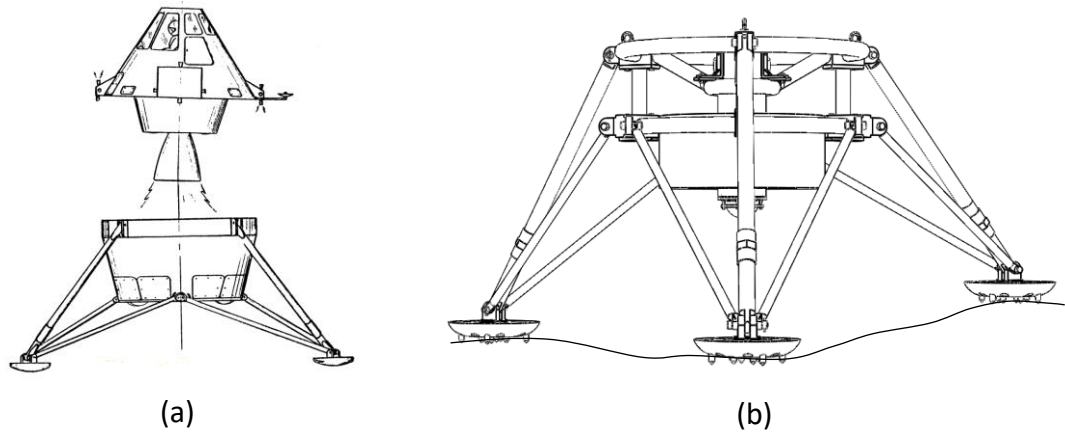


Fig. 19 (a) Concept for the Lunar Excursion Module (May of 1962) [60]; (b) Main support variable length, AMALIA Lander concept

Material

For these components, the considered material will be metallic and polymeric, already described. Supports will have to bear a high impact during the structure landing, so, they must be resistant, yet, low weight.

3.2.3 Buoyancy

A main feature for the lander is the ability to maintain its buoyancy variable according to its movement's requirements: negative when landing, positive when climbing. For this, we considered variable buoyancy systems that could fit this requisite. As presented on the state-of-the-art, the existent variable buoyancy systems are *Discharge Variable Buoyancy Systems*; *One-way Tank Flood Variable Buoyancy System*; *Pumped Oil Variable Buoyancy System*, and *Pumped Water Variable Buoyancy System* [39]. These will be presented next.

Discharge Variable Buoyancy Systems

The most commonly used system is the discharge variable system, where the release of ballast and the use of floating equipment alternates the system buoyancy from negative to positive [39].

An example of this technology is *three stage drop-weight system* used in Promare's 11k. The first release is after descent, just above the sea floor or after the crash. After that, in order to obtain neutral buoyancy small weight are dropped, and, at the end of the dive, the last weight is release, leading to the vehicle ascent [24].

The most used floating equipment are glass spheres (Fig. 20 (a)) caused by its relatively low cost. Although they can operate at full ocean depth, there is the risk of implosion at big depth-rates due to hydrostatic pressure, which can trigger the collapse of all structure [9]. To protect glass, is usually used neutrally buoyant polyethylene *Hard Hats* (Fig. 20 (b)). These consist of two flanged parts bolted together with stainless steel hardware [61].



Fig. 20 (a) TELEDYNE Glass Sphere; (b) Hard Hats Shapes

Another widely floating material used is syntactic foam that, although more expensive than glass sphere, it does not present the risk of implosion, and it can be custom made. Another disadvantage of syntactic foam is that in depth-rates to 6000 m, the air weight can be more than the double that of glass buoyancy. For shallower depths, less than 2000 m, the foams do not present this disadvantage [9].

Other two alternative floating materials still under development are titanium spheres and density thixotropic liquids [9].

One-way Tank Flood Variable Buoyancy System

This system lays in the simplicity of filling an empty tank in order to increase the negative buoyancy. This is an effective method, and, as an additional advantage, it does not discharge ballast material into the ocean [39], yet, it needs more energy in order to remove it from the water.

Pumped Oil Variable Buoyancy System (POVBS)

This system, similar to pumped water system, changes its buoyancy by pumping the liquid in and out of a pressure chamber. Thus, this system is capable of achieving two-way buoyancy changes [39]. The main difference between POVBS and PWVBS is that the first presents fixed mass, and thus is the adjustment of the dislocation of the vehicle that controls its buoyancy. Said so, in order to increase the buoyancy, the oil is pumped from a pressure chamber to an external flexible bladder. As this bladder expands, and the water is pushed out of the housing and thus the buoyant Force increases. To decrease it, a valve is opened and the oil goes back into the chamber due to water pressure forces.

This system performance is highly limited by the power source and the available space. Moreover, all the equipment associated, such as the motor and piston, must be contained in a pressure housing, consequently increasing the total volume and weight of the system.

Pumped Water Variable Buoyancy System (PWVBS)

This VBS is a highly flexible method used to control lander buoyancy. It lays in a fixed volume tank, where the water addition or removal changes the buoyancy of the system. Originally, the tank is full filled with one atmosphere pressurized air, then, it allows the entrance of water,

decreasing its buoyance. In order to re-increase it, the water is pumped out. A common change is the use of compressed air instead of a pump, in order to force water out of the container [39].

The main advantage of PWVBS is its design flexibility, since it can be custom made, meeting all specifications needed by the designer. However, the power source available limits the system's operations and cycles, in addition, the energy required increases with depth [39].

AMALIA Lander Buoyancy System

After analysing pros and cons from different buoyancy systems, the one that better fits our requirements is the pumped water variable system. The possibility of adapt the water volume that works as a ballast weight fits with the modularity feature of the lander. Furthermore, it is capable, as well, to control the landing velocity, preventing possible damage by impact.

This set comprises three main components: Pressure tank, Pump, and Valve System.

Table 13 Pumped Water Variable Buoyancy System Valves functioning

Pressure	Flow	Valves Open
$P_{\text{water}} < P_{\text{tank}}$	Pump in	D & B
$P_{\text{water}} < P_{\text{tank}}$	Flow out	A & B or C & D
$P_{\text{water}} > P_{\text{tank}}$	Pump out	C & A
$P_{\text{water}} > P_{\text{tank}}$	Flow in	A & B or C & D

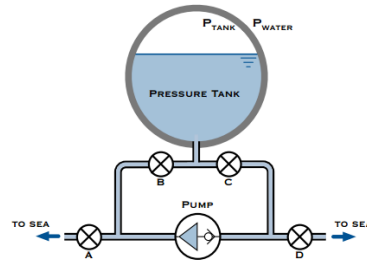


Fig. 21 Pumped Water Variable Buoyancy System;

For this vessel, the considered materials will be the same as for the central module. Due to its great volume, it must be a low density material, yet, strong enough to bear big pressure differentials, once it will be exposure to both internal and external pressure.

3.2.4 Modularity

The whole idea behind this geometry concept was the lander's modularity. This means, the fully possibility to join different components according to the operation needs. Circular rings, linked through height profiles columns, essentially compose the main frame of this lander. This feature creates the concept of *levels*.

Although the main hardware for this application is comprised in one main module, the central one, for hypothetical future purposes and application the lander may need extra space for payload and equipment, this concept fits this possibility.

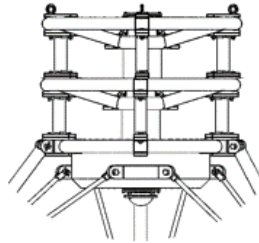


Fig. 22 AMALIA Lander modularity feature

4 Structure design

The dimensioning on the lander is mainly directed to their main components, i.e., the components that will bear critical loads during the lander operations. The two modules that will be given special focus are the central module and the outer frame.

4.1 Central Module Housing design

4.1.1 Coordinate System

Once presented the different alternatives for housing materials and the application requirements, the structure will be now analysed considering the dimensions and applied loads.

For the axisymmetric geometry of the tube, the coordinates system adopted was cylindrical. As said by Eric Weisstein on *Wolfram MathWorld*, “Cylindrical coordinates are a generalization of two-dimensional polar coordinates to three dimensions by superposing a height (z) axis.” [62]. Hence, the stress comes as:

- σ_t – Circumferential Stress
- σ_r – Radial Stress
- σ_l – Axial Stress

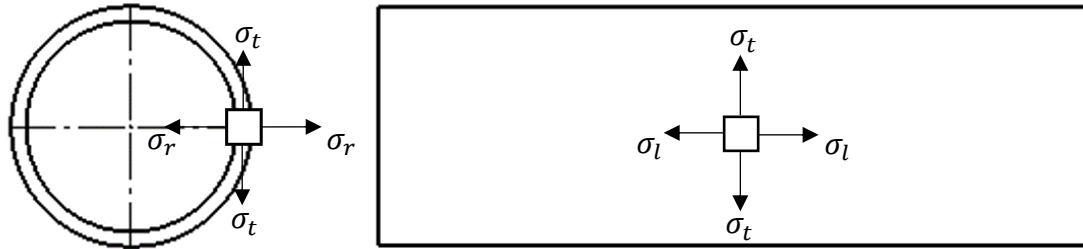


Fig. 23 Stress Representation

4.1.2 Theoretical Approach

Cylindrical tubes may be defined as constant ring shaped section solids and they can be classified by the ratio between the thickness and inner diameter as thick-walled cylinders or thin walled cylinders, and by the existence, or not, of tops. This last feature defines the presence of axial stresses [63]. Thus, if the following condition is verified [63]:

$$\frac{t}{d_i} < 0.1 \quad (9)$$

Where

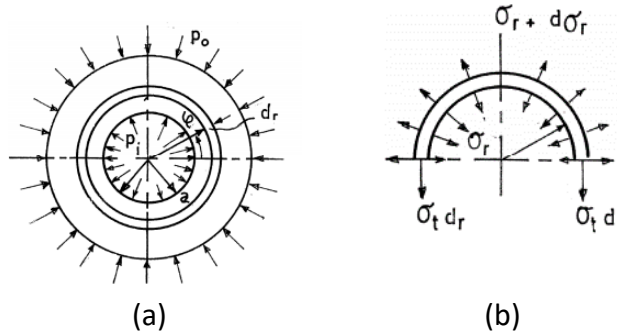
- t – Cylinder thickness
- d_i – Inner Diameter

We can consider it as a thin-walled cylinder and, as such, we can admit that the circumferential stress value, σ_t , is a constant through the wall and it can be obtain from the following equation.

$$\sigma_t = -p_o r / t \quad (10)$$

The axial stress, σ_l , is given by

$$\sigma_l = -p_o r / 2t \quad (11)$$



Where p_o is the external pressure.

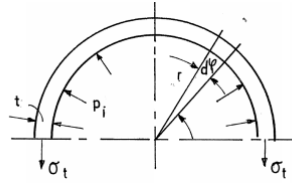


Fig. 24 Thin-walled cylinder cross section representation

Fig. 25 (a) Cross section of a thick-walled cylinder loaded by both internal and external pressure; (b) Elementary ring with thickness dr .

If $\frac{t}{d_i} > 0.1$ we have a thick-walled cylinder. In this case, the circumferential and radial stresses are depending on the radius. To determine theses stress values, it is assumed that all cross sections maintain perpendicular to the rotation axis [63].

For this particular case, the internal pressure is null, $p_i=0$, and the external pressure is depending on the depth that the lander will operate, thus, it will be maintained as p_o . Said so, the stresses can be obtained from Lamé's Equations, which, for this study, can be formulated as [63]:

$$\sigma_t = \frac{-p_o d_e^2}{d_e^2 - d_i^2} \left(1 + \frac{d_i^2}{r^2} \right) \quad (12)$$

$$\sigma_r = \frac{-p_o d_e^2}{d_e^2 - d_i^2} \left(1 - \frac{d_i^2}{r^2} \right) \quad (13)$$

The stress variations over the radius can be represented and is on the following figure, where it is shown, as well, the σ_t and σ_r values for inner and outer diameter, a and b , respectively [63].

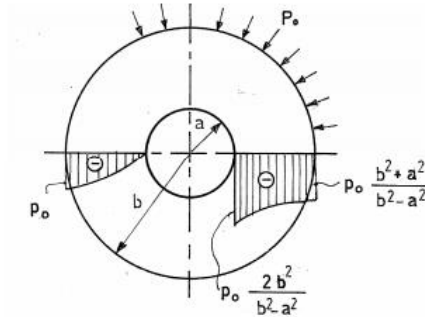


Fig. 26 Circumferential and Axial Stresses distribution through the cylinder thick wall.

From the Generalized Hooke's Law we can relate the three stresses that the cylinder is subjected and the resulting strain. Therefore, it comes as:

$$\varepsilon = \frac{1}{E} [\sigma_1 - \nu(\sigma_r + \sigma_t)] \quad (14)$$

For last, we can determinate the radial displacement from [64]:

$$\delta_r = \frac{rp_o d_e^2}{E(d_e^2 - d_i^2)} [(1 - \nu) + (1 + \nu) \frac{d_i^2}{r^2}] \quad (15)$$

4.1.3 Open Cylinder Dimensioning

The central module's analysis was made based on the, already mentioned, *project requirements*- maximum cylinder weight of 6 kg, outer diameter of 200 mm, 600 mm length and minimum depth range of 200 m. For that, the analysis comprised a set of operations that went from finding the minimum thickness to knowing the maximum external pressure that the tube can withstand. The material choice was made based on the ratio €/m, i.e., the less amount of money that we can pay for each meter of depth that the lander can operate, and on the risk of damage, such as corrosion, and.

Therefore, for each material listed above (*Steel, Stainless Steel, Titanium, Aluminium, POM, PTFE, FEP, PFA, LDPE, HDPE, UHMW-PE, PBI, 94Al₂O₃, 96Al₂O₃, Si₃N₄, and MZ 24*), we first found the tube thickness. For that, we have

$$\rho = \frac{m}{V} \quad (16)$$

Where, for a cylindrical tube, the *volume* comes as

$$V = \pi (r_e^2 - r_i^2)l \quad (17)$$

For this application:

- $m = 6 \text{ kg}$
- $l = 600 \text{ mm}$
- $r_e = 100 \text{ mm}$

Once ρ depends on each material, the equation for thickness, t , with ρ as a variable is

$$\pi (0.1^2 - (0.1 - t)^2) 0.6 \rho = 6 \quad (18)$$

For each material, rounding the thickness values to ease future processing, we came to:

Table 14 Tube thickness for each material

	Steel	S. Steel	Titanium	Aluminium	POM	PTFE	FEP	PFA	HDPE
$t \text{ (mm)}$	2.048	2.075	3.506	5.921	12.009	7.643	7.718	7.718	18.392
$\approx t \text{ (mm)}$	2	2	3.5	5.5	12	7.5	7.5	7.5	18

	LDPE	UHMW-PE	PBI	94Al2O3	96Al2O3	Si3N4	MZ 24
$t \text{ (mm)}$	19.188	18.798	13.100	4.441	4.386	5.112	59.708
$\approx t \text{ (mm)}$	19	18.5	13	4	4	5	60

Known the thickness, using the software *SOLIDWORKS* it was modelled each tube made from the respective material and then, using the software *ANSYS*, the hydrostatic pressure was simulated. Using POM as an example, the model used and mechanical properties inputted on the software were:

- $E = 2.9 \text{ GPa}$
- $\nu = 0.3935$



Fig. 27 POM cylindrical tube

On ANSYS, it was simulated the stress, strain, displacements of the tube when subjected to five different cases of constant load over the surface. Once the length of the tube can be considered as relatively small, it was not considered the pressure differential, resultant from different depth ranges. For each three different external pressure, it was given the maximum Von Mises equivalent stress, σ_{eq} .

For boundary conditions on ANSYS simulations, for this, it was automatically generated a tetrahedral mesh, with 1596 elements, and it was considered that radial displacements on

each top surface were null, once the cylinder will have two rings on each top, for assembly purposes. Both tops were free for axial displacements.

Table 15 Equivalent Stress resultant from different loads

$P_e(MPa)$	1	2	3	4	5
$\sigma_{eq}(MPa)$	10.719	21.438	32.157	42.876	53.595

From the result's table we can draw the following graphic.

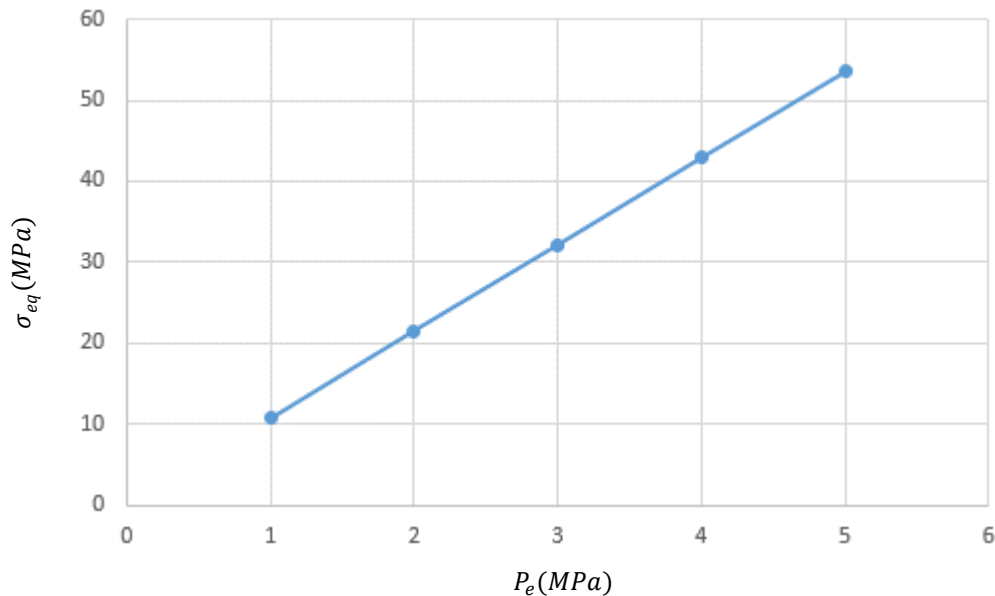


Fig. 28 Impact of External Pressure on Equivalent Stress

Using the software Excel, we got the equation for the linear regression. With it, knowing the Yield point of each material, we can know the depth range that the cylinder can operate before collapsing. For POM, the relation between External Pressure and Stress is given by:

$$\sigma_{eq} = 10.719 P_e \quad (19)$$

With $R^2=1$.

When projecting load withstanding structures it must be used a *safety factor*. The value for this factor depends on a large range of variables, such as the probability of failing during operation, or a whole set of uncertainties during calculations or poorly sustained assumptions regarding geometry and stress during early calculations. Other important aspect that affect the magnitude of the safety factor is the seriousness of damage that would happen in case of fail.

For Gunter Erhard, in "*Designing with Plastics*", the safety factor to consider when projecting load bearing plastic structures should be congruent with the following guide [65].

- $S_{min} \geq 2$ for calculations preventing fracture;

- $S_{min} \geq 3$ for calculations preventing bending and buckling;
- $S_{min} \geq 1.2$ for calculations preventing fracture stresses due to cracking.

As it was said previously, the risk of buckling represents a major concern due to hydrostatic pressure. Hence, following the guide, the minimum value for the safety factor is two. Once the housing stores a set of expensive electronic devices, it was assumed that the safety factor is three. Although this guide is targeted to *plastic design*, this value was used for every material to keep the analysis coherent.

Said so, to know the depth range, we achieve the maximum external pressure that the tube can withstand, replacing the σ_{eq} by one third of the Yield's. For POM, the Yield's point is given for 71.7 MPa, considering the safety factor, the equation comes as:

$$23.9 = 10.719 P_e \quad (20)$$

$$P_e = 2.23 \text{ MPa} \quad (21)$$

Using the software ANSYS, now loading the cylinder surface with 2.23 MPa, for POM, we got the stress, strain, and displacements graphic representations.

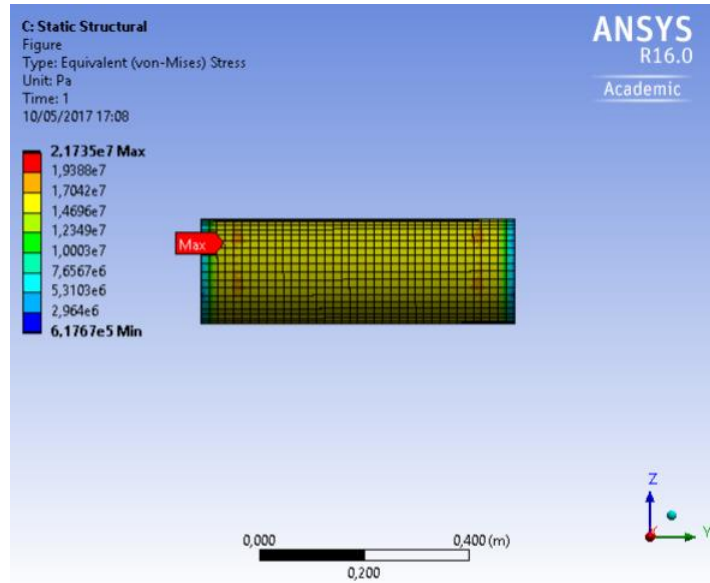


Fig. 29 Equivalent (von-Mises) Stress (Pa) for 2.23 MPa external pressure – POM

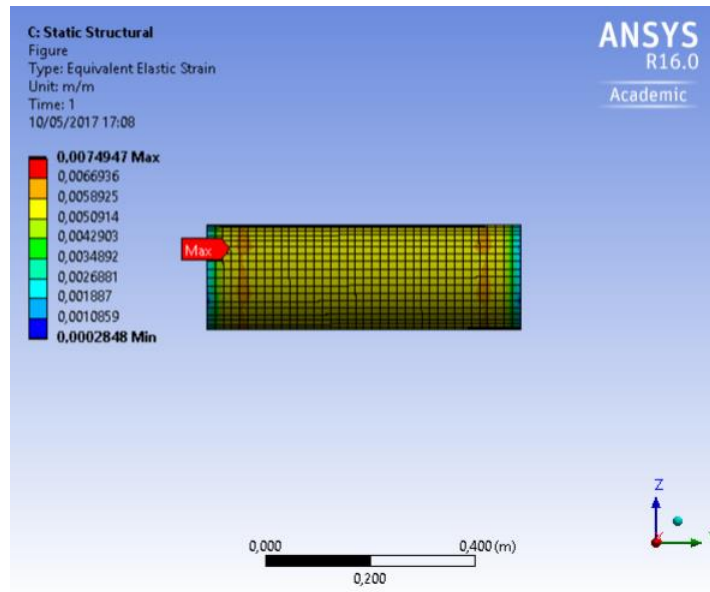


Fig. 30 Equivalent Elastic Strain (m/m) for 2.23 MPa external pressure - POM

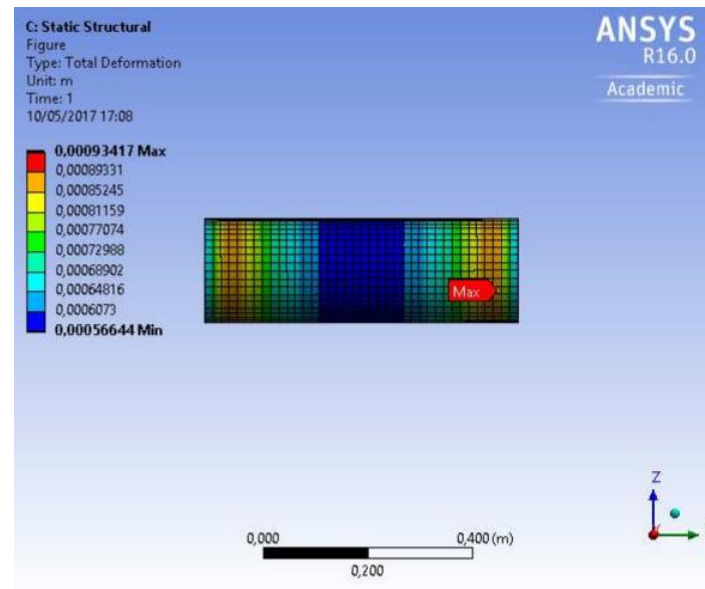


Fig. 31 Total Deformation (m) for 2.23 MPa external pressure - POM

The maximum values, observed from the graphic representations above, are:

Table 16 Maximum ANSYS values for POM

<i>Equivalent (von-Mises) Stress (MPa)</i>	<i>2.1735 e 7</i>
<i>Equivalent Elastic Strain (m/m)</i>	<i>0.0074947</i>
<i>Total Deformation (m)</i>	<i>0.00093417</i>

The following equation gives us the relation between hydrostatic pressure and depth in a liquid [66].

$$P = \gamma h \quad (22)$$

Where P is the pressure in fluid, h the depth in the fluid where we want to measure the pressure, and γ is the specific weight of a fluid, it is defined as the fluid weight per volume unit. The specific weight can be related to the water density through the equation [66]

$$\gamma = \rho g \quad (23)$$

Where g is the local acceleration of gravity, and ρ the water density. The density is defined as mass per volume unit of water [66] and depends on its purity, and temperature. The most common value for water density is 1000 kg/m^3 , this value is verified for pure water at 4°C . The mean density of ocean water at the surface is 1027 kg/m^3 . The two main factors that make ocean water denser than pure water is its salinity, and the lower temperatures. As we go to the bottom of the ocean, the temperature decreases, which means the density increases as we reach the seafloor [67].

Hence, the depth range the lander would achieve before collapsing, using POM, once again as example, we have the depth range

$$h = 221.39 \text{ m} \quad (24)$$

The depth range for each material was calculated using the same procedure and the results are listed on the following table where the green cells represent depth ranges higher than the project required, i.e., the cylinder can achieve, at least, 200 m , and the red cells represent those cylinders that do not. For syntactic foams, the analysis was different due to its low density, thus, the results will be presented on the next point.

Table 17 Depth range for each material

	Steel	S. Steel	Titanium	Aluminium	POM	PTFE	FEP	PFA	HDPE
$h \text{ (m)}$	140.42	292.48	858.22	585.33	221.39	39.58	31.60	28.62	107.06

	LDPE	UHMW-PE	PBI	94Al2O3	96Al2O3	Si3N4
$h \text{ (m)}$	57.69	117.01	158.72	234.21	234.21	532.75

Considering the estimated price per unit of mass listed on the tables above, we can achieve a final price for each material. The criteria for the material choice, as mentioned previously, will be the ratio price per range depth. Said that, the result ration comes on the following table.

Table 18 Ratio €/m for each material for maximum depth

	<i>Steel</i>	<i>S. Steel</i>	<i>Titanium</i>	<i>Aluminium</i>	<i>POM</i>	<i>PTFE</i>	<i>FEP</i>	<i>PFA</i>	<i>HDPE</i>
€/m	0.02	0.12	0.14	0.02	0.07	1.81	4.55	8.38	0.11

	<i>LDPE</i>	<i>UHMW-PE</i>	<i>PBI</i>	<i>94Al2O3</i>	<i>96Al2O3</i>	<i>Si3N4</i>
€/m	0.21	0.13	7.92	0.26	0.4	0.45

Note that this ratio was calculated for each material maximum depth, however, other components, such as monitoring equipment, will not be able to achieve these depth ranges, once it will be designed to withstand up to 2 MPa.

Since Syntactic Foams have a relatively low density, the mass could not be the primary criteria, otherwise the tube would have 60 mm thickness letting us with only a 40 mm open cylinder for equipment. Thus, the iteration started considering the maximum external pressure of 2 MPa. For this, using the software ANSYS, we obtained the Von Mises equivalent stress for three tubes with different thickness.

Table 19 Syntactic Foam ANSYS analysis

t (m)	m (kg)	P _e (MPa)	σ _{eq} (MPa)
0.04	4.584212	2	7.633
0.035	4.136535	2	8.396
0.03	3.653044	2	9.639

Using this values on Excel, we come to a stress/thickness relation that can be observed on the following graphic.

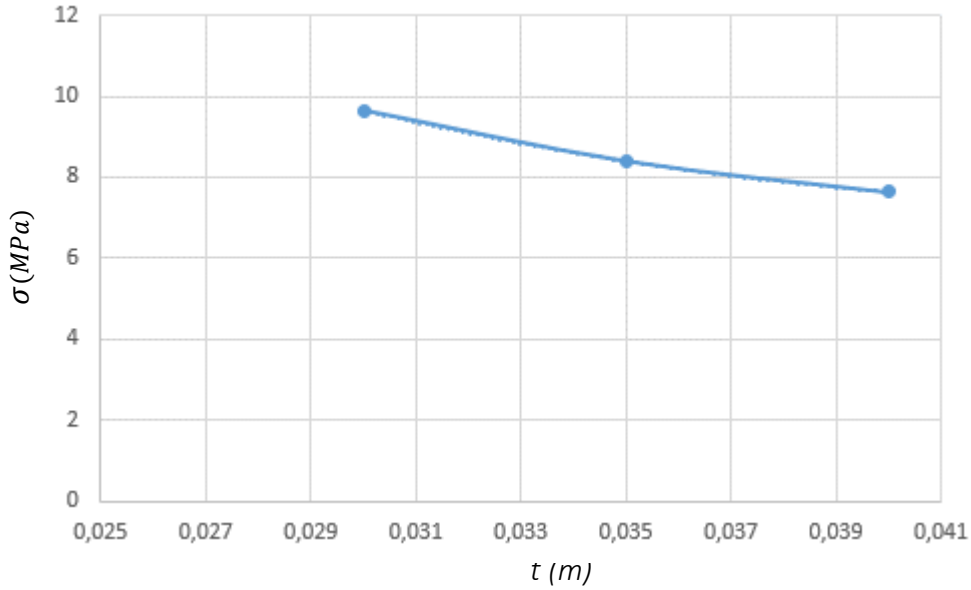


Fig. 32 Stress/thickness relation on Syntactic Foams

Using a polynomial regression of second order, the equivalent stress due to thickness variation on syntactic foams is given through

$$\sigma = 9600t^2 - 872,6t + 27,177 \quad (25)$$

With $R^2=1$.

For $\sigma = \frac{\sigma_{ced}}{3}$, where, for MZ – 24, $\sigma_{ced} = 22.063 \text{ MPa}$, we obtained the minimum tube thickness before collapse. After calculations, we came to 0.043 m for a thickness value.

Once it is an engineered material it is not easily found on the market, and, for the same reason, its price has to be quoted for this specific geometry. Said that, some quotations for a tube with outer diameter 0.2 m, inner diameter 0.114 m and 0.6m length were asked and the final price for the cylinder would be \$ 1394.00 ($\approx 1239.51\text{€}$). After we have done the same ratio as for the previous materials, we got that, for 200 m depth range, the cost per meter would be 6.19 €/m.

After analysing the final table, we can choose the best of the three ratios; these will be *Steel* (0.02), *Aluminium* (0.02) and *POM* (0.07). As seen previously, it is not impossible to achieve 200m depth range using *Steel* housing; therefore, it can be discarded. Leaving us with two main options: *POM* and *Aluminium*. Considering that the material price for each material, we would spend 12.15 € for *Aluminium* and 15.42€ for *POM*, this price is an estimated price only for raw material, not considering machining, welding, and an anticorrosion coat, the last two referred to *Aluminium*. That said, the final cost, if we choose *Aluminium*, would exceed *POM*'s cost, once machining this copolymer is an easy and low-cost process. Hence, the material selected for the central module open cylinder was *POM*.

Elastic Instability

As said by Paulo Tavares de Castro in '*Reservatórios Metálicos*' [68] (*Metallic Reservoirs*), every dimensioning of any mechanical component must obey criteria that underlies in ensure and fit the structure purpose, preventing possible failures. For that, it is usual to limit elastic instability, plastic instability, rupture by fatigue, fragile rupture, creep or corrosion.

When talking about vessels subjected to external pressure, the dominant criteria is elastic instability. This phenomenon can be defined as an unacceptable shape alteration due to low structural rigidity [68].

Critical length

If the tube's length exceeds a specific value, the cylinder's tops do not have any influence on the central part. This value is commonly called *critical length*, l_c . In this case, the pressure at which the collapse begins is the same as it is for an infinite length tube [69].

To avoid collapse, some features that will increase the structure rigidity are used. These can be tops or rigidity rings. Yet, this will only have impact on a limited extension of the tube, since only rings will not collapse. Therefore, when using those, the length among them must not exceed the critical length. As seen on the following image.

To find the critical length, we have the equation [68]

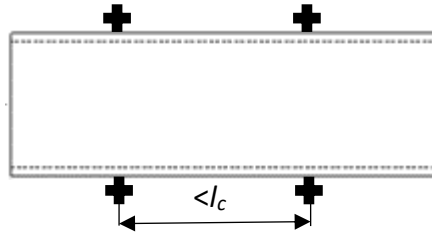


Fig. 33 Representation of rigidity rings

$$l_c = \frac{4\pi\sqrt{6}}{27} (\sqrt[4]{(1-\nu^2)}) (d \sqrt{\frac{d}{t}}) \quad (26)$$

Substituting the variables for this particular case, we have that the critical length is 736.8 mm, which is higher than 600 mm. This means that both tops affect the central cross section and, therefore, there is no need for extra features in order to increase the cylinder rigidity.

Theoretical Critical Pressure

For a cylindrical surface under an external pressure, the collapse begins for a specific load magnitude. This value is commonly known as *Theoretical Critical Pressure* and it depends on the ratio thickness/diameter and on the material properties (E and ν) [68]. For a tube with

length superior than the critical, this pressure can be obtained from Bresse and Bryan equation [70]:

$$P_{tc} = \frac{2E}{1 - \nu^2} \left(\frac{t}{d}\right)^3 \quad (27)$$

For tubes with length inferior than the critical, Southwell came with an equation that allows us to find this pressure [69]:

$$P_{tc} = \frac{8\pi\sqrt{6}}{27} \frac{E}{(1 - \nu^2)^{\frac{3}{4}}} \frac{\left(\frac{t}{d}\right)^{\frac{5}{2}}}{\frac{1}{d}} \quad (28)$$

For POM cylinder values we have that the Theoretical Critical Pressure is 28 MPa, which means that this cylinder fits the depth range requirement. Comparing these values with ANSYS results, we have:

Table 20 ANSYS and theoretical values comparison for POM cylinder

<i>ANSYS</i>	<i>Theoretical Values</i>	<i>Difference</i>
2229685.60	2827943	21.1%

4.2 Frame dimensioning

The outer frame can be divided in several components, as well. These are: the main supports, secondary supports, columns, and hoops, depending on these described there is the buoyancy tank. For each one of these, there are specific criteria for their material and dimensions, which selection based on load cases for the lander during its working life cycle.

For these lander components, it was considered, analysed and compared two distinct materials: steel and POM. As seen previously, POM is a high-performance polymer, whose characteristics such as non-expensive, and low-weight make it a desirable material to work with, while steel, although heavier, is a valuable alternative due to its low-price and good mechanical properties.

The dimensioning of the structure is an iterative process, as such, it was needed a starting point: an assumption that will be then verified. Said that, it was considered that the main module has a total weight of 30 kg (*self-weight and payload*), and that the frame is made of steel, once it is the heaviest of the alternatives.

Therefore, for steel, we have a starting self-weight of 849.16 kg for the main frame; this does not consider the ballast tank nor the main module. The buoyancy tank main purpose

is to keep the lander at the surface when filled with air, hence, as a rough approximation for its volume, to start the iteration, considering 889.16 kg of steel, with density equal of 7750 kg/m³, we have a total volume of 0.11473 m³. Once the density of seawater is 1027 kg/ m³, we have an up thrust created by the structure equal to 117.82 kg. The difference between the self-weight and the frame buoyancy, multiplied by the seawater density gives us the ballast volume.

$$889.16 - 117.82 = 771.34 \text{ kg} \quad (29)$$

$$d_{\text{seawater}} = 1027 \frac{\text{kg}}{\text{m}^3} \quad (30)$$

$$V_{\text{buoyancyank}} = 0.751 \text{ m}^3 \quad (31)$$

Once again, this is merely an approximation to start the iterative dimensioning process.

With the consideration above referred, we design the main frame as it follows.

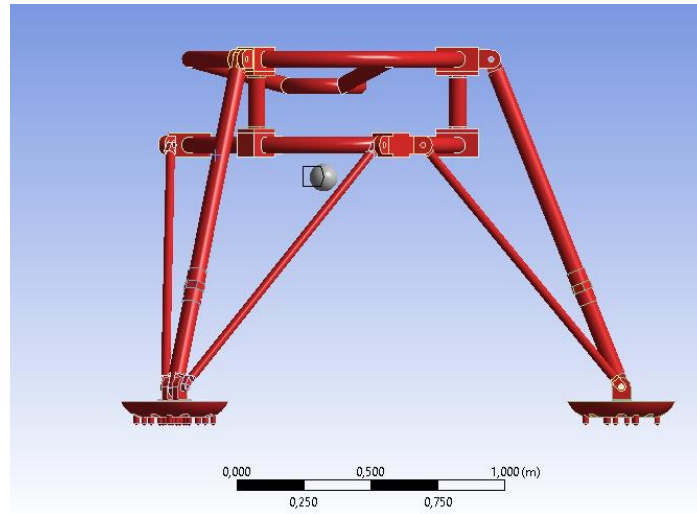


Fig. 34 First model Assumed for the main structure

The worst case and, therefore, the first we are analysing, is when the ballast tank is fully loaded with water, which, even if should not happen, a valve malfunctioning may occur, on the surface, i.e., no up thrust forces will occur.

Once the lander has an axisymmetric shape, the weight will be equally distributed in three bases, as so, the analysis of one support will, analogously give us an analysis of the remaining supports.

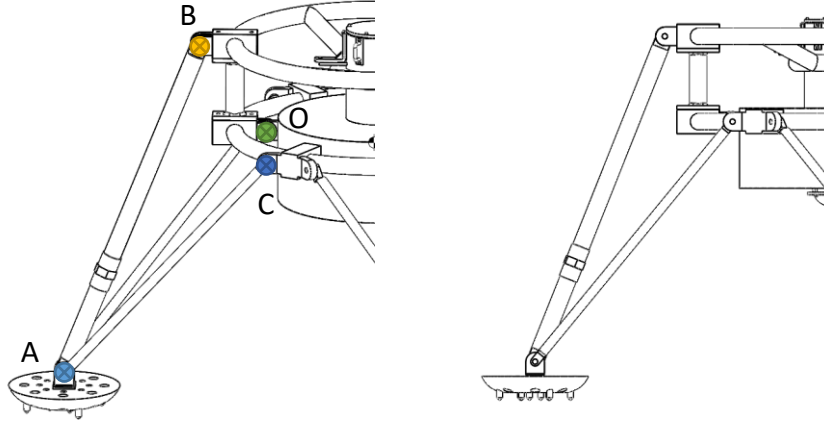


Fig. 35 Relative coordinates of the main joints

As said previously, the main purpose for this geometry on the support is to be adjustable to the seafloor, maintaining the lander vertical, consequently, the loads applied on each support leg of the Lander structure may vary according to the chosen angle. To obtain the loads on each support, after a geometric analysis of the structure it was possible to obtain the directional vectors of each force. Considering the relative coordinates of the points O, A, B, and C, we have

$$\begin{aligned}
 O &= (0,0,0) \\
 A &= (r \sin \theta, -r \cos \theta, 0) \\
 B &= (254.52, 320, 0) \\
 C &= (0, 0, 450.25) \\
 D &= (0, 0, -450.25)
 \end{aligned}$$

The two secondary supports rotate over an axis with a constant length from the point O to the support base, point A. This length is referred as support radius, r_s , and its value is 1166.60 mm; θ is the angle that r_s makes with a vertical axis in a bi-dimensional approach.

For the equilibrium of the forces on one basis, we have that

$$\sum \vec{F} = \vec{0} \quad (32)$$

$$\begin{Bmatrix} 0 \\ P/3 \\ 0 \end{Bmatrix} = N_1 \overrightarrow{AB} + N_2 \overrightarrow{AC} + N_3 \overrightarrow{AD} \quad (33)$$

$$\overrightarrow{AB} = \frac{B - A}{\|AB\|} = \frac{(254.42 - r_s \sin \theta, 320 + r_s \cos \theta, 0)}{\sqrt{(254.42 - r_s \sin \theta)^2 + (320 + r_s \cos \theta)^2}} \quad (34)$$

$$\overrightarrow{AC} = \frac{C - A}{\|AC\|} = \frac{(-r_s \sin \theta, r_s \cos \theta, 450.25)}{\sqrt{r_s^2 + 450.25^2}} \quad (35)$$

$$\overrightarrow{AD} = \frac{D - A}{\|AD\|} = \frac{(-r_s \sin \theta, r_s \cos \theta, -450.25)}{\sqrt{r_s^2 + 450.25^2}} \quad (36)$$

For $\theta=45^\circ$, as an examples, the direction vectors are

$$\overrightarrow{AB} = (-0.62052, 0.784191, 0)$$

$$\overrightarrow{AC} = (-0.79383, 0.490087, 0.360064)$$

$$\overrightarrow{AD} = (-0.79383, 0.490087, -0.360064)$$

The equilibrium equation is therefore

$$\begin{pmatrix} 0 \\ P/3 \\ 0 \end{pmatrix} = N_1 \begin{pmatrix} -0.62052 \\ 0.784191 \\ 0 \end{pmatrix} + N_2 \begin{pmatrix} -0.79383 \\ 0.490087 \\ 0.360064 \end{pmatrix} + N_3 \begin{pmatrix} -0.79383 \\ 0.490087 \\ -0.360064 \end{pmatrix} \quad (37)$$

For $P = 889.16 + 1027 * 0.751 = 1660.427 \text{ kg}$, hence, the third of the weight will be 553.4757 kg , multiplied by gravity acceleration.

$$\begin{pmatrix} 0 \\ 5424.062 \\ 0 \end{pmatrix} = N_1 \begin{pmatrix} -0.62052 \\ 0.784191 \\ 0 \end{pmatrix} + N_2 \begin{pmatrix} -0.79383 \\ 0.490087 \\ 0.360064 \end{pmatrix} + N_3 \begin{pmatrix} -0.79383 \\ 0.490087 \\ -0.360064 \end{pmatrix} \quad (38)$$

With these conditions, the forces that the supports are subjected are:

$$N = \begin{pmatrix} 13522.95 \\ -5285.3 \\ -5285.3 \end{pmatrix} \text{ N} \quad (39)$$

This means that when the lander is on the ground, the main support is subjected to a compressive load, while both secondary supports are tensioned, thus, the main support may suffer from buckling.

Considering an ideal slim column, with restrained displacements and rotation on one end, and free on the other, when subjected to an axial compression load, it will have a linear-elastic behaviour. If this load does not exceed the critical pressure value, the column will remain straight, i.e., the column is stable if when any transversal load is applied, the beam deforms in the same direction but, as the load is took off, the column will reconfigure to its straight shape. Buckling phenomenon will be the first analysed.

4.2.1 Buckling Analysis

Either main supports or secondary are subjected to alternate compression and tension loads, depending on if the structure is landed or not. Once these are slender structures, there is the risk of buckling and, said so, this will be the criteria for dimensioning. For this, it was followed the Eurocode 3. This applies for steel structures that aim to obey security and application standards and requirements, afterwards verified by EN 1990.

The first approach for solving buckling instability problems in slender columns were introduced by Leonard Euler in 1744. Back then, the usual materials for construction were stones and wood, whose low resistance values led to use thick elements, thus, non-slender. Hence, elastic instability was not a problem. With the spread of steel construction on civil applications, buckling analysis started to gain a major role on slender columns subjected to compression loads, once the material started to collapse not because the stress was higher than the Yield Point, but due to elastic instability.

In a successful attempt to standard the different criteria adopted by different European countries, the Convention Européenne de la Construction Métallique led a set of trials and numerical simulations to determinate the curves used to design columns subjected to compression loads. Unlikely what it used before Eurocode, when there was only one curve to find the buckling limit point, there are now five curves that comprises residual stresses during the fabrication, the non-linearity of the beam axis, and the Young Module values scattering. The EC curves give the buckling coefficient, χ , also called buckling resistance reduction factor.

$$\chi = \frac{\sigma_{cr}}{f_y} \quad (40)$$

Where σ_{cr} is the the buckling resistance and f_y the Yied point. This value is based on the reduced slenderness $\bar{\lambda}$. This non-dimensional value is the quotient of the bar slenderness, λ , and the Euler slenderness, λ_E .

Hence

$$\bar{\lambda} = \frac{\lambda}{\lambda_E} \quad (41)$$

The bar slenderness can be obtained through

$$\lambda = \frac{l_E}{i} \quad (42)$$

where i is the gyration radius, which is the square root of the quotient between the inertia moment the cross section area, and l_E is the buckling length and it depends on the boundary conditions on its endpoints, and it can be obtained multiplying the bar length by a factor, μ .

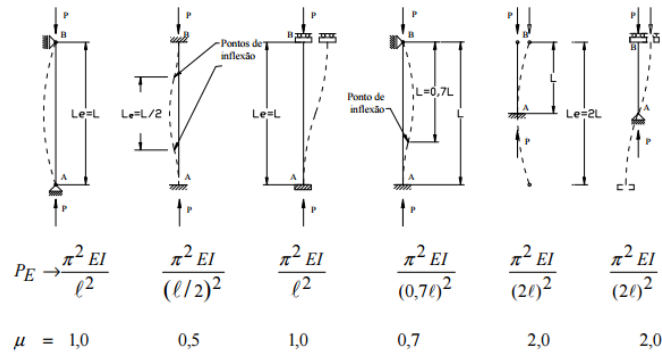


Fig. 36 Buckling factor function of boundary conditions

For the structure, it was considered two rotational joints for each end, as such, the μ factor will take the unity value, so l_e as the same value as L .

The Euler's slenderness, λ_E , may be defined as the slenderness where the Euler's critical stress is equal to the material yield point. It is given by:

$$\lambda_E = \pi \sqrt{\frac{E}{f_y}} \quad (43)$$

After that, in order to include imperfections on the dimensioning process, we use the imperfection coefficient. This is defined through the buckling curves, which are function of cross section, fabrication process, among others. The following table presents the correspondent curve to each profile.

As seen, all circular profiles may be class a, b, or c, depending on its fabrication method, said that, considering the following table, the value for the imperfection coefficient used was 0.49.

Table 21 Imperfection Factors

<i>Buckling curve</i>	a_0	a	b	c	d
<i>Imperfection factor, α</i>	0.13	0.21	0.34	0.49	0.76

Known α , we can obtain φ as

$$\varphi = 0.5. (\bar{\lambda}^2 + \alpha. (\bar{\lambda} - 0.2) + 1) \quad (44)$$

We can now achieve the plastic resistance reduction coefficient, χ , as

$$\chi = \frac{1}{\varphi + \sqrt{\varphi^2 + \bar{\lambda}^2}} \quad (45)$$

The resistance axial stress is, then

$$N_{b,Rd} = \chi \beta_A \cdot \frac{A \cdot f_y}{\gamma_{M1}} \quad (46)$$

Where

- γ_{M1} is the partial coefficient to consider on buckling, which is recommended by Eurocode to assume the unitary value;
- β_A for 1,2, or 3 cross section classes, or the relation between the effective section area and the cross section area for class 4.

The four cross section classes may be defined as:

- Class 1 – It is possible to create a ball joint, with the necessary rotation for a plastic analysis. There is no resistance reduction;
- Class 2 – These may achieve the plastic resistant momentum, however its rotation is limited by local buckling;
- Class 3 – Those where the axial stress were compressed, whose calculations are based on a linear-elastic stress distribution may achieve the Yield point, however, the local buckling prevents that the plastic resistant momentum to be achieved;
- Class 4- Those where local buckling occurs before achieving the Yield point.

That said, we must assure that the chosen profile does not reach the class 4. For tubular sections, the ration between diameter and thickness must obey the following relations in order to be defined as class 1, 2 or 3.

Table 22 different classes of profiles

<i>Class</i>	<i>relation</i>
<i>Class 1</i>	$d/t \leq 50\varepsilon^2$
<i>Class 2</i>	$d/t \leq 70\varepsilon^2$
<i>Class 3</i>	$d/t \leq 90\varepsilon^2$

Where ε is given by

$$\varepsilon = \sqrt{235/f_y} \quad (47)$$

4.2.2 Steel Structure

Primarily, the whole structure will be dimensioned in Stainless Steel, according to the assumptions made.

Main supports

For the main supports, in steel, the profiles chosen had standard dimensions with outer diameter of 60.3 mm and 48.3 mm.

Repeating the Eurocode 3 process for each profile, it was obtained the following table for Stainless Steel.

Table 23 Buckling analysis for steel frame main supports

<i>OD</i> <i>(mm)</i>	<i>t</i> <i>(m)</i>	<i>le</i> <i>(m)</i>	<i>A</i> <i>(m²)</i>	<i>I</i> <i>(m⁴)</i>	<i>i</i> <i>(m)</i>	λ
60,3	0,005	1,3	0,000869	0,000000335	0,019634169	66,21110517
60,3	0,004	1,3	0,000707	0,000000282	0,019971691	65,09213329
60,3	0,0036	1,3	0,000641	0,000000259	0,020101148	64,67292225
60,3	0,003	1,3	0,00054	0,000000222	0,020275875	64,11560505
48,3	0,005	1,3	0,00068	0,000000162	0,015434873	84,2248607
48,3	0,004	1,3	0,000557	0,000000138	0,015740262	82,59074939
48,3	0,003	1,3	0,000427	0,00000011	0,016050272	80,99551054

OD (mm)	t(m)	λ_E	$\bar{\lambda}$	α	ϕ	χ	$N_{b,Rd}$
60,3	0,005	63,13321	1,04875245	0,49	1,235112	0,522146	191961,9
60,3	0,004	63,13321	1,03102847	0,49	1,215494	0,530962	149094,2
60,3	0,0036	63,13321	1,02438836	0,49	1,226661	0,525917	175298,8
60,3	0,003	63,13321	1,01556072	0,49	1,257885	0,512182	231444,8
48,3	0,005	63,13321	1,33408178	0,49	1,667737	0,374739	132507,7
48,3	0,004	63,13321	1,30819823	0,49	1,6272	0,385375	111620
48,3	0,003	63,13321	1,28293041	0,49	1,588273	0,396103	87950,69

As we can see, the reduced resistance axial stress resulted from buckling instability is not a limiting factor when considering steel for the main support. Considering the slenderness, λ , it does not exceed the recommended value 180, as well. Therefore, the chosen profile is the one with the highest χ , i.e., the profile that will maximize axial resistance stress.

The main supports are then circular tubes with an outer diameter of 60.3 mm, with 3 mm thickness.

Table 24 Buckling analysis for steel frame main supports - chosen profile

OD (mm)	t(m)	le(m)	A(m ²)	I(m ⁴)	i(m)	λ
60,3	0,003	1,3	0,00054	0,00000022	0,02027587	64,1156050
λ_E	$\bar{\lambda}$	α	ϕ	χ	$N_{b,Rd}$ (N)	
63,13321	1,0155607	0,49	1,21549	0,530962	149094,2	

Obtaining ε for stainless steel, we have

$$\varepsilon = \sqrt{235/f_y} = 0.672 \quad (48)$$

d/t	$50\varepsilon^2$	$70\varepsilon^2$	$90\varepsilon^2$
20.1	22.596	31.635	40.673

Once $20.1 < 22.596$, it is a Class 1 cross section and there is not the risk of local buckling when using this profile subjected to compression loads.

The oversizing of these structural components are desirable considering diverse factors: the possibility of landing with only one support, the lander's modularity capacity, which may increase considerably the structure self-weight.

Secondary Supports

For these, the same approach was made and for steel we obtain:

Table 25 Buckling analysis for steel frame secondary supports

OD (mm)	T (m)	le (m)	A (m ²)	I (m ⁴)	i (m)	λ
21,3	0,00023	1,145	0,00137	6,29E-09	0,002143	534,368
21,3	0,00032	1,145	0,000182	7,68E-09	0,006496	176,2628
26,9	0,00023	1,145	0,000178	1,36E-08	0,008741	130,9924
26,9	0,00032	1,145	0,000238	0,000000017	0,008452	135,4782
33,7	0,0003	1,145	0,000289	3,44E-08	0,01091	104,9482
33,7	0,00036	1,145	0,00034	3,91E-08	0,010724	106,7718
42,4	0,00026	1,145	0,000325	6,46E-08	0,014099	81,214

OD (mm)	$t(m)$	λ_E	$\bar{\lambda}$	α	φ	χ	$N_{b,Rd}$
21,3	0,00023	63,13321	8,464135	0.49	38,34551	0,013202	9405,218
21,3	0,00032	63,13321	2,791918	0.49	5,032424	0,108467	10265,35
26,9	0,00023	63,13321	2,074857	0.49	3,111856	0,184127	17042,78
26,9	0,00032	63,13321	2,145911	0.49	3,279215	0,173647	21490,61
33,7	0,0003	63,13321	1,66233	0.49	2,239942	0,267289	40168,13
33,7	0,00036	63,13321	1,691215	0.49	2,295451	0,259908	45951,79
42,4	0,00026	63,13321	1,286391	0.49	1,593567	0,394613	66689,63

In these case we obtain higher slenderness values, however, apart from OD 21.3 mm profiles, are not critical, as the axial resistant stress values. Just as made with the main supports, it was consider the reduction coefficient, χ , values. The two outer diameter profiles that have, at least, 0.3 χ value are 33.7 mm and 42.4 mm. Once these structural components will not be subjected to compression loads while the lander is on operation, an oversizing is not as valuable as it is for the main supports, said so, the choice will be based on minimizing the mass. For that reason, it will be a circular profile with OD of 33.7 mm and 3mm thickness.

Table 26 Buckling analysis for steel frame secondary supports - chosen profile

$OD (mm)$	$t(m)$	$le(m)$	$A(m^2)$	$I(m^4)$	$i(m)$	λ
33.7	0,003	1,145	0,00029	3,44E-08	0,01091	104,9482
λ_E	$\bar{\lambda}$	α	φ	χ	$N_{b,Rd}$	
63,1332	1,66233	0,49	2.23994	0.267289	40168.13	

For local buckling, comparing the ratio between the outer diameter and thickens, we conclude that it is a Class 1 cross section, as well.

d/t	$50\varepsilon^2$	$70\varepsilon^2$	$90\varepsilon^2$
11.23	22.596	31.635	40.673

Columns

The critical load case to consider to dimension this feature will be when the structure is suspended, before released into the sea, where the columns will be tensioned. The relation among the axial stress and yield stress is given by

$$\sigma = \frac{N}{A} \quad (49)$$

Considering the purchase of different components, it is highly desirable to have the same profile for different applications. In terms of final cost, it can be convenient. For this reason the chosen profile was OD 63.05 mm with 3 mm thickness giving a higher range for the structure weight,

$$N = \sigma \cdot A = 520\,000\,000 \cdot 0.0054 = 280\,800\,N \quad (50)$$

Hoops

The main obstacle with this feature is its geometry, once there are not important loads for which this component is subjected.

For the chosen geometry, it must be consider the bending radius. When deforming a tube, bending it, the outside wall thickness is considerably reduced do to stretching the material, while the inside wall thickness, due to compression, increases.

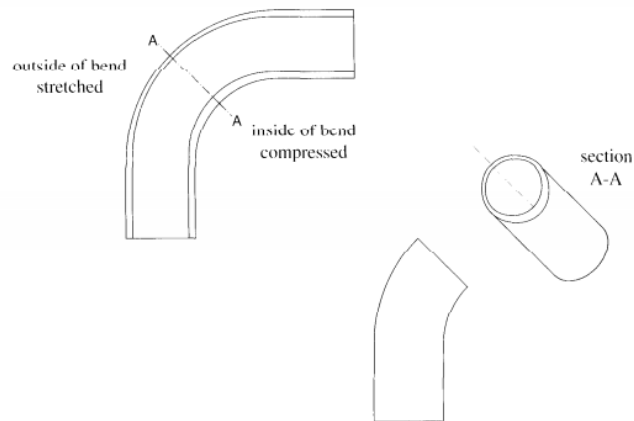


Fig. 37 - Representation of thickness deformation

For the landers geometry, the critical bending radius will be $100+R$ mm for the small superior hoop, where the profile radius gives R . The following table gives us the minimum bending radius achieved when using a mandrel.

Table 27 Bending radius for steel profiles

OD (inches)	OD (mm)	Thickness (mm)					
		0.889	1.2446	1.651	2.1082	2.362	3.05
0.5	12.7	25.4	22.224	19.05	15.875	-	-
1	25.4	114.3	98.425	82.55	73.025	63.5	-
1.5	38.1	203.2	177.8	155.575	133.35	111.125	92.075
2	50.8	304.8	266.7	228.6	152.399	127	101.6
2.5	63.5	609.6	508	431.8	355.6	279.4	228.6

Buoyancy tank

For the already chosen OD 60.3 mm profile, interpolating the given values we obtain for 3 mm thickness the minimum bending radius will be 127.77 mm, which is inferior to $100 + R$ mm, where R is 30.15 mm.

For the given dimensions is obtained the structure represented on the image of Fig. 38.

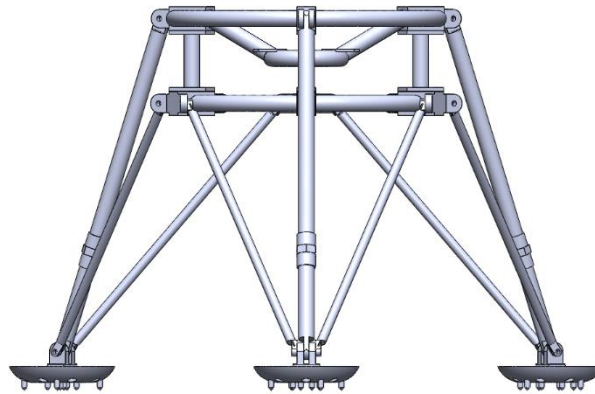


Fig. 38 Steel Structure

This structure, made of steel, has material volume of 0.031644 m^3 , with three concrete bases, gives a total weight of 286 kg. Once steel has a poorly corrosion resistance all volumes must be watertight in order to increase their durability. In order to compute the dimension of the buoyancy tank we have to calculate the immersed volume of the structure.

Table 28 Steel Structure submersed volume

<i>Component</i>	<i>Quantity</i>	<i>Unitary Volume</i> <i>(m³)</i>	<i>Volume</i> <i>(m³)</i>
<i>Main supports</i>	3	0,003713	0,011138
<i>Secondary Supports</i>	6	0,022007	0,132041
<i>Hoop</i>	2	0,004486	0,008972
<i>Small Hoop</i>	1	0,000897	0,000897
<i>Hoops Links</i>	3	0,000571	0,001713
<i>Links</i>	6	0,000896	0,005376
<i>Collumns</i>	3	0,001428	0,004284
<i>Central Module</i>	1	0,01885	0,01885
<i>Concrete Base</i>	3	0,005684	0,017052

Doing the sum of the different components volume we have a total immerse volume of 0.2 m^3 . Multiplying this volume value for the seawater density we obtain the upthrust force applied on the structure.

$$I = 0.2 * 1027 * 9.8 = 2016.16 \text{ N} \quad (51)$$

Restraining the volume of the tank with the outer radius of 345 mm, and a bare cylinder of 200 diameter in order to assembly the central module, the buoyancy tank height is given by

$$I_{\text{tank}} + I_{\text{structure}} = W \quad (52)$$

$$\pi(OR^2 - IR^2) * h * 1027 * 9.8 = (286 + 30) * 9.8 - 2016.16 \quad (53)$$

$$h = 0.551 \text{ m} \quad (54)$$

Once the tank will be fulfilled with water at the same pressure during operation, it is not considered the elastic instability that occurs on thin wall cylinders are subjected to external pressure.

4.2.3 Structure Material Alternative - POM

One of the goals of this project is the minimization of the final weight, allied with a low-cost product. As seen on the design of the *central module*, POM and aluminium are two low cost alternatives that still gives the necessary structural strength and stability, however, for the aluminium frame would be needed an in-depth study regarding corrosion problems. Said so, for this thesis POM will be considered as a structural material for the main frame.

Main Supports

The same procedure made for steel was adopted for POM although Eurocode 3 is mainly directed for steel structures. However, for the buckling analysis, most of the parameters are non-dimensional and they are given through relations with material properties and, for this reason, the Eurocode will be use with POM as a first approach.

Said so, once POM will be extruded for this application, it was considered different outer diameters with thickness varying between 5 and 10 mm. Considering the Yield stress as 71.7 MPa and a Young's Module of 2.9 GPa, we can obtain the needed parameters for the different components.

Table 29 Buckling analysis for POM frame main supports

OD (m)	t (mm)	le (m)	A (m ²)	I (m ⁴)	i (m)	λ
0,06	0,005	1,3	0,00086393	6,3612E-07	0,027135367	47,90795773
0,06	0,01	1,3	0,00157079	6,3562E-07	0,020116846	64,62245566
0,07	0,005	1,3	0,00102101	1,1756E-06	0,033974945	38,26349115
0,07	0,01	1,3	0,00188495	1,1781E-06	0,025	52
0,08	0,005	1,3	0,00117809	2,0105E-06	0,041311507	31,46822977
0,08	0,01	1,3	0,00219911	2,0101E-06	0,030233467	42,99870799
0,09	0,005	1,3	0,00133517	3,2209E-06	0,04911323	26,46944614
0,09	0,01	1,3	0,00251327	3,2201E-06	0,035794553	36,31837538

OD (m)	t (mm)	λ_E	$\bar{\lambda}$	α	φ	χ	$N_{b,Rd}$
0,06	0,005	19,97972	2,39783	0,49	3,913262	0,142738	8841,817
0,06	0,01	19,97972	3,234403	0,49	6,474111	0,082765	9321,511
0,07	0,005	19,97972	1,915117	0,49	2,75404	0,211274	15466,7
0,07	0,01	19,97972	2,60264	0,49	4,475513	0,123206	16651,51
0,08	0,005	19,97972	1,575009	0,49	2,077204	0,291418	24615,92
0,08	0,01	19,97972	2,152118	0,49	3,294075	0,172773	27242,29
0,09	0,005	19,97972	1,324816	0,49	1,653149	0,378506	36235,18
0,09	0,01	19,97972	1,817762	0,49	2,548482	0,230698	41572,11

For the final choice of the profile's dimensions, the resistant stress must be considered. As said by Gunter Erhard in 'Designing with Plastics' [65], "(...) *for calculations safeguarding against bending and buckling*" the safety factor must be superior or equal to three, therefore, some of this values will be considerably low for this application.

Table 30 Safety factor for polymeric buckling analysis

$OD\ (m)$	t (mm)	$N_{b,Rd}$	$N_{b,Rd}SF$
0,06	0,005	8841,817	2947,272
0,06	0,01	9321,511	3107,17
0,07	0,005	15466,7	5155,568
0,07	0,01	16651,51	5550,504
0,08	0,005	24615,92	8205,307
0,08	0,01	27242,29	9080,765
0,09	0,005	36235,18	12078,39
0,09	0,01	41572,11	13857,37

Considering the values after being reduced to its third, the only reasonable stresses, considering that must be a wide range values due to the landers modularity, correspond to profiles with 0.09 m outer diameter. For these two, considering χ , we have 0.378506 for 0.09x0.005 and 0.230698 for 0.09x0.01, this means that the first referred has a greater use of its cross-area, and hence it is preferable.

Considering the dimensions of the tube, as the mechanical properties of the POM, it must be considered, as well, its elastic stability due to the external hydrostatic pressure. For this, taking up the concept of critical length and the theoretical critical pressure for the collapse of the tube.

For an OD of 0.09 m and a thickness of 0.005 m, we have

$$l_c = \frac{4\pi\sqrt{6}}{27} (\sqrt[4]{(1-\nu^2)})(0.090) \sqrt{\frac{0.090}{0.005}} = 0.41738\text{ mm} \quad (55)$$

This value is inferior than the support length, 1,3 m. For this tubes the P_{tc} comes as:

$$P_{tc} = \frac{2E}{1-\nu^2} \left(\frac{t}{d}\right)^3 = 1176719\text{ Pa} \quad (56)$$

Which is minor than the required 2 MPa, correspondent to 200 depth range. For this, it will be analysed an OD 0.09 m tube with 0.006 m thickness. For this dimensions, the critical pressure is 2033370 Pa > 2000000 Pa.

Other criteria that must be considered is the local buckling, for this, we must assure that the profile fits, at least, the class 3 requirements. This can be written as

$$\frac{d}{t} < 90\varepsilon^2 \quad (57)$$

Where

$$\varepsilon = \sqrt{235/f_y} \quad (58)$$

It then comes as

$$\frac{0.090}{t} < \frac{90 \times 235}{71.7} \quad (59)$$

$$t > 0.000271 \text{ m} \quad (60)$$

The minimum thickness required to avoid local buckling is 0.000271 m, which is inferior than the actual thickness of 0.006 m.

Therefore, for the main supports we have an OD 0.09 m tube with 0.006 m thickness.

Table 31 Buckling analysis for POM frame main supports - chosen profile

$OD (m)$	$t(m)$	$le(m)$	$A(m^2)$	$I(m^4)$	$i(m)$	λ
0.09	0,006	1,3	0,001583363	3,22056E-06	0,045099889	28,82490456
λ_E	$\bar{\lambda}$	α	φ	χ	$N_{b,Rd} (N)$	$N_{b,Rd,SF} (N)$
19,9797	1,44270	0,49	1,845167	0,333836	37899,38	12633,13

Secondary supports and columns

As already mentioned, from the point of the view of the acquisition of these products from the market it is highly desirable that all the tubes have the same dimensions. It can bring some advantages from the economic standpoint, and, since POM's density is considerably low, the oversizing of the tubes will not reflect in a significant increase of the structural weight.

This way, for the secondary supports, it is already known that the tube length exceeds the critical length and, as such, the equation for the critical pressure is the same as it is for the

main supports, wherefore we know that the tube bears the require external pressure for the depth range of 200 m without collapsing due to elastic instability.

Regarding the Eurocode analysis for buckling, once tube length is different, consequently other buckling parameters will change, and the obtained values are now presented on Table 32.

Table 32 Buckling analysis for POM frame secondary supports - chosen profile

$OD(m)$	$t(m)$	$le(m)$	$A(m^2)$	$I(m^4)$	$i(m)$	λ
0.09	0,006	1,145	0,001583363	3,22056E-06	0,045099889	25.38808902
λ_E	$\bar{\lambda}$	α	φ	χ	$N_{b,Rd}$	$N_{b,RdSF}$
19,9797	1,20693	0,49	1,56965	0.401423	45572.37	15190.79

For the columns, we must know the axial resistance when tensioned.

$$717000000 = \frac{N}{0.00158336} \quad (61)$$

$$N = 113526.9 \text{ N} \quad (62)$$

Hoops

Once again, the main obstacle with hoops is its own geometry. The dimensions for this feature were considered an OD of 60.3 mm, with 3 mm thickness, the same as used in steel. Considering the critical external pressure, for this dimensions the value is 845088,2105 Pa, inferior to the required. Said so, increasing one mm thickness obtains 2003172,054 Pa, which is suitable for the application.

To obtain the designed geometry, several polymer fabrication processes must be considered. While steel hoops were normally produced by deforming it on the plastic domain, the same procedure is not valid in polymers once it would seriously decrease its mechanical properties and the effect on the cross-area thickness would be more significant than it is on steel hoops, as so, bending is not an option.

We must consider processes that will give us the accurate shape at the time the polymer is synthesized. For both hoops (1 m diameter and 0.2 m diameter) it is an option to fabricate it hollow or full. While a hollow geometry would reflect in a lower structural weight, a solid body could improve the mechanical resistance of these components. Considering processes for both cases we have:

Solid body

- Extrusion - For the bigger hoop we can obtain 3 arches with 120° using bending rollers after extrusion, yet, it does not grant precise dimensions, mainly on the profile diameter;
- Injection moulding – for this process it would be needed a mould able of bearing the high pressures needed for this, what could be a major expense. In addition, due to residual stress it is more likely to have geometric anomalies due to residual stress.

Hollow body

- Extrusion – This process is suitable for either solid or hollow bodies, however, the geometric inconsistencies would be more significant for this instance.
- Blow moulding – this process could be a suitable option for manufacturing the arch of the bigger hoop. In this process, pressurized air is used to inflate soft thermoplastic into a mould cavity;
- Gas-assist injection moulding – this process is also suitable for the small hoop. Here, pressurized nitrogen is introduced into a mould cavity previously fulfilled with thermoplastic. For this, high resistant moulds would be needed, what would reflect in a cost increase, as well.
- Rotation moulding – in this process, the gravity is used to achieve a hollow geometry on a rotational mould. This could be applicable for both bigger and smaller hoop and, when compared with previous listed processes, this is the most inexpensive.

The hoops will be treated as hollow bodies, considering that the best process to fabricate them is rotational moulding, with an outer diameter of 60.3 mm and 4 mm thickness.

Buoyancy tank

For a POM structure, due to its voluminous profiles, there is a higher up thrust force. Doing the same procedure as recommended previously, we have a total up thrust force of 557.8 N, to obtain the buoyancy necessary to keep the lander on the surface, the buoyancy tank must have at least 0.2 m of diameter, however, and oversizing this component it was considered a value of 0.3 m.

This final POM structure, with three concrete bases as well, has a total volume of 0,05542 Cubic meters and weighting 109.39 kg.

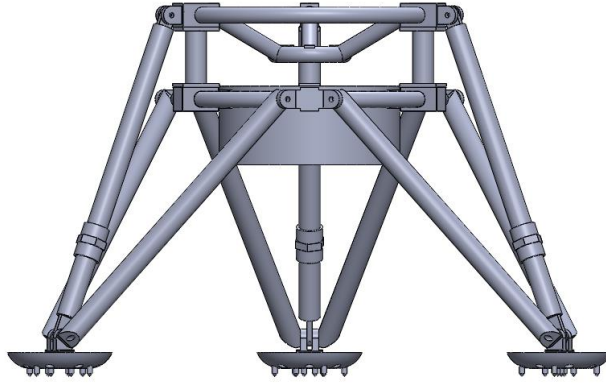


Fig. 39 POM structure

5 Drag Force

Although bot structures are considered open framed, the geometry of the tank responsible for the buoyancy is the main contribution to increase the projected area and so for the drag force. The drag, D , is defined as the resultant force in the direction of the upstream velocity, considered for this project with the intensity of 1 m/s.

The drag can be obtained through [71]

$$D = \frac{1}{2} C_D \rho U^2 A \quad (63)$$

Where

- C_D is the drag coefficient;
- U is the upstream velocity;
- ρ is the water density;
- A is the projected area.

For these calculations, it will be used as an approach the geometry of a smooth cylinder and, for those, the drag coefficient can be obtained from a relation with the Reynolds number. This is a dimensionless parameter used as a criterion to distinguish between laminar and turbulent flow.

This can be computed as;

$$Re = \frac{\rho U D}{\mu} \quad (64)$$

Where

- D is the cylinder diameter;
- μ is the dynamic viscosity of the seawater, considered 0.00108 Ns/m² [72] for 20°C;

The Reynolds number is, then, for both POM and steel structure, equal to

$$Re = \frac{1027 * 1 * 0.345 * 2}{0.00108} = 6.6 \times 10^5 \quad (65)$$

Considering the Fig. 40, we can obtain the Drag coefficient as a function of the Reynold's number and geometry.

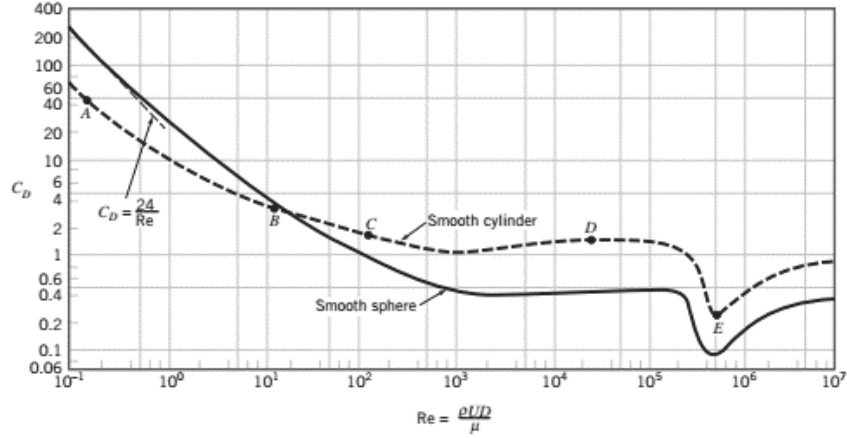


Fig. 40 Drag coefficient as a function of the Reynold's number and geometry

The drag coefficient considered to oversizing the structure can be considered 1.5. Said so, for the steel structure, the drag force can be estimated as;

$$D = \frac{1}{2} C_D \rho U^2 A \quad (66)$$

$$D = \frac{1}{2} * 1.5 * 1027 * 1^2 * 0.345 * 2 * 0.6 = 318,8 \text{ N} \quad (67)$$

Using this value for the drag force, it is possible to know the maximum upstream velocity bearable by the structure without rotating over one base. Once the drag coefficient is a function of the Reynold's number, this is an iterative process, yet, it will be considered a value of 1.5 for the coefficient.

Having this in mind, for a 50° between the support radius and the vertical axis, we have;

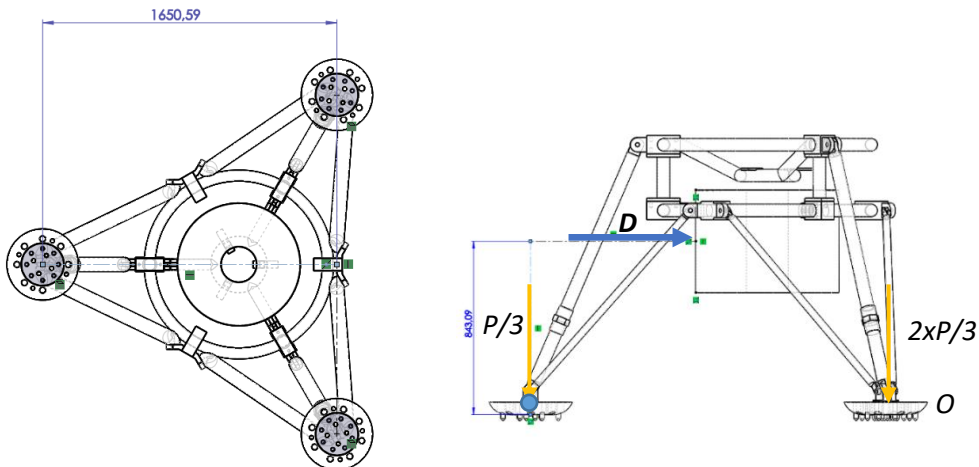


Fig. 41 Structure dimensions - Steel

In order to do not flip, the sum of momentums in the point O must be null. It comes, in the scalar form as:

$$\sum M_O = 0 \quad (68)$$

$$0.84309 * D - \frac{P}{3} * 1.65059 = 0 \quad (69)$$

$$D = \frac{1}{3} * 316 * 9.8 * \frac{1}{0.84309} \quad (70)$$

$$D = 1224.39 \text{ N} \quad (71)$$

Once

$$D = \frac{1}{2} C_D \rho U^2 A \quad (72)$$

The maximum upstream velocity, U, is given by

$$U = \sqrt{\frac{2D}{C_D \rho A}} = 2.77 \text{ m/s} \quad (73)$$

For the POM structure, the same analysis can be done. Considering the same drag coefficient, the resultant force is given by;

$$D = \frac{1}{2} * 1.5 * 1027 * 1^2 * 0.345 * 2 * 0.3 = 159,4 \text{ N} \quad (74)$$

To determine the maximum upstream velocity bearable by the structure, we have, once again to sum the momentums on one single point of the base.

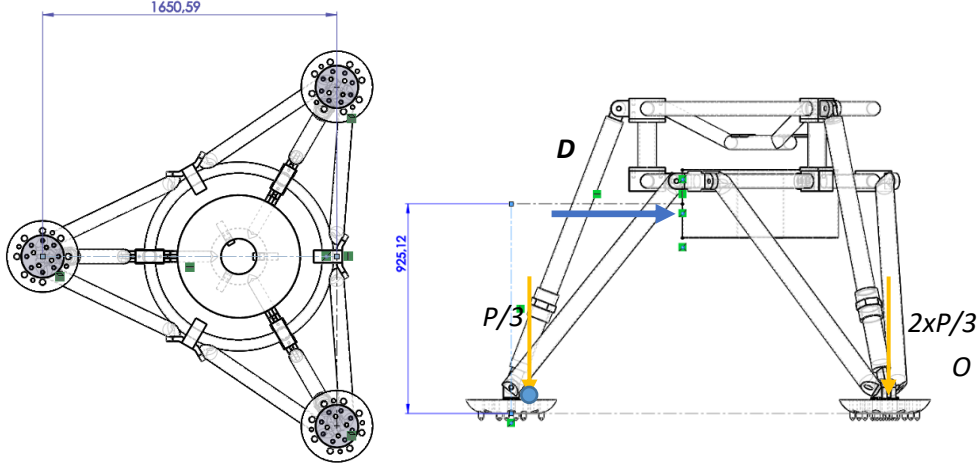


Fig. 42 Structure dimensions - POM

In the scalar form, we can write

$$\sum M_O = 0 \quad (75)$$

$$0.92512 * D - \frac{P}{3} * 1.65059 = 0 \quad (76)$$

$$D = \frac{1}{3} * 109.39 * 9.8 * \frac{1}{0.84309} \quad (77)$$

$$D = 855.5642 \text{ N} \quad (78)$$

The maximum upstream velocity, is then give by

$$U = \sqrt{\frac{2D}{C_D \rho A}} = 2.316463 \text{ m/s} \quad (79)$$

6 Finite Element Analysis - Simulation

After dimensioning the main structural components of the lander, it was made a numeric simulation using the software ANSYS to evaluate its structural behaviour. This is a software mainly used to “to simulate interactions of all disciplines of physics, structural, vibration, fluid dynamics, heat transfer and electromagnetic for engineers” as it is stated in ANSYS website [73].

To simulate the distinct scenarios that the structure may be subjected during its working life, it was assumed that five different load cases may occur:

- Load Case no 1 – the structure is landed on the seafloor. Here it was considered 2 MPa uniform pressure applied to all the surface except for the buoyancy tank, due to hydrostatic pressure. This will be filled with water at the same pressure, therefore, the pressure differential is null.
- Load Case no 2 – when deployed into the water, the lander will be carried by a crane that will be interacting with the structure through three eyebolts dispersed axisymmetric on the superior loop.
- Load Case no 3 – in this case, as the previously mentioned, concerns the relocation of the structure, however, with the main objective with this load case is to simulate the structural response in case of one of the interaction with crane breaks, i.e., the whole structure is supported by two eyebolts non-axisymmetric placed.
- Load Case no 4 – this case is also to simulate the structural response in case of the structure is only supported by one eyebolt.
- Load Case no 5 – like load case 1, this pretends to simulate what is the structural response to the drag forces for a current velocity of 1 m/s.

Each one of these load cases were simulated on the software for both structures, it will be shown the Total Deformation – that represent the displacements, shown in meters, Equivalent Strain – dimensionless, and the Equivalent von-Mises Stress represented in the whole structure in Pa. For each load case, some specific structural components are more solicited, the graphic representation of these will be on Annex, yet, the values obtained will be presented.

For load cases no 3 and 4, in a real life occasion, the structure would move and, as so, its centre of mass would create different loads as the considered in this simulation. Here, once the boundary condition is applied on the drill holes, it is assumed that the structure remains fixed.

Using the ‘Modal’ feature of ANSYS Workbench, it was possible to simulate the dynamic response of the structure, as well.

For both structures it was automatically generated a tetrahedral mesh, comprising 105958 elements for the POM structure, and 126358 for the steel structure.

6.1 Load Case no 1

This first load case expresses the load that the lander will be exposed during most of its operation time, when it is monitoring the seafloor. For this case it was consider that the lander bases are fixed to the seabed, while this is subjected to a uniform pressure on all surfaces, except on the tank, as referred, and its self-weight, plus the 30kgf assumed for the central module.

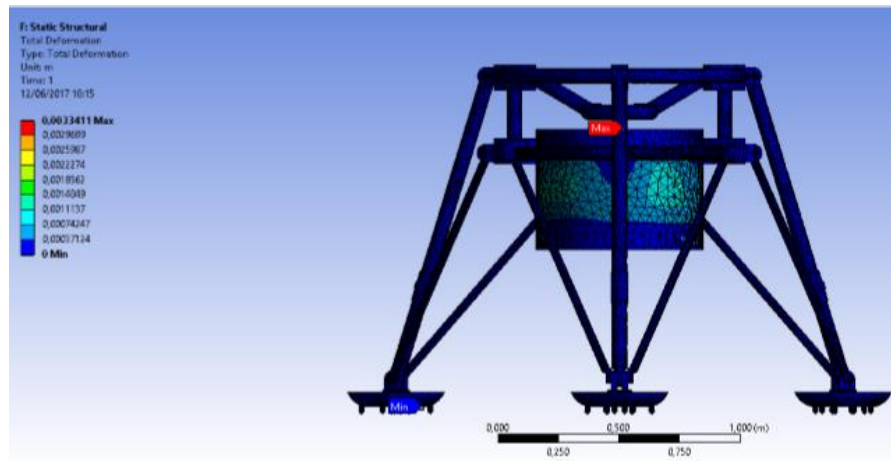


Fig. 43 Total Deformation (displacements in m) on Load Case no 1 - Steel Structure

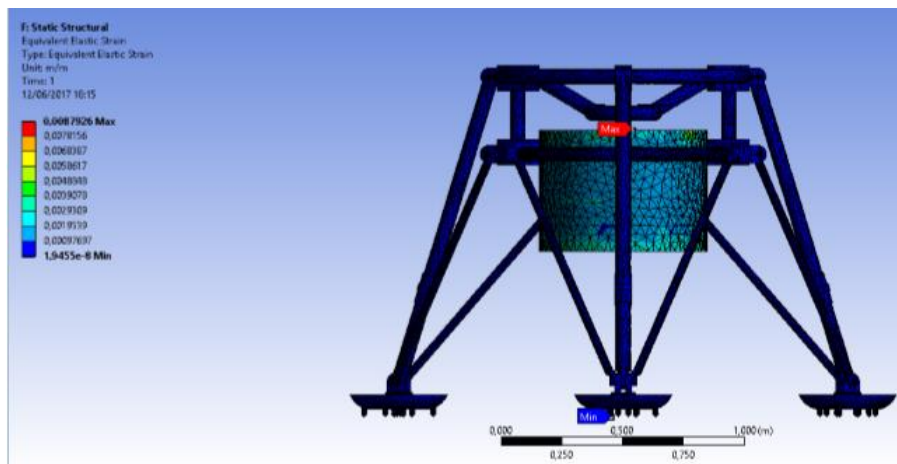


Fig. 44 Equivalent Elastic Strain on Load Case no 1- Steel Structure

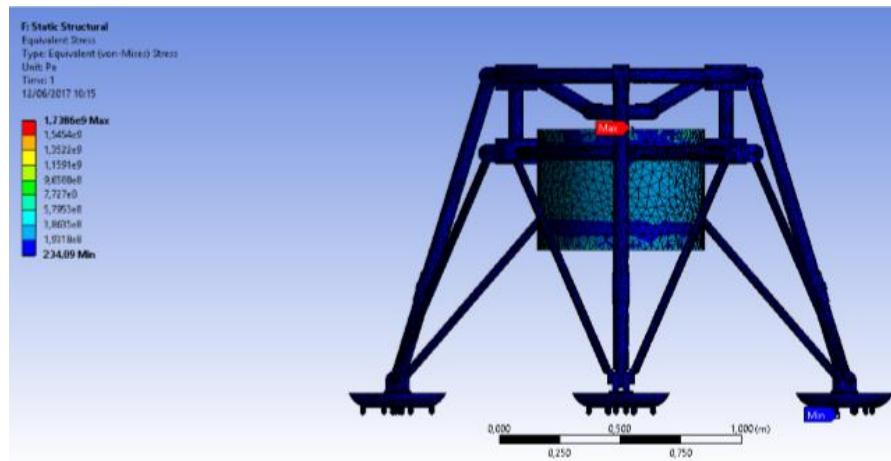


Fig. 45 Equivalent von-Mises Stress (Pa) on Load Case no 1- Steel Structure

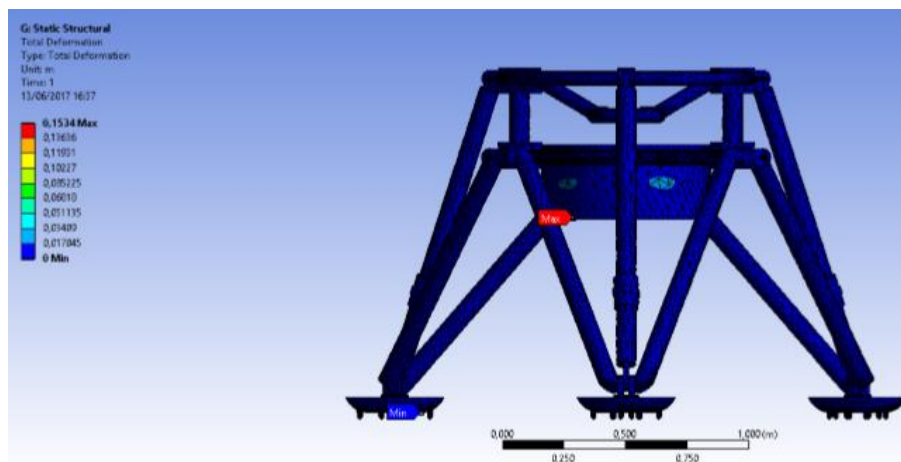


Fig. 46 Total Deformation (displacements in m) on Load Case no 1 - POM Structure.

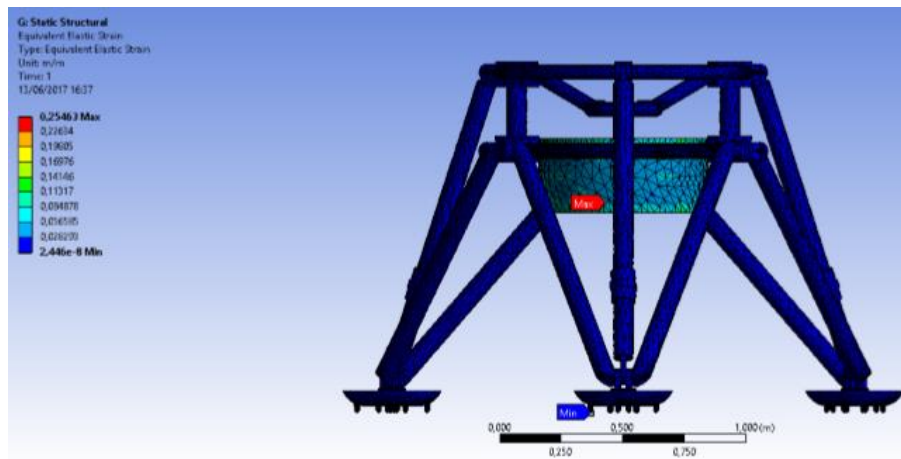


Fig. 47 Equivalent Elastic Strain on Load Case no 1- POM Structure.

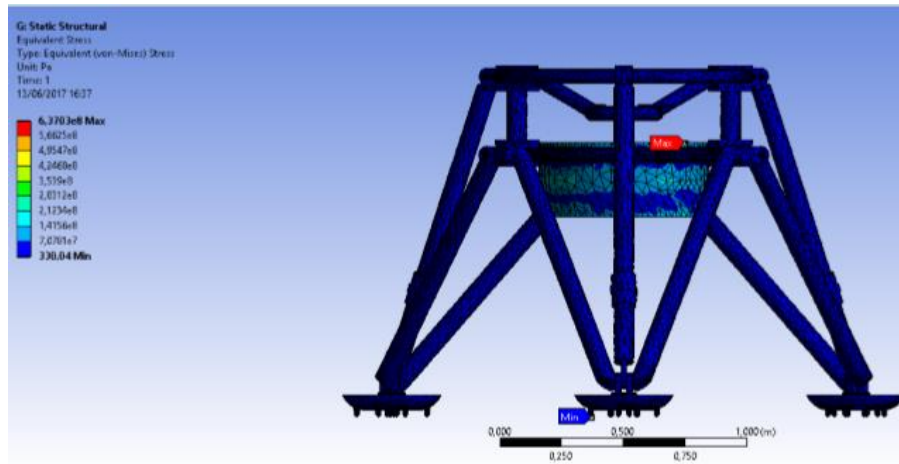


Fig. 48 Equivalent von-Mises Stress (Pa) on Load Case no 1- POM Structure.

Components Stress

The Table 33 presents the maximum and minimum stress values for which the main structural components are being subjected to Load Case 1.

Table 33 Structural Components Stress - Load Case 1

Component	Steel Structure (MPa)		POM Structure (MPa)	
	Min	Max	Min	Max
Main support – upper part	0.1305	3.9346	0.01465	0.8825
Main support – lower part	0.14466	4.6162	0.01613	0.8660
Secondary	0.3498	16.2380	0.03897	5.3938
Superior Hoop	0.01311	18.4360	0.1308	14.0730
Column	0.08025	1.38740	0.9287	16.4170

6.2 Load Case no 2

As mentioned, the objective with Load Cases number 1, 2 and 3 is to simulate what will be the structure behaviour during relocation with a crane. During this process, the only loads that the structure is subjected is the weight. It was assumed that the structure is fixed on three drill holes that exist in each block on the superior hoop. Fig. 49, Fig. 50 and Fig. 51 represents graphically the simulation when it is the steel structure, while Fig. 52, Fig. 53 and Fig. 54 the POM structure simulation.

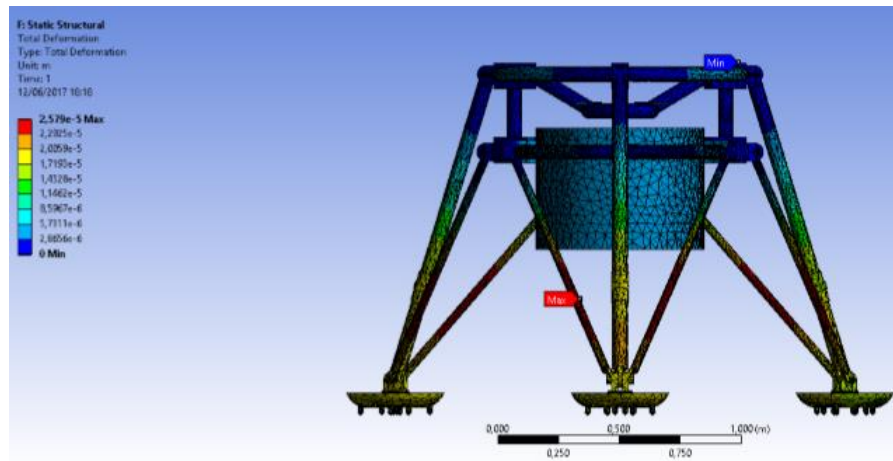


Fig. 49 Total Deformation (displacements in m) on Load Case no 2 - Steel Structure.

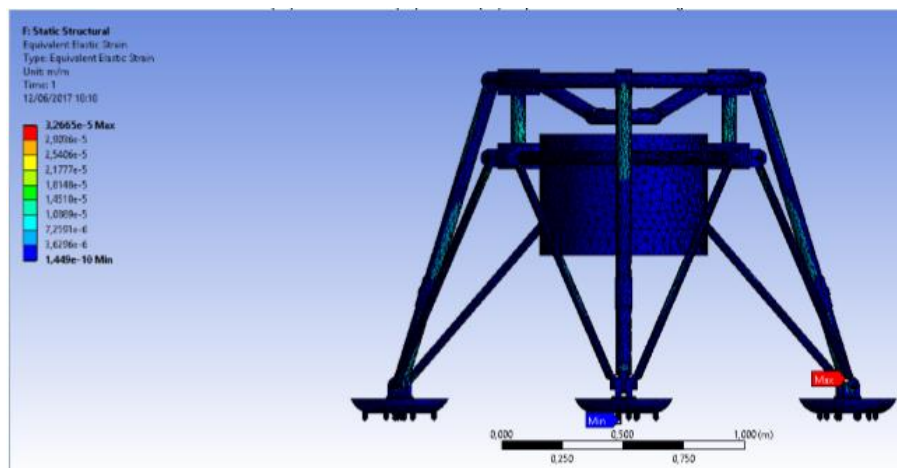


Fig. 50 Equivalent von-Mises Stress (Pa) on Load Case no 2- Steel Structure.

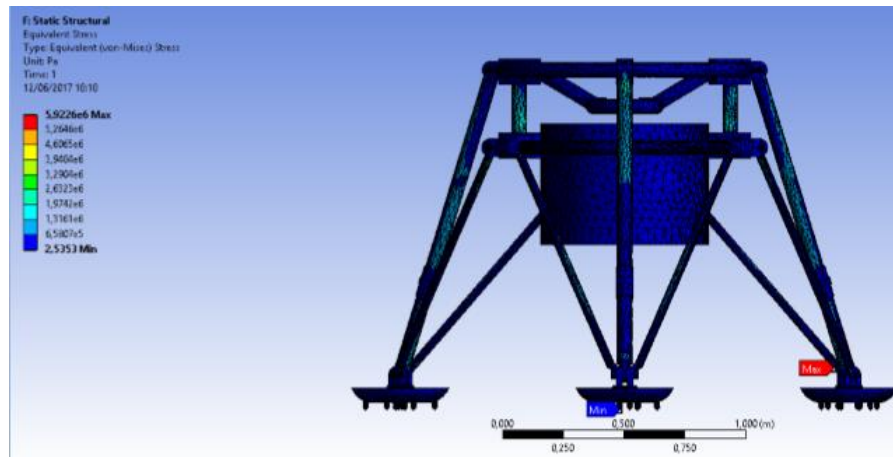


Fig. 51 Equivalent von-Mises Stress (Pa) on Load Case no 2- Steel Structure.

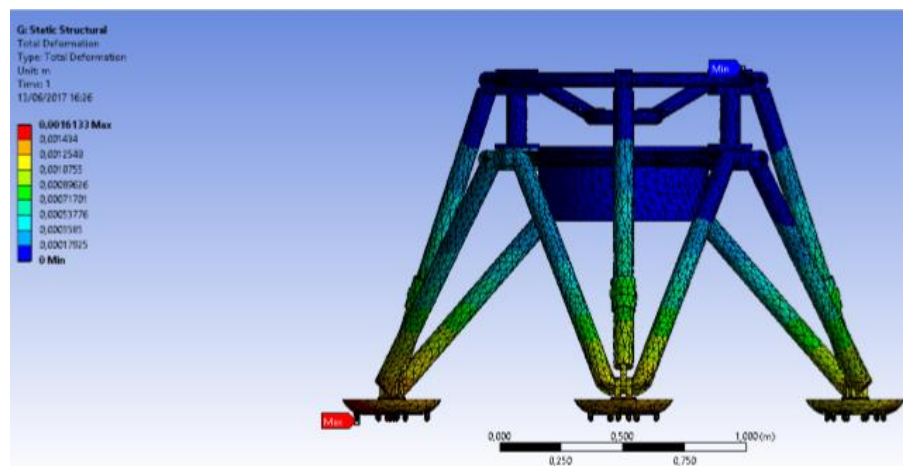


Fig. 52 Total Deformation (displacements in m) on Load Case no 2 - POM Structure.

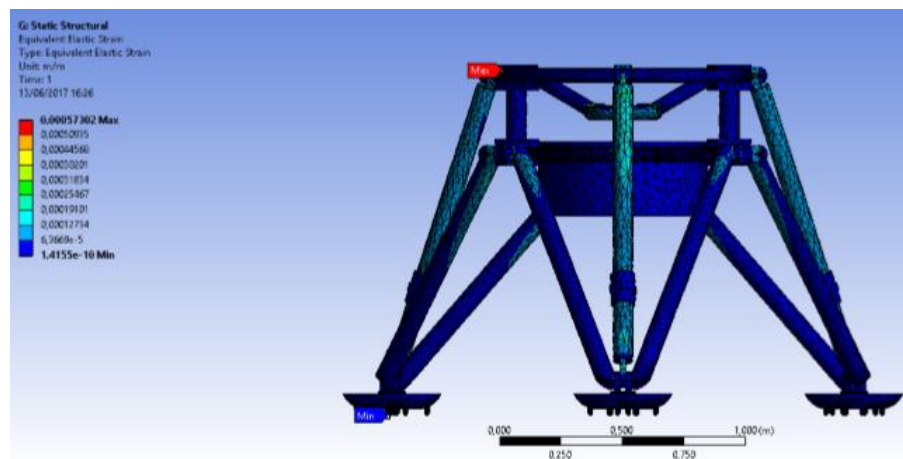


Fig. 53 Equivalent Elastic Strain on Load Case no 2- POM Structure.

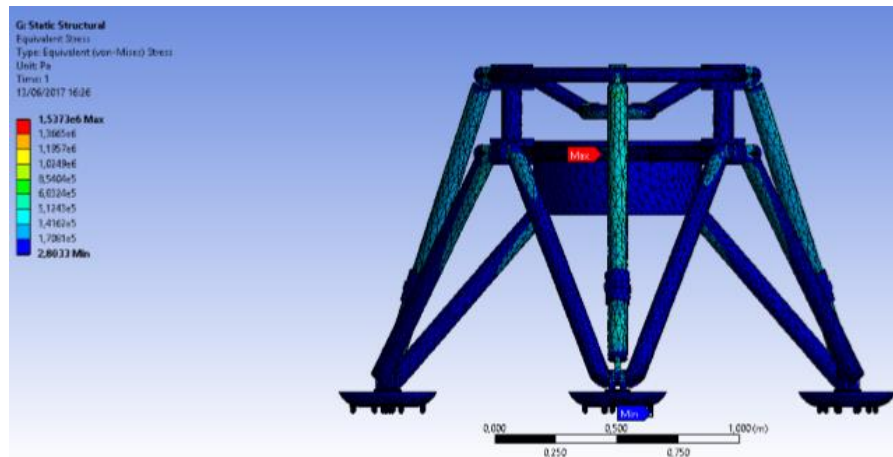


Fig. 54 Equivalent von-Mises Stress (Pa) on Load Case no 2- POM Structure.

Components Stress

The Table 34 presents the maximum and minimum stress values for which the main structural components are being subjected to Load Case 2.

Table 34 Structural Components Stress - Load Case 2

Component	Steel Structure (MPa)		POM Structure (MPa)	
	Min	Max	Min	Max
Main support – upper part	0.069967	2.2925	0.040401	0.67092
Main support – lower part	0.090921	1.657	0.021319	0.41762
Secondary	0.0047653	57.667	0.004752	0.1246
Superior Hoop	0.000581	3.9452	0.008273	1.3478
Column	0.078039	2.1112	0.00048535	1.1231

6.3 Load Case no 3

For this case, the structure was fixed to one of the drill holes of the two blocks. The Deformation, Equivalent Elastic Strain and Equivalent von-Mises Stress are represented on Fig. 55, Fig. 56 and Fig. 57.

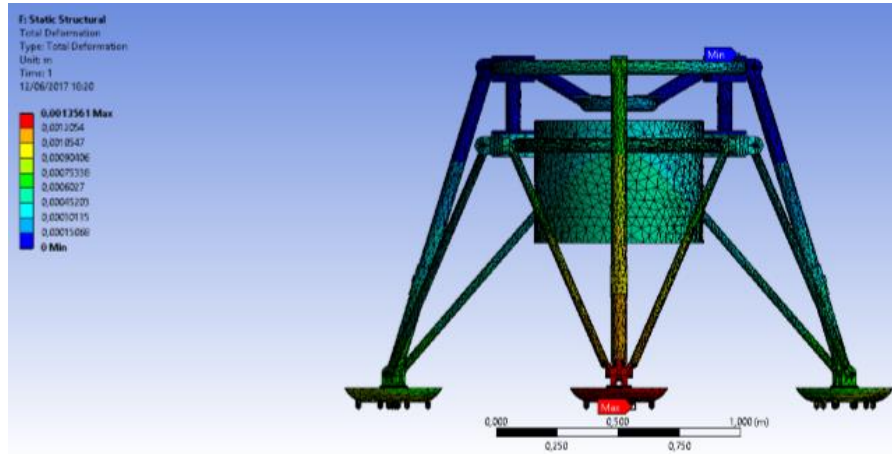


Fig. 55 Total Deformation (displacements in m) on Load Case no 3 - Steel Structure.

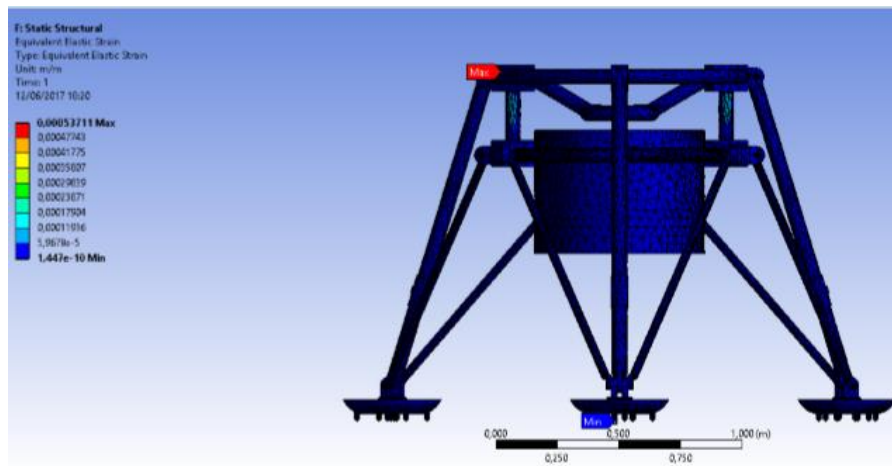


Fig. 56 Equivalent Elastic Strain on Load Case no 3- Steel Structure.

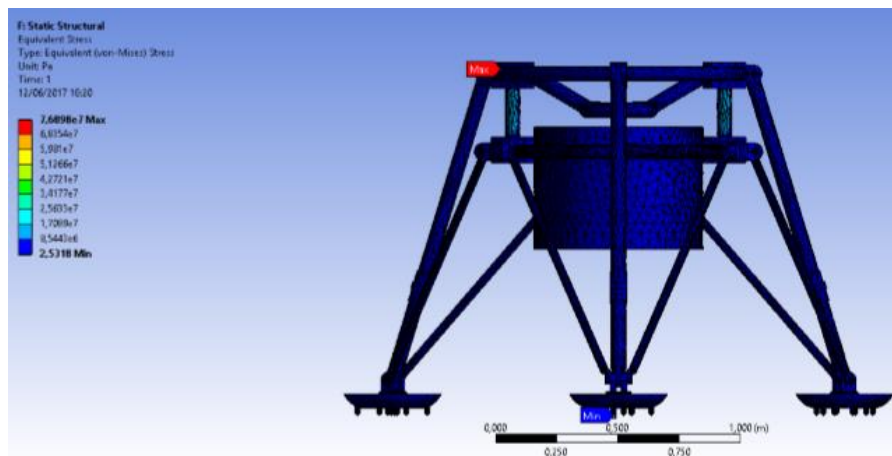


Fig. 57 Equivalent von-Mises Stress (Pa) on Load Case no 3- Steel Structure.

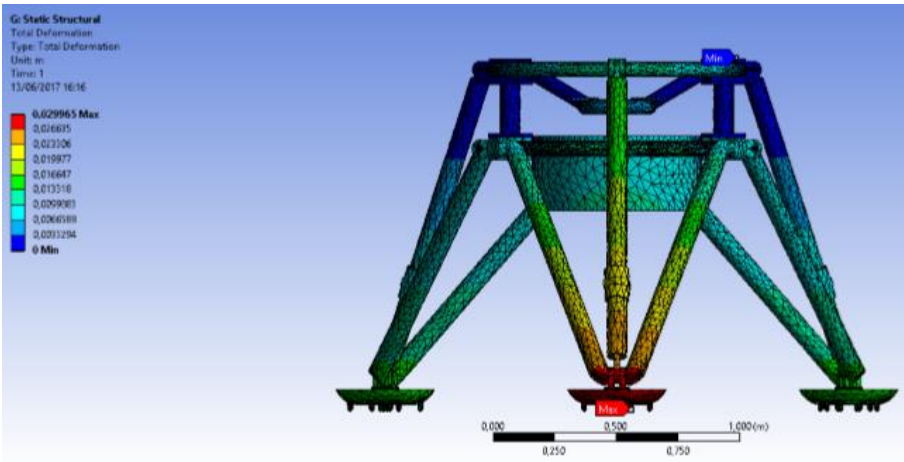


Fig. 58 Total Deformation (displacements in m) on Load Case no 3 - POM Structure.

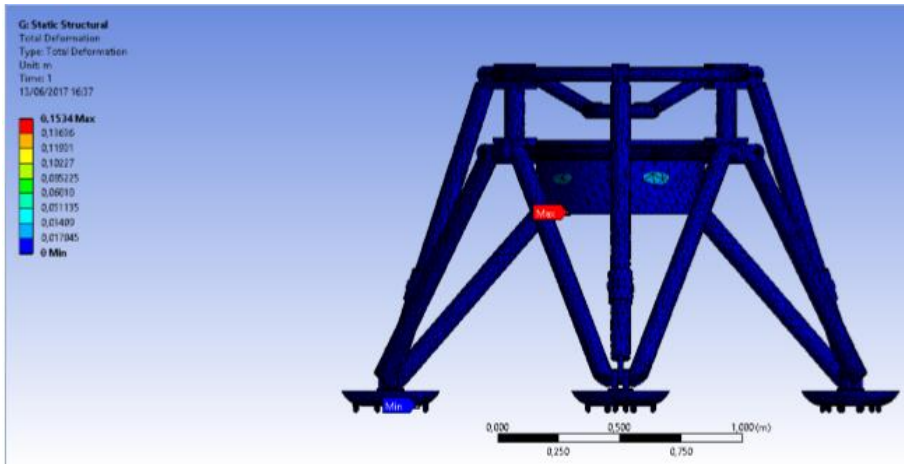


Fig. 59 Equivalent Elastic Strain on Load Case no 3- POM Structure.

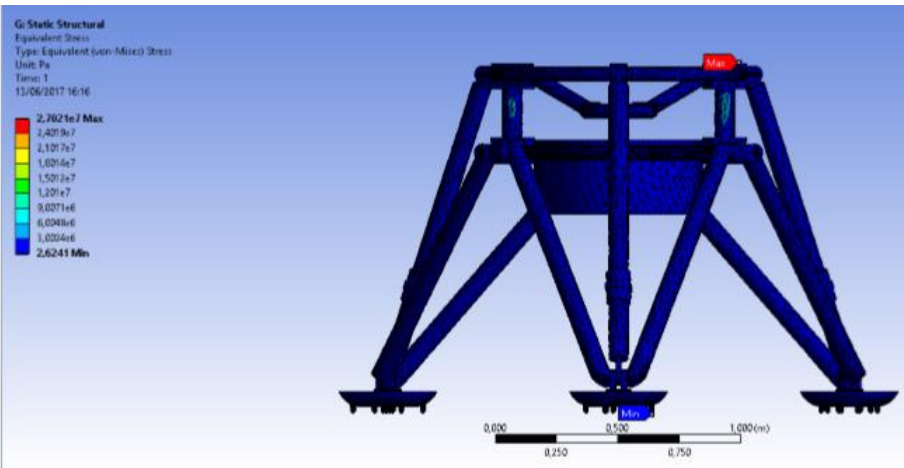


Fig. 60 Equivalent von-Mises Stress (Pa) on Load Case no 3- POM Structure.

Components Stress

The Table 35 presents the maximum and minimum stress values for which the main structural components are being subjected to Load Case 3.

Table 35 Structural Components Stress - Load Case 3

<i>Component</i>	<i>Steel Structure (MPa)</i>		<i>POM Structure (MPa)</i>	
	Min	Max	Min	Max
<i>Main support – upper part</i>	0.002954	2.448	0.042825	0.06733
<i>Main support – lower part</i>	0.080353	1.3874	0.017203	0.56159
<i>Secondary</i>	0.011433	4.215	0.0041598	1.8914
<i>Superior Hoop</i>	0.0038708	76.898	0.0043613	27.021
<i>Column</i>	0.26006	12.462	0.01268	3.4923

6.4 Load Case no 4

Like previous load cases 2 and 3, however the structure is merely fixed to one drill grill. The graphic representation for the steel structure and POM structure are represented on Fig. 66 Fig. 61 and Fig. 62, and Fig. 63 Fig. 58 Fig. 59, respectively.

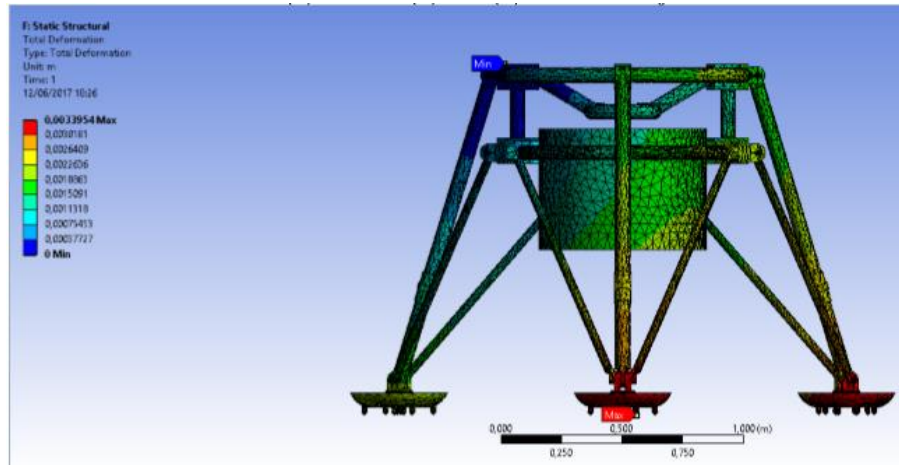


Fig. 61 Total Deformation (displacements in m) on Load Case no 4 - Steel Structure.

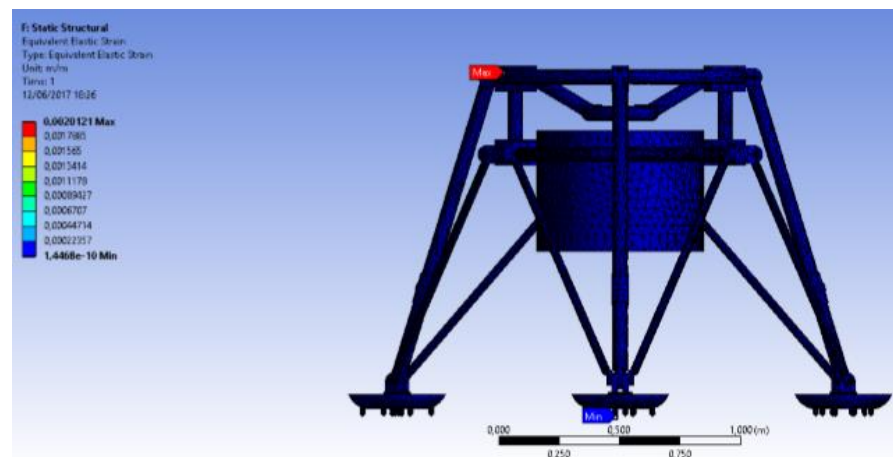


Fig. 62 Equivalent Elastic Strain on Load Case no 3- Steel Structure.

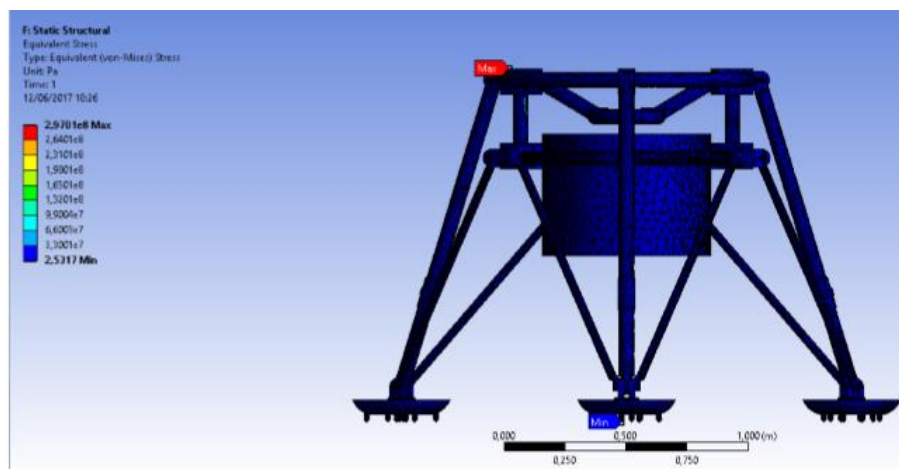


Fig. 63 Equivalent von-Mises Stress (Pa) on Load Case no 4- Steel Structure.

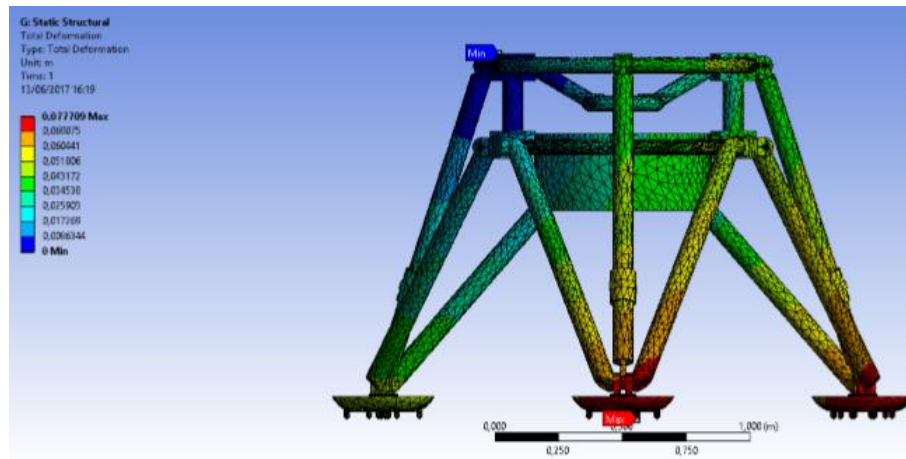


Fig. 64 Total Deformation (m) on Load Case no 4 - POM Structure.

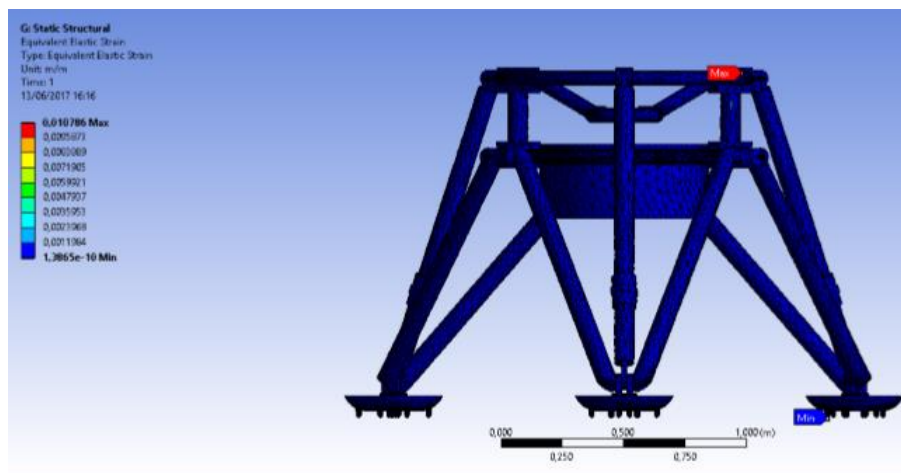


Fig. 65 Equivalent Elastic Strain on Load Case no 4- POM Structure.

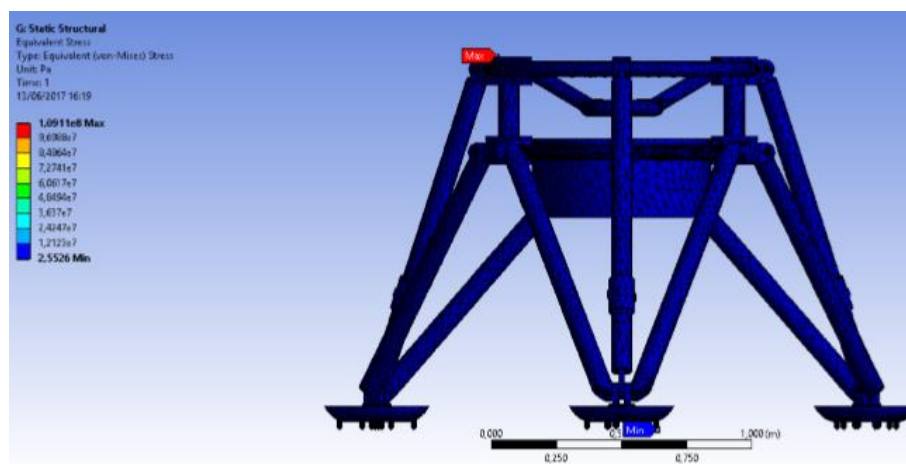


Fig. 66 Equivalent von-Mises Stress (Pa) on Load Case no 4 - POM Structure.

Components Stress

The Table 36 presents the maximum and minimum stress values for which the main structural components are being subjected to Load Case 4.

Table 36 Structural Components Stress - Load Case 4

<i>Component</i>	<i>Steel Structure (MPa)</i>		<i>POM Structure (MPa)</i>	
	Min	Max	Min	Max
<i>Main support – upper part</i>	0.030937	3.5378	0.013092	1.0503
<i>Main support – lower part</i>	0.056089	1.4266	0.010160	0.8949
<i>Secondary</i>	0.021474	5.2419	0.071103	1.198
<i>Superior Hoop</i>	0.04618	297.0	0.0057667	109.118
<i>Column</i>	0.43897	17.845	0.071858	10.783

6.5 Load Case no 5

The main goal of simulating this load case it to know the whole structure deformation due to drag effect. For this, on the structural simulation of ANSYS, it was considered the self-weight of the structure, with the addition of 30 kg of the central module, the hydrostatic pressure, and the drag force resultant from the buoyancy tank for each structure. It was assumed that all bases were fixed to the seafloor, with a horizontal current orientation.

The resultant deformation, strain, and stress are now shown, first on the steel structure (Fig. 67, Fig. 68, Fig. 69) and then on the POM structure (Fig. 70, Fig. 71, Fig. 72).

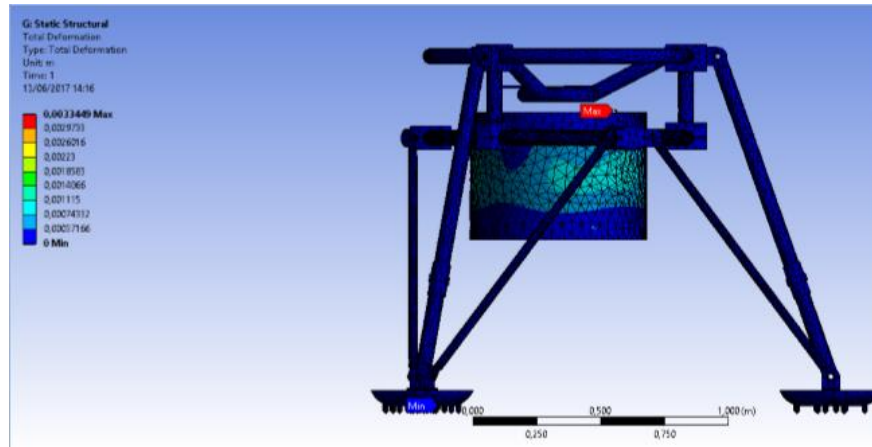


Fig. 67 Total Deformation (displacements in m) with Drag force- Steel Structure.

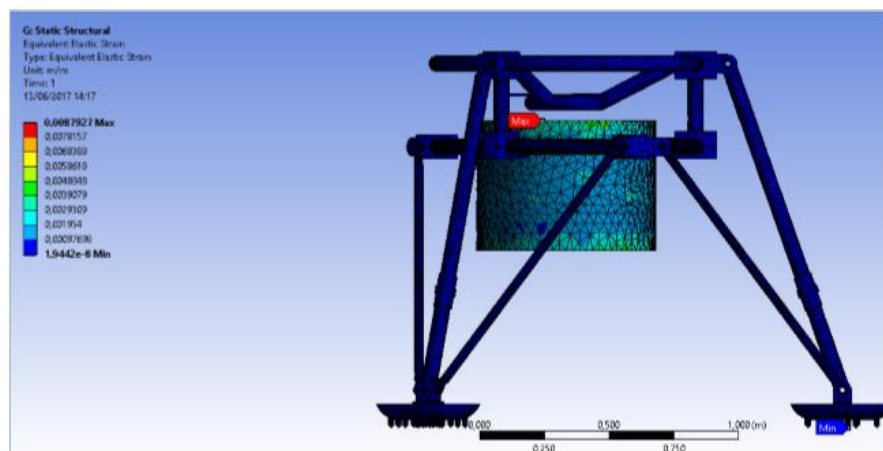


Fig. 68 Equivalent Elastic Strain with Drag force - Steel Structure.

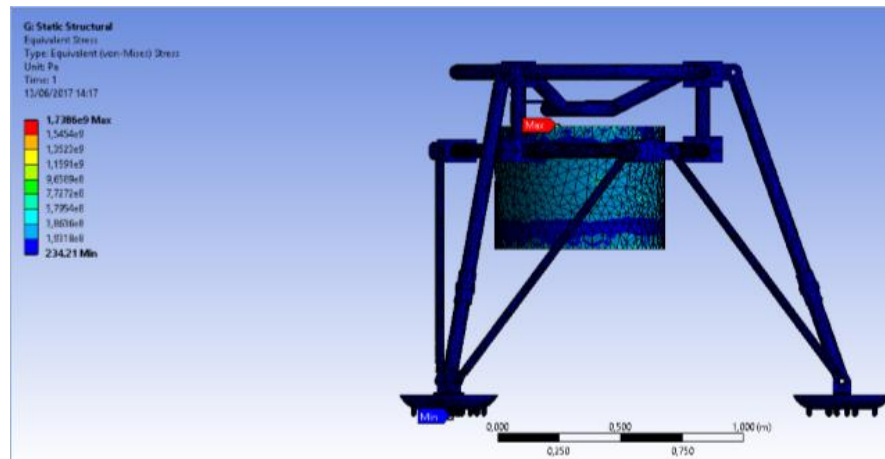


Fig. 69 Equivalent von-Mises Stress (Pa) with Drag force - Steel Structure.

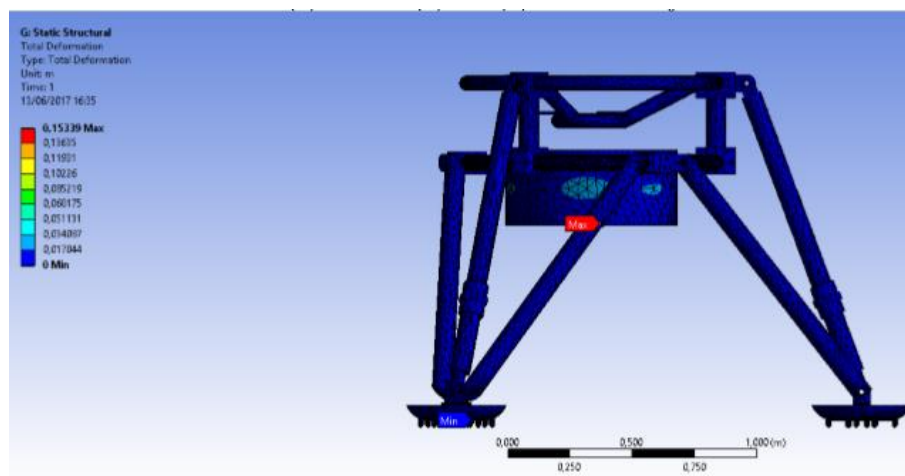


Fig. 70 Total Deformation (displacements in m) with Drag force - POM Structure.

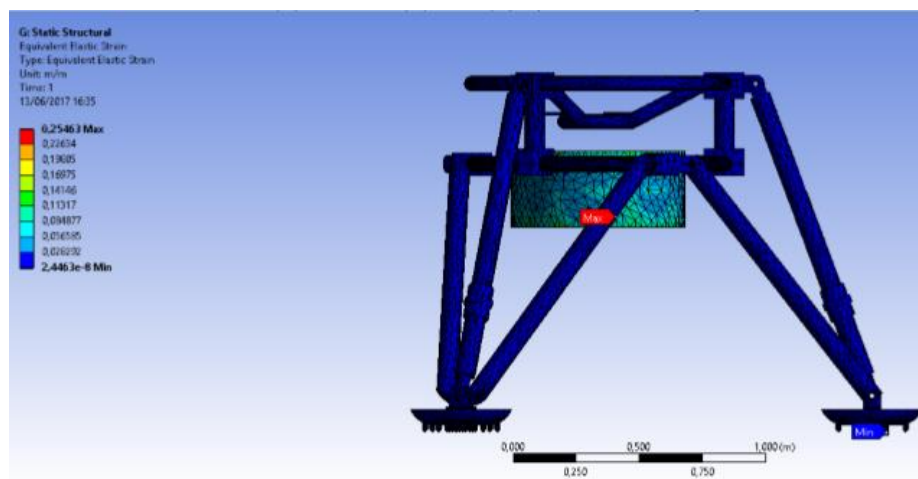


Fig. 71 Equivalent Elastic Strain with Drag force - POM Structure.

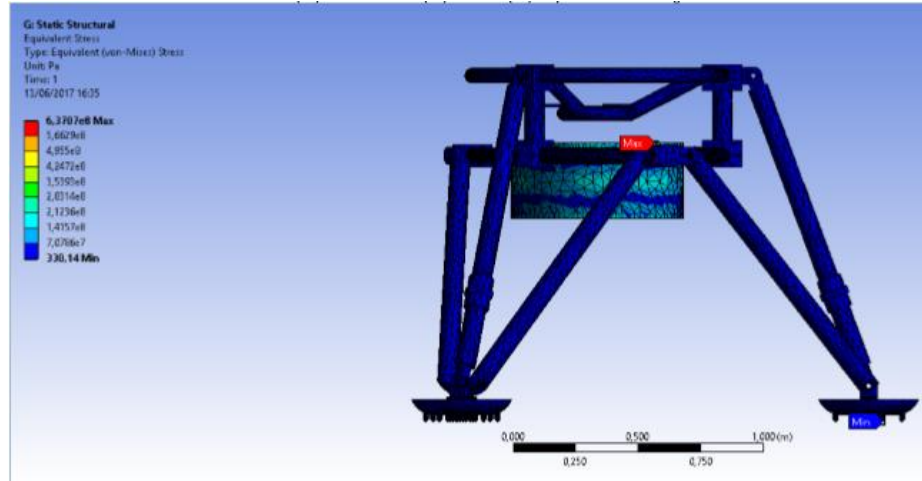


Fig. 72 Equivalent von-Mises Stress (pa) with Drag force - POM Structure.

6.6 Modal Analysis

Using ANSYS Modal Analysis feature, it is possible to study and determine the vibratory response of a structure under a dynamic load. Considering first of all the structure operating out of the water, where it is weakly damped. A simplification for the dynamic response calculations can be done, where the damping component of the equilibrium equations is neglected. Under these circumstances, to perform a modal analysis the program must first solve the following eigenvalue problem:

$$[K - \omega^2 M] \cdot \{u\} = 0 \quad (80)$$

Where K and M are the stiffness and mass matrices respectively and are calculated using FEM, and u corresponds to the displacements of the structure. Solving this equation, the obtained eigenvalues correspond to the natural frequencies of the structure, and the eigenvectors to the modal amplitudes for each frequency.

The natural frequencies of a structure correspond to the frequency that the structure vibrates after it is disturbed and the modal amplitudes correspond to the specific pattern that the structure vibrates under those frequencies. Thus, if a harmonic solicitation with a frequency that is equal to one of the natural frequencies of a structure is applied, the structure will enter in resonance mode, increasing the amplitude of vibration which can lead to serious damage on the structure.

For that reason, a modal analysis must be done to assure that the frequency range domain at which the frame will operate does not include any natural frequencies of the same. It should be stated that when the structure was immersed the surrounding water will induce a high damping effect which attenuated the effect of vibrations induced by currents.

Steel

Table 37 and Fig. 73 present the correspondent frequencies, shown in Hertz, for the first six vibration modes for the steel structure.

Table 37 Steel Structure Vibration modes' frequencies

<i>Mode</i>	<i>Frequency [Hz]</i>
1	48.283
2	48.397
3	79.789
4	81.643
5	108.87
6	108.96

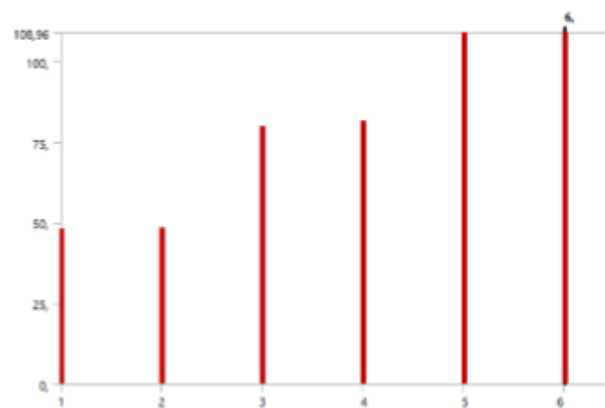


Fig. 73 The first 6 eigen-frequencies for the proposed Steel Structure.

Next, on Fig. 74 Mode 1 Shape – 48.283 Hz Fig. 74, Fig. 75, Fig. 76, Fig. 76, Fig. 77, Fig. 78, and Fig. 79, will be shown the diferent vibration patterns for each mode.

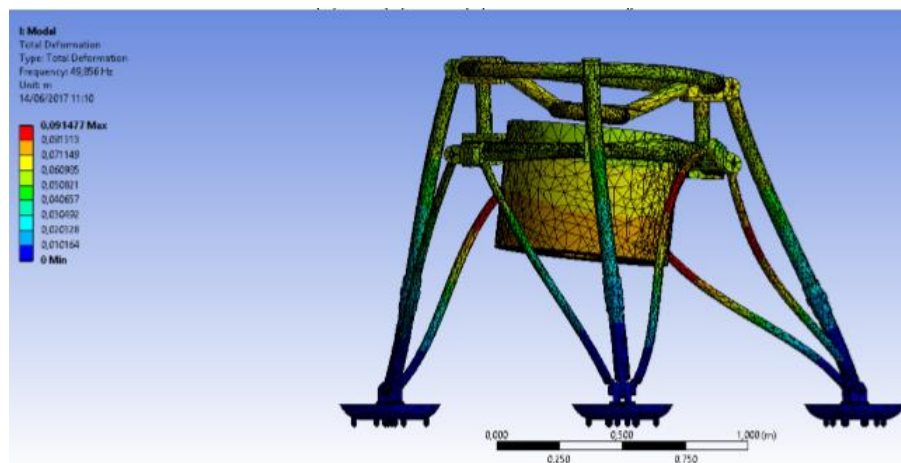


Fig. 74 Mode 1 Shape – 48.283 Hz

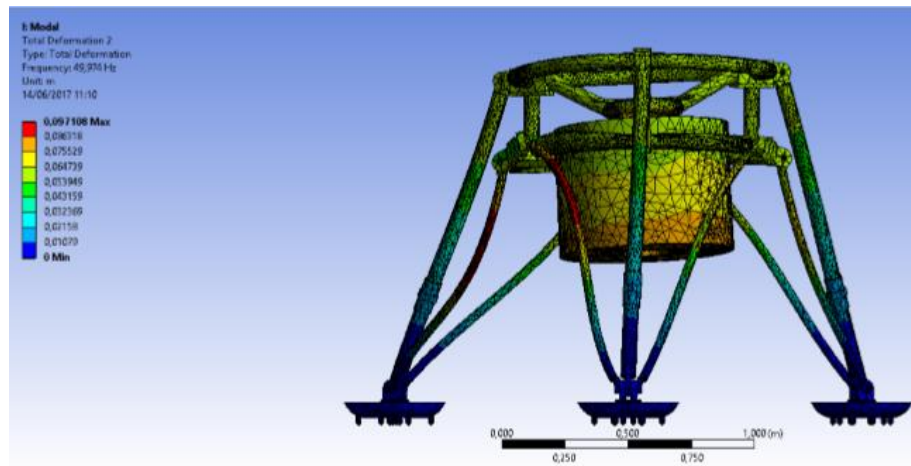


Fig. 75 Mode 2 Shape - 48.397 Hz

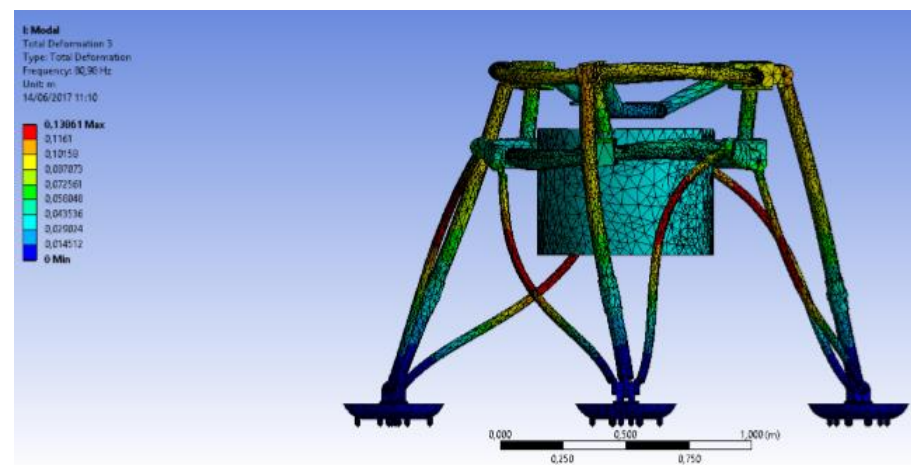


Fig. 76 Mode 3 Shape – 79.789 Hz

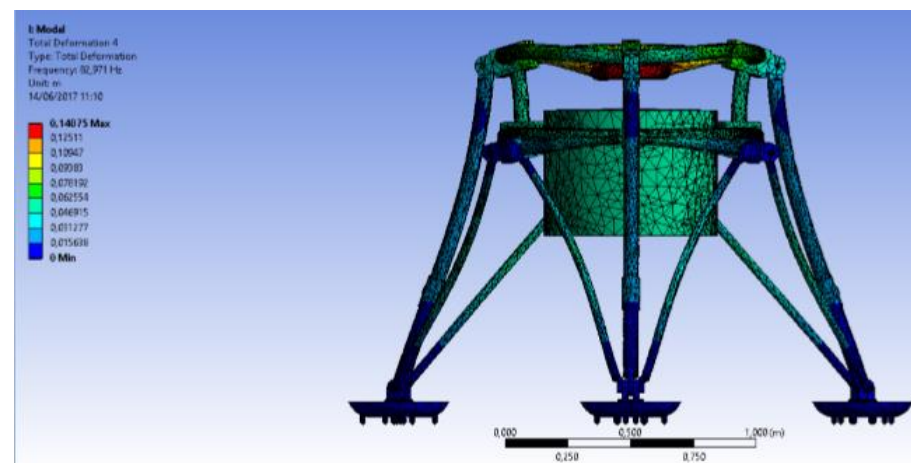


Fig. 77 Mode 4 Shape – 79.789 Hz

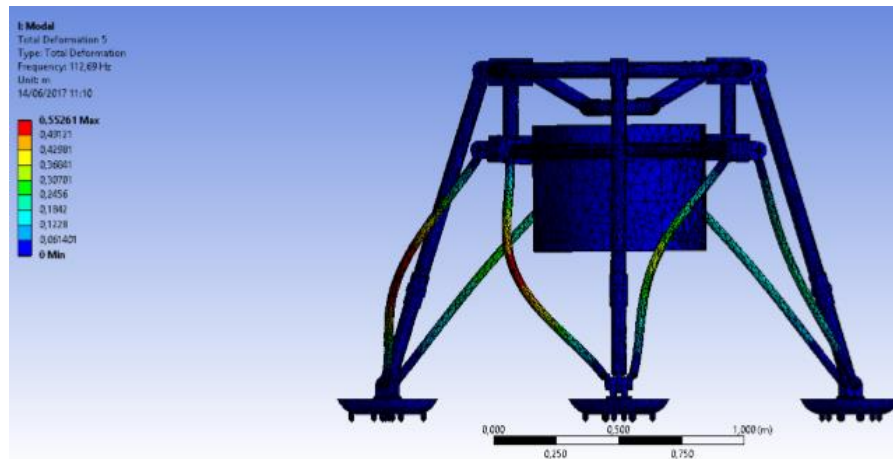


Fig. 78 Mode 5 Shape – 108.87 Hz

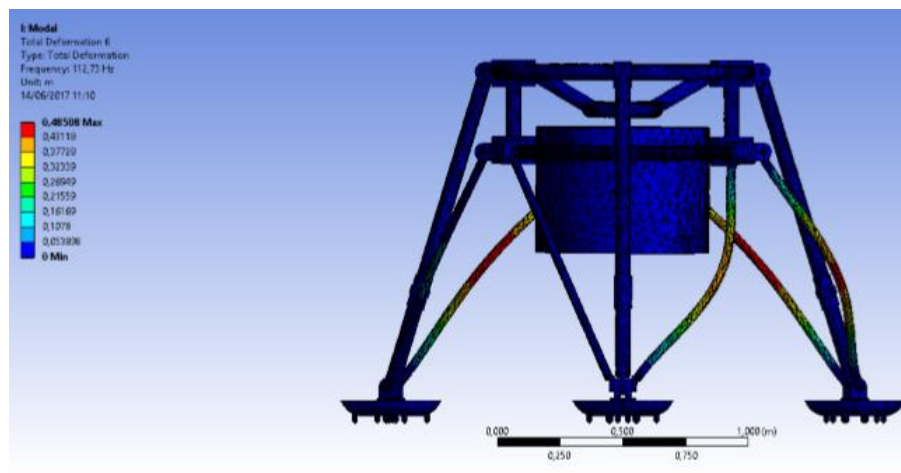


Fig. 79 Mode 6 Shape – 108.96 Hz

POM

Table 38 and Fig. 80 present the correspondent frequencies, shown in Hertz, for the first six vibration modes for the POM structure.

Table 38 POM Structure Vibration modes' frequencies

Modes	Frequencies (Hz)
1	23.301
2	23.4
3	32.279
4	32.327
5	42.927
6	43.037

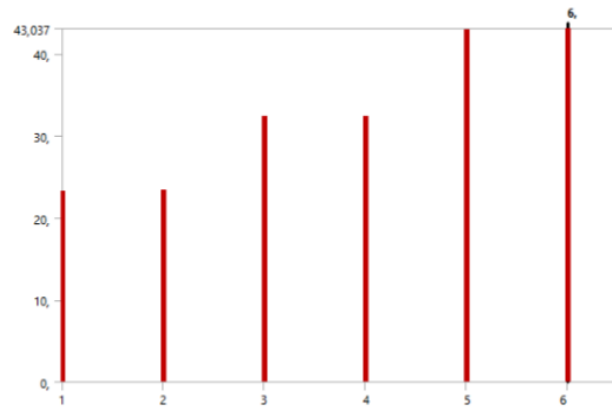


Fig. 80 The first 6 eigen-frequencies for the proposed POM Structure.

Next will be shown the different vibration patterns for each mode.

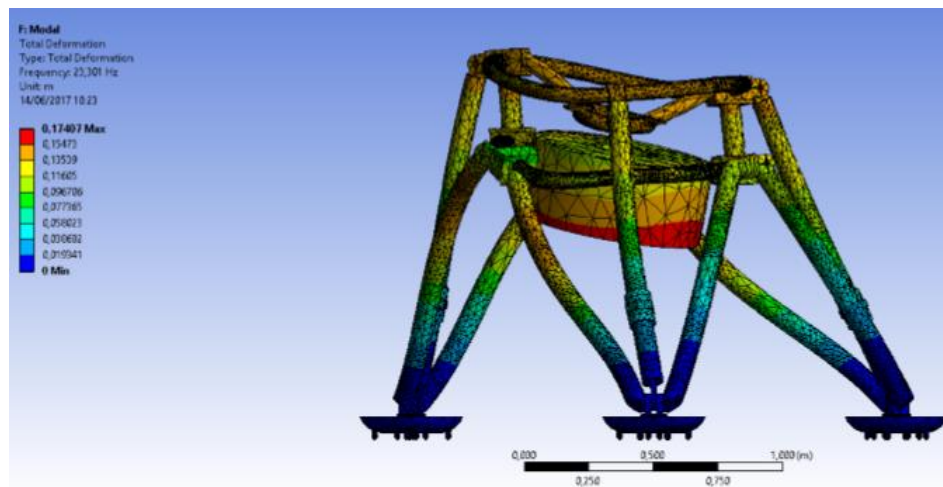


Fig. 81 Mode 1 Shape – 23.301 Hz

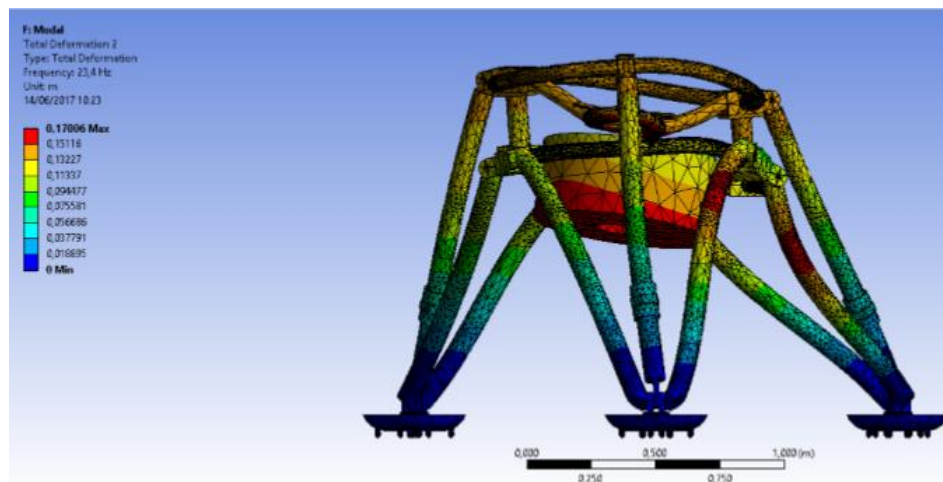


Fig. 82 Mode 2 Shape – 23.4 Hz

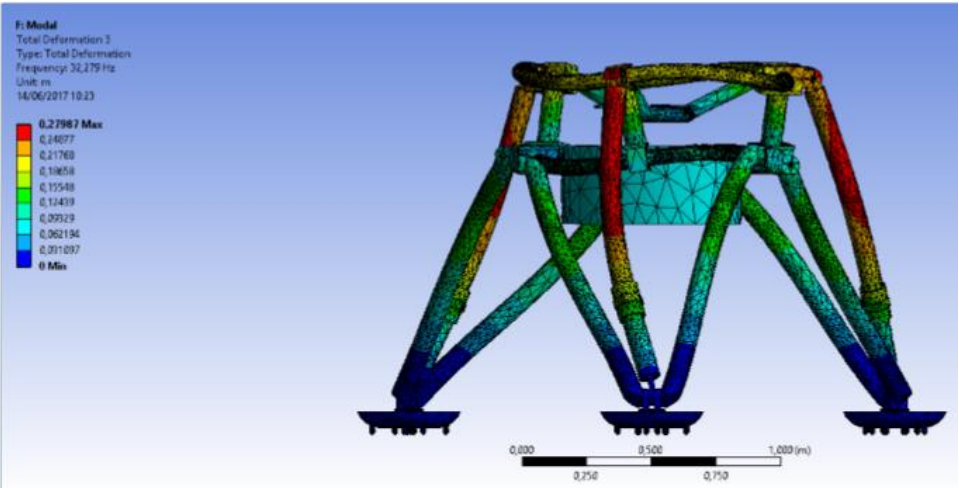


Fig. 83 Mode 3 Shape – 32.279 Hz

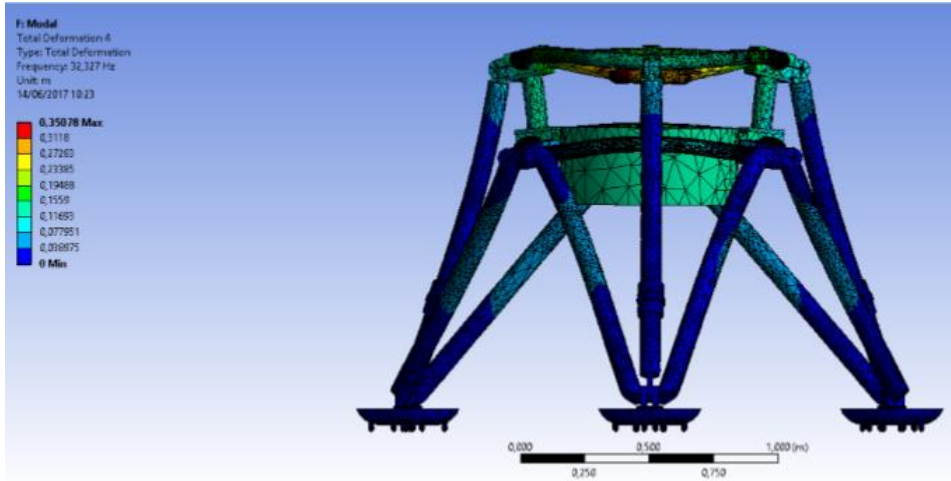


Fig. 84 Mode 4 Shape – 32.327 Hz

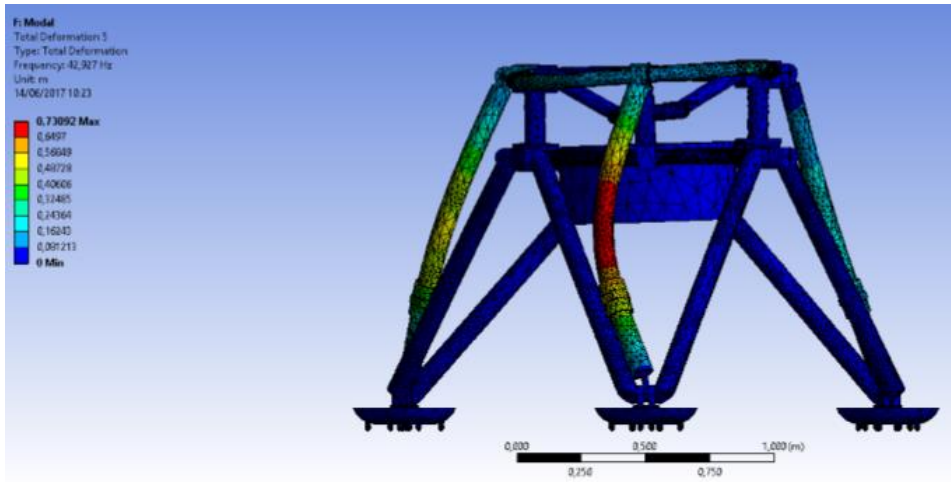


Fig. 85 Mode 5 Shape - 42.927 Hz

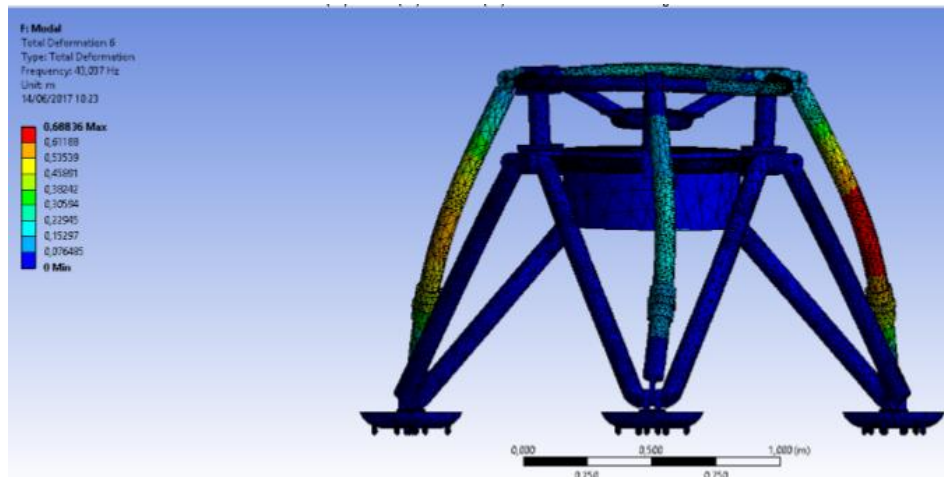


Fig. 86 Mode 6 Shape - 43.037 Hz

According to professor António Falcão, in 'Modelling of Wave Energy Conversion' [74], the typical values for wave period values are comprised between 4.4 s to 13.9 s, i.e., between a frequency range of 0.0719 Hz to 0.2273 Hz, which are considerably inferior to vibration modes for both structures, therefore, the risk of resonance is low.

7 Analysis Discussion

The design of the different components comprised some assumptions that, after the numeric analysis, using a proper software, may require some refinements.

Comparing the maximum stress resistance values given by the structure dimension with the maximum stress values for different load cases, it is possible to understand the structural behaviour and if the dimensioning is enough accurate and what is the safety factor in each case.

Therefore, for the steel structure, we have

Table 39 Maximum Stress Values - Steel Structure

<i>Component</i>	$\sigma_{max} \text{ (MPa)}$ <i>Load Case 1</i>	$\sigma_{max} \text{ (MPa)}$ <i>Load Case 2</i>	$\sigma_{max} \text{ (MPa)}$ <i>Load Case 3</i>	$\sigma_{max} \text{ (MPa)}$ <i>Load Case 4</i>
<i>Main support – upper part</i>	3.9346	2.2925	2.448	3.5378
<i>Main support –lower part</i>	4.6162	1.657	1.3874	1.4266
<i>Secondary</i>	16.2380	5.7667	4.215	5.2419
<i>Superior Hoop</i>	18.4360	3.9452	76.898	297.0
<i>Column</i>	1.38740	2.1112	12.462	17.845

If we compare this values with the obtained after the buckling analysis, we have for main supports, considering the maximum value for all the load cases, 3.936 MPa,

$$\sigma = \frac{N}{A} \quad (81)$$

We have, then, a resistance stress resultant of 2782.752 Pa, which is considerably lower than the obtained value from the software, safety factor of 1414. For the secondary supports we obtained 1672.343 Pa, which, once again is considerably lower than the results given by ANSYS.

For the hoop, the value 297 MPa, although it is the critical value, it is lower than the Stainless steel Yield Stress, hence, the Steel structure is suitable for all load cases that the lander may be subjected.

The same comparison can be made for the POM structure, as it is shown on Table 40.

Table 40 Maximum Stress Values - POM Structure

<i>Component</i>	σ_{max} (MPa) <i>Load Case 1</i>	σ_{max} (MPa) <i>Load Case 2</i>	σ_{max} (MPa) <i>Load Case 3</i>	σ_{max} (MPa) <i>Load Case 4</i>
<i>Main support – upper part</i>	0.8825	0.67092	0.06733	1.0503
<i>Main support –lower part</i>	0.8660	0.41762	0.56159	0.8949
<i>Secondary</i>	5.3938	0.1246	1.8914	1.198
<i>Superior Hoop</i>	14.0730	1.3478	27.021	109.118
<i>Column</i>	16.4170	1.1231	3.4923	10.783

For the supports, either main supports or secondary ones, the structure is oversized, as well. However, when considering the superior hoop, the stress value at which this component is subjected is considerably higher than its Yield Stress. However, this value appears not on the tube itself, but on the place that the structure is fixed, as so, it is not a dimensioning consequence but a resultant from the mechanical properties of the polymer.

The whole structure is oversized, yet, it is highly desirable for future applications where, due to the modularity feature of the system, it may be overloaded.

8 Conclusions and Future works

With this work, it was possible to conclude that the insertion of *high-performance* polymeric materials for underwater operations is a benefit when it is intended to obtain resistant and anti-corrosive structures. However, the process to obtain complex geometries with this type of materials is more expensive when compared to steel. Said so, the suggested geometry, mainly the hoops, is only enforceable if it is meant to do more than 1000 units, otherwise, the process of only one piece would turn out in a major expensive. To minimize this expense, I suggest that, as a future work, to rethink of the hoops geometry, comprising inexpensive profiles linked through standard pieces, easily found on market.

Although it has only been considered one-material only structure, a hybrid one, which could comprise POM and steel components, could bring benefits to the structure response to extreme load cases, as it happened on Load Case no 4, where the POM hoop stress value was higher than the yield stress, while the steel component did not. That said, a study of hybrid structures considering different configurations, could optimize the structural performance.

In the future, is also intended to study the different structural links, which, given the deadline, was not possible to study yet, however, some of this links will be subjected to high load values and, as such, it is relevant to analyse each one of them.

Regarding the numeric simulation done on the structure, ANSYS has shown as a useful and user-friendly software. As a future analysis, it is intended to use the feature AQUA to study the structural behaviour of the lander when it bounces on the surface. It has been tried under this thesis; however, many errors showed up regarding the complexity of the geometry, hence, it was not possible for a deeper study of this software to found ways to keep the study. With this analysis, it would possible to include the water damping effect on the structure, which has been neglected during the presented modal analysis.

Initially it was intended to make an economic analysis of the materials and processes for the lander conception, however, it has become a hard task to complete within the deadlines. Although several emails have been sent to specific companies that could help during this analysis, there were few answers and, as so, there were not enough data to make a solid comparison between different materials and processes. Yet, based on estimated prices given by different software's, such as CES Edupack, and websites, a rough approximation has been made. Said so, the final cost of the structure may not correspond to presented values.

It is also desired to create, as concluded the structural design, a prototype and test it in a wave controlled tank to predict the structural behaviour under marine conditions.

Bibliography

- [1] N. O. and A. A. US Department of Commerce, "What is ocean exploration and why is it important?"
- [2] The Ocean Portal Team, "The Deep Sea | Smithsonian Ocean Portal," 2016. [Online]. Available: <http://ocean.si.edu/deep-sea>. [Accessed: 06-Apr-2017].
- [3] "Macroalgae as a new source of bio-based products | Solution | Aarhus University | State of Green." [Online]. Available: <https://stateofgreen.com/en/profiles/aarhus-university/solutions/macroalgae-as-a-new-source-of-bio-based-products>. [Accessed: 06-Apr-2017].
- [4] D. van der Wal, J. van Dalen, A. Wielemaker-van den Dool, J. T. Dijkstra, and T. Ysebaert, "Biophysical control of intertidal benthic macroalgae revealed by high-frequency multispectral camera images," *J. Sea Res.*, vol. 90, pp. 111–120, 2014.
- [5] V. P. (Vikrant P. Shah, "Design considerations for engineering Autonomous Underwater Vehicles," 2007.
- [6] "A brief history of deep sea exploration." [Online]. Available: <http://dujs.dartmouth.edu/2016/02/8271/#.WOZU7IgrKUK>. [Accessed: 06-Apr-2017].
- [7] "INEGI - Instituto de Ciência e Inovação em Engenharia Mecânica e Engenharia Industrial." [Online]. Available: <http://www.inegi.pt/>. [Accessed: 30-May-2017].
- [8] M. E. Ç. Stelios Katsanevakis, Inger Wallentinus, Argyro Zenetos, Erkki Leppäkoski and A. C. C. Bayram Oztürk, Michal Grabowski, Daniel Golani, "Impacts of Invasive alien marine species on ecosystem services and biodiversity: a pan-European review."
- [9] A. Tengberg *et al.*, "Benthic chamber and profiling landers in oceanography — A review of design, technical solutions and functioning," *Prog. Oceanogr.*, vol. 35, no. 3, pp. 253–294, Jan. 1995.
- [10] E. Silva *et al.*, "TURTLE - a robotic autonomous deep sea lander," in *OCEANS 2016 MTS/IEEE Monterey*, 2016, pp. 1–5.
- [11] "MTS Manned Underwater Vehicles Committee." [Online]. Available: <http://www.mtsmuv.org/>. [Accessed: 20-Feb-2017].
- [12] R. A. Geyer, *Submersibles and their use in oceanography and ocean engineering*. Elsevier Scientific Pub. Co., 1977.
- [13] "DEEPSEA CHALLENGE – National Geographic Explorer James Cameron's Expedition." [Online]. Available: <http://www.deepseachallenge.com/>. [Accessed: 20-Feb-2017].
- [14] "Remotely Operated Vehicle Committee of the Marine Technology Society." [Online]. Available: <http://www.rov.org/>. [Accessed: 20-Feb-2017].
- [15] "Introduction to Remote Sensing." [Online]. Available: <http://www.seos-project.eu/modules/remotesensing/remotesensing-c01-p05.html>. [Accessed: 20-Feb-2017].
- [16] "Overview « ROV KIEL 6000 « Technology & Logistics Centre « Central Facilities « Centre « GEOMAR - Helmholtz Centre for Ocean Research Kiel." [Online]. Available: <http://www.geomar.de/en/centre/central-facilities/tlz/rovkiel6000/overview/>. [Accessed: 20-Feb-2017].
- [17] "VICTOR 6000 - Ifremer Fleet." [Online]. Available:

- <http://flotte.ifremer.fr/fleet/Presentation-of-the-fleet/Underwater-systems/VICTOR-6000>. [Accessed: 20-Feb-2017].
- [18] R. B. Wynn *et al.*, “Autonomous Underwater Vehicles (AUVs): Their past, present and future contributions to the advancement of marine geoscience,” *Mar. Geol.*, vol. 352, pp. 451–468, 2014.
 - [19] “Woods Hole Oceanographic Institution.” [Online]. Available: <https://www.whoi.edu/>. [Accessed: 20-Feb-2017].
 - [20] “Bluefin-21 » Bluefin Robotics.” [Online]. Available: <http://www.bluefinrobotics.com/vehicles-batteries-and-services/bluefin-21>. [Accessed: 20-Feb-2017].
 - [21] “DEPTHX.” [Online]. Available: <http://stoneaerospace.com/depthx/>. [Accessed: 20-Feb-2017].
 - [22] A. D. A. D. Bowen *et al.*, “Navigation and control of the Nereus hybrid underwater vehicle for global ocean science to 10,903 m depth: Preliminary results,” *Sea Technol.*, vol. 285–286, no. 3, pp. 1–10, 2007.
 - [23] “Robotic Deep-sea Vehicle Lost on Dive to 6-Mile Depth : Woods Hole Oceanographic Institution.” [Online]. Available: <https://www.whoi.edu/news-release/Nereus-Lost>. [Accessed: 20-Feb-2017].
 - [24] “11K Deepwater Submersible | Promare.” [Online]. Available: <http://www.promare.org/promares-deepwater-submersible/>. [Accessed: 03-Apr-2017].
 - [25] “HADAL-Lander A :: HADES.” [Online]. Available: <http://web.whoi.edu/hades/hadal-lander-a/>. [Accessed: 20-Feb-2017].
 - [26] “HADAL -Lander B :: Hades.” [Online]. Available: <http://web.whoi.edu/hades/hadal-lander-b/>. [Accessed: 20-Feb-2017].
 - [27] “Case Study: The ROBIO Lander | Oceanlab | The University of Aberdeen.” [Online]. Available: <http://www.abdn.ac.uk/oceanlab/business/case-study-robio.php>. [Accessed: 20-Feb-2017].
 - [28] N. O. and A. A. US Department of Commerce, “NOAA Ocean Explorer: Bioluminescence and Vision on the Deep Seafloor 2015: Background Resources: Medusa Lander.”
 - [29] “Meet The Medusa, The Latest In Deep-Sea Tech | Oceanscape Network.” [Online]. Available: http://oceanscape.aquarium.org/explore/general_articles/meet-the-medusa-the-latest-in-deep-sea-tech. [Accessed: 20-Feb-2017].
 - [30] “Lander K/MT 100.”
 - [31] “DELOS - Deep-ocean Environmental Long-term Observatory System | The University of Aberdeen.” [Online]. Available: <http://www.delos-project.org/>. [Accessed: 20-Feb-2017].
 - [32] “OBSEA.” [Online]. Available: <http://www.upc.edu/cdsarti/OBSEA/>. [Accessed: 20-Feb-2017].
 - [33] J. Aguzzi *et al.*, “The New Seafloor Observatory (OBSEA) for Remote and Long-Term Coastal Ecosystem Monitoring,” *Sensors*, vol. 11, no. 12, pp. 5850–5872, May 2011.
 - [34] K-lander, “K-LANDER AUTONOMOUS SEABED SENSOR CARRIER.”

- [35] "Online Materials Information Resource - MatWeb." [Online]. Available: <http://www.matweb.com/index.aspx>. [Accessed: 20-Feb-2017].
- [36] "Daily Exchange Data." [Online]. Available: <https://www.metalprices.com/dailyexchangedata/Index>. [Accessed: 20-Feb-2017].
- [37] "CES EduPack." 2016.
- [38] S. C. Dexter, *Handbook of oceanographic engineering materials : volume I - metals and alloys*. Woods Hole, MA: Woods Hole Oceanographic Institution, 1972.
- [39] H. F. Jensen Iii, "MASTER OF SCIENCE THESIS," 2009.
- [40] "ISO shipping container size and dimension specifications | Container Solutions." [Online]. Available: <http://containersolutions.net/specifications/>. [Accessed: 03-Apr-2017].
- [41] "TRAINING | How do Mooring Systems Work? | Rigzone." [Online]. Available: http://www.rigzone.com/training/insight.asp?insight_id=358. [Accessed: 13-Mar-2017].
- [42] J. Babicz, *Encyclopedia of Ship Technology*. 2013.
- [43] "SP Systems Guide to Composites."
- [44] N. K. Naik and R. S. Kumar, "Compressive strength of unidirectional composites: evaluation and comparison of prediction models," *Compos. Struct.*, vol. 46, no. 3, pp. 299–308, 1999.
- [45] C. Yang, S.-S. Pang, and Y. Zhao, "Buckling Analysis of Thick-Walled Composite Pipe under External Pressure," *J. Compos. Mater.*, vol. 31, no. 4, pp. 409–426, Feb. 1997.
- [46] J. D. Stachiw, S. Associates, D. Peters, and G. McDonald, "CERAMIC EXTERNAL PRESSURE HOUSINGS FOR DEEP SEA VEHICLES Woods Hole Oceanographic Institution Woods Hole, MA."
- [47] G. McDonald, "Operations to 11,000m: Nereus Ceramic Housing Design and Analysis."
- [48] "Fluoropolymer Plástico | UL Prospector." [Online]. Available: <https://plastics.ulprospector.com/pt/generics/14/fluoropolymer>. [Accessed: 10-Apr-2017].
- [49] "Aetna Plastics : Industrial Plastics : Plastic Fabrication : Plastic Piping Systems." [Online]. Available: <http://www.aetnaplastics.com/>. [Accessed: 10-Apr-2017].
- [50] "TECAFORM™ Acetal Copolymer - Modern Plastics." [Online]. Available: <http://www.modernplastics.com/products/tecaform-acetal-copolymer/>. [Accessed: 10-Apr-2017].
- [51] "POM | Tipos de polímeros - Resinex." [Online]. Available: <http://www.resinex.pt/tipos-de-polimeros/pom.html>. [Accessed: 10-Apr-2017].
- [52] "What is Syntactic Foam? - Engineered Syntactic Systems," 2016. [Online]. Available: <http://esyntactic.com/products-solutions/what-is-syntactic-foam/>. [Accessed: 22-May-2017].
- [53] "MZ Grade," vol. 145, p. 3907, 2014.
- [54] "Chromatic Aberration | Understanding Chromatic Aberration from Nikon." [Online]. Available: <http://www.nikonusa.com/en/learn-and-explore/a/products-and>

innovation/chromatic-aberration.html. [Accessed: 26-May-2017].

- [55] G. Paredes, "E Studo Experimental De Um," 2008.
- [56] J. Heller, D. Henrion, and T. Pajdla, "Stable radial distortion calibration by polynomial matrix inequalities programming," *Lect. Notes Comput. Sci. (including Subser. Lect. Notes Artif. Intell. Lect. Notes Bioinformatics)*, vol. 9003, pp. 307–321, 2015.
- [57] "Angle of coverage - Cameras Underwater Ltd." [Online]. Available: <http://www.camerasunderwater.info/optics/lens-fov>. [Accessed: 29-May-2017].
- [58] "Optics of Dome Ports." [Online]. Available: <http://scubageek.com/articles/wwwdome.html>. [Accessed: 29-May-2017].
- [59] "Glass vs. Acrylic - The First Tank Guide - The Differences Between and the Debate About Glass Aquariums and Acrylic Aquariums." [Online]. Available: <http://www.firsttankguide.net/tanktype.php>. [Accessed: 30-May-2017].
- [60] "Aerospace Projects Review Blog – Page 49 – Unbuilt aircraft and Spacecraft Projects." [Online]. Available: <http://www.aerospaceprojectsreview.com/blog/?paged=49>. [Accessed: 29-May-2017].
- [61] "A Teledyne Marine Systems Company."
- [62] E. W. Weisstein, "Cylindrical Coordinates."
- [63] P. Tavares de Castro, "Tensões em Peças Cilíndricas," vol. 1, p. 62, 2016.
- [64] "Thick Walled Cylinders."
- [65] G. Erhard, "Designing with Plastics 5C alculations for Structures under Mechanical Load – Examples of Geometrically Simple Structural Parts under Static Loads 5.1 Specific Materials and Processing Problems Deformation Behavior under Uniaxial Dynamic Tensile Stress."
- [66] B. R. Munson, T. H. Okiishi, W. W. Huebsch, and A. P. Rothmayer, "Fundamentals of fluid mechanics 7th Edition," *J. Mech. Work. Technol.*, pp. 1–196, 2013.
- [67] "Density of Ocean Water - Windows to the Universe." [Online]. Available: <http://www.windows2universe.org/earth/Water/density.html>. [Accessed: 19-May-2017].
- [68] P. Tavares de Castro, *Reservatórios metálicos*. 2015.
- [69] D. W. Igh, T. F. W. In, B. U. Rg, C. T. R. Il, L. In, and D. C. Washington, "Collapse b y In stab ility o f T h in C y lin d rica l Shells U n d er E x te r n a l P re ssu re," *N. Y*, vol. 53, 1931.
- [70] S. P. Timoshenko and J. M. Gere, *Theory of Elastic Stability*. 1963.
- [71] B. R. Munson, T. H. (Theodore H. Okiishi, W. W. Huebsch, and A. P. Rothmayer, *Fundamentals of fluid mechanics*. John Wiley & Sons, Inc, 2013.
- [72] "Sea Water Properties." [Online]. Available: http://www.engineeringtoolbox.com/sea-water-properties-d_840.html. [Accessed: 13-Jun-2017].
- [73] "Simulation Driven Product Development | ANSYS." [Online]. Available: <http://www.ansys.com/>. [Accessed: 26-Jun-2017].
- [74] A. F. O. Falcão, "Modelling of Wave Energy Conversion," pp. 1–38, 2014.

Appendix A - Different Materials Behaviour under External Pressure

Table 41 Structural Steel Analysis Values

t	Re	Ri	V	P	m	Pe	σ
(m)	(m)	(m)	(m ³)	(kg/ m ³)	(kg)	(MPa)	(MPa)
0,002	0,1	0,098	0,000746	7850	5,859573	1	59,041
0,002	0,1	0,098	0,000396	7850	5,859573	1,5	88,562
0,002	0,1	0,098	0,000396	7850	5,859573	2	118,08
0,002	0,1	0,098	0,000396	7850	5,859573	2,5	147,6
0,002	0,1	0,098	0,000396	7850	5,859573	3	177,12

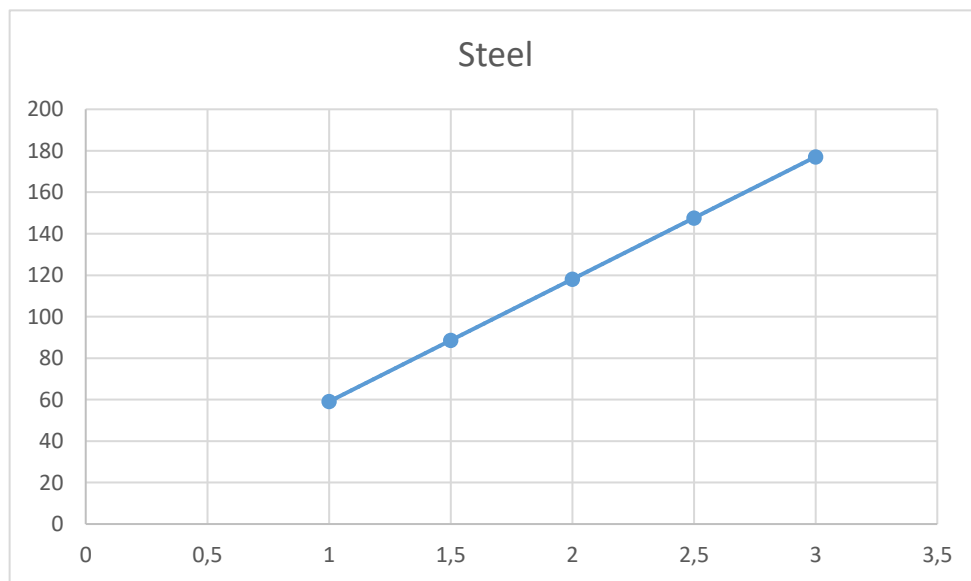


Fig. 87 Stress in function of External Pressure - Structural Steel

Table 42 Stainless Steel Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,002	0,1	0,098	0,000746	7750	5,7849287	1	58,947
0,002	0,1	0,098	0,000396	7750	5,7849287	1,5	88,421
0,002	0,1	0,098	0,000396	7750	5,7849287	2	117,89
0,002	0,1	0,098	0,000396	7750	5,7849287	2,5	147,37
0,002	0,1	0,098	0,000396	7750	5,7849287	3	176,84

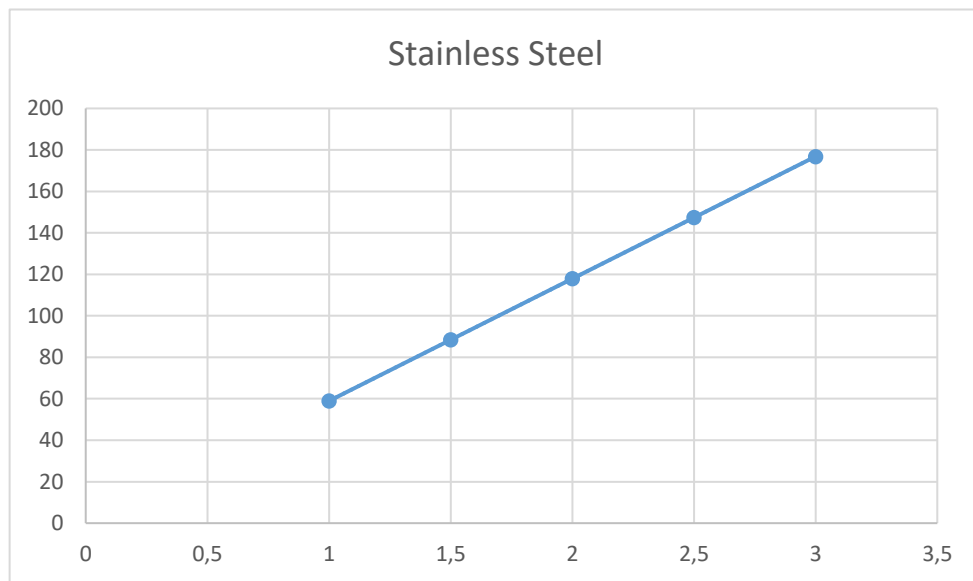


Fig. 88 Stress in function of External Pressure - Stainless Steel

Table 43 Titanium Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,0035	0,1	0,0965	0,001296	4620	5,989267	1	0,3492
0,0035	0,1	0,0965	0,000688	4620	5,989267	5	174,71
0,0035	0,1	0,0965	0,000688	4620	5,989267	10	349,42
0,0035	0,1	0,0965	0,000688	4620	5,989267	15	524,13
0,0035	0,1	0,0965	0,000688	4620	5,989267	20	698,84

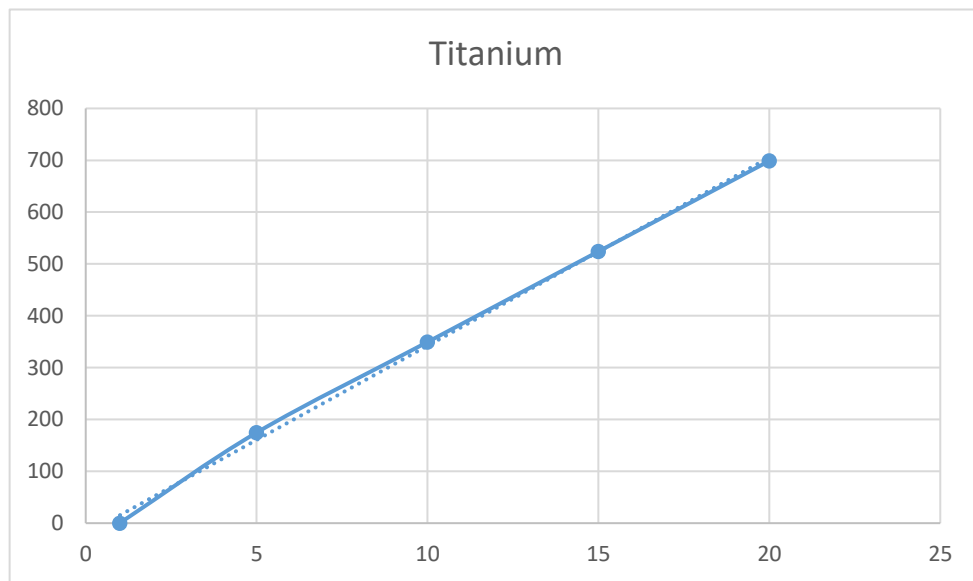


Fig. 89 Stress in function of External Pressure - Titanium

Table 44 Aluminium Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,0055	0,1	0,0945	0,002016	2770	5,5855145	1	23,455
0,0055	0,1	0,0945	0,00107	2770	5,5855145	5	117,27
0,0055	0,1	0,0945	0,00107	2770	5,5855145	10	234,55
0,0055	0,1	0,0945	0,00107	2770	5,5855145	15	351,82
0,0055	0,1	0,0945	0,00107	2770	5,5855145	20	469,09

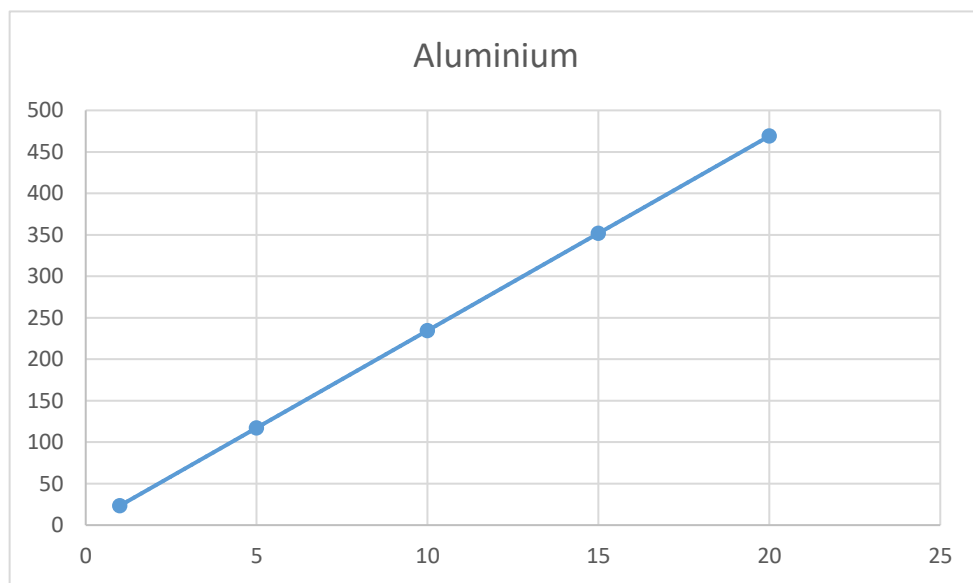


Fig. 90 Stress in function of External Pressure - Aluminium

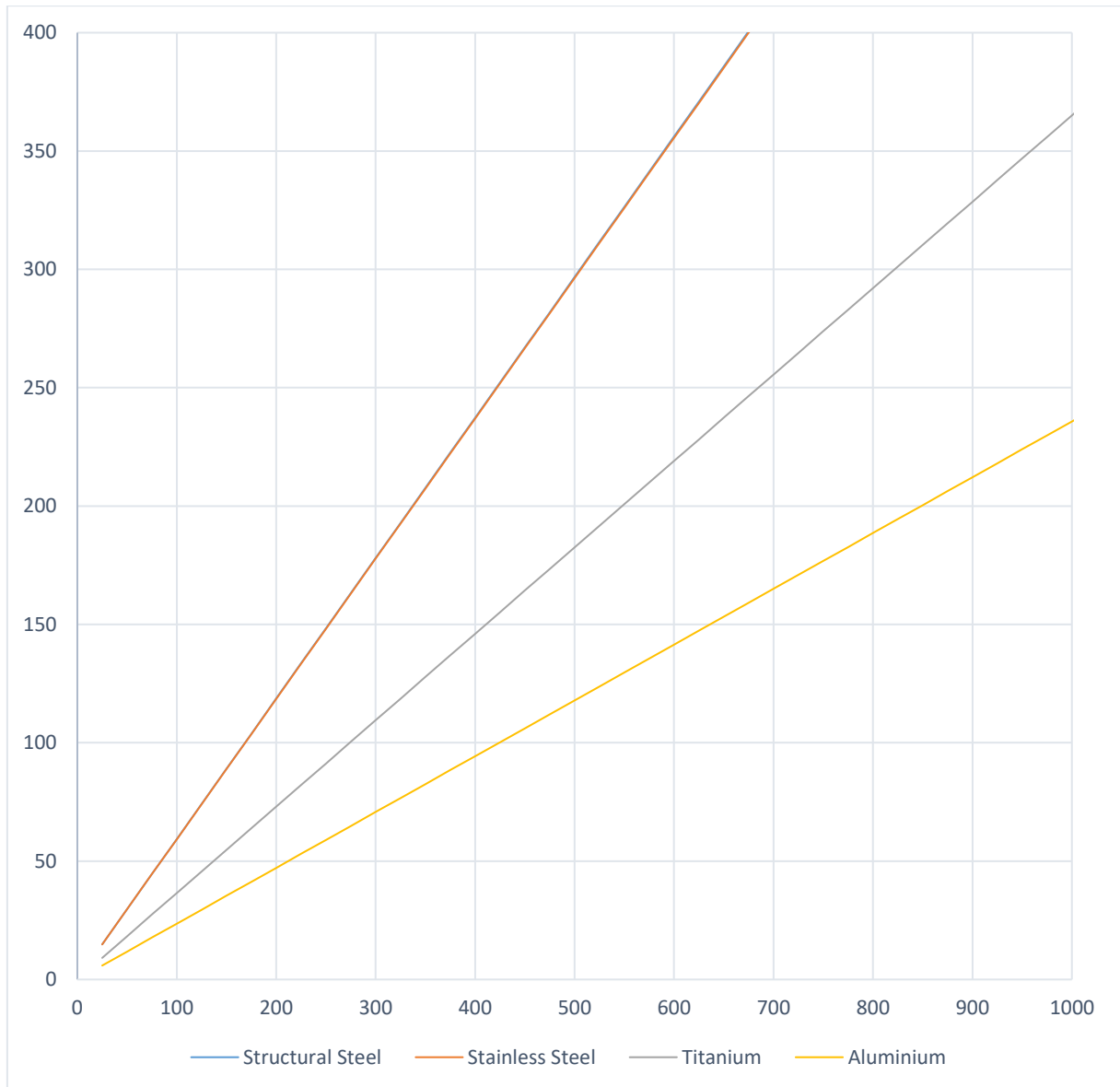


Fig. 91 Comparison of metal material stress

Table 45 PTFE Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,0075	0,1	0,0925	0,002721	2165	5,891841	0,001	0,016763
0,0075	0,1	0,0925	0,001444	2165	5,891841	0,005	0,083814
0,0075	0,1	0,0925	0,001444	2165	5,891841	0,01	0,16763
0,0075	0,1	0,0925	0,001444	2165	5,891841	0,1	1,6763
0,0075	0,1	0,0925	0,001444	2165	5,891841	0,2	3,3526

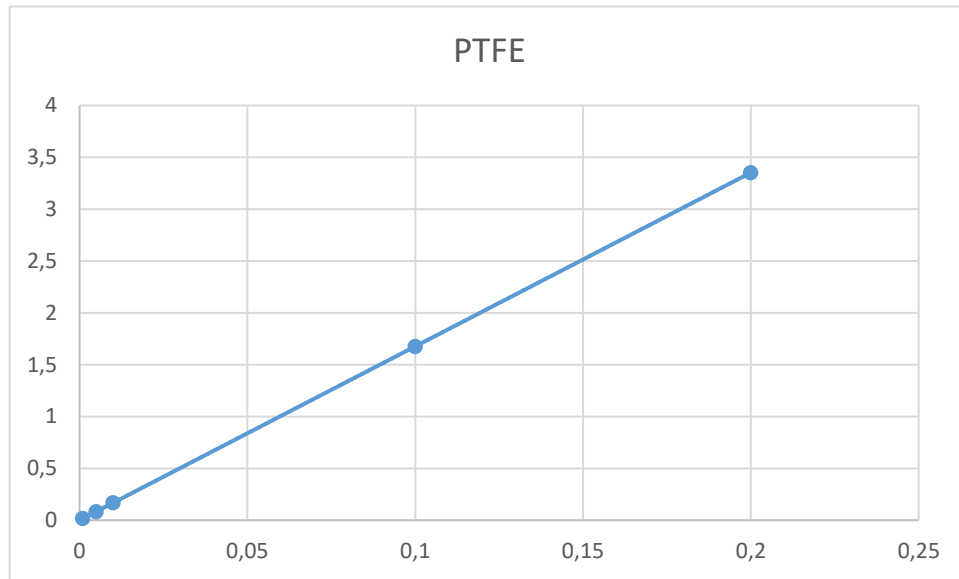


Fig. 92 Stress in function of External Pressure – PTFE

Table 46 FEP Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,0075	0,1	0,0925	0,002721	2145	5,837413	0,02	0,33556
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,04	0,67112
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,06	1,0067
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,08	1,3422
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,1	1,6778

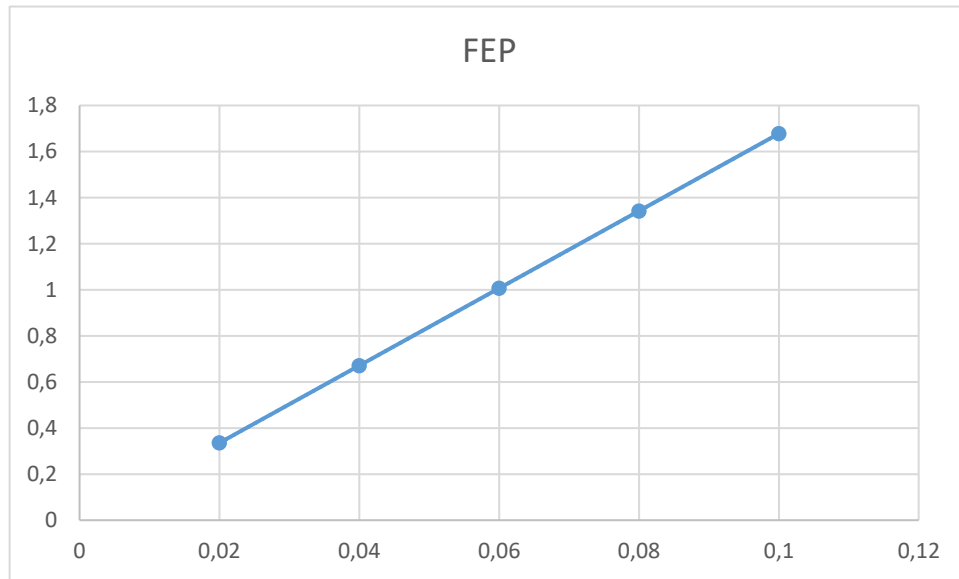


Fig. 93 Stress in function of External Pressure – FEP

Table 47 FPA Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,0075	0,1	0,0925	0,002721	2145	5,837413	0,02	0,33579
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,04	0,67158
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,06	1,0074
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,08	1,3432
0,0075	0,1	0,0925	0,001444	2145	5,837413	0,1	1,6789

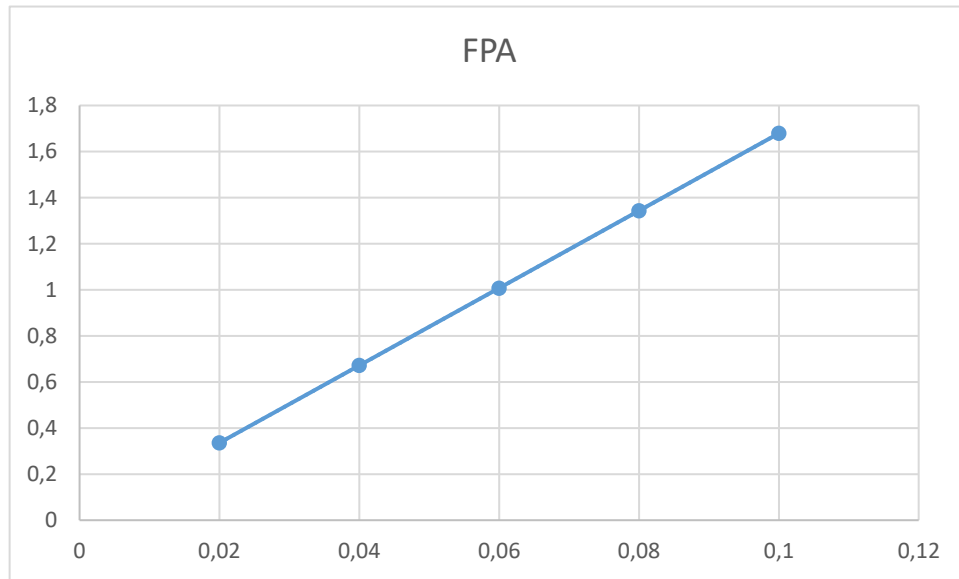


Fig. 94 Stress in function of External Pressure - FPA

Table 48 LDPE Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,019	0,1	0,081	0,006482	917,5	5,947567	0,1	0,6743
0,019	0,1	0,081	0,003439	917,5	5,947567	0,15	1,0115
0,019	0,1	0,081	0,003439	917,5	5,947567	0,2	1,3486
0,019	0,1	0,081	0,003439	917,5	5,947567	0,25	1,6858

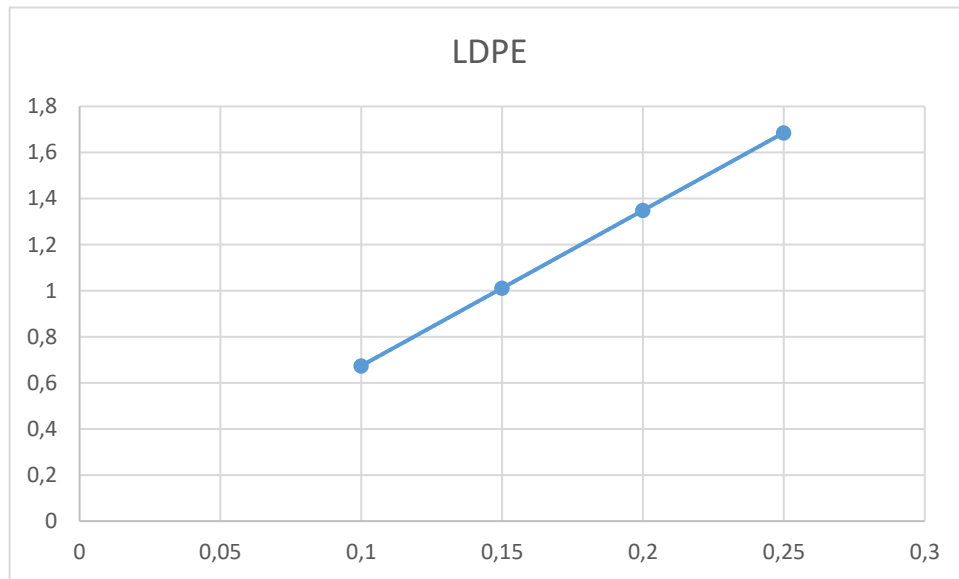


Fig. 95 Stress in function of External Pressure – LDPE

Table 49 HDPE Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,018	0,1	0,082	0,006175	953	5,884884	0,1	0,71552
0,018	0,1	0,082	0,003276	953	5,884884	0,2	1,431
0,018	0,1	0,082	0,003276	953	5,884884	0,3	2,1466
0,018	0,1	0,082	0,003276	953	5,884884	0,4	2,8621
0,018	0,1	0,082	0,003276	953	5,884884	0,5	3,5776

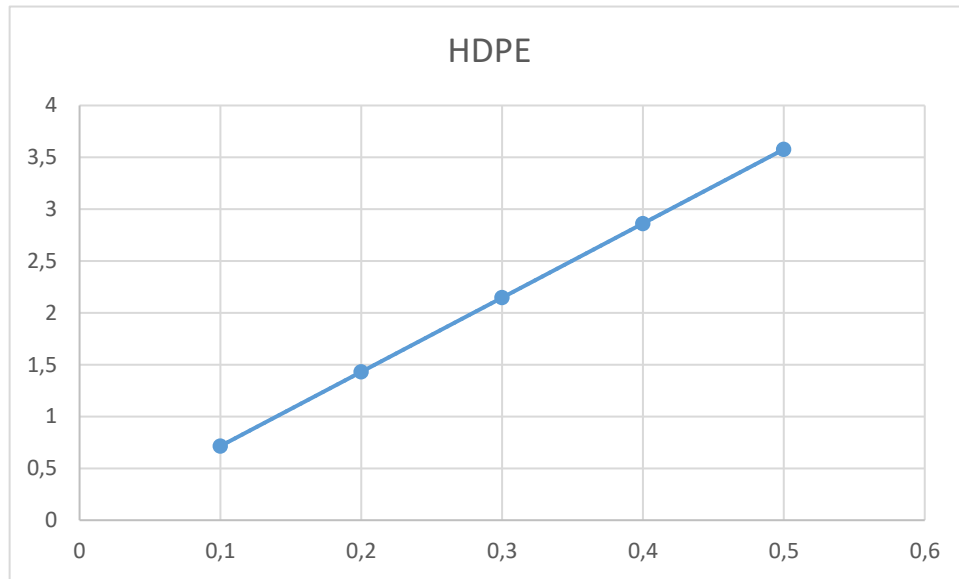


Fig. 96 Stress in function of External Pressure HDPE

Table 50 UHMW-PE Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,0185	0,1	0,0815	0,006329	934,5	5,914646	0,1	0,69435
0,0185	0,1	0,0815	0,003358	934,5	5,914646	0,2	1,3887
0,0185	0,1	0,0815	0,003358	934,5	5,914646	0,3	2,083
0,0185	0,1	0,0815	0,003358	934,5	5,914646	0,4	2,7774
0,0185	0,1	0,0815	0,003358	934,5	5,914646	0,5	3,4717

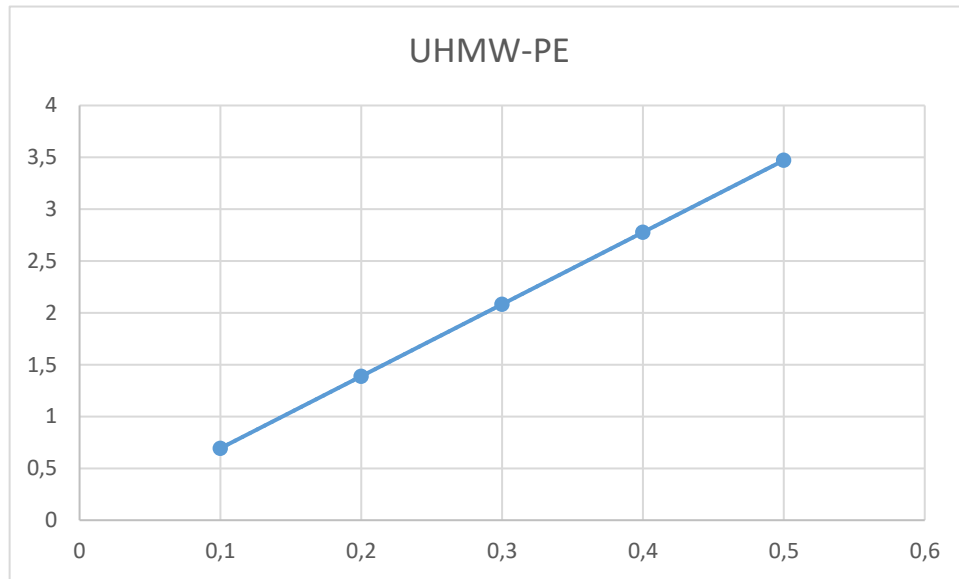


Fig. 97 Stress in function of External Pressure - UHMW-PE

Table 51 PBI Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,013	0,1	0,087	0,004582	1300	5,957025	0,1	1,0097
0,013	0,1	0,087	0,002431	1300	5,957025	0,5	5,0483
0,013	0,1	0,087	0,002431	1300	5,957025	1	10,097
0,013	0,1	0,087	0,002431	1300	5,957025	1,5	15,145
0,013	0,1	0,087	0,002431	1300	5,957025	2	20,193

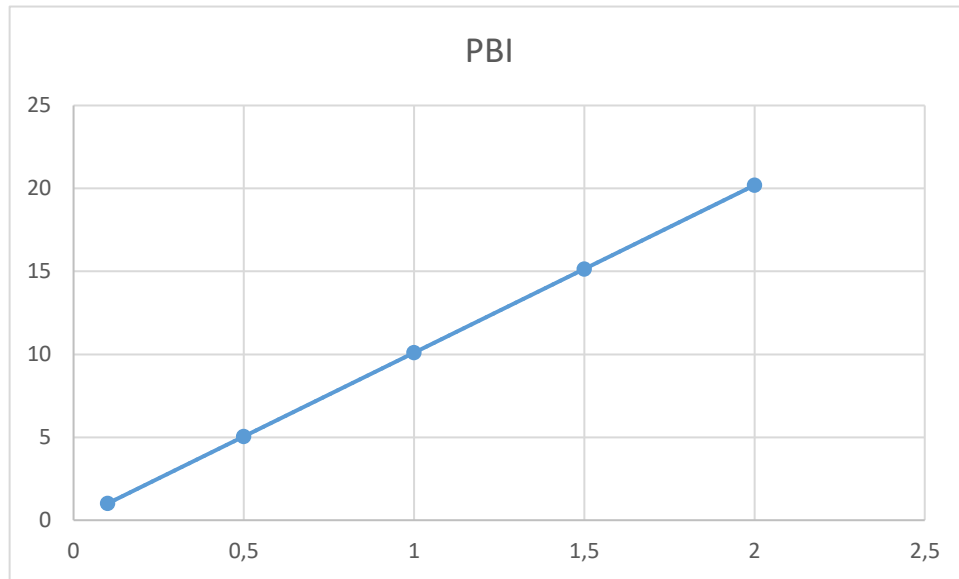


Fig. 98 Stress in function of External Pressure - PBI

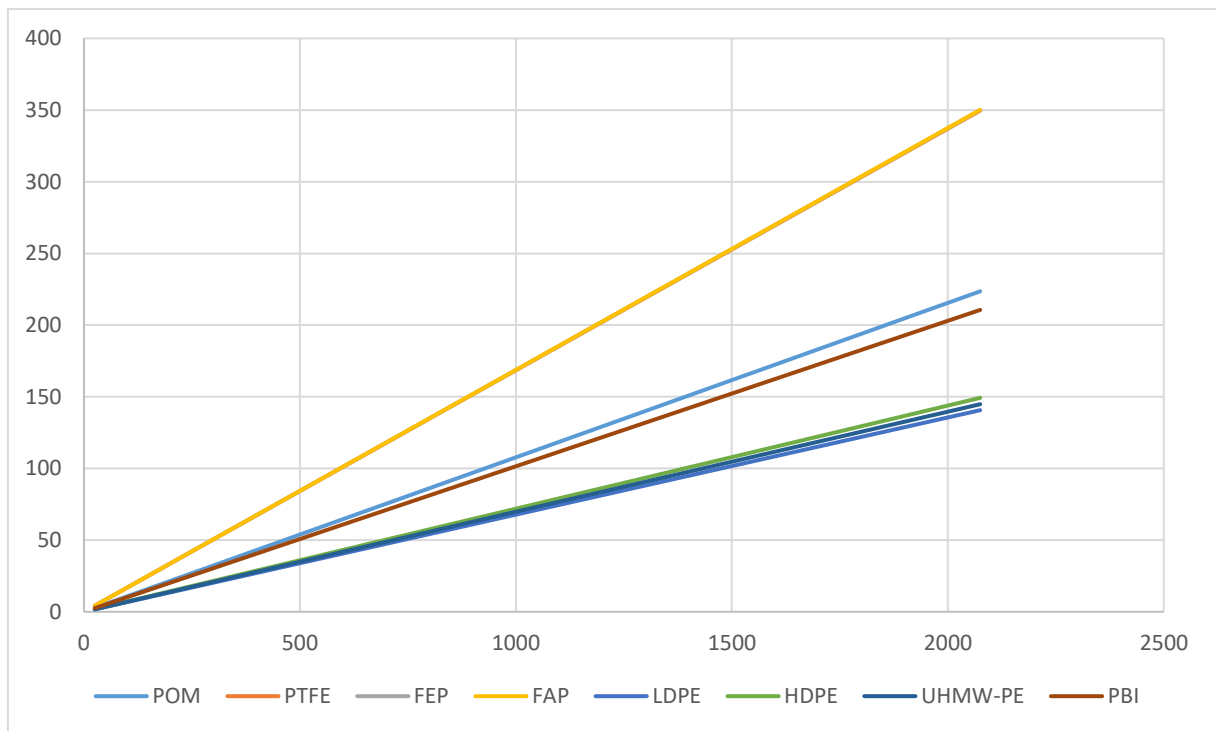


Fig. 99 Comparison of polymeric material stress

Table 52 94Al2O3 Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,004	0,1	0,096	0,001478	3665	5,416156	1	31,219
0,004	0,1	0,096	0,000784	3665	5,416156	2	62,439
0,004	0,1	0,096	0,000784	3665	5,416156	3	93,658
0,004	0,1	0,096	0,000784	3665	5,416156	4	124,8797
0,004	0,1	0,096	0,000784	3665	5,416156	5	156,0997

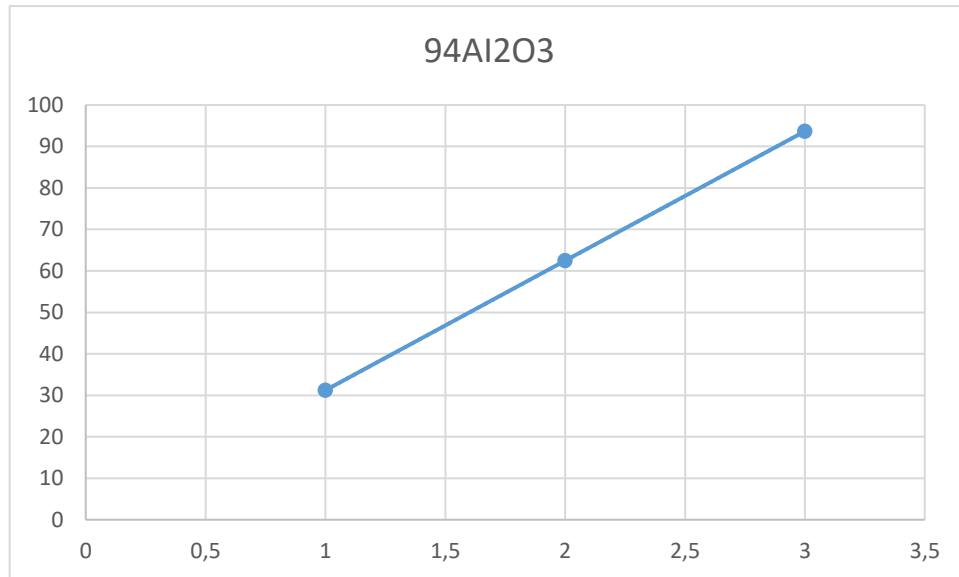


Fig. 100 Stress in function of External Pressure - 94Al2O3

Table 53 96Al2O3 Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,004	0,1	0,096	0,001478	3710	5,482657	1	31,219
0,004	0,1	0,096	0,000784	3710	5,482657	2	62,439
0,004	0,1	0,096	0,000784	3710	5,482657	3	93,658
0,004	0,1	0,096	0,000784	3710	5,482657	4	124,8797
0,004	0,1	0,096	0,000784	3710	5,482657	5	156,0997

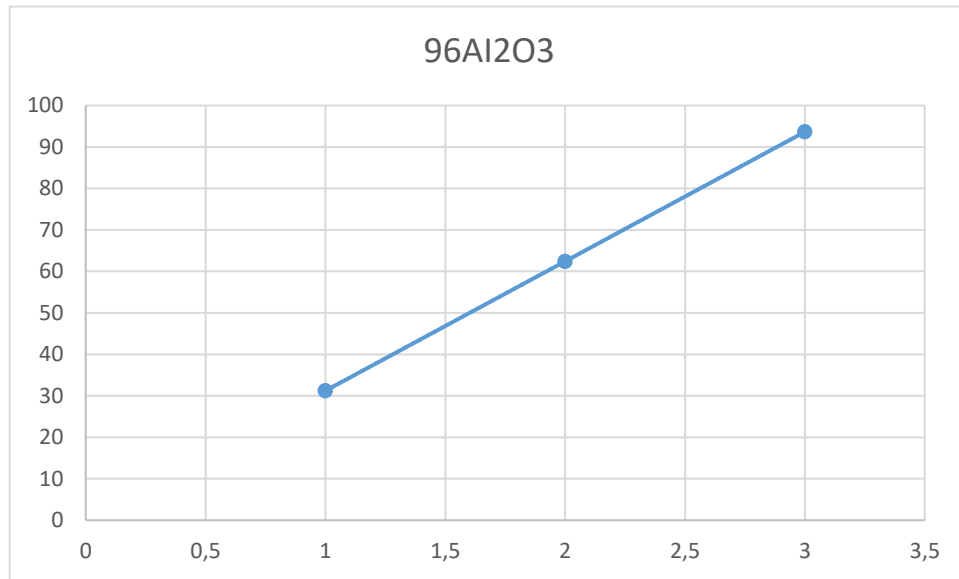


Fig. 101 Stress in function of External Pressure - 96 Al2O3

Table 54 Si3N4 Analysis Values

t (m)	Re (m)	Ri (m)	V (m ³)	P (kg/ m ³)	m (kg)	Pe (MPa)	σ (MPa)
0,005	0,1	0,095	0,001838	3195	5,871872	1	31,153
0,005	0,1	0,095	0,000975	3195	5,871872	2	62,307
0,005	0,1	0,095	0,000975	3195	5,871872	3	93,46
0,005	0,1	0,095	0,000975	3195	5,871872	4	124,6157
0,005	0,1	0,095	0,000975	3195	5,871872	5	155,7697

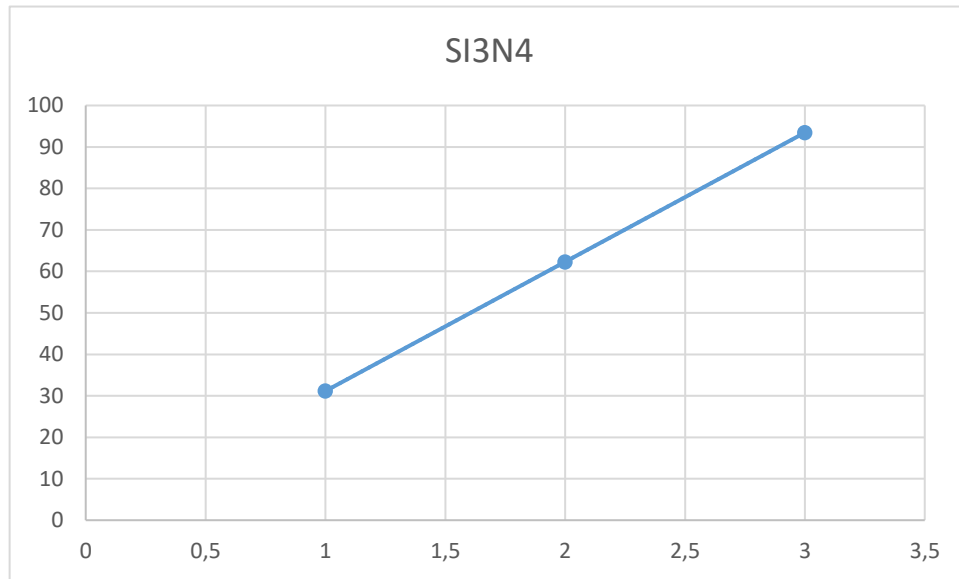


Fig. 102 Stress in function of External Pressure - Si3N4

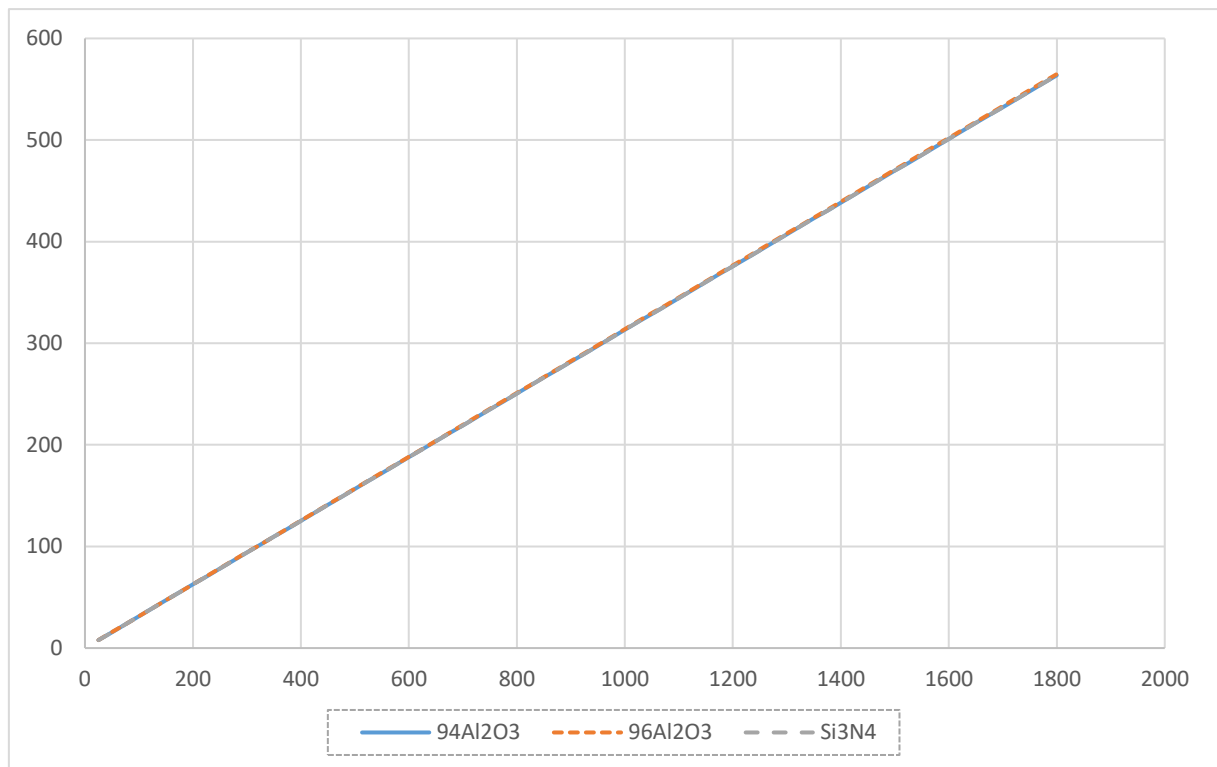


Fig. 103 Comparison of ceramic material stress

Appendix B - Different components stress representation for distinct Load Cases

Main Supports - Upper

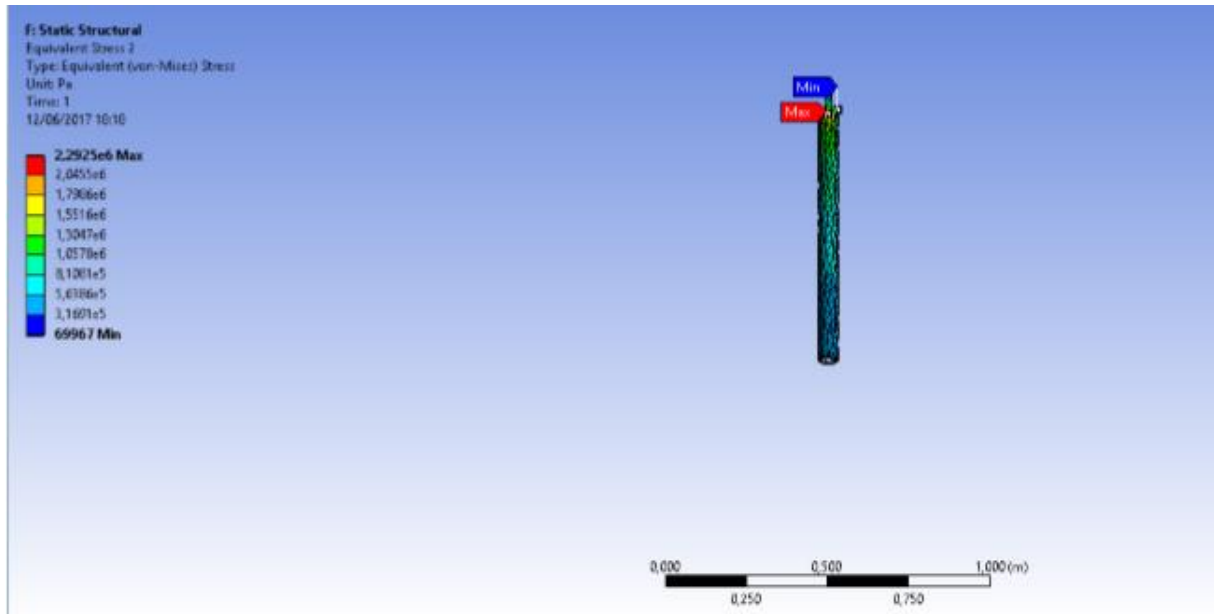


Fig. 104 Main Support (Upper Part) - Load Case 1 - Steel

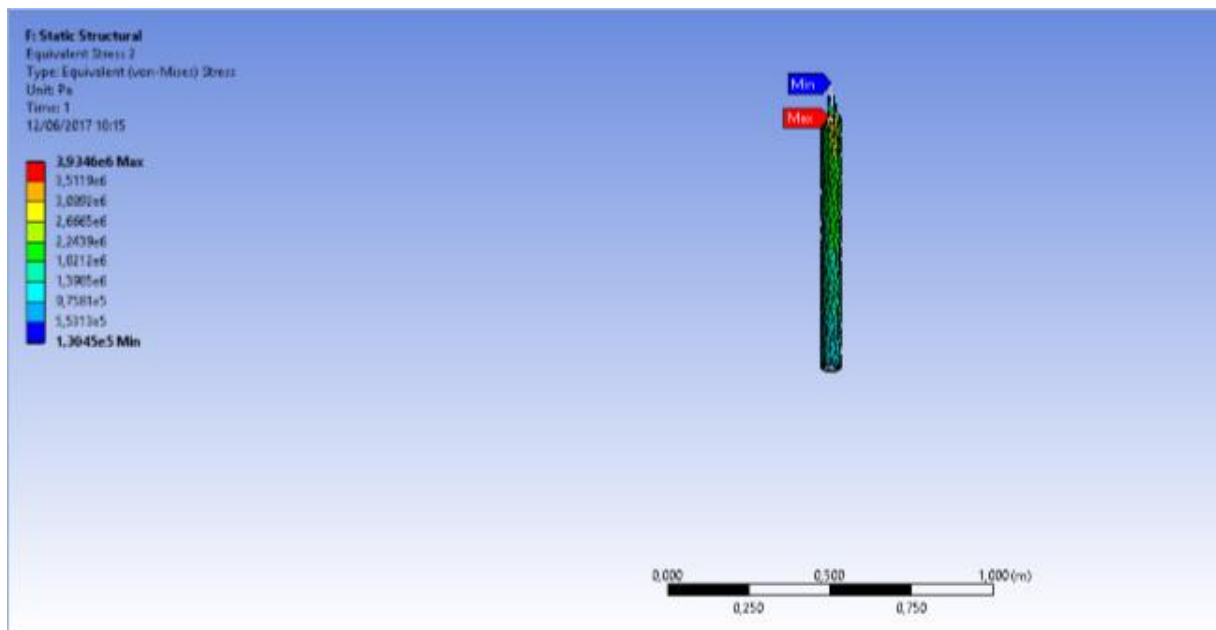


Fig. 105 Main Support (Upper Part) - Load Case 2 - Steel

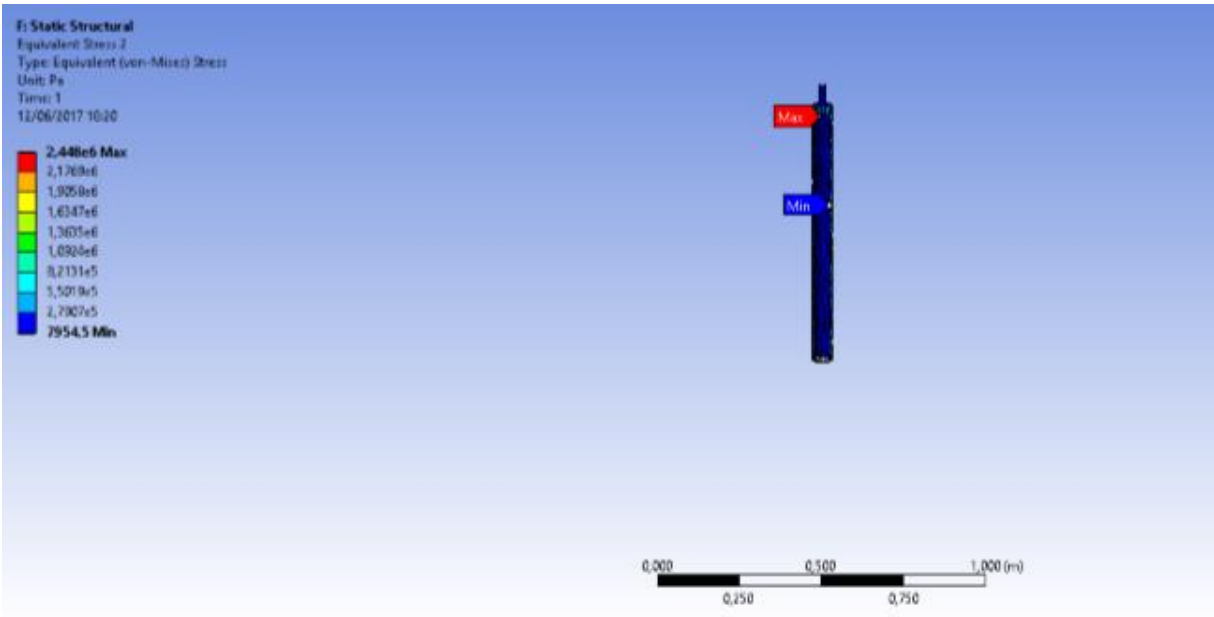


Fig. 106 Main Support (Upper Part) - Load Case 3 - Steel

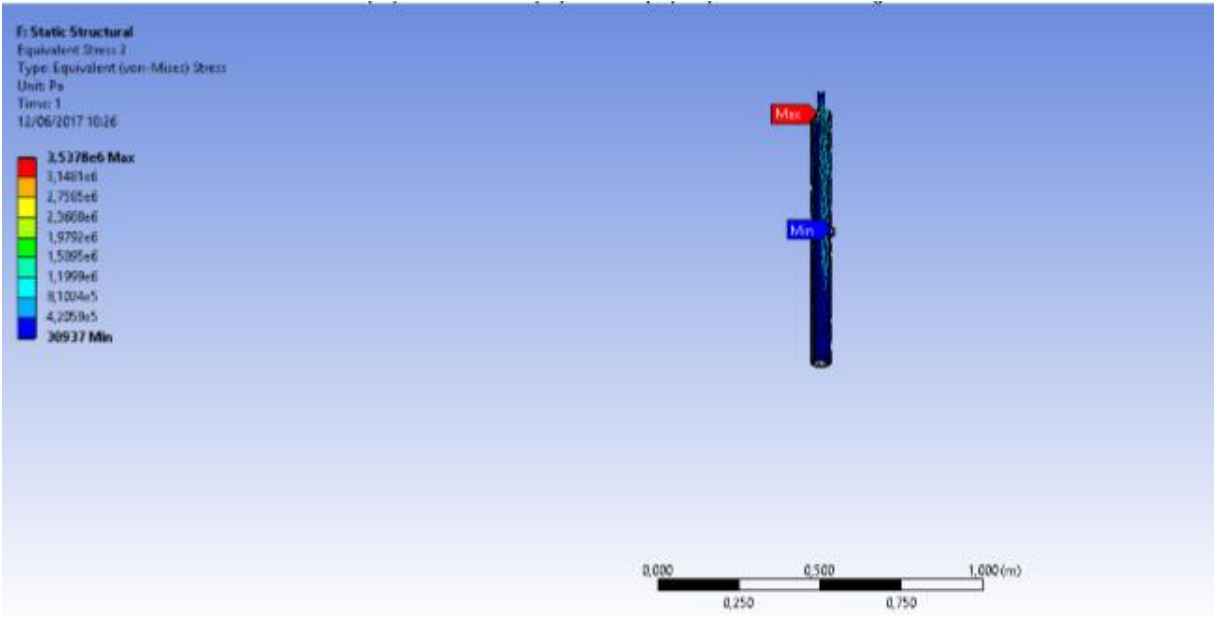


Fig. 107 Main Support (Upper Part) - Load Case 4 - Steel

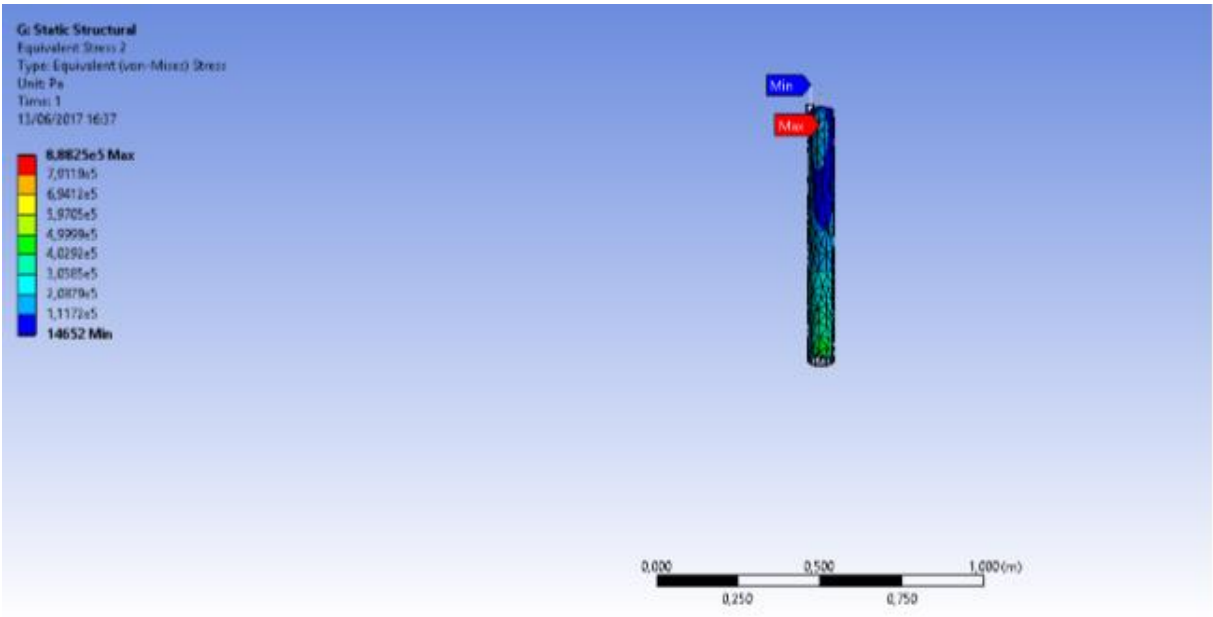


Fig. 108 Main Support (Upper Part) - Load Case 1 - POM

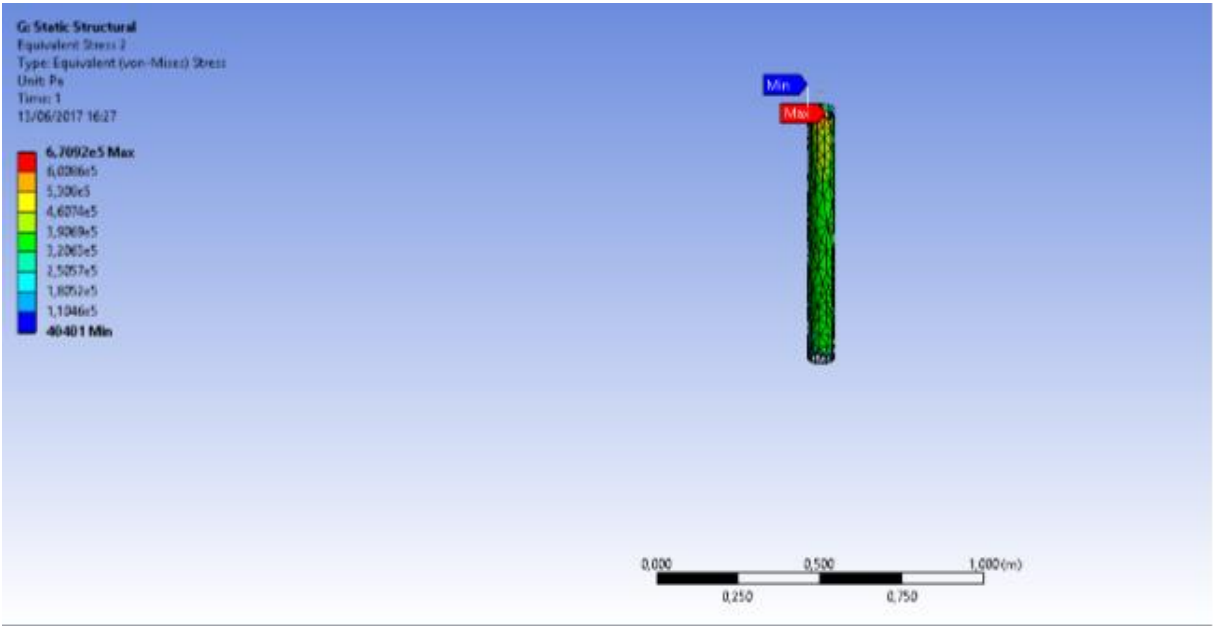


Fig. 109 Main Support (Upper Part) - Load Case 2 - POM

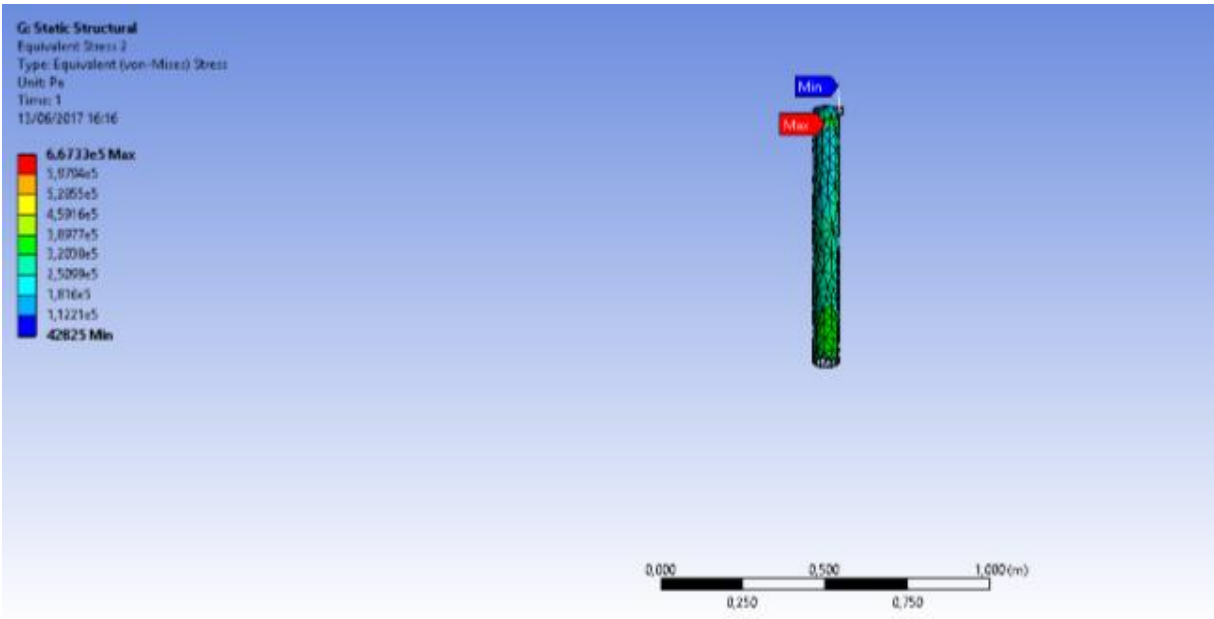


Fig. 110 Main Support (Upper Part) - Load Case 3 - POM

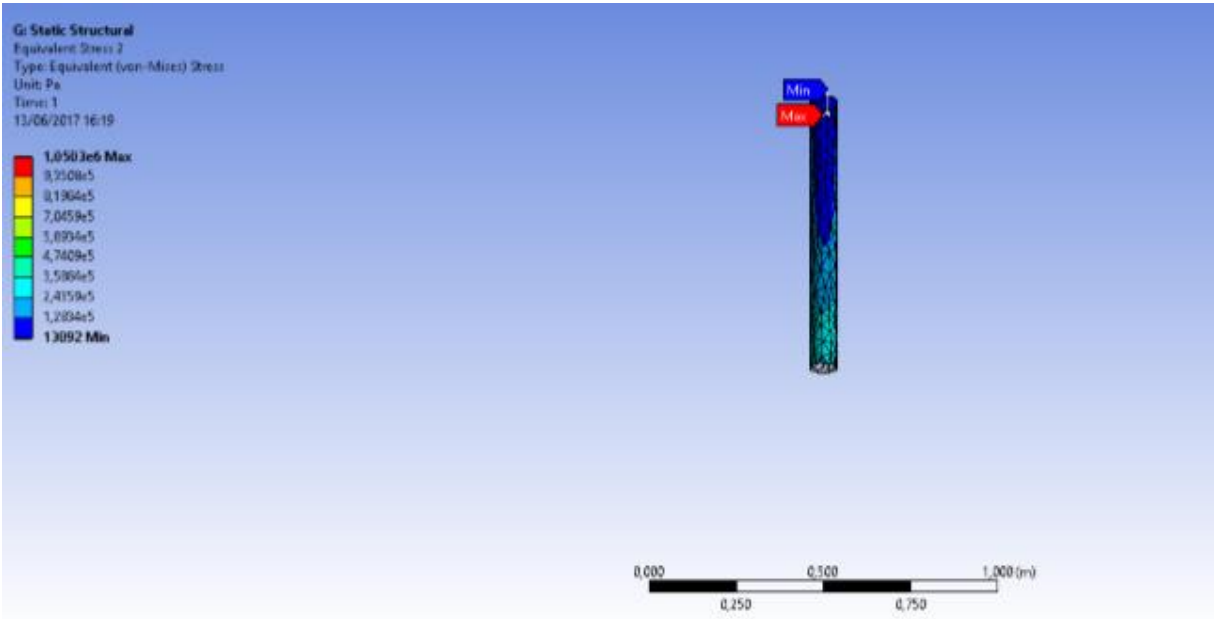


Fig. 111 Main Support (Upper Part) - Load Case 4 – POM

Main Support - Inferior Part

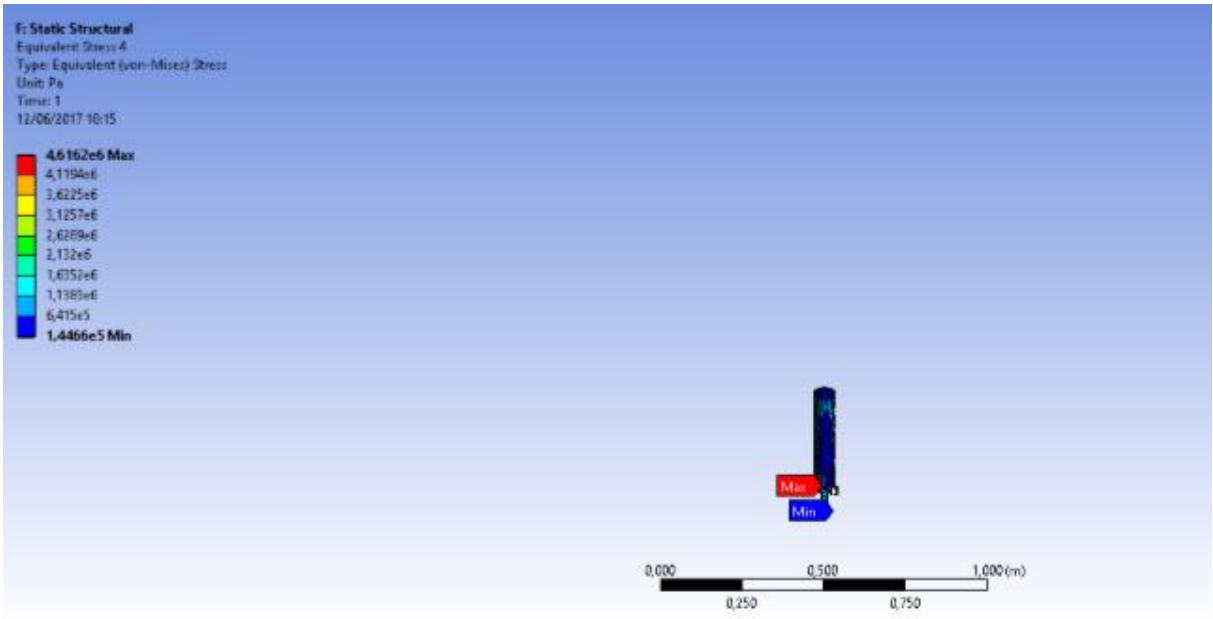


Fig. 112 Main Support (Inferior Part) - Load Case 1 - Steel

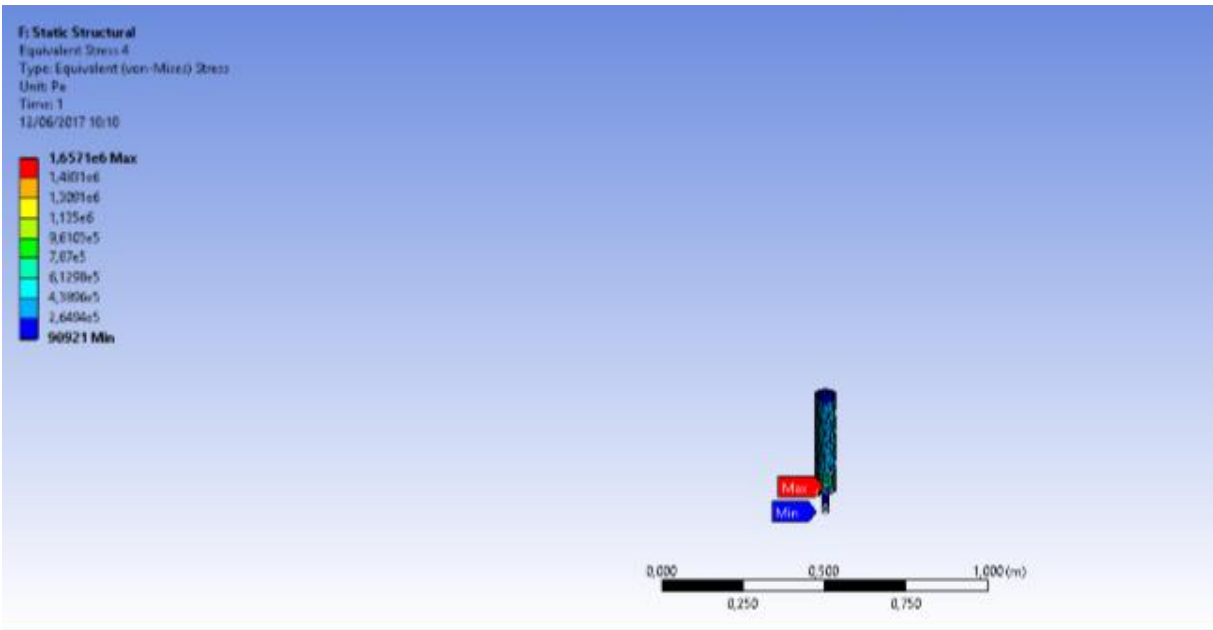


Fig. 113 Main Support (Inferior Part) - Load Case 2 - Steel

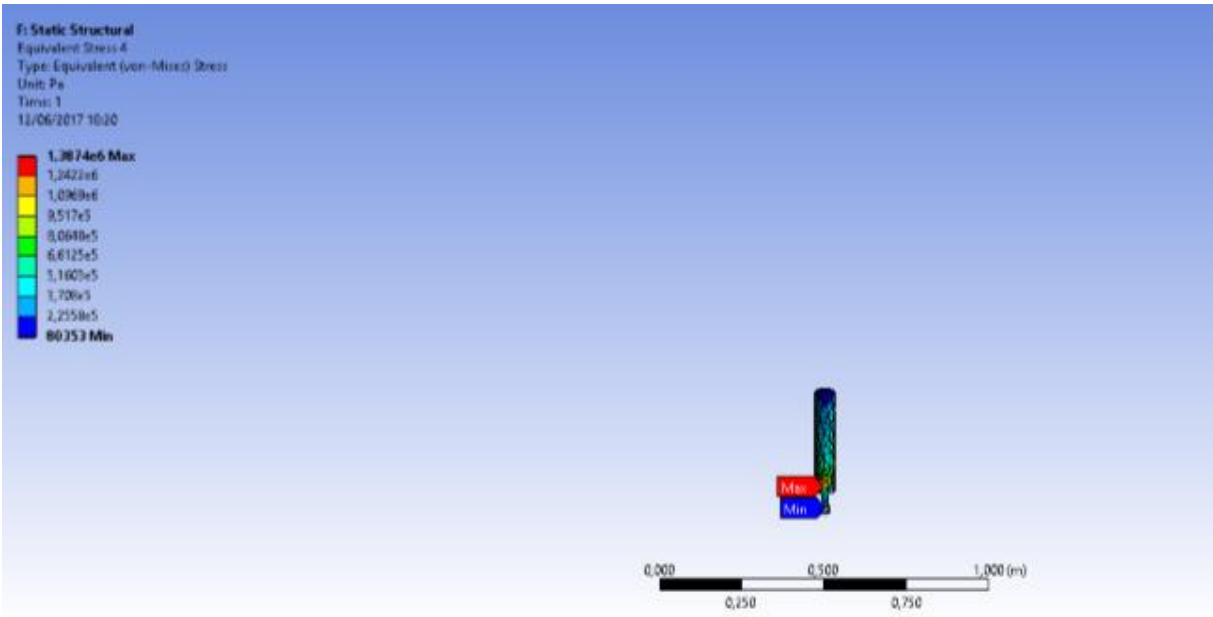


Fig. 114 Main Support (Inferior Part) - Load Case 3 - Steel

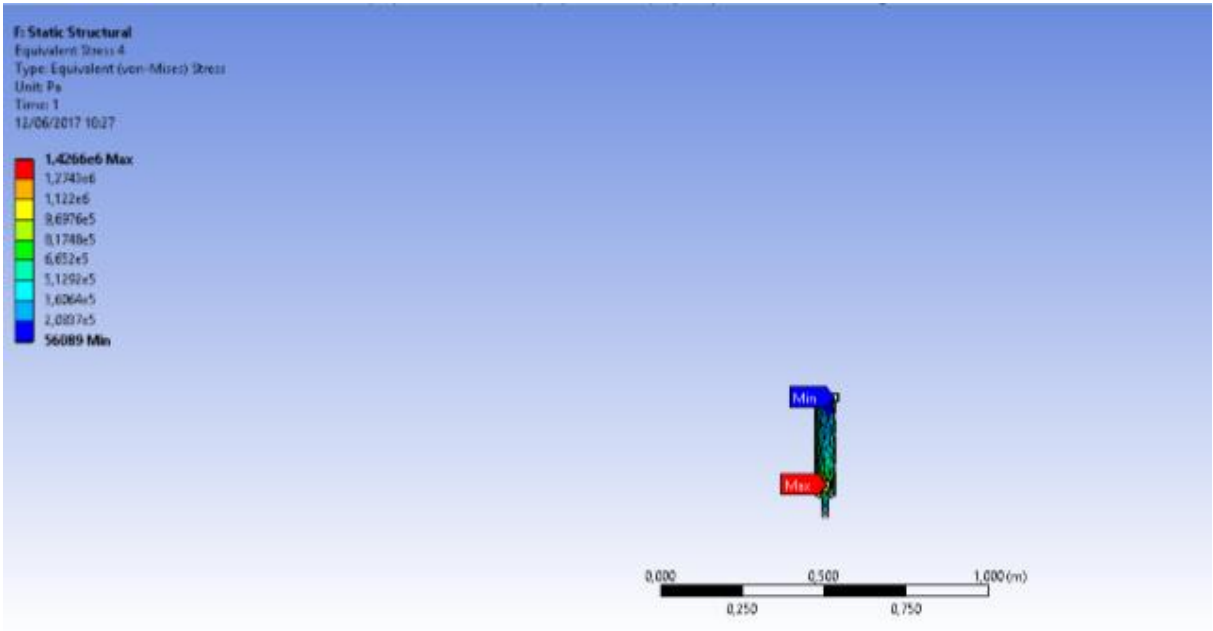


Fig. 115 Main Support (Inferior Part) - Load Case 4 - Steel

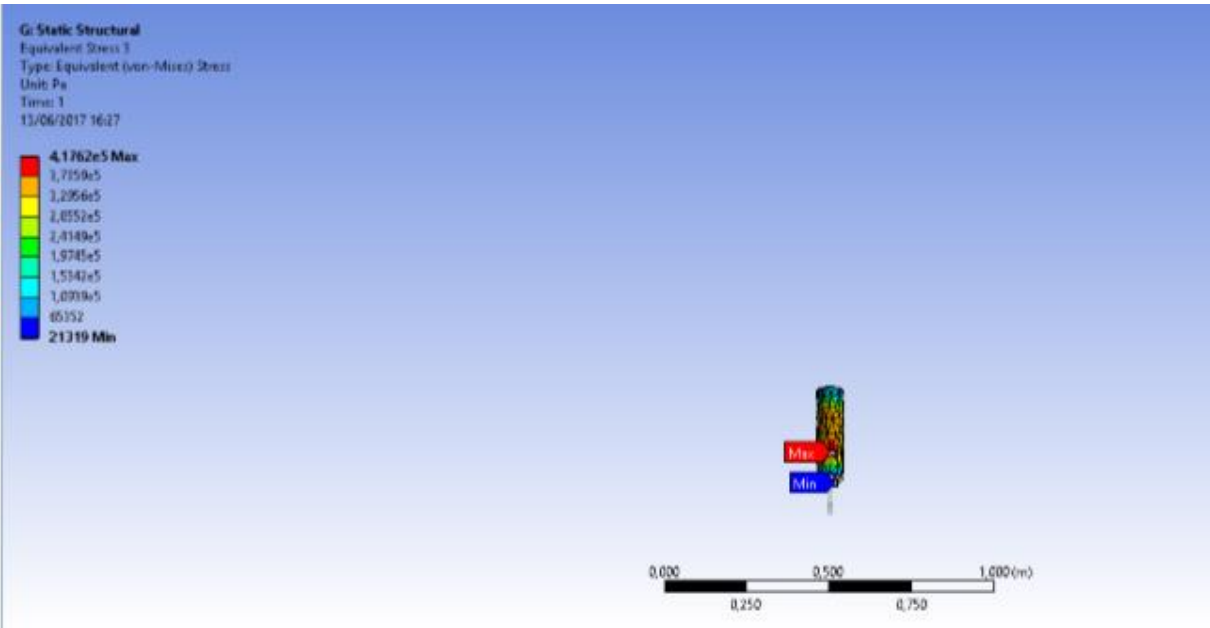


Fig. 116 Main Support (Inferior Part) - Load Case 1 - POM

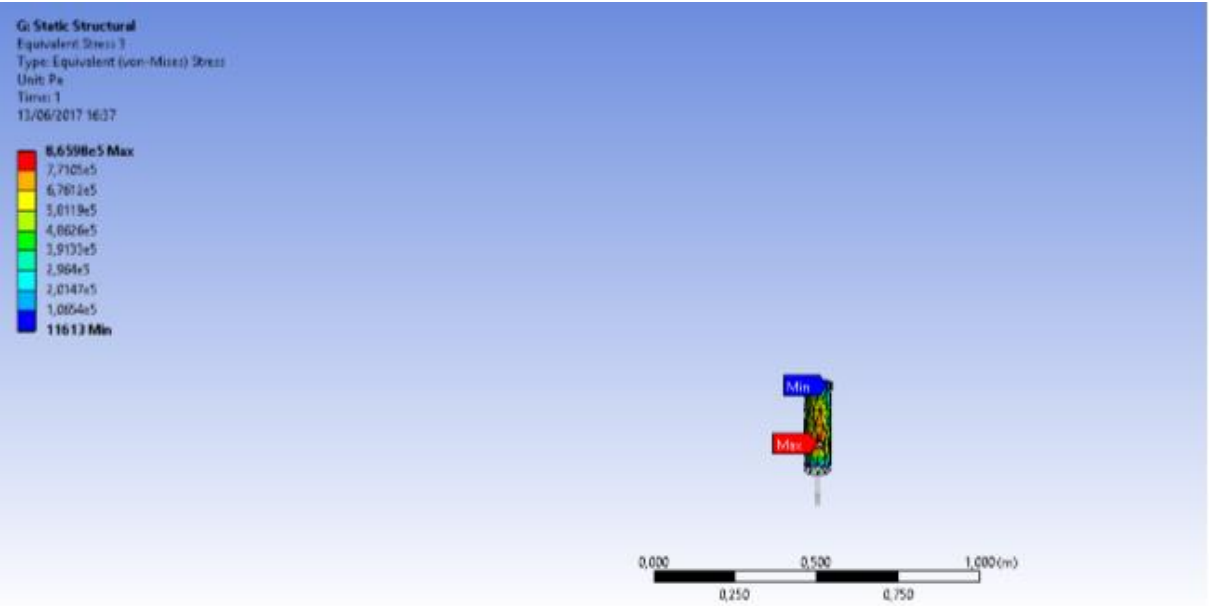


Fig. 117 Main Support (Inferior Part) - Load Case 2 - POM

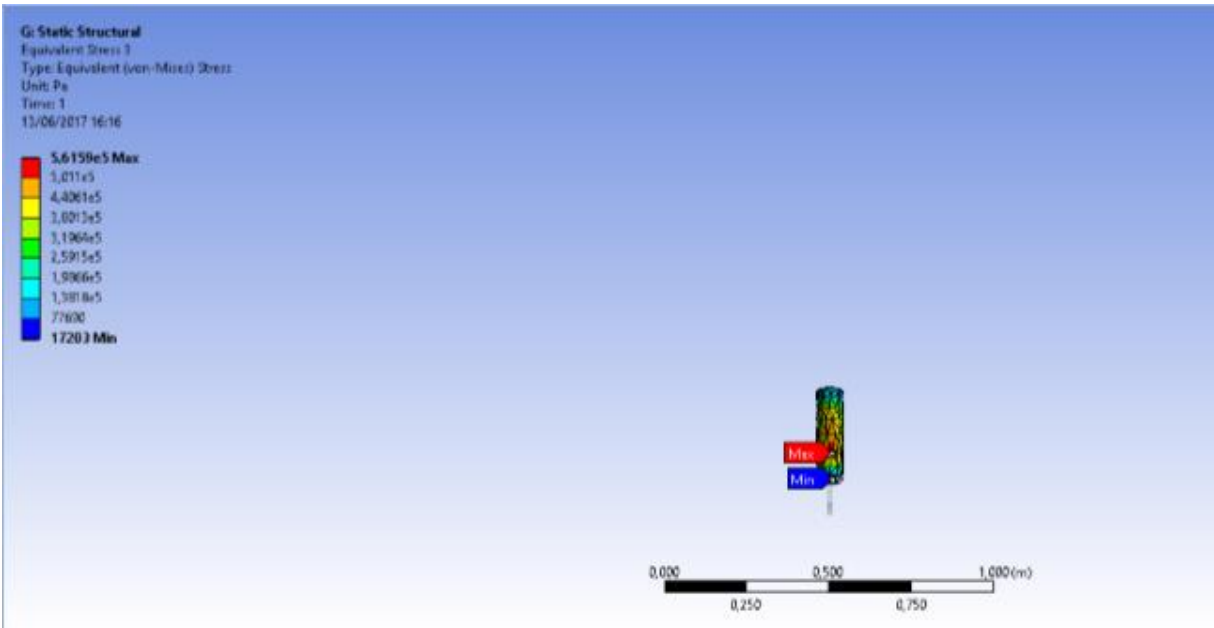


Fig. 118 Main Support (Inferior Part) - Load Case 3 - POM

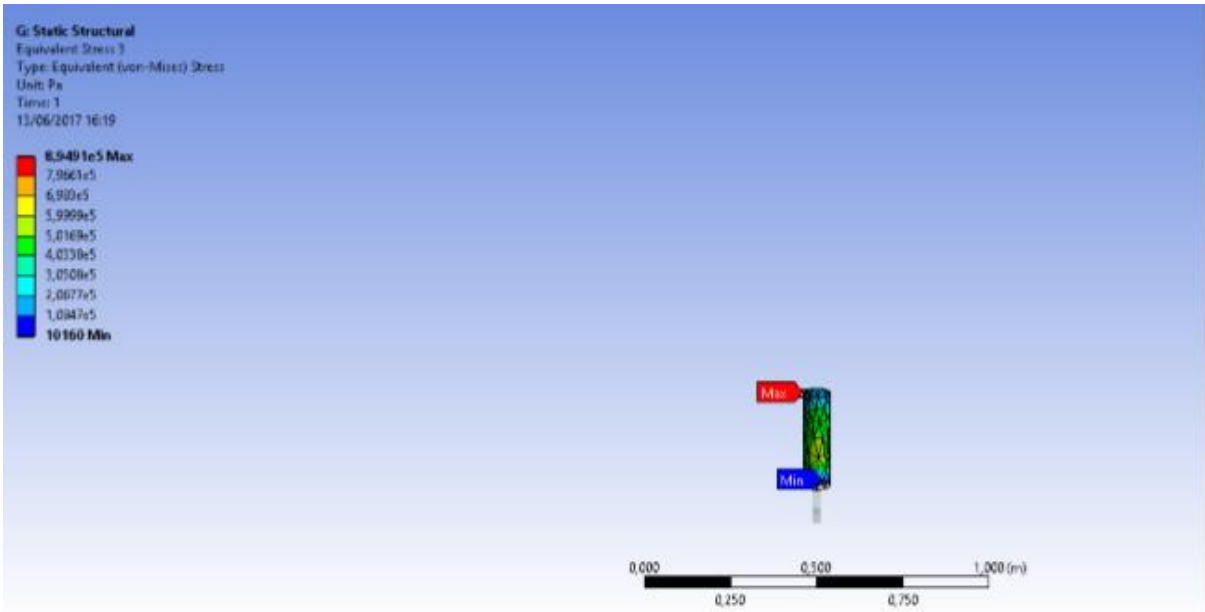


Fig. 119 Main Support (Inferior Part) - Load Case 3 – POM

Secondary Supports

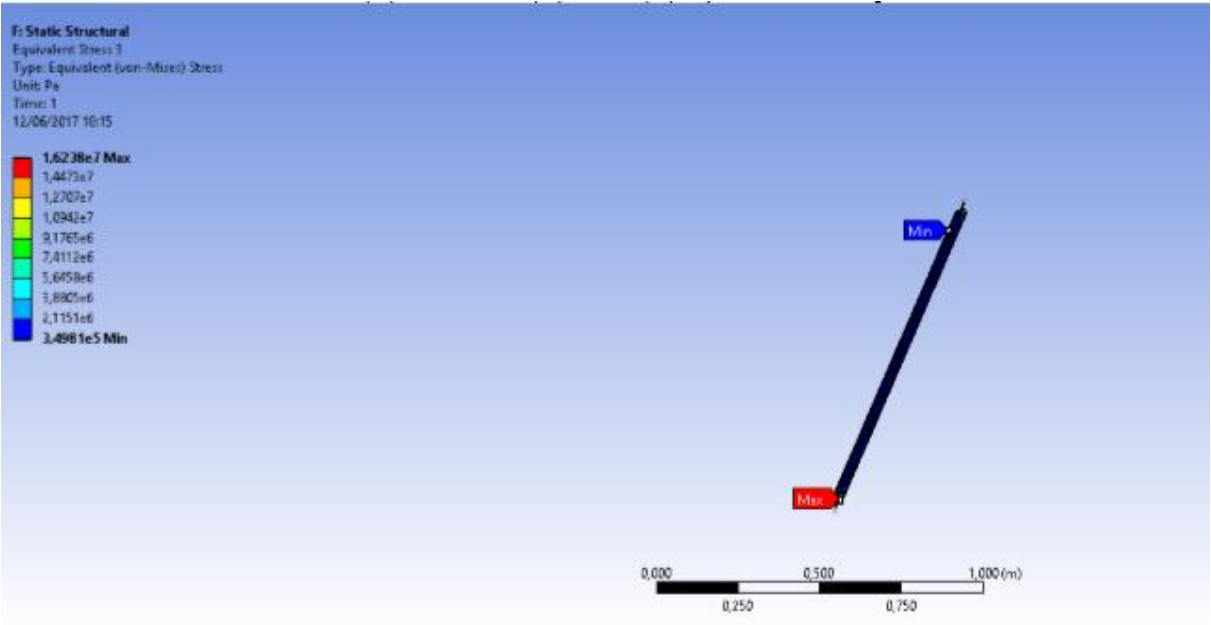


Fig. 120 Secondary Support - Load Case 1 - Steel

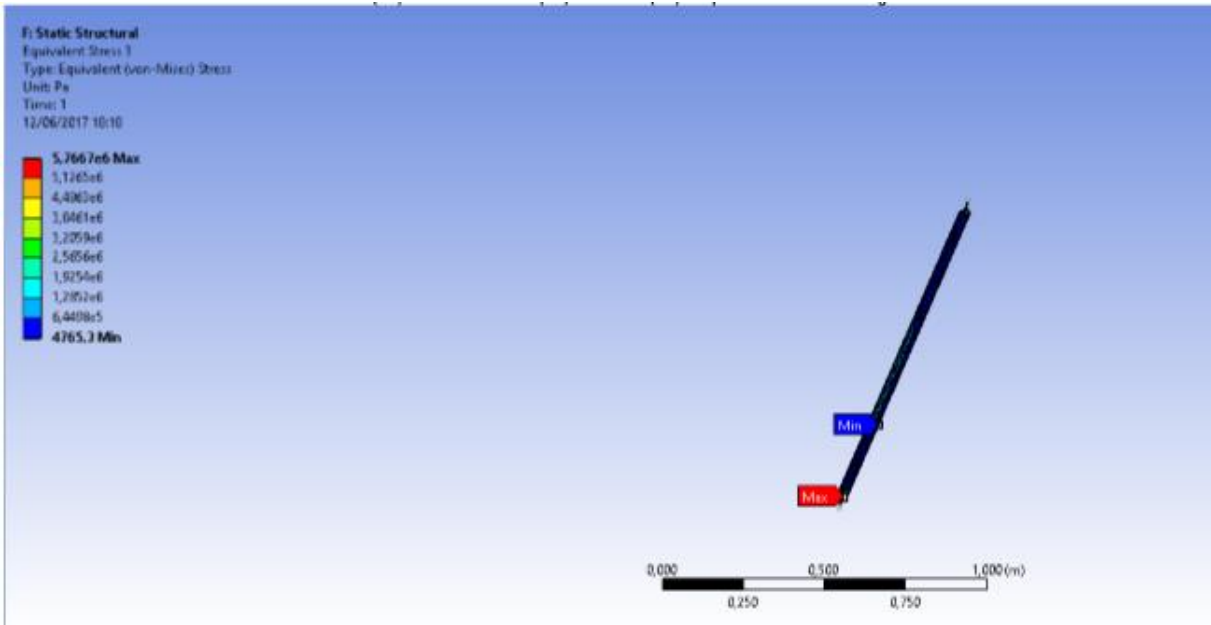


Fig. 121 Secondary Support - Load Case 2 - Steel

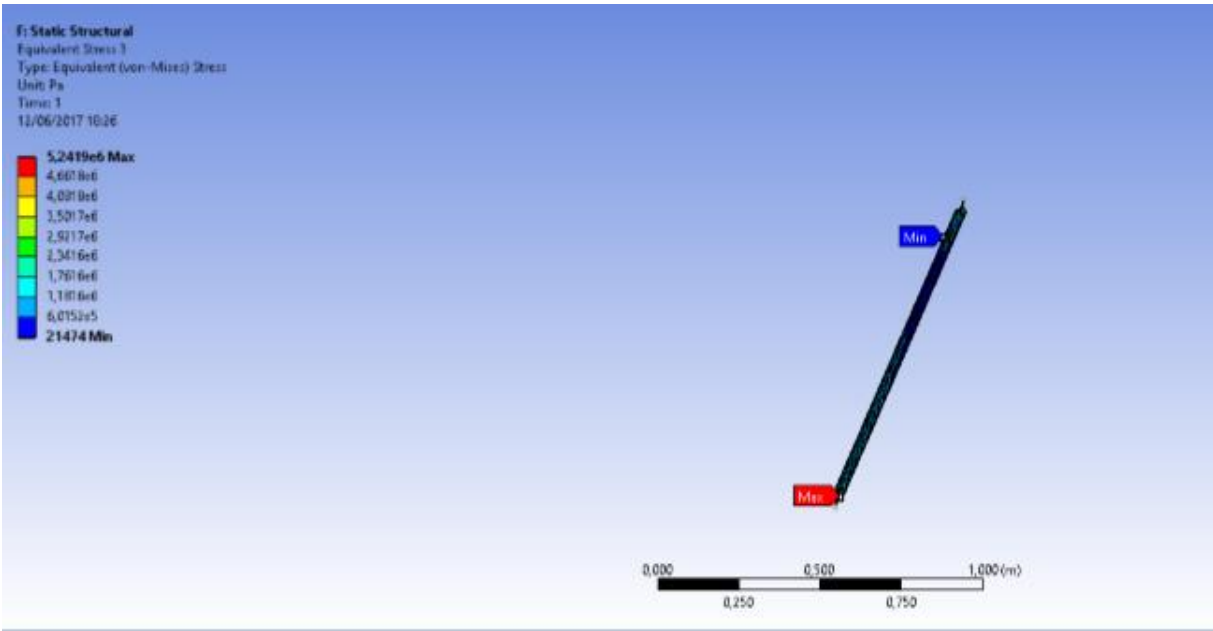


Fig. 122 Secondary Support - Load Case 3 - Steel

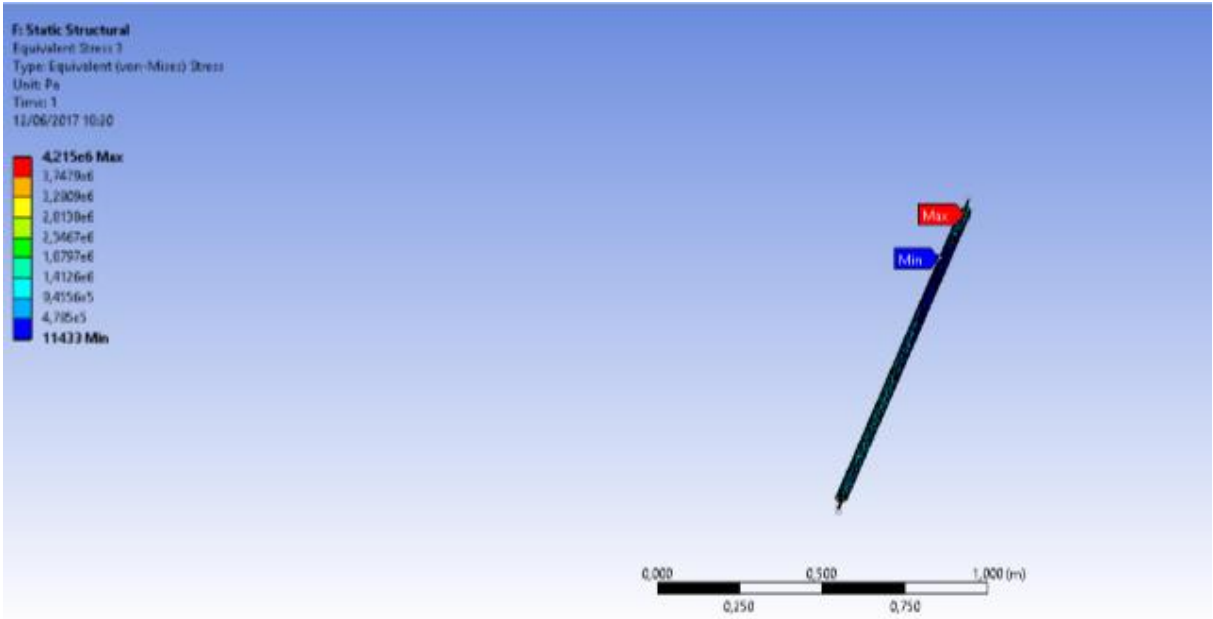


Fig. 123 Secondary Support - Load Case 4 - Steel

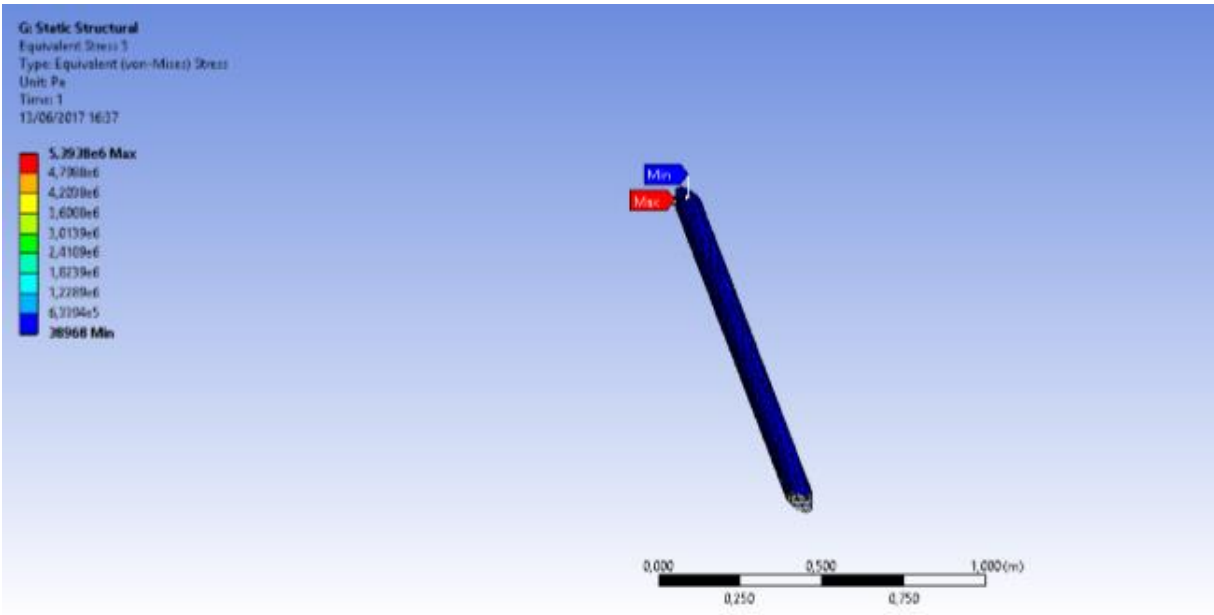


Fig. 124 Secondary Support - Load Case 1 - POM

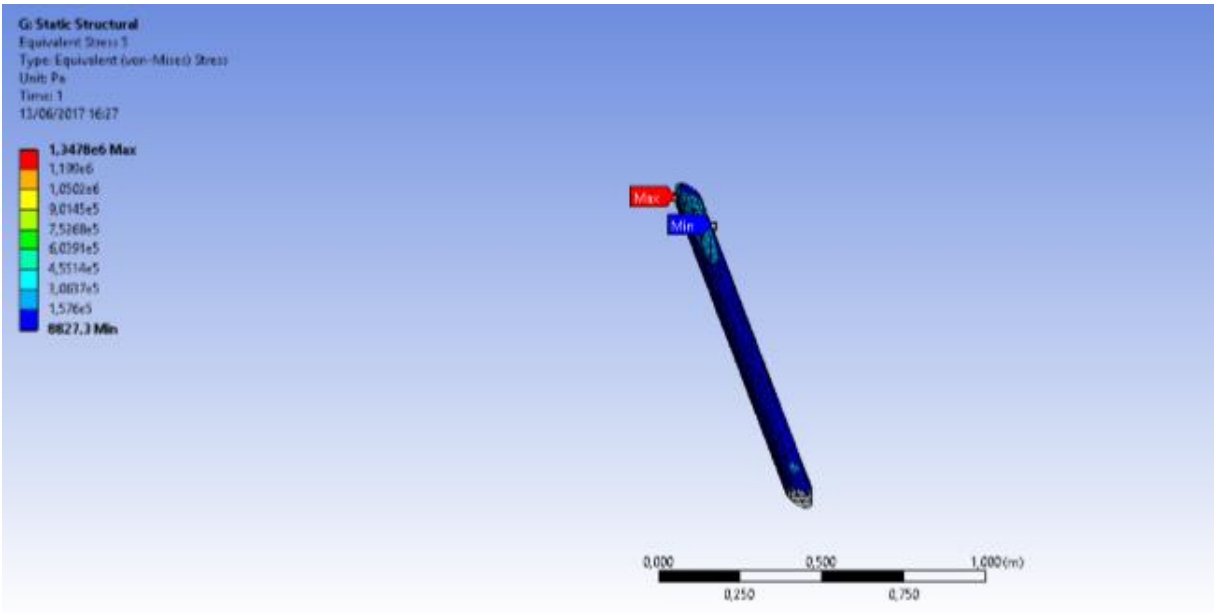


Fig. 125 Secondary Support - Load Case 2 - POM

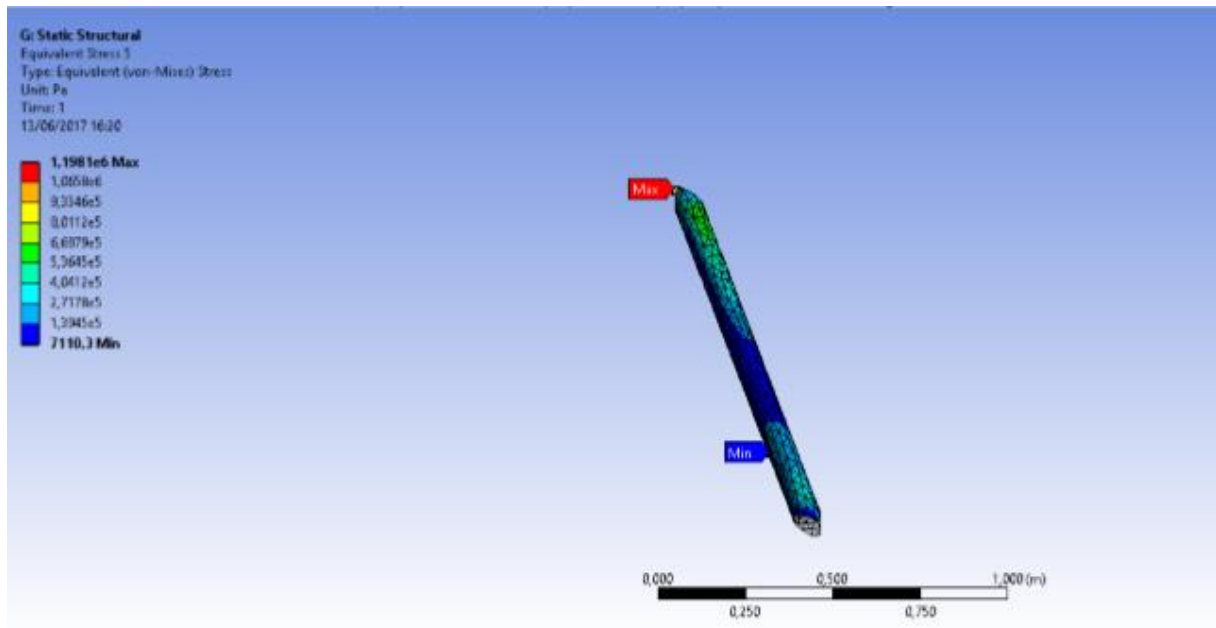


Fig. 126 Secondary Support - Load Case 3 - POM

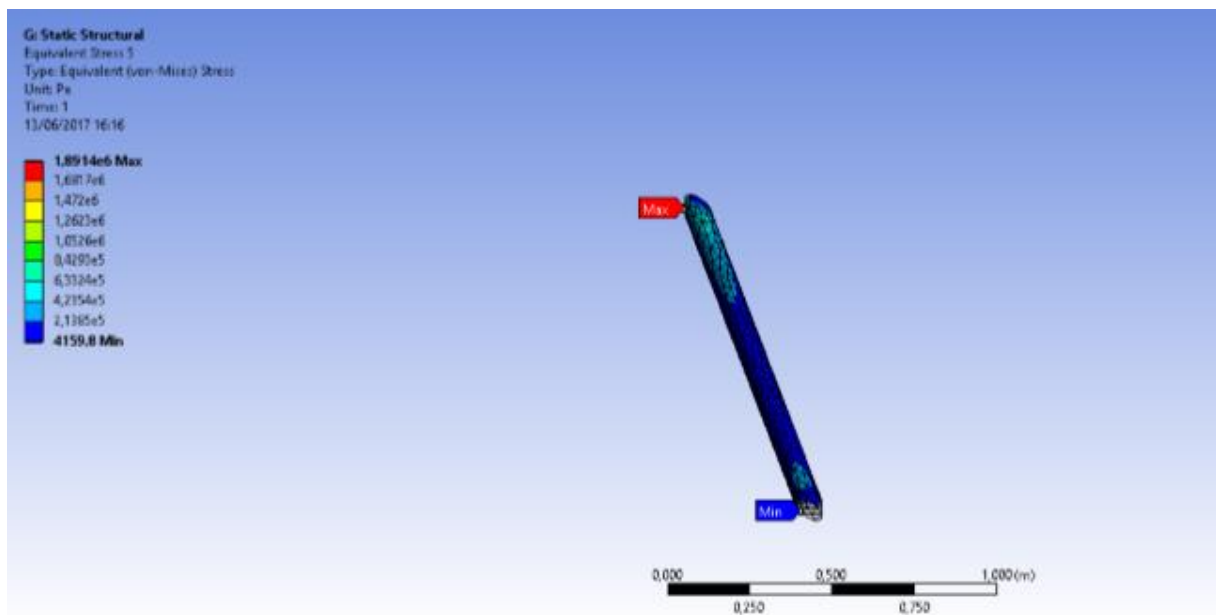


Fig. 127 Secondary Support - Load Case 4 – POM

Columns

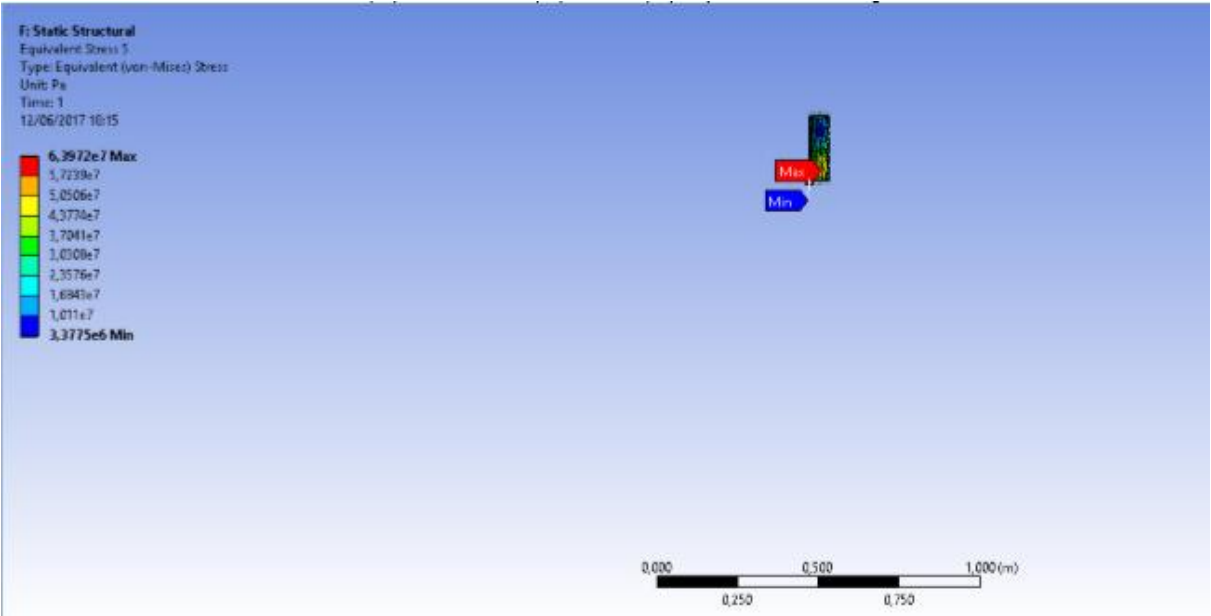


Fig. 128 Columns - Load Case 1 - Steel

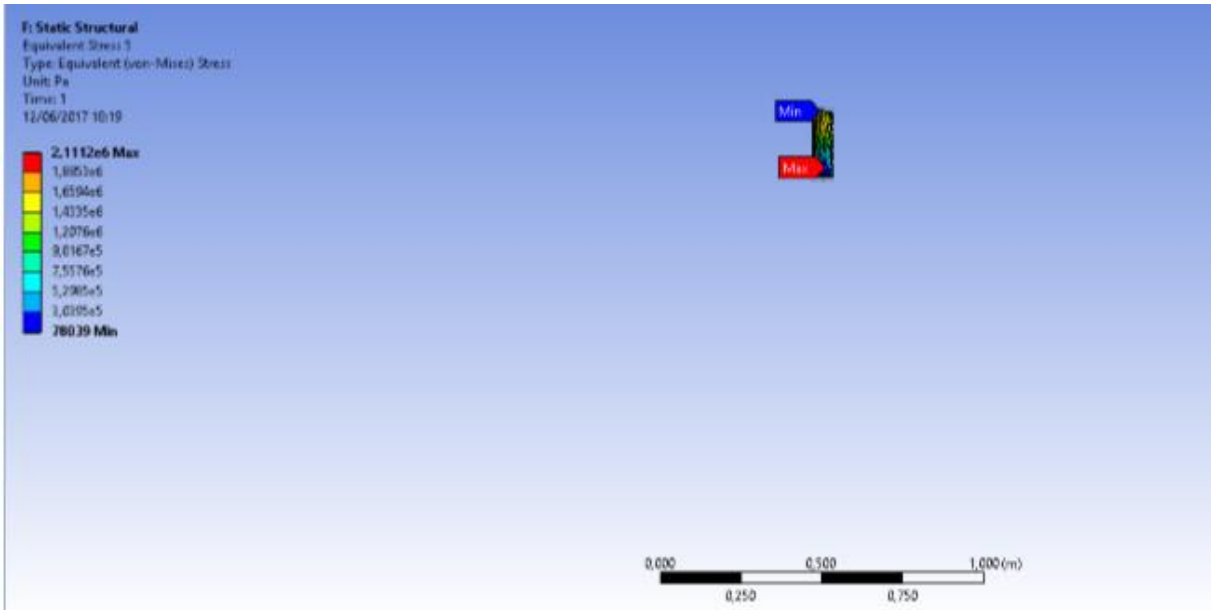


Fig. 129 Columns - Load Case 2 - Steel

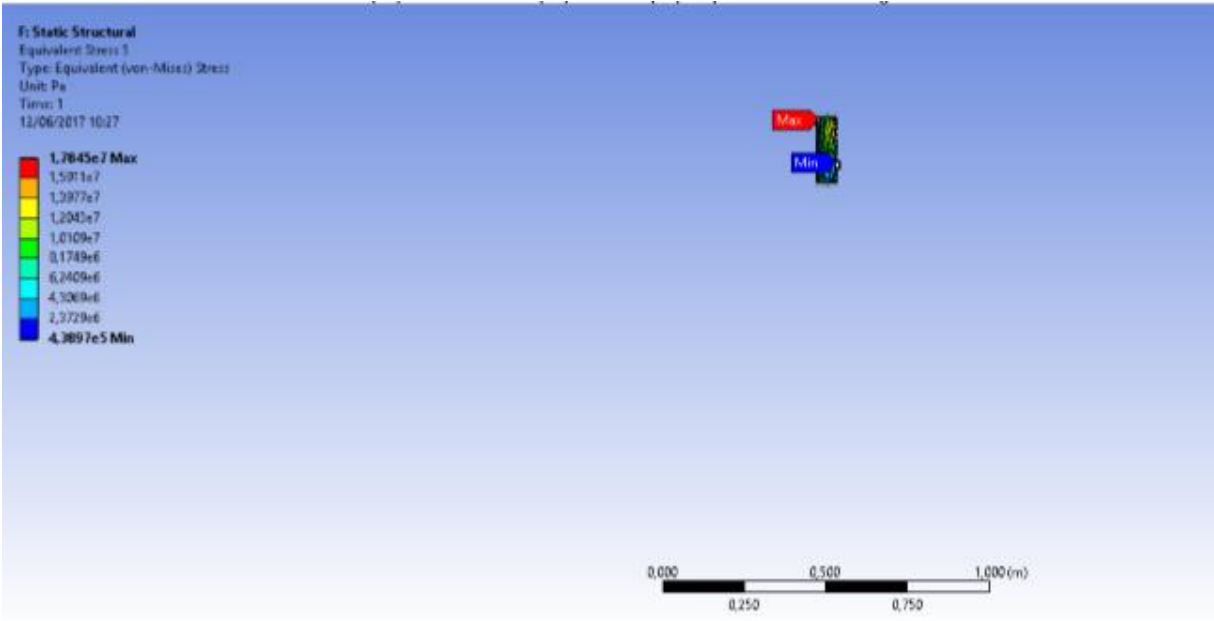


Fig. 130 Columns - Load Case 3 - Steel

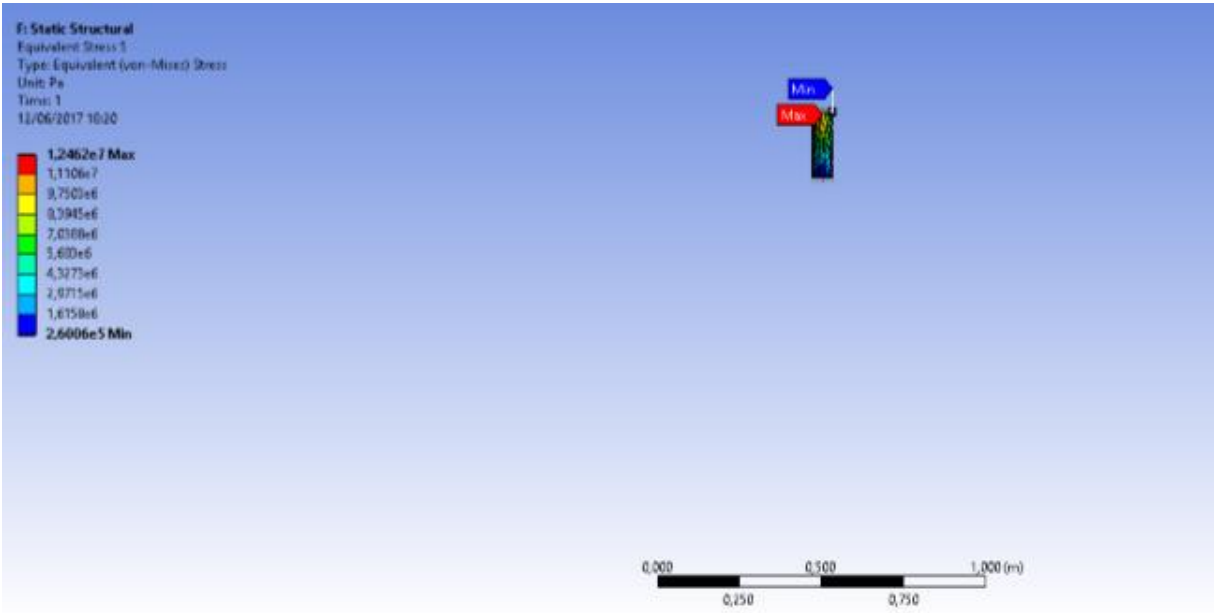


Fig. 131 Columns - Load Case 4 - Steel

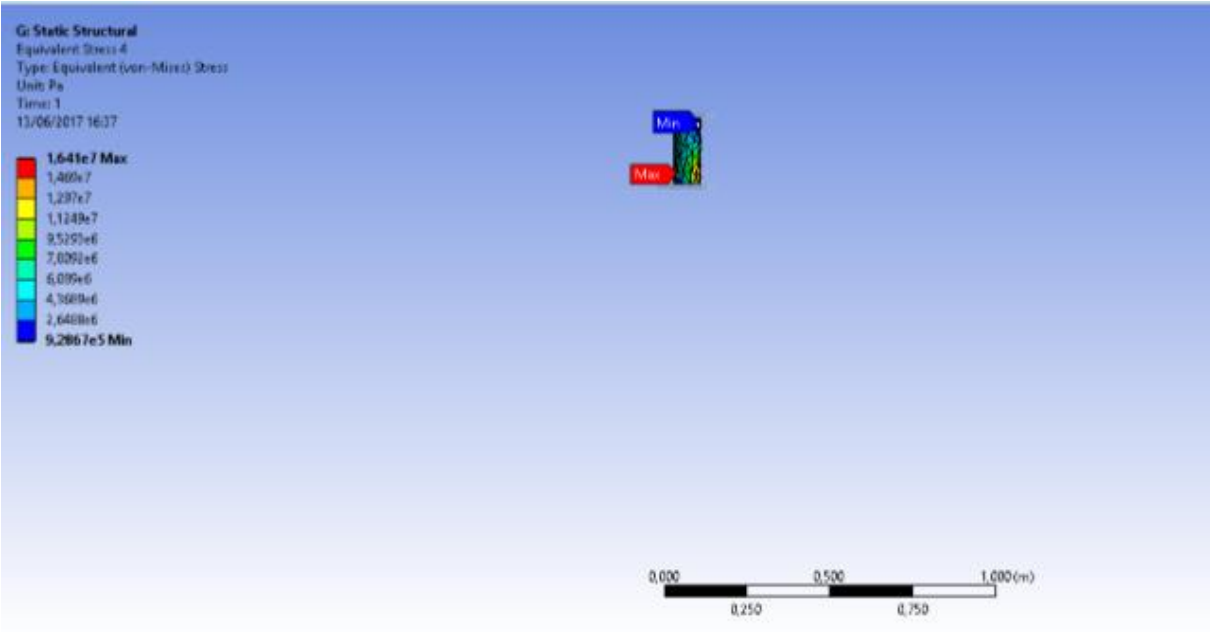


Fig. 132 Columns - Load Case 1 - POM

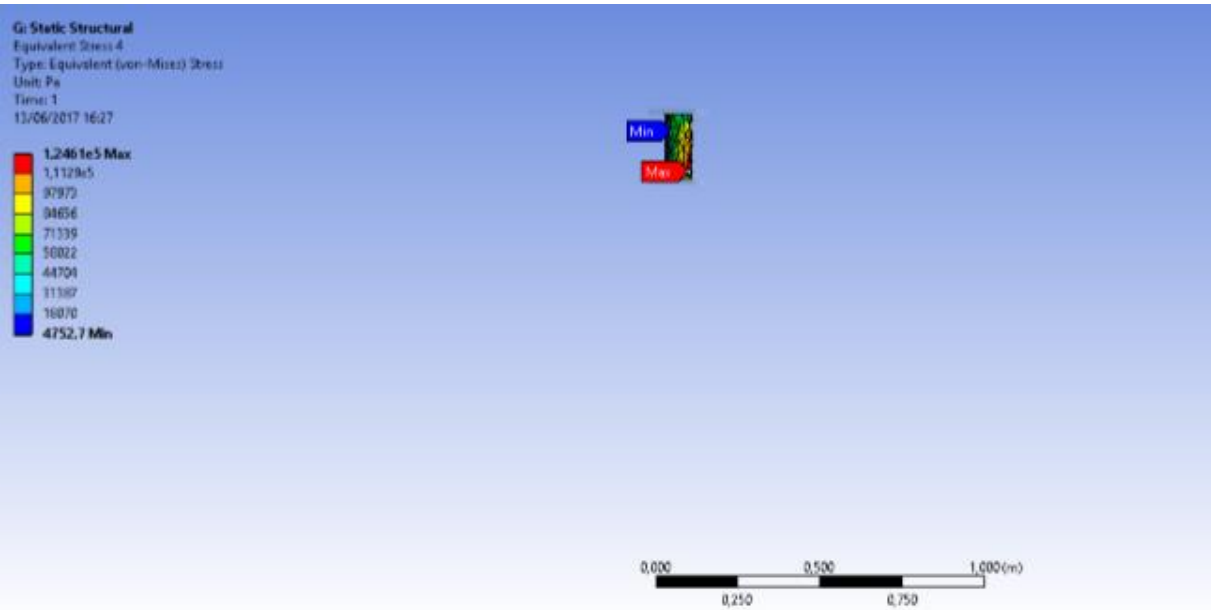


Fig. 133 Columns - Load Case 2 - POM

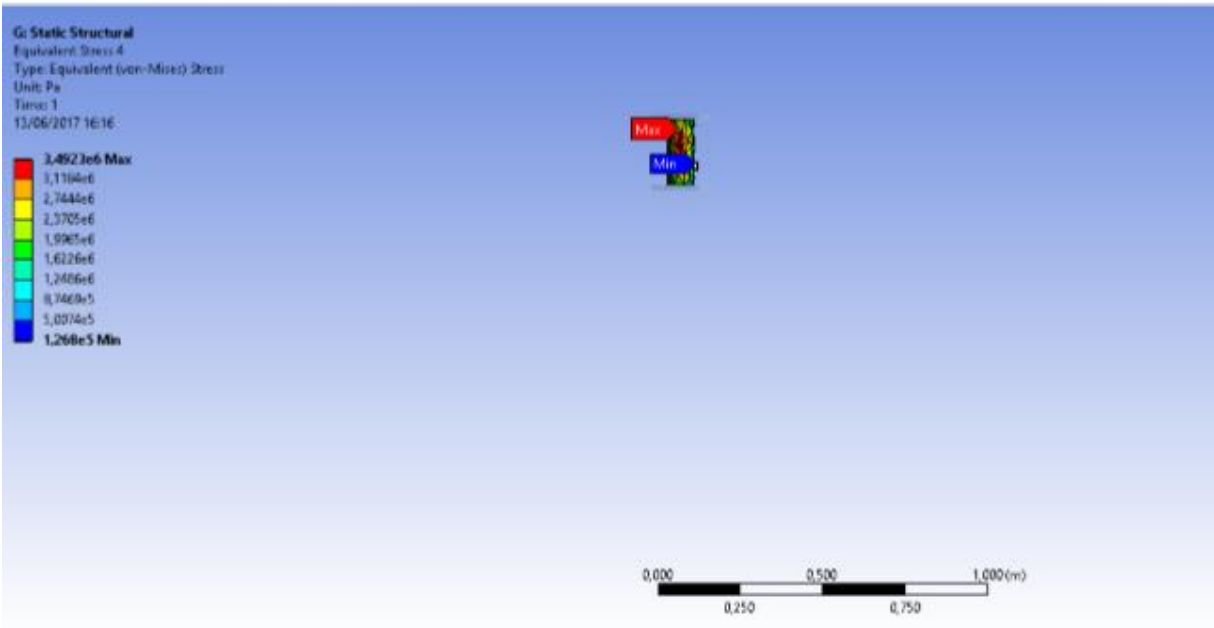


Fig. 134 Columns - Load Case 3 - POM

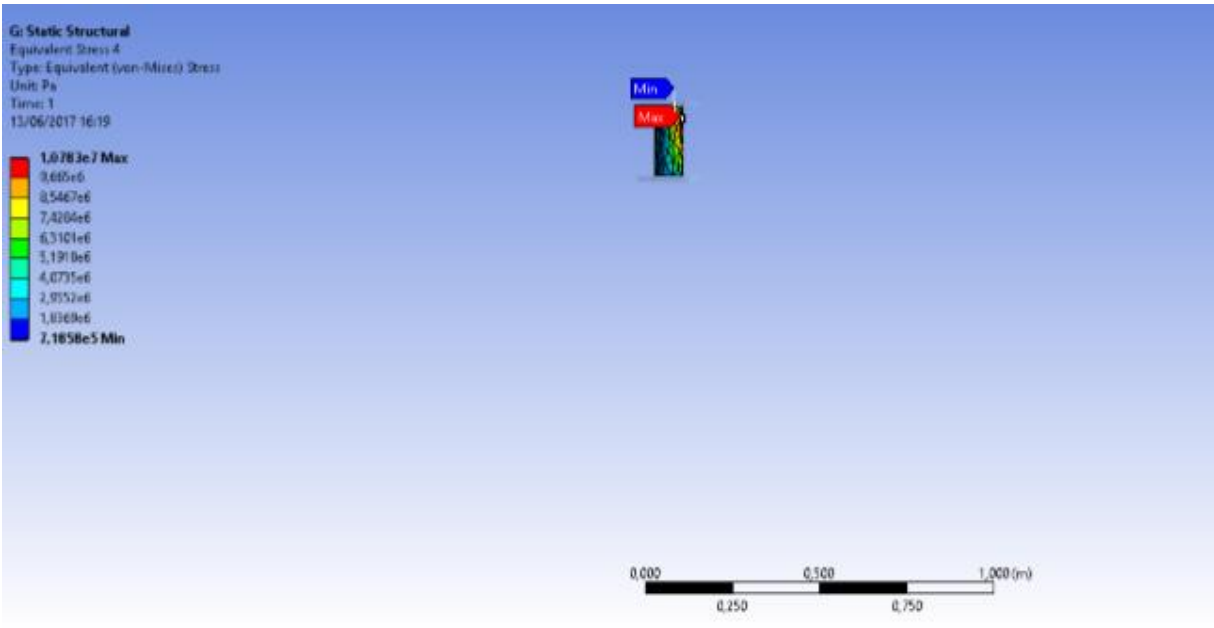


Fig. 135 Columns - Load Case 4 - POM

Superior Hoop

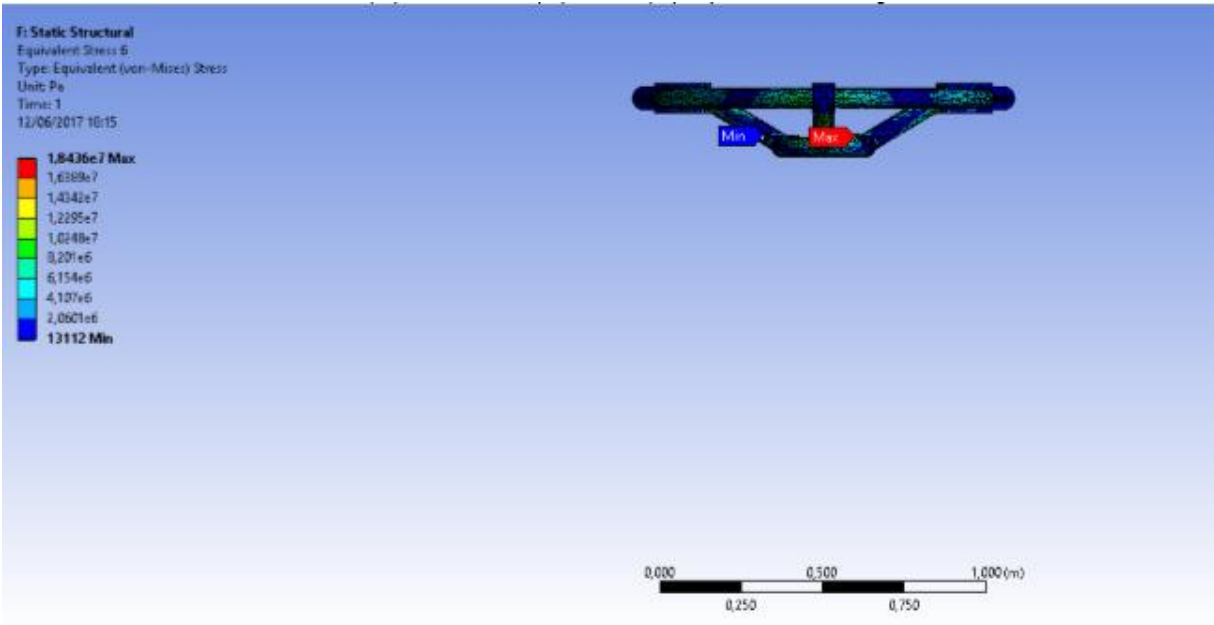


Fig. 136 Superior Hoop - Load Case 1 - Steel

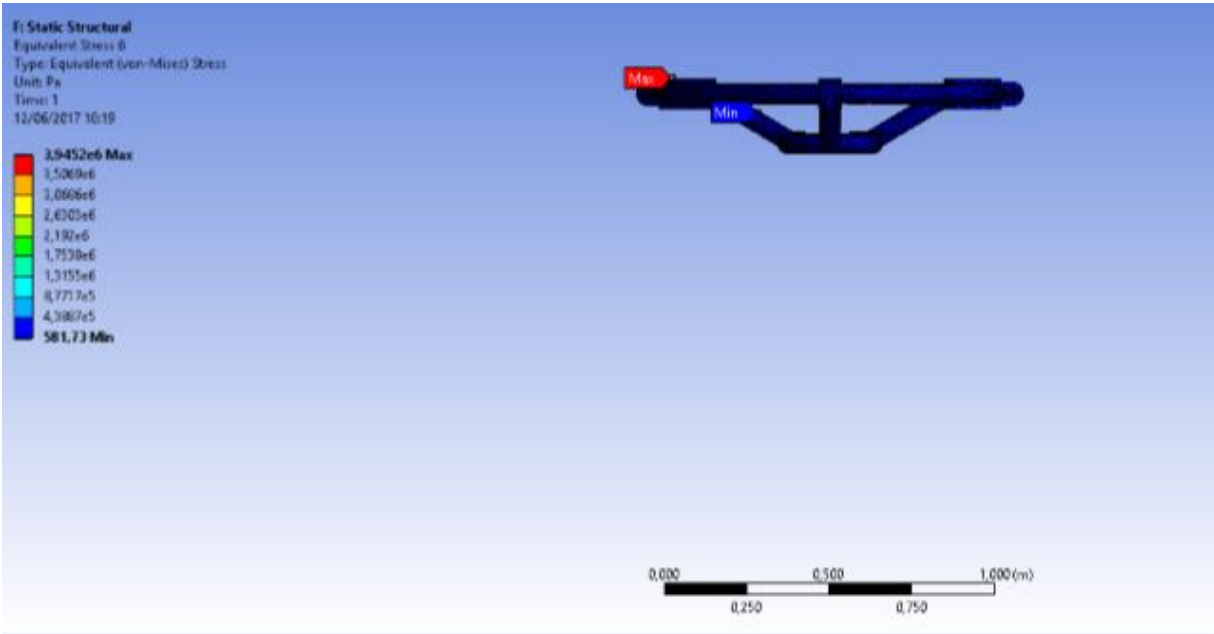


Fig. 137 Superior Hoop - Load Case 2 - Steel

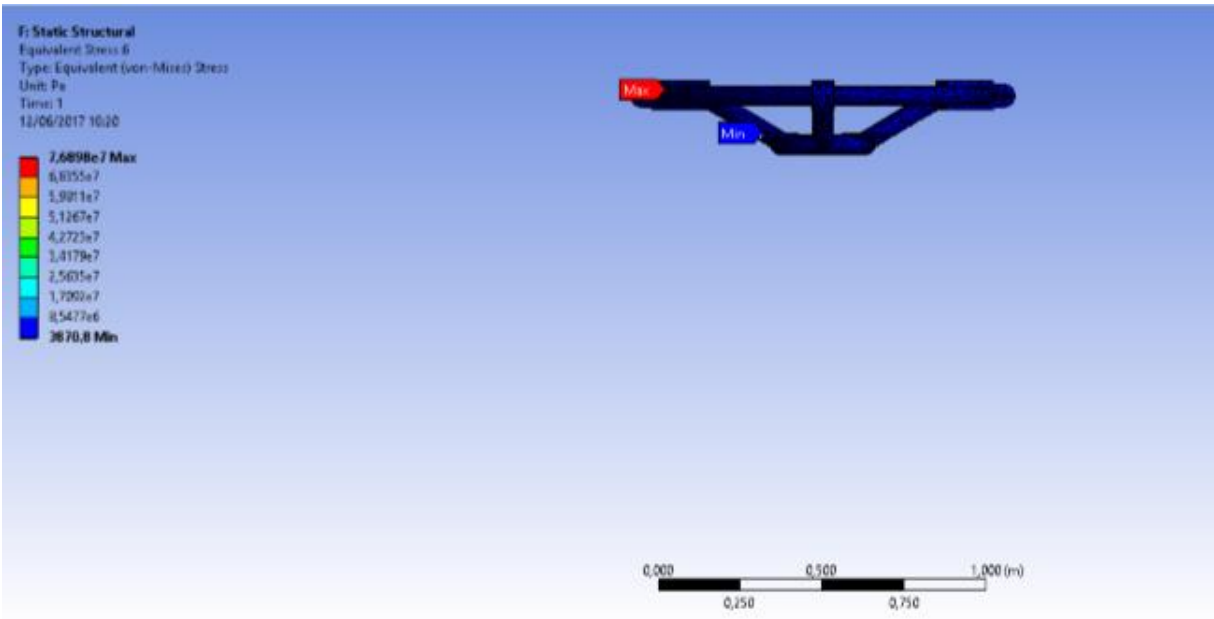


Fig. 138 Superior Hoop - Load Case 3 - Steel

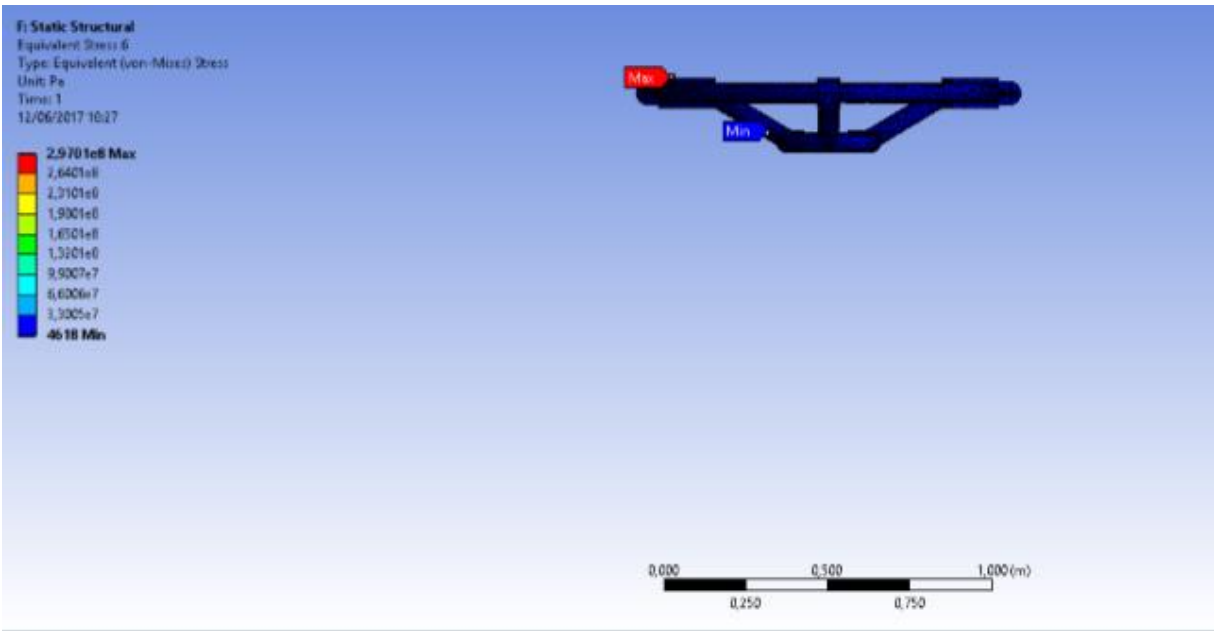


Fig. 139 Superior Hoop - Load Case 4 - Steel

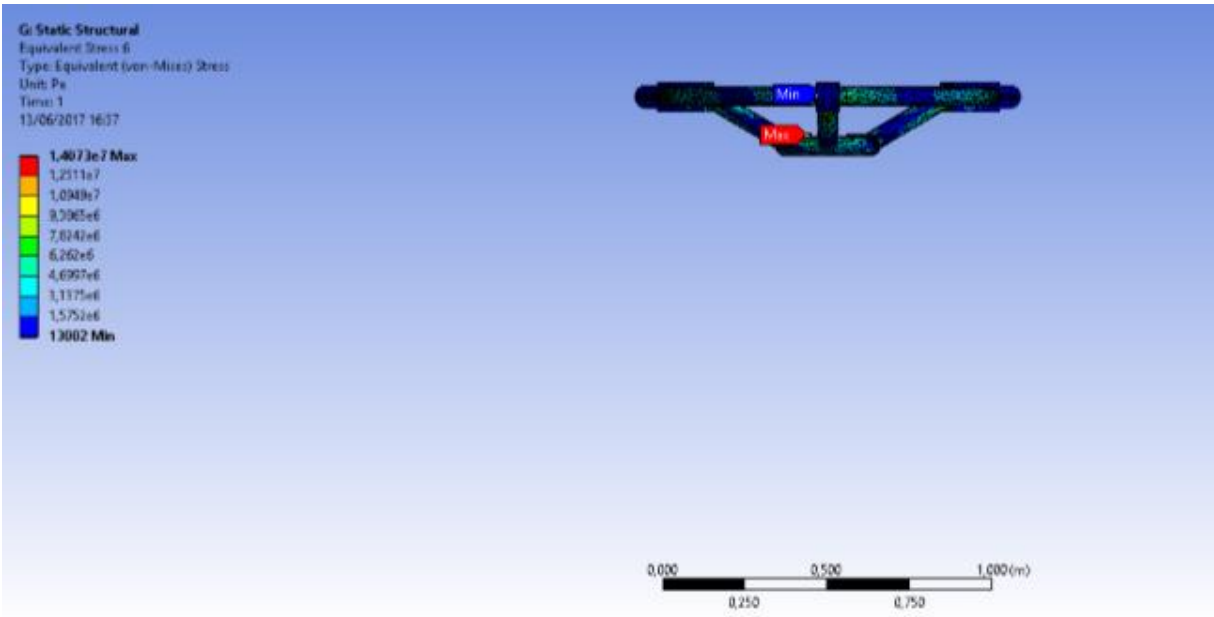


Fig. 140Superior Hoop - Load Case 1 - POM

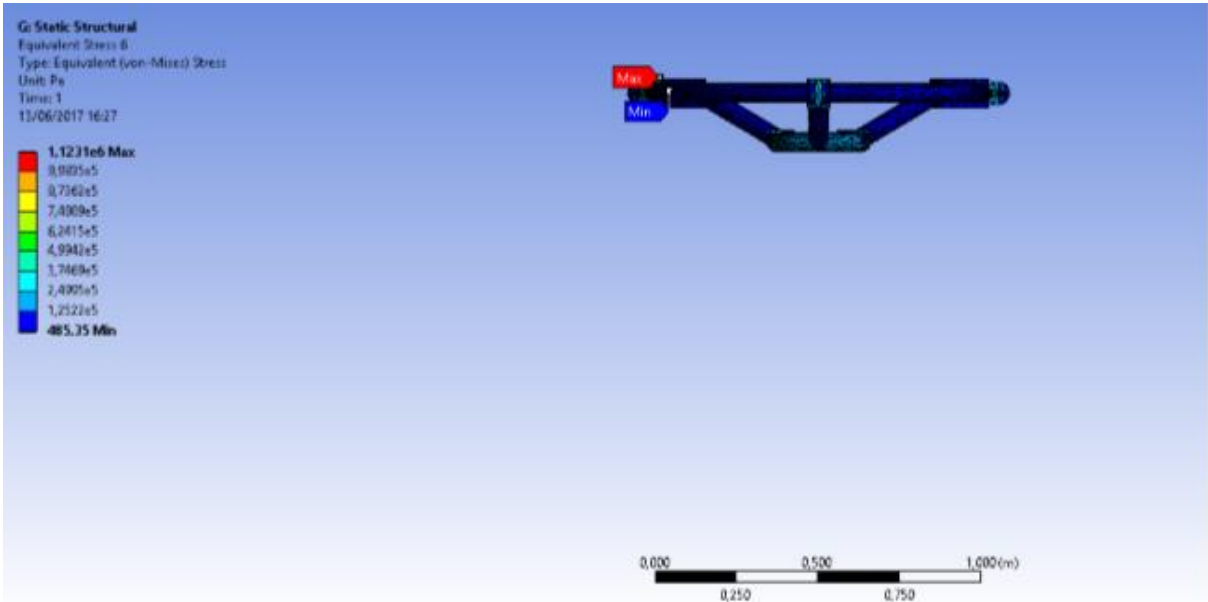


Fig. 141 Superior Hoop - Load Case 2 - POM

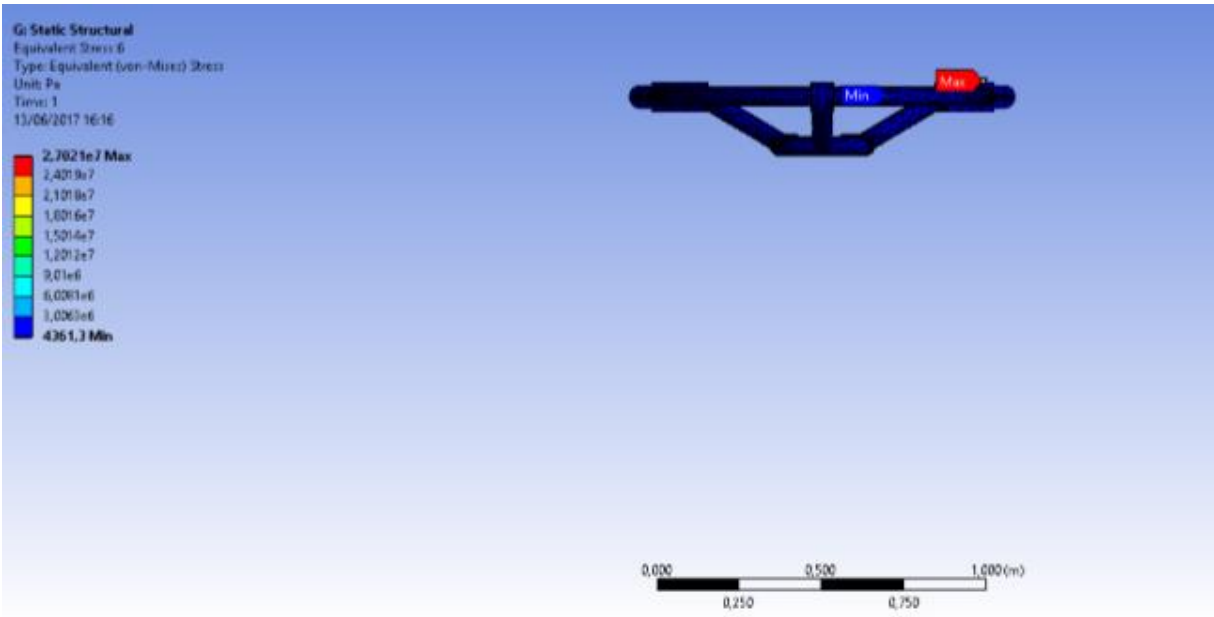


Fig. 142 Superior Hoop - Load Case 3 - POM

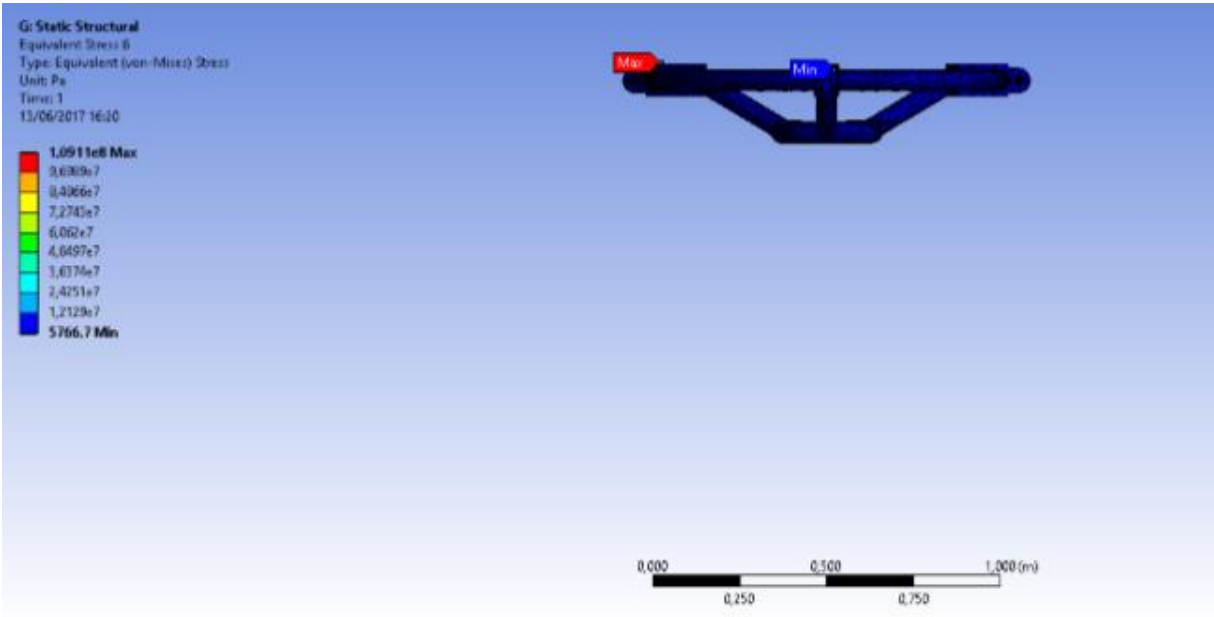


Fig. 143 Superior Hoop - Load Case 4 - POM