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Design and Analyze of a Pressure Vessel for an Underwater Remotely Operated Vehicle Produced by Injection Molding

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for

Stud. Techn. Audun W. Scheide

Design and analyze of a pressure vessel for an underwater remotely operated vehicle produced by injection molding

BlueEye is a company originated from AMOS by Norwegian University of Science and Technology (NTNU). They aim to develop and provide the world's best underwater drone for the global consumer market. Professional underwater vehicles are priced too high to be affordable for the common consumer, and often cost hundreds of thousand dollars. A price limit for the BluEye explorer are NOK 20 000, which would enable private enthusiast, oceanographers, professors and students to invest in equipment to further explore the marine environment.

For the next generation prototype, BluEye want the possibility of mass producing the entire hull, and are looking into a more cost effective production method than machining. Injection molding is a commercially well-known method for plastic mass production, and the main objective of this thesis is to investigate the possibilities of operating an ROV produced by injection molding, down to 100 m depth.

The main objective for this thesis is to investigate the possibility for mass producing the P1 prototype by injection molding. In order to answer this, the following objectives are to be met in this thesis:

- Describe operational and environmental conditions that may influence the material selection and the ROV performance.
- Evaluate polymers that can be used in injection molding and select an appropriate material suitable for the pressure vessel design.
- Identify and present important design aspects connected to injection molding.
- Modify and design the pressure vessel to be suited for injection molding by using SolidWorks.
- Describe and create an illustrative presentation of the work process.
- Analyse the re-designed pressure vessel and evaluate its performance.
- Discuss and conclude on the results of hull performance and material selection.
- Address challenges connected to the P1 design and propose future modifications.

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This project is working towards the next generation prototype. If all analyses indicate that it may be possible to mold a large quantity with sufficient structural integrity, the next phases would be prototyping and testing.

The work scope may prove to be larger than initially anticipated. Subject to approval from the supervisors, topics may be deleted from the list above or reduced in extent.

In the thesis the candidate shall present her personal contribution to the resolution of problems within the scope of the thesis work.

Theories and conclusions should be based on mathematical derivations and/or logic reasoning identifying the various steps in the deduction.

The candidate should utilise the existing possibilities for obtaining relevant literature.

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The thesis should be organised in a rational manner to give a clear exposition of results, assessments, and conclusions. The text should be brief and to the point, with a clear language. Telegraphic language should be avoided.

The thesis shall contain the following elements: A text defining the scope, preface, list of contents, summary, main body of thesis, conclusions with recommendations for further work, list of symbols and acronyms, references and (optional) appendices. All figures, tables and equations shall be numerated.

The supervisors may require that the candidate, in an early stage of the work, presents a written plan for the completion of the work. The plan should include a budget for the use of computer and laboratory resources which will be charged to the department. Overruns shall be reported to the supervisors.

The original contribution of the candidate and material taken from other sources shall be clearly defined. Work from other sources shall be properly referenced using an acknowledged referencing system.

The report shall be submitted in two copies:
- Signed by the candidate
- The text defining the scope included
- In bound volume(s)
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    - The report shall also be submitted in pdf format along with essential input files for computer analysis, spreadsheets, MATLAB files etc in digital format.

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Thesis supervisor:

Professor Martin Ludvigsen

Contact person BluEye, Rune Hansen

Deadline: June 11, 2016

Trondheim, June, 2016

Preface

This thesis present a design and analysis study of the pressure vessel for a remotely operated vehicle (ROV), produced by injection molding. The thesis is a part of my Degree in Master of Science with specialization in Marine Technology at the Norwegian University of Science and Technology (NTNU). The thesis has been written entirely during the spring semester in 2016, and carried out in cooperation with BluEye Robotics.

BluEye Robotics is a newly established company that produce consumer market ROVs. They originated from AMOS by NTNU, and are working close with professors and students at the Department of Marine Technology (IMT). Professor Martin Ludvigsen, my supervisor, presented some challenges and desires for their next generation prototype. This was a very interesting topic, which I believe could be a large contribution to the company.

It is assumed that the reader are familiar with basic hydrodynamics and have some knowledge of material behavior. Also, it is assumed that the reader know the basics behind the finite element method and linear analyses.

Trondheim, 09.06.2016

Audun W. Scheide

Acknowledgment

I would like to thank professor Martin Ludvigsen who has been my excellent supervisor, for invaluable guidance, support and feedback throughout this thesis. I would also like to thank designer Rune Hansen at BlyEye Robotics, one of the creators of the BluEye Explorer P1 prototype, for always being available for discussions and for providing assistance, ideas and suggestions during the design phase. I am also grateful for the assistance provided by professor Bernt Johan Leira, who encouraged to reflective conversations and gave inputs with respect to stiffener solutions, buckling and model behavior.

Thanks to all colleagues and professors at the Department of Marine Technology, especially my colleagues at office A2.011, for making my five years at NTNU a joyful experience. I wish them all the best for the future.

A.W.S

Abstract

BluEye Robotics aims to create a mass produced remotely operated vehicle for the consumer marked, that can be used in all parts of the world. For the next generation prototype they want the possibility of mass producing the entire hull, and are looking into a more cost effective production method than machining. Injection molding is a commercially well-known method for plastic mass production, and the main objective of this thesis is to investigate the possibilities of producing the P1 prototype by injection molding, and operate down to 100 m depth.

The vehicle is produced in two parts, where the front part is pressure resistant and the back part is used for thruster assembly. Designing for the global environment required that the analyses included all relevant material properties that varied with temperature. This required a thorough material selection process and evaluation of the structural capacity of the pressure vessel. Material selection was based on a number of criteria, that was weighted on the importance of the various properties, and with respect to BluEyes interests. Many different polymers were available, with a large range of possible reinforcement combinations, and the selection was done by evaluating the price, water absorption, mechanical properties, density and how well the polymer could be manufactured by injection molding. Polyamide 66 (PA) with 30% glass fibers reinforcement was chosen for the pressure vessel design, mainly due to its combination of mechanical properties, price and suitability for injection molding.

Injection molding is a process where the polymer is heated, before injected into a mold cavity and kept in place over time. The part then cools off, before being ejected from the mold. Designing the pressure vessel for injection molding required care in the dimensioning, as there were several design limitations and requirements that had to be followed for the hull to behave as designed. As the heated polymer are injected into the mold, it requires a smooth flow that fills all the corners and cavitation of the mold. If the ROV exceeded the design limitations, it could result in improper filling of the mold and a vehicle with flaws and reduced strength. In addition, sink holes, warping and deformation of the geometry could occur, if the stiffeners of the vessel were designed with improper dimensions related to the hull thickness. A maximum nominal thickness of 2.92 mm was recommended for the PA 66 30%GF plastic. Restrictions were sat for the filet radius, height and width of the stiffeners and minimum distance between adjacent stiff-

eners. This put a severe limitation on the design of stiffeners for the pressure vessel. Therefore, the model was designed with maximized height and width of the stiffeners, while minimizing the distance between them. After the cooling process the part is pulled from its mold, and all stiffeners should be aligned with the pulling direction to avoid being destroyed in the process. To avoid additional tools for the manufacturing process it was not possible to stiffen all parts of the vessel in two directions.

Both static and buckling analyses were performed to calculate the structural capacity of the vessel. In all analyses the connection between the front and back compartments was assumed to be perfect, to give an indication of the structural performance of the pressure vessel itself. The static analyses indicated that there were large stress concentrations present, and that displacements were too large for the design to operate down to 100 m depths. A hull with a dry PA66 30%GF was able to operate down to 37 m before the material yielded, while in fully conditioned state the hull was only able to resist the pressure down to 24.5 m. PA66 30%GF was four times as strong as the ABS plastic used in the P1 prototype, but the pressure vessel could not be modified for injection molding and stiffened enough to meet the operational requirements. As the maximum nominal thickness put restrictions on the stiffeners, there were no possibilities for adding additional stiffness to the design. The geometry had large stress concentrations, especially in the back compartment, due to the filet radius limitation of 0.25 times the nominal thickness. The front compartment had a large global radius and the stresses were below the yield stress, both in dry and fully conditioned state. This indicated that a larger global radius on the back compartment would distribute the stresses better, and decrease the stress concentrations. Even if the back compartment was designed with the same shape as the front compartment there would still be significant displacements, and buckling analyses indicated that buckling would become a problem between 37 m and 53 m depth, depending on the amount of absorbed water. Additional, internal stiffening devices would reduce the displacements, but the lack of stiffness in the walls orthogonal to the pulling direction would result in buckling before reaching the design depth. It is necessary to invest in additional equipment to manufacture stiffeners in the walls where the buckling occurred, if the present design shall meet the requirements. This will be an benefit versus investment evaluation BluEye have to do, before designing the next prototype.

Sammendrag

BluEye Robotics har som mål å skape en masseprodusert undervannsfarkost for det globale for-brukermarkedet. Pr. 1.06.2016 tester BluEye sin seneste prototype, P1. For neste generasjon pro-totype ønsker de muligheten for å masseprodusere hele skroget, og ønsker å undersøke en mer kostnadseffektiv produksjonsmetode enn maskinering, som er brukt for de nåværende proto-typene. Injeksjonsstøping er en kommersielt velkjent metode for å masseprodusere plastdeler, og formålet med denne avhandlingen er å undersøke mulighetene for å produsere P1 proto-typen ved bruk av injeksjonsstøping. Farkosten skal være i stand til å operere ned til 100 m dyp.

Undervannsfarkosten er konstruert i to deler, hvor den fremre delen er trykk-tett og den bakre delen er brukt for montering av thrustere, og for å skape en hydrodynamisk geometri. Prosessen ved å designe en farkost for det globale markedet krevde at alle materialegenskapene som varierte med temperatur, og andre miljøfaktorer, ble inkludert i analysene. Valg av ma-teriale for bruk på trykkammeret var en omfattende prosess, som baserte seg på en rekke kri-terier. Kriteriene ble vektet med hensyn på de materielle egenskapene til de ulike polymerene, samt hvilke aspekter som var av interesse for BlyEye. Mange ulike polymerer var tilgjengelige, med et stort utvalg av mulige forsterkninger, og materialvalget ble gjort ved å evaluere pris, vannabsorbasjon, mekaniske egenskaper, tetthet og hvor godt egnet polymeren var for injek-sjonsstøping. Polyamid 66 (PA) med 30 % glassfiber-fylling (GF) ble valgt som materiale for neste generasjons prototype, hovedsakelig på grunn av dens kombinasjon av mekaniske egenskaper, pris og egnethet for injeksjonsstøping.

Injeksjonsstøping er en prosess hvor polymeren oppvarmes, før den injiseres i en støpe-form og holdes under trykk over tid. Støpeformen avkjøles før delen blir dratt ut av formen. Prosessen ved å dimensjonere farkosten for injeksjonsstøping krevde nøye dimensjonering, og det var flere krav og anbefalte begrensninger som måtte oppfylles for at fartkosten skulle ha de ønskede egenskapene. Plasten som skal sprøytes inn i formen blir først varmet opp til den er fly-tende, og det var nødvendig å følge anbefalingene for at plasten skal få en jevn, feilfri fylling av formen. Dersom kravene for injeksjonsstøp ikke blir fulgt, kan det resultere i vridninger, synke-merker, luftbobler og redusert styrke av farkosten. I tillegg kan selve geometrien bli deformert og få redusert styrke, dersom avstiverene ikke ble dimensjonert korrekt. Maksimalt nominell

tykkelse på 2,92 mm ble anbefalt for PA 66 30 % GF. Restriksjoner ble satt for avrunding av hjørner, høyde og bredde på stivere , samt minimum avstand mellom stivere. Dette satte en begrensing for hvordan farkosten kunne avstives. Modellen ble modellert med maksimal høyde og tykkelse på stiverene, samt minimal avstand mellom dem. For å unngå å bruke ekstra produksjonsutstyr, ble kun stivere i "tekkretningen" til støpet utformet. Grunnet høy vannabsorbasjon for polymeren ble strukturanalyser utført båre i tørr og mettet tilstand, for å få en indikasjon på om designkriteriene kunne bli oppfyllt.

Både statiske analyser, og analyser av muligheten for knekking, ble utført, for å beregne den strukturelle kapasiteten til farkosten. Det ble antatt en perfekt sammenkobling mellom den fremre og bakre delen av trykkammeret, i alle analyser, for å gi en indikasjon på kapasiteten til selve trykkbeholderen. I de statiske analysene ble store spenningskonsentrasjoner observert, samt store deformasjoner. Resultatene viste at designkravet om å operere ned til 100 m dyp, ikke kunne oppfylles. En farkost med en tørr PA66 30 % GF var i stand til å operere ned til 37 m dyp, før materialet nådde sin flytgrense. I mettet tilstand var fartøyed kun i stand til å motstå trykket ned til 24,5 m. PA66 30 % GF er fire ganger så sterk som ABS-plasten som ble brukt i tidligere prototyper. Det var ikke mulig å modifisere og avstive trykkammeret for injeksjonsstøping, og oppnå de samme strukturelle egenskapene som den foregående prototypen. Siden den maksimale nominelle tykkelsen la begrensinger på stiverene, var det ikke mulig å avstive fartøyet til noe større grad uten å overskride de gitte anbefalingene. Spesielt begrensningene avrunding av hjørner var kritiske, da spenningskonsentrasjonene oppsto i størst grad i hjørnene. Den fremre delen av trykkammeret hadde en stor global radius, og spenningene var under flytgrensen i dette området. Dette indikerte at en større global radius på den bakre delen ville fordelt spenningene i større grad. Selv om det bakre rommet blir utformet med samme form som det fremre, vil det fortsatt være betydelige utbøyninger, og knekkingsanalysene indikerte at knekking kan bli et problem på mellom 37 m og 53 m dyp, avhengig av mengden av absorbert vann. Ekstra, indre avstivningsmekanismer vil redusere utbøyningene, men mangelen på stivhet i veggene vinkelrett på trekkretningen vil føre til knekking før farkosten når design-dybden på 100 m. Det er derfor nødvendig å investere i ekstra utstyr for å produsere stivere i veggene, der knekking forekommer, hvis den nåværende utformingen skal kunne oppfylle designkravene. Dette vil være en investeringsevaluering BluEye må gjøre, før de utformee neste generasjons prototype.

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Chapter 1

Introduction

Chapter one provides an introduction to the objectives of designing and analyzing a pressure vessel produced by injection molding, project background and literature survey, main objectives and limitations. The approach of reaching the thesis objectives are covered, before briefly summarizing how the thesis is structured.

1.1 Introduction

Remotely operated vehicles (ROVs) exist in a broad size and price range. Underwater robotics are extensively used for commercial and defense operations, but are now experiencing increasing interest from the private consumer marked. A consumer marked vehicle is easy and intuitive to maneuver, and can be carried and handled by one or two operators without any additional equipment. Continuous improvements and simplifications are done to adapt the changing needs of the global marked, and the potential in the marked lies in transmitting high quality video from the underwater vehicle, and send back in high-definition real time.

BlueEye is a company originated from AMOS by Norwegian University of Science and Technology (NTNU). They aim to develop and provide the world's best underwater drone for the global consumer market. Professional underwater vehicles are priced too high to be affordable for the common consumer, and often cost hundreds of thousand dollars. A price limit for the BluEye explorer is NOK 20 000, which would enable private enthusiast, oceanographers, professors and students to invest in equipment to further explore the marine environment.

Blueye aims to make underwater exploration possible for everyone, with user friendly and supreme underwater drones that let you discover and learn about the world hidden below the surface. We are passionate about the ocean, and want our drones to enable users worldwide to explore the oceans and waters around. This way we will build an ecosystem of explorers and "citizen scientists".

— BluEye Robotics, 2016

Many of the current underwater vehicles for the consumer marked are only able to operate between 20 m and 50 m depth, and BluEye are extending the operational depth to 100 m. In addition, with high resolution cameras installed, the vehicle could to be used to replace humans in some inspection-related tasks. There is an increasing focus on the ocean environment, and it is predicted an increasing marked potential for consumer marked ROVs. With increasing demand and increasing technological advances within subsea robotics, the ROV becomes more and more important in the journey to explore the ocean space.

1.2   Background

In the development process of creating the BlyEye Explorer ROV a number of different prototypes have been manufactured and tested. The latest version of the prototype, BluEye P1, are manufactured by machining and have been delivering promising testing results. Machining is not a production methodology suited for mass production, and in order to manufacture a large quantity the production costs have to be reduced. Injection molding is a well known and efficient technique to manufacture a large quantity of plastic parts, and for the next generation vehicle the goal is to mass produce the entire vehicle by using injection molding. Manufacturing prototypes have proved to be too expensive, and using the injection molding technique would strongly contribute to reducing the total vehicle cost below the NOK 20 000 limit.

Design regulations and proper dimensioning of plastic parts produced by injection molding are discussed thoroughly in Rosato (2000) and Zhou (2013). Restrictions and limitations connected to the molding technique should be applied to the P1 prototype, and the entire hull requires modification to be prepared for computer-aided analyses (Rosato and Rosato (2003)).

ABS plastic has been used for the P1 prototype, and require a 15 mm wall thickness to be pressure resistant at 100 m depth. Injection molding has restrictions for the nominal thickness, depending on the polymer used, and the maximum thickness will be lower than the current 15 mm. It would therefore be necessary to select a material suited for injection molding, with appropriate characteristics, and re-design the pressure vessel to be both strong enough and possible to manufacture without flaws.

Morgan (2005) provided an overview over typical properties for unreinforced polymers, and Hough (1998) discussed and compared a number of polymers, with respect to production methodology and properties. Integration of theory and modeling methods are discussed in Rong Zheng and Fan (2011). Kazmer (2007) further discuss how to best design and produce identical molded parts for the cyclic injection molding process. The final material selection, and results from analysis performed in this thesis, creates a fundament for the next generation prototype. It would give an indication of whether the current prototype could be modified to be produced by injection molding. If the design proves not to be strong enough, this thesis will address challenges and possible solutions for future prototype designs.

1.3 Objectives

The main objective for this thesis is to investigate the possibility for mass producing the P1 prototype by injection molding and operate the ROV down to 100 m depth. In order to answer this, the following objectives are to be met in this thesis:

1. Describe operational and environmental conditions that may influence the material selection and the ROV performance.

2. Evaluate polymers that can be used in injection molding and select an appropriate material suitable for the pressure vessel design.

3. Identify and present important design aspects connected to injection molding.

4. Modify and design the pressure vessel to be suited for injection molding by using Solid-Works.

5. Describe and create an illustrative presentation of the work process.

6. Analyze the re-designed pressure vessel and evaluate its performance.

7. Discuss and conclude on the results of hull performance and material selection.

8. Address challenges connected to the P1 design and propose future modifications.

1.4 Limitations

This thesis was carried out in cooperation with BluEye, who gave continuous inputs to the work. During the process some objective changed to cover the most interesting topics for BluEye, which created some delays in the work process. Since a significant part of this thesis was to re-design the pressure vessel, a large amount of time was spent learning how to use SolidWorks and the theory and processes behind each analysis. By selecting a new material a complete re-design of the internal arrangement was required, and led to creation of multiple models. Also, the SolidWorks Education license did not support non-linear elastic material models when performing non-linear analyses, which was undesirable when analyzing the vessel performance.

1.5 Approach

The work process towards selecting the most appropriate material and design solution was based on the desired outcome of the thesis. Selection criteria was established, based on inputs from BluEye and by comparing weighted criteria connected to the end-use experience. Injection molding constraints for plastic parts was thoroughly evaluated, and design and analyses were performed as an iteration process, between stiffening solutions and the corresponding results.

The traditional IMRAD organizational structure was used as research approach in this thesis, which include an introduction, method, result and discussion. Since this thesis have multiple topics under the "method" section of the IMRAD approach, it was distributed into phases. The phases present relevant topics that had to be done prior to the analyses. All phases where depended on the previous, and the chronological framework was as listed below:

1. Material selection

2. Establishing the theoretical background for injection molding designs.

3. Design and modeling of the pressure hull in SolidWorks.

4. Perform finite element analyses of the modified geometries.

The first phase was done in cooperation with designer Rune Hansen in BluEye Robotics, who gave inputs based on previous experience and research done in this thesis. Relevant design constraints were evaluated with basis in the given constraints, before establishing the theoretical background of injection molding, parallel to selecting the stiffening solutions. Professor Leira gave inputs on design configurations and stiffening of the structure, in the design and modeling phase of the thesis. The detailed design and analysis was performed using Computer Aided Design (CAD) tools available at the Norwegian University of Science and Technology (NTNU). SolidWorks was used due to its simplicity and good integration with other analysis software. BluEye is also using SolidWorks in their design, which simplified sharing of data.

1.6   Structure of the Report

The rest of the report is organized as follows; Chapter 2 gives an introduction to the requirements and constraints in the project. This includes operational, geometrical and production constraint, and the chapter sets the basis for the design limitations. Environmental and operational conditions that may have an influence on the material selection is also briefly covered.

Further, Chapter 3 covers the material selection process. Here different polymeric materials are discussed along with the benefit of using various fillers and reinforcements. Selection criteria are explained, and specific polymers are evaluated with respect on the given constraints. A decision gate towards the material selection is explained, and the chosen materials are described.

In Chapter 4 the concept of the BluEye Explorer is presented. How to design for injection molding is covered, and the work process is illustrated. Further some model simplifications are

explained, along with some structural design aspects used in the modeling. Also a brief overview of some applicable plastic joining techniques are included.

A short description of the theory behind the analyses is covered in Chapter 5, along with mesh convergence tests, element selection, applied boundary conditions and load cases.

Chapter 6 presents the results and address problems related to static pressure, buckling and connection methods. In Chapter 7 the thesis is summarized and results discussed. Also, challenges with the current design are addressed along with some recommendations for future work.

Chapter 2

Constraints and Requirements

2.1 Operating Environment

The main target groups for the BluEye Explorer are researchers, oceanographers, private enthusiasts and schools located all over the world. Mass producing a ROV for use in such diverse geographical areas require a product that are capable of performing in accordance to its promised characteristics, in a vast range of external conditions. Different geographical locations will have distinct variations for both the temperature and salinity content. Local conditions, as currents, wind and waves, will be less predictable to take into consideration when designing the vehicle. Mixing due to differences in heat and salinity content will give local forces on the vehicle, and are also difficult to include in the design phase. Some conditions will have influence on the operational performance of the vehicle. Forces connected to surface waves will have a different impact on the ROV, when operating in deep compared to shallow water (Faltinsen (1990)). In the vehicle user manual there will be notes with respect to the performance under various currents and sea states, but these aspects are not included as additional loads in the structural analyses of this thesis.

A large variety of people will handle the vehicle and it is expected that many adverse situations will occur. Operators will handle the vehicle differently, both on land and in water. Improper storage may degrade the material, while dropping and improper launch and recovery of the vehicle could crack the hull and make it weaker. When operating the vehicle subsea collisions and seabed interactions may occur, and reduce the structural capacity and total vehicle

performance. These aspects of the operating environment are very individual between users. When receiving feedback from the current prototype, these aspects could be estimated and included in the analyses.

Temperature variations is an important aspect to include when selecting material, and calculating the structural performance of the vehicle. Many plastics have large variations in their performance and properties, as the temperature varies. Extremal temperatures may vary between -2°C and 35°C, depending on where in the world the vehicle is operated (NASA (2016)). As the material behavior changes with the temperature it will be necessary to evaluate all possible variations of the environmental conditions when selecting a material.

BluEye Explorer should be able to operate down to 100 m depth. At such depths the vehicle require a well designed body to minimize the drag and thrust enough to operate as desired. Vehicle thrust is covered in a separate master thesis. General loads of interest for a submerged vehicle are:

1. Gravitational loads - forces due to the ROV weight

2. Hydrostatic loads - resultant of hydrostatic pressure

3. Thrust loads - change in angular momentum produced by the thrusters

4. Inertia loads - resistance to changes in accelerations

5. Contact loads - forces due to waves-wind-current and accidental loads

Gravitational loads will change throughout this project, as the vehicle is being re-designed. BluEye is working parallel with this project, to modify and reduce the internal equipment, and specified that the equipment should be left out of all analyses. Analyses in this project are limited to calculate if the design is pressure resistant down to 10 bar pressure. The results should be used to indicate whether the pressure vessel is capable to withstand the hydrostatic pressure forces at 100 m depth. Additional loads, as contact loads and inertia loads, will be included in future analyses, when the vehicle is proved to be pressure resistant and is being evaluated for prototype production.

2.2 Geometric Requirements and Maneuverability

The vessel geometry is shaped to have four thrusters installed. One thruster will go vertically through the entire hull, to provide trust in the heave direction. A lateral thruster go through the middle section of the vehicle, to ensure thrust in sway direction. The pressure vessel is formed around the lateral thruster, and in the modification process the external geometry of the pressure vessel is restrained from modification. As this study is performed to investigate if the current prototype could be modified and manufactured, BluEye did not want the layout of the vessel to change. Two thrusters will be installed in the back to create surge and yaw motion. Axis definition, and a figure of the BluEye P1 prototype, can be seen in Figure 2.1.
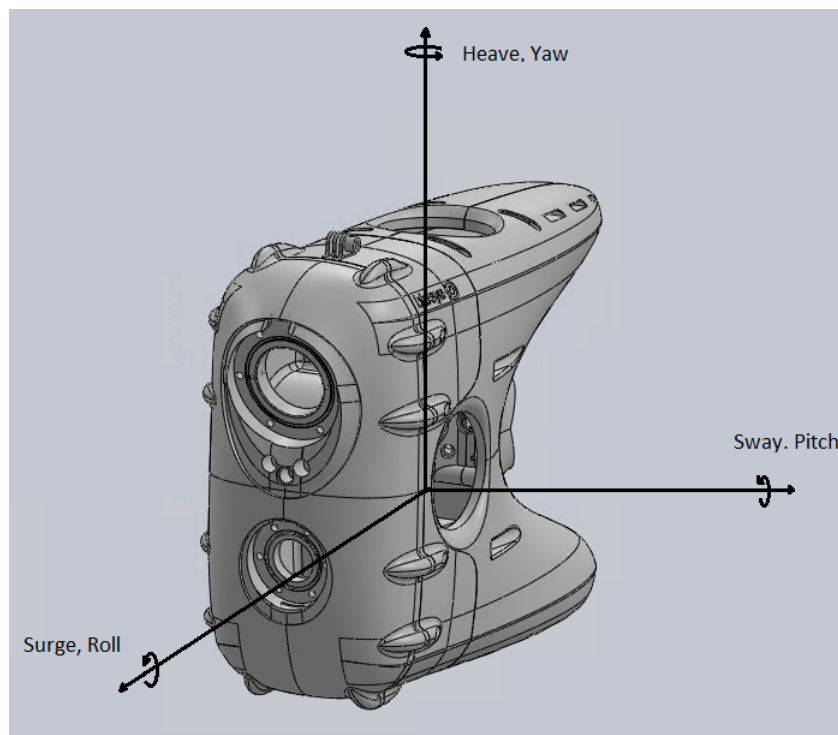
Figure 2.1: BluEye Explorer P1 prototype with axis definitions

BluEye's current P1 prototype design was used as a basis in the entire design and modification phase of this thesis. P1 has a 15 mm thick ABS (acrylonitrile butadiene styrene) hull and it was desired that the total volume of the vehicle remained unchanged. In order to produce the vehicle by injection molding the thickness had to be reduced, depending on the selected material, while the external dimensions remained the same. Interior layout would be fitted after the geometry is fully stiffened and the vehicle made ready for prototyping. It is necessary to have

the gravitational center a distance below the buoyancy center in order to ensure a stable equilibrium position. Position of the gravity center is calculated when installing the inner equipment and payload, and depend on the shape of the hull, weight of equipment and placement of buoyancy elements. Inner equipment will not set any limitations on the pressure hull modification process.

2.3 Production and Molding Process

Investigating the possibilities for producing the vehicle by injection molding requires a carefully considered design. When designing the vehicle, a number of requirements and limitations were given with respect to thickness and design configuration. A visualization and brief explanation of the production process are shown in Figure 2.2. Granulates of plastic are inserted into the feed hopper, before forced through the feed one by an extrusion screw. The granulates are then heated and forced under pressure. Melted plastic are injected with high pressure into the die cavity. The cavity then cools off and it is possible to extrude a rigid part. A full description of the process are not covered in this thesis and it is assumed that a production engineer will control the productional considerations when prototyping becomes relevant. A detailed description of the complete injection molding process can be found in Rong Zheng and Fan (2011). Recommendations for the mold design and manufacturing setting are also covered, along with some challenges that may occur.

Restraints and tolerances for the plastic part are dependent on a number of parameters, and will vary with the selected polymeric resin group. Process variations will also have an impact on the end product, and to avoid design errors, that may influence the structural capacity of the vehicle, it was important to have control of the maximum and minimum constraints for all parts of the design.

Construction and design of the mold itself, machine capabilities and process variables were not considered in the design phase. These considerations are done by the process engineer and tooling engineer in a later stage of the project, when prototyping become relevant. To ease the molding process, and make it efficient, the design should simplify the filling operation and avoid
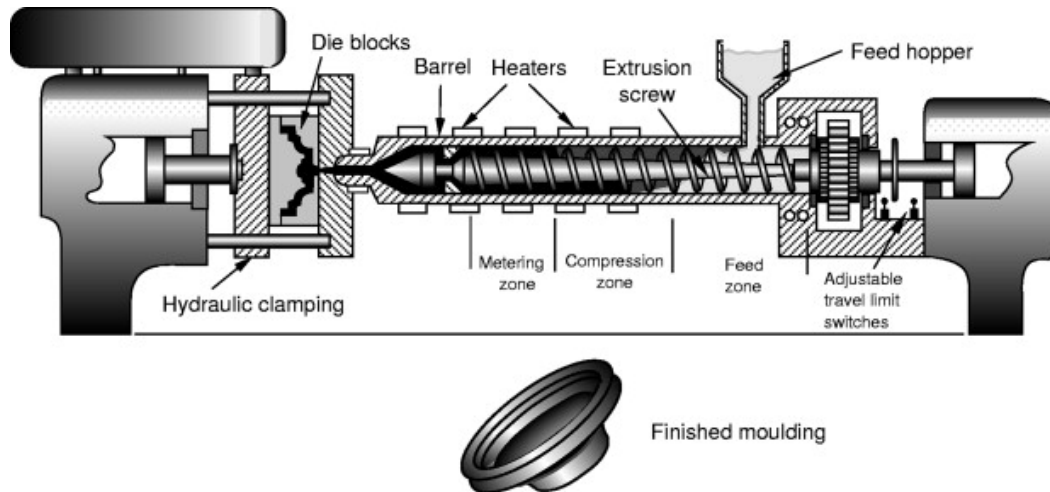
Figure 2.2: Injection Molding Process layout (Rong Zheng and Fan (2011))

production flaws. According to Zhou (2013), common areas where design errors usually occur are;

- Parts too thick or thin to be molded

- Bad transition designs

- Interference fits and hinges

- Slender bails and handles

- Draft angles

- Sharp corners

- Rib design and thread inserts

Areas with common design errors were handled with additional care, and conservative limitations were used to reduce the possibilities of a faulty design. The recommended maximum and minimum nominal thickness will vary based on the selected material. Typical thicknesses for injection molded parts are between 0.4 mm and 13 mm (Swift and Booker (2013)). Based of the part thickness and dimensions, the design was based upon the tolerances in Figure 2.3. This shows that the pressure vessel could, for most polymers, be produced with relatively low tolerances.
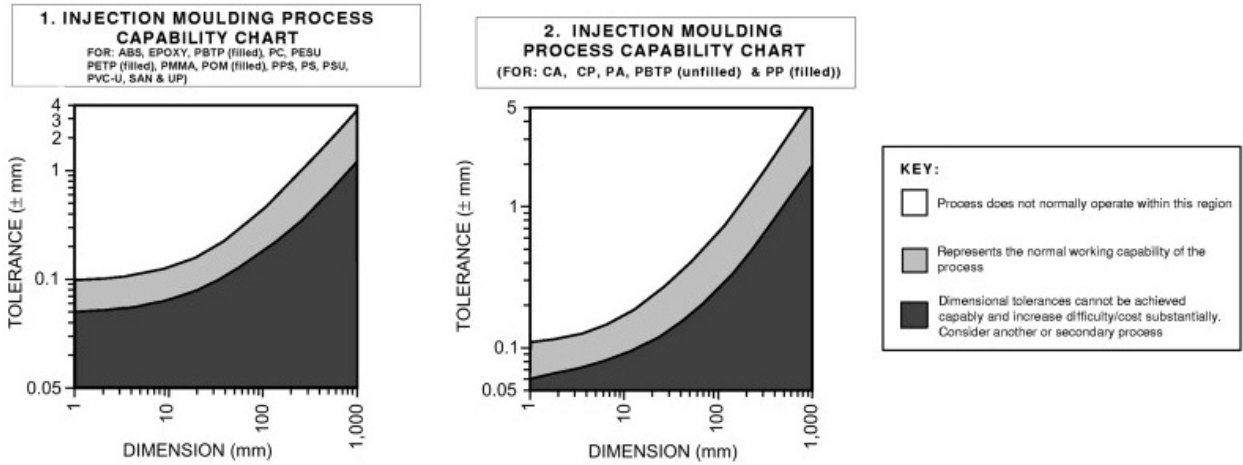
Figure 2.3: Injection molding process capability chart for a selection of polymers (Swift and Booker (2013))

The design aspects of the pressure vessel include complex shapes and intricate details. Having an uniform wall thickness is generally preferred in injection molding, and transitions between thin and thick walls should be avoided. If unavoidable, the transition should be as gradual as possible. In addition, corners should have constant thickness. The inside radius should not be lower than half the thickness of the primary wall. Outside radius is then required to be equal the inside radius, plus the wall thickness.

Table 2.1: Basic design limitations for injection molded parts

Wall thickness [mm]	Dependent on material and process
	·Uniform thickness
	·Maximum 3:1 thickness transitions [ratio]
Draft Angles	·1/2° for unreinforced materials
	·1.5° for glass-reinforced materials
	·Use ribs and bosses to reduce ejection pressure
Corner radius [mm]	·0.25 x thickness for inner radius
	·1.25 x thickness for outer radius
	·Use genereous radii when possible
Stiffeners	·Thickness = 0.5 - 0.75 x nominal thickness
	·Height < 3 x nominal thickness
	·Distance between projections
	minimum 2.5 x nominal thickness

Conservative values were used for the part design limitations, and Table 2.1 list some of the minimum requirements. More details connected to the specific design principles and limitations are discussed in Section 4.3.

Chapter 3

Material Selection

Polymers and plastics are widely used materials for injection molding, and provides properties that are favorable for a light weight pressure vessel. Plastic is a term used for describing the compound of a polymer with one or more additives. Such additives are used to enhance the performance of the polymer, and achieve desirable material properties. The properties of plastic is what makes them economically favorable compared to other materials. It is possible to assess a wide range of low price polymers, that are easy to process by injection molding, with high strength, low density and high chemical resistance (Zhou (2013)).

3.1   Available Plastics for Injection Molding

Plastics may be classified in different ways, depending on the criteria used. Morphology is often used as a way to describe polymers, and refers to the shape and arrangement of the molecule structure. Morphology indicate the molecular dimensions and the proportion between amorphous and crystalline phases, in addition to the amount of fillers present. Mechanical properties are changing with the degree of crystallinities present in the material, as crystalline structures generally are stiffer than the random oriented amorphous structure. Table 3.1 shows a comparison between the basic performance of crystalline and amorphous plastics.

Generally, amorphous materials are easier to process than the crystalline. Crystalline materials have a dense and ordered structure, with the molecules closely aligned. When heating a crystalline polymer the structure remain solid until reaching their sharp melting point, then the

Table 3.1: Comparison of crystalline and amorphous plastics (Zhou (2013))

Amorphous	Crystalline
Low density	High density
Low tensile strength and modulus	High tensile strength and modulus
Low dhrinkage	High shrinkage
High ductility and elongation	Low ductility and elongation
Low viscosity	High viscosity
Broad softening range	Sharp melting point
Low chemical resistance	High chemical resistance

material becomes a flowing liquid substance. Such crystalline polymers undergo larger volumetric changes when heated and formed, and require more precise control during fabrication. Perfect crystalline materials are not produced for commercial use, but many plastics are semi-crystalline. Such plastics act as a composite of crystalline and amorphous polymers, and the amount of each phase characterize the overall properties of the composite structure. Amorphous structures have a broad softening range and do not actually melt. The material is rigid, before becoming a stiff flowing substance that eases up as the temperature increases. Most plastics, independent on form and properties, fall into one of two groups; thermoplastics and thermosets. These two groups will be briefly explained in Section 3.1.1 and Section 3.1.2

Most polymers may be used in injection molding, which include both thermoplastics and thermosets, and to some extent elastomers. Elastomers are allowed to undergo large deformations, and will not be evaluated for use for the pressure vessel. According to Biron (2013b), at least 80 wt% (weight percentage) of the injection molding plastics are thermoplastics, and most literature are directed towards thermoplastics. The range of available materials increases yearly, and there are now over 18000 different types available (Kauffer (2011)). Due to the vast amount of different available polymeric compounds, it is important to include all requirements and constraints when evaluating possible materials for use for the pressure vessel. The availability of a material is closely connected to its price, which is a crucial factor in the material selection. Molding parameters depend on the selected material, and decisions are based on the end-use performance of the pressure vessel.

3.1.1 Thermoplastics (Engineering Plastics)

Thermoplastics are polymers that only have secondary links between their molecule chains. Engineering plastics are often used as a loose description of thermoplastics that are suitable for engineering processes and holds adequate mechanical properties (Crawford (1998)). Engineering polymers may go through repeated cycles of melting and solidification, and are in some cases also referred to as linear polymers. The term linear is connected to the molecule structure and must not be confused with the mechanical properties.

Thermoplastics are sensitive to temperature and will change its properties with varying temperatures. When heated above the melting temperature the material becomes soft and moldable. A thermoplastic that may be used as a substitute for a metal, such as aluminum, at a given temperature, may be an unsatisfying substitute at a different temperature. The term engineering plastics are governing the thermoplastics that are able to support loads more or less indefinitely. Generally these engineering materials are easy to process, have low density and are resistant to many corrosive liquids. Disadvantages is inferior strength compared to metals, and low time-dependent moduli (Crawford (1998)). Thermoplastic polymers may be either amorphous or crystalline, but are rarely above 50 % crystalline. Amorphous polymers have random oriented molecular chains and are often less dense than crystalline. Crystalline thermoplastics have closer packed chain molecules, which enhances the corrosion resistance and the resistance to environmental stress, cracking, hardness, friction and wear (Brinson and Brinson (2015)). A list of common thermoplastics used in engineering design are listed in table 3.2

Table 3.2: Thermoplastics used in engineering design (Brinson and Brinson (2015)).

Amorphous	Crystalline
Polyvinyl Cloride (PVC)	Polypropylene (PE)
Polystyrene (PS)	Acetals (POM)[*]
Polycarbonate (PC)[*]	Polyesters (PETP, PBTP)[*]
Acrylic (PMMA)	Polyamides (PA / nylon)[*]
Acrylonitrile-butadiene-styrene (ABS)[*]	Polyterafluoroethylene (PTFE)
Polyethersulphone (PPO)[*]	Fluorcarbons (PFA, FEP and ETFE)[*]

[*] Materials regarded as engineering plastics

3.1.2 Thermosets

Thermosetting polymers are often used when high strength and modulus are important in addition to high thermal and dimensional stability. Thermosets have primary bonds in addition to secondary bindings, and are often identified as cross-linked polymers. These cross-linked molecular chains adds strength and stops creep in the material, but also prevents the polymer from being reshaped once it has been molded or processed. The resin blend is typically liquid at room temperature and requires a hardener to create solidification. This solidification process is irreversible, in contrast to the thermoplastic. Thermosetting materials have high thermal and chemical resistance, surface hardness, stiffness and dimensional stability (Crawford (1998)). These properties make thermosets preferred for many technical applications, and in many cases thermoplastics cannot match the properties of thermosets. Common thermosets used in engineering applications are:

- Aminos

- Polyurethanes

- Epoxides

- Phenolics (Bakelite)

- Polyesters

3.1.3 Additives, Fillers and Reinforcements

A material made from two or more constituent materials is called a composite material. Pure polymers may not have the desired characteristics for use in the pressure vessel. It is possible to modify the material properties of a polymer by adding additives, or by blending multiple polymers in an alloy. Additives are modifying the properties of the base polymer, and the filling degree determines to what extent. Fibre composites are the most attractive for engineering applications, and both thermoplastics and thermosets get enhanced properties through fibre reinforcements. Strong, stiff fibers are bonded together in the polymeric matrix and creates a

strong and lightweight material. Normally these particles are not covalently bonded to the matrix phase, but form a secondary phase. Reinforced materials offers good combinations of stiffness and strength, and are less expensive compared to traditional materials with comparable properties.

Alloying, or using fillers, in plastics are done to achieve the advantages of several polymers in one material. Their features are combined and result in a material with different characteristics from the individual base materials. There are certain building blocks in polymeric alloys, and Table 3.3 shows some typical fillers and additives, along with their benefits and disadvantages. The blend ratio is adjustable to achieve a specific ratio of properties, and it is possible to combine reinforcements.

Table 3.3: Fillers and additives typically used for engineering applications (Crawford (1998), Wellmann (2009)).

Additive or Filler	Advantage	Disadvantage
Glass Fiber	·Increase strength, stiffness and dimensional stability ·Reduce shrinking time	·Decrease true toughness and flexibility. · Increase density and can cause warpage
Glass Beads	·Increase compressive strength, stiffness, and dimensional stability ·Reduce shrinking, warpage and cycle time	·Decrease toughness and flexibility ·Increase brittleness
Metallics	·Increase thermal and electrical conductivity	·Reduce toughness and flexibility. ·Increase density and cost
Impact Modifiers	·Increase impact resistance, flexibility and toughness	·Decrease stiffness and tensile strength ·Increase melt viscosity
Minerals	·Increase stiffness and dimensional stability Enhances the surface finish ·Reduce cost, warpage and shrink marks	·Decrease toughness and flexibility Rougher surface

Glass fibers are currently the most widely used additive, along with carbon fibers and aramid. Glass fibers account for nearly 95 % of the reinforcements, while carbon and aramid for the remaining 5 % (Biron (2013a)). Figure 3.1 shows how the typical increase in modulus of elasticity,
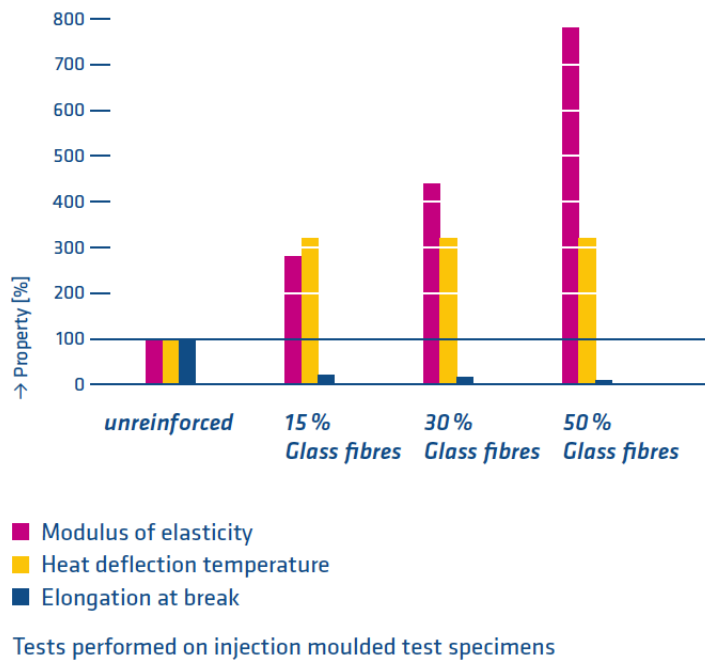
Figure 3.1: Glass fiber reinforcements for an injection molded test specimen (Ensinger (2012))

elongation at break and heat deflection temperature is, for an injection molded specimen with glass fiber reinforcements. Typically, glass fiber reinforcements will have a huge impact on the material strength, compared to elongation at break, and will increase the heat deflection temperature.

Carbon fibers provide similar effects as glass fibers, but give a better weight-to-strength ratio. However, carbon fibers are more expensive than glass fibers and will not be evaluated as an primary option. In evaluation of reinforcements, glass fiber is selected before evaluating the need for carbon fiber reinforcements. The main reason for this is economical considerations, and carbon fiber reinforcements becomes relevant if there is need for a stronger material without increasing the material weight. An example of how different types and amounts of reinforcements change the mechanical properties of polypropylene (PP) can be seen in Table 3.4. Such reinforcements can be done for most polymers, but with different outcomes. There are also upper limits of possible reinforcements, especially with respect to production considerations.

Table 3.4: Examples of reinforced mechanical properties of PP (Crawford (1998)).

	Reinforcements	Average tensile strength		Average impact strength	
		[MPa]	[% variation][*]	[J/m]	[% variation][*]
PP Homopolymer	None	30	-	40	-
PP 10-20 % GF[**]	Fibers	45	+50	97	+142
PP 30-40 % GF[**]	Fibers	56	+87	102	+155
PP 10-40 % Mineral	Spherical particles	21	-30	74	+85
PP 10-40 % talc	Platelets	24	-20	115	+187
PP high impact	Dispersed polymer	30	0	554	+1285

[*] Variation in % relative to homopolymer PP
[**] Glass fiber

3.2   Selection Criteria

Selecting the best material for injection molding required good knowledge of the design limitations of injection molded parts, and how materials and their properties varies with the operational environment. This was essential for knowing how the functionality of the end product would be. Standardized methods are developed to ensure that tests from different manufacturers are comparable. These tests assume that the production are done under best possible conditions (Biron (2013b)). Firstly, the polymeric resin group was selected, based on production characteristics and polymeric behavior. After this initial selection, the polymers were evaluated for use under different conditions to address deviations and the need for additional fillers or reinforcements. This section will discuss the meaning of some physical properties and how to evaluate each selection criteria.

3.2.1   Thermal Behavior

Thermal behavior of polymers characterize how the properties change with varying temperature. As the temperature rise the material tends to soften and the tensile stress and elasticity modulus will decay. It was necessary that the selected polymer was able to withstand all end-use temperatures around the world. This means that the material could be subjected to cold water and hot storage, within a relative short time span. Thermal expansion of the hull could lead to irreversible shrinkage or geometrical discrepancies. Coefficient of linear thermal expansion

(CLTE) defines the ratio of dimensional change per degree Celsius, or Kelvin (ASTM D696). The chemical structure in plastics leads to a higher CLTE than e.g. metals. It is especially important to be aware of the CLTE for:

1. Components with narrow tolerances

2. Areas with high temperature fluctuations

Productional tolerances for plastics evaluated are relatively small, as previously shown in Figure 2.3. The design required the geometry to fit all assembly details, and temperature tolerances would affect the end-product. Sea water temperatures would not have rapid temperature variations, but differences between air and sea temperatures may be critical for polymers with high CLTE. The CLTE can be reduced by introducing reinforcements, but it was necessary to also control the thermal dimensional stability, to ensure that the stiffness in the pressure vessel were intact (Ensinger (2012)).

Material data sheets give information about the heat deflection temperature (HDT), which measures the high temperature behavior of the material, and to what extent the stiffness is affected (ISO 75). This temperature characterize the upper limit temperature the material could be exposed to for a longer time period. With increasing temperatures the internal molecular chain bonds becomes weaker compared to the thermal energy of the molecules, and large deformations may occur. This cannot be directly used to characterize the material, but is important when comparing the expected behavior for available plastics. Two aspects are especially important in determining the thermal dimensional stability of a plastic, namely;

1. Production technique

2. Polymeric structure

Ensinger (2012) states that data determined by measuring test specimens from machining and milling deviate from results from injection molded specimens. Therefore it had to be kept in mind that the material for a pressure vessel produced by injection molding could not be directly comparable to the machined prototype. Another considerable aspect of the polymer behavior was how the mechanical properties varied with short-term changes in temperature. Short-term

is referring to time spans from minutes to occasionally hours, and the plastic should not take considerable damage from being exposed to any short-time temperature fluctuations. Long term thermal effects also had to be considered, even if the vehicle was designed to operate for just a few hours. Maximum service temperature depend on where in the world the ROV is used. Using the vehicle in colder water would result in an more brittle material with increased modulus and rigidity, which eventually would lead to a reduction in impact resistance. The ROV could be stored in variable environments over longer time and should not have material degradation if stored properly. In addition, future upgrades may have the possibility of changing the battery and operate for several hours.

3.2.2 Density

Density is the measure of mass per unit volume. This property is important since weight saving is one of the main reasons for selecting plastics, compared to other materials. Material suppliers often use the specific gravity to determine the material weight and cost. Specific gravity (s.g) are defined as in Equation 3.1

$$\text{Specific gravity} = \frac{\rho_{material}}{\rho_{water}} \tag{3.1}$$

Prices are often based on the total weight of raw material. Foams and cellular materials may have specific gravity down to 0.01, but denser thermoplastics range from 0.8 to 2.0. The pressure vessel was designed to have a slightly positive buoyancy to ensure surface return if the power was cut. To reduce the need for additional buoyancy elements it was desirable to use low density polymers for the pressure vessel, as it would give a positive buoyancy effect. BluEye communicated that plastics with specific gravity between 1.0 and 1.7 were preferred, mainly due to these buoyancy considerations. Figure 3.2 shows a selection of polymers within the desired density range.

3.2.3 Mechanical Properties

Mechanical properties were one of the most essential criteria when choosing a material for the pressure vessel, as the main structural goal was to withstand the external pressure forces. The
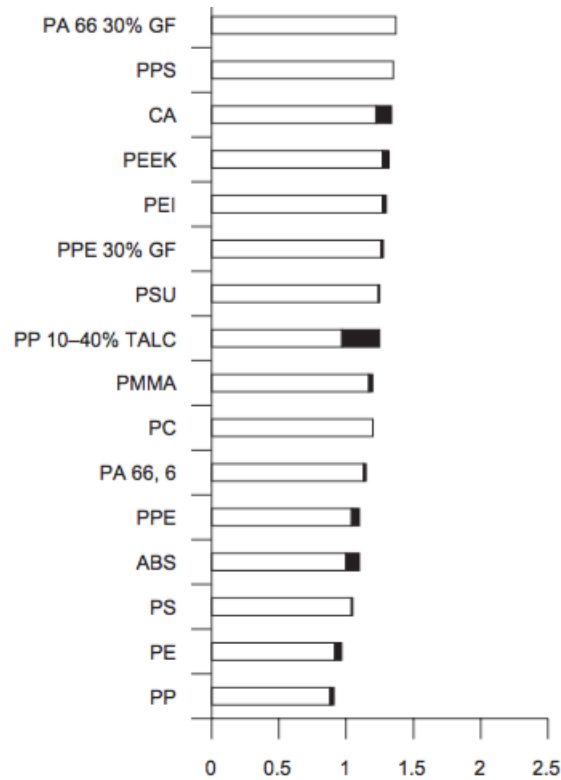
Figure 3.2: A selection of polymers with specific gravity close to seawater (Biron (2013b))

fundamental selection criteria were the material strength, toughness, formability and rigidity. Strength typically refers to the material resistance against stresses, while the toughness is the energy absorption capacity under these stresses. Formability is connected to material deformation under stress, while rigidity describe the resistance against deformation (Ensinger (2012)). A basic figure of how to read the material behavior from a stress-strain curve is shown in Figure 3.3. A material with large formability is able to undergo large deformations before breaking. The material strength characterize the stress at which breaking occurs. A rigid material is often referred to as brittle, and does not allow for much deformation before failing. With reduced rigidity, the toughness increases. The pressure vessel required a combination of these properties. Strength and rigidity were most important, as the hull should was required to withstand large pressure forces without having too much deformation.

Data used to evaluate the mechanical properties may be influenced by the test environment, and production methodology of the test specimen. To ensure compatibility between mechanical properties for all plastics, it was important that the data had been obtained by performing

proper test. Tests had to be done in accordance with American Society for Testing Materials (ASTM) or the International Organization for Standardization (ISO). Some mechanical evaluation criteria are listed in Table 3.5, along with their respective ASTM and ISO test.

Table 3.5: Important mechanical properties and their standard ASTM and ISO tests

Property	ASTM	ISO
Tensile modulus	ASTM D638	ISO 527
Tensile strength	ASTM D638	ISO 527
Elongation at break	ASTM D638	ISO 527
Flexural strength	ASTM D790	ISO 178
Flexural modulus	ASTM D790	ISO 178
Impact strength	ASTM D256	ISO 180

Flexural strength is an indication of the material stiffness, and signify how well the material resist bending. The pressure hull was allowed to bend, but not get any permanent deformation. Permanent deformation in a material is closely connected to the yield stress, and the flexural modulus, which is a number that measures the resistance agains elastic bending deformation in a material. In ideal theory the flexural modulus is equal to the tensile modulus. Flexural modulus is also known as Young's modulus of elasticity. However, for some plastics these values are dissimilar and have to be evaluated separately (Campo (2008a)). The elasticity modulus define the stress to strain ratio below the yield point. At this point the specimen starts deforming and the cross-section is decreasing. For reinforced polymers the yield stress are equal to the ultimate stress, as the material becomes brittle and does not yield in the same way as their unreinforced resin does.

ISO 527 measures the elongation at break, tensile strength and tensile modulus by stretching a bar. Tensile strength is a measure of the materials ability to resist being pulled apart. The ratio of tensile stress and corresponding strain before deformation begins, is called the tensile modulus. Resulting stress-strain curves deduce a lot of characteristics of a polymer. Elongation at break is the total elongation that occur before reaching the break point (ultimate elongation). An explanation of the terms can be seen in Figure 3.4. The elasticity modulus is the slope of the stress-strain curves in this figure. A brittle plastic will have a small elongation at break and a high

S Strength
F Formability
R Rigidity
T Toughness

σ_B Ultra-high tension ε_B Elongation under ultra-high tension
σ_R Tensile strength at break ε_R Elongation at break
σ_S Tensile strength at yield ε_S Elongation at yield
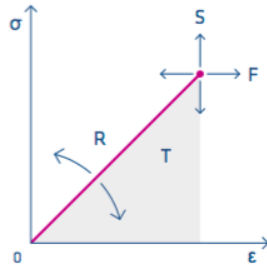
Figure 3.3: Brief description of a polymer stress-strain curve (Ensinger (2012))
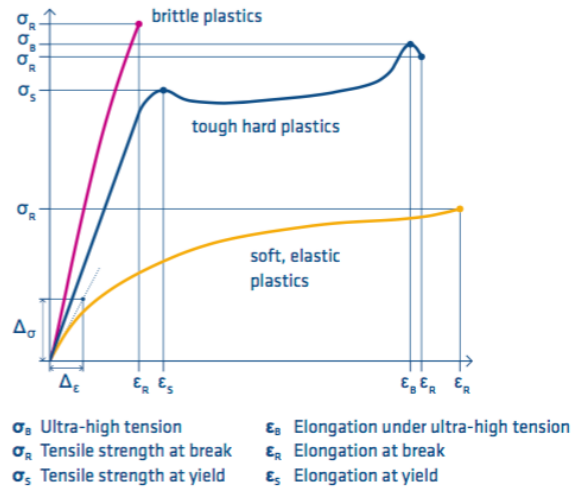
Figure 3.4: Typical stress-strain curve for linear elastic materials (Ensinger (2012))

elastic modulus. Softer, more elastic, plastics will undergo large deformations before reaching the tensile stress which it breaks.

Plastics have lower yield strength and elastic modulus compared to many other materials, but have high strain to failure ratio (Crawford (1998)). In the case of a vehicle submitted only to external pressure, the stiffness and density were important. By also including the aspect of collision, it was desired to have a high stiffness to weight ratio, and a high impact strength to weight ratio. Stiffness is proportional to the moment of inertia, and will also be a function of the hull thickness. Kutz (2011) derives how it is possible to evaluate the stiffness and weight by, considering the elasticity modulus and density. Equation 3.2 shows how to relate the stiffness per weight proportional to the elastic modulus, E, and the density, ρ. S is the stiffness and W is the polymer weight.

$$\frac{S}{W} \propto \frac{E}{\rho} \tag{3.2}$$

Equation 3.3 can be used as an indication of how tough the material is, by comparing its impact strength, I, per weight in proportion to the elongation at failure, ε_{max} times the modulus divided by the density (Kutz (2011)). Toughness may also be read from the material data sheet, and origins from the loss of energy from a pendulum impact on a material specimen.

$$\frac{I}{W} \propto \frac{E * \varepsilon_{max}}{\rho} \tag{3.3}$$

By using these equations it was possible to quickly compare properties of materials by comparing their respective ratios.

3.2.4   Water Absorption and Ultra Violet Resistance

Polymers are organic materials that are sensitive to UV sources. The ROV is made for outdoor activities and will experience both natural and artificial light. Aging may occur if the vehicle is left too long in sunlight. A result may be degradation of the mechanical properties, discoloration or surface crazing and cracking. According to Biron (2013b) it is difficult to interpret the behavior of the polymer due to climate diversity in the areas of use, and the lack of correlation between natural and artificial aging. UV stabilizers can be added to enhance the protection and avoid aging, but this is not considered as a evaluation criteria in this thesis.

An important aspect for the structural integrity of the pressure vessel was the amount of water absorbed by the plastic. Plastic materials may swell up when submerged into water, which would result in loss of buoyancy. The water absorption is the percentage weight increase of the material after being submerged in water for 24 hours. It is important to notice that the actual amount of absorbed water depends on the part geometry, and environmental factors like temperature, humidity and time (Campo (2008d)).

Moisture will affect the mechanical properties for most plastics by reducing the stiffness, tensile strength and elasticity module, based on the amount of water present in the plastic. In addition, the water captured inside the material could freeze if the vehicle are stored improperly, and result in cracking of the hull. Absorption characteristics depend on the polymer composition, and the resin group. Engineering plastics containing hydrogen or carbon, such as polyethylene (PE) and polystyrene (PS), will be very water resistant. On the opposite, nylon and other plastics that contains oxygen groups, will absorb a high amount of water. PTFE and other materials containing fluorine, chlorine or bromine will be water repellent, independent on the reinforcements (Campo (2008b)).

3.3   Decision Gate and Proposed Materials

Selecting the best material was done by first filter out all materials that were able to be injection molded. Properties and characteristics of basic resins was then compared. As discussed, it was preferred to select a plastic with high tensile strength and a good combination of rigidity and flexibility. In Figure 3.5 some resin groups that are suitable for injection molding are visually compared according to their characteristics and properties.
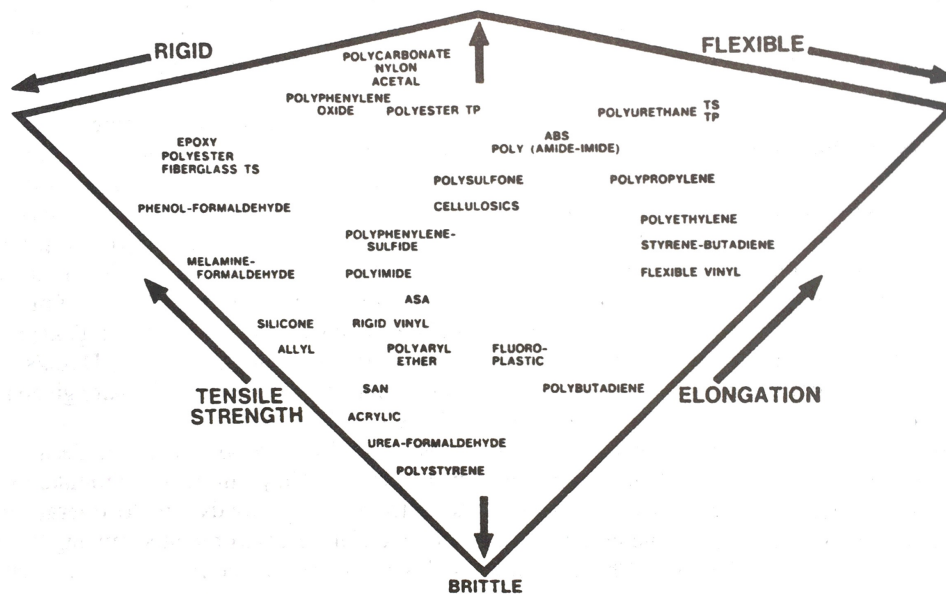
Figure 3.5: Range of mechanical properties for injection molding plastics (Rosato (2000))

The table in Figure 3.6 rate injection molded plastics based on relevant criteria. Physical properties are given a score based on its performance, and an explanation of the parameters are given in the figure. In the selection process parameters were weighted based on the criteria and requirements presented in Chapter 2 and Chapter 3. In general, the overall considerations that had an impact on the material selection were:

1. The end-use performance and targeted life span of the vehicle

2. Operating environment and ocean temperatures

3. Mechanical properties

4. Dimensional tolerances, rigidity and deformability

5. Density and possible reinforcements

6. Economical considerations

G/R Resin Groups / Design Criteria	Strength & Stiffness	Toughness	Short Term Heat Resistance	Long Term Heat Resistance	Environ-mental Resistance	Dimensional Accuracy In Molding	Dimensional Stability	Wear & Frictional Properties	Cost
Styrenics ABS SAN Polystyrene	2 1 3 — **3**	1 2 3 — **6**	1 2 3 — **6**	1 2 3 — **6**	1 2 3 — **6**	3 1 2 — **1**	2 1 3 — **5**	3 1 2 — **6**	3 2 1 — **2**
Olefins Polyethylene Polypropylene	2 1 — **5**	2 1 — **4**	2 1 — **4**	2 1 — **5**	2 1 — **3**	1 1 — **5**	1 1 — **5**	2 1 — **3**	1 2 — **1**
Other Crystalline Resins Nylons 6 6/6 6/10, 6/12 Polyester Polyacetal	2 1 3 4 5 — **1**	2 3 1 4 5 — **1**	2 1 3 2 5 — **2**	2 1 3 1 2 — **4**	5 4 3 2 1 — **4**	1 2 2 2 3 — **4**	4 3 2 1 2 — **4**	3 2 3 4 1 — **2**	1 2 4 1 1 — **3**
Arylates Modified PPO Polycarbonate Polysulfone Polyethersulfone	4 2 2 1 — **3**	3 1 2 3 — **2**	4 3 2 1 — **3**	4 3 2 1 — **3**	3 4 2 1 — **5**	4 1 2 3 — **1**	4 3 2 1 — **2**	4 3 1 2 — **4**	1 2 3 4 — **4**
High Temp. Resins PPS Polyamide-Imide	1 2 — **2**	2 1 — **4**	2 1 — **1**	2 1 — **1**	1 2 — **2**	1 2 — **4**	2 1 — **1**	2 1 — **4**	1 2 — **5**
Fluorocarbons FEP ETFE	2 1 — **6**	1 2 — **2**	2 1 — **2**	1 2 — **1**	1 2 — **1**	2 1 — **6**	2 1 — **6**	1 2 — **1**	2 1 — **6**

Ratings: 1—most desirable, 6—least desirable. Large numbers indicate group classification; small numbers are for the specific resins within that group.
Strength & stiffness: The ability to resist instantaneous applications of load while exhibiting a low level of strain. Materials that demonstrate a proportionality between stress and strain have been assigned better relative ratings.
Toughness: The ability to withstand impacting at high strain rates.
Short-term heat resistance: The ability to withstand exposure to elevated temperatures for a limited period of time without distortion.
Long-term heat resistance: The ability to retain a high level of room-temperature mechanical properties after exposure to elevated temperature for a sustained period.
Environmental resistance: The ability to withstand exposure to solvents and chemicals.
Dimensional accuracy in molding: The ability to produce wrap-free, high tolerance molded parts.
Dimensional stability: The ability to maintain the molded dimensions after exposure to a broad range of temperatures and environments.
Wear and frictional properties: The ability of the plastic to resist removal of material when run against a mating metal surface. The lower the frictional values, the better the relative rating.
Cost: The relative cost per cubic inch.

Figure 3.6: Resin groups selection scheme for injection molding (Rosato (2000))

Total deformation of the pressure hull should not reduce the total volume by more than 3%. This was an absolute limit set by BluEye. The previously listed criteria and considerations had different weight factors, and the most important material characteristics are ranked below;

1. Stiffness and Strength

2. Cost and raw material price

3. Toughness and dimensional stability

4. Short and long term heat resistance

5. Environmental resistance

In Biron (2013c), Rosato (2000) and Hough and Dolbey (1998) it was possible to find detailed overviews of the properties of the most common plastic resins. Hough and Dolbey (1998) graphically visualized all the criteria by using bar charts, and made it possible to quickly compare unmodified resins. In coordination with BluEye and professor Ludvigsen, the resin groups in Table 3.6 were selected for further evaluation. In this first material evaluation it was of the company's interest to select a polymer that was economically favorable, and a cost factor (Morgan (2005)), was included in the initial selection. This factor depended on the availability, demand and current oil price and was a relative number of how cost effective the polymer was, compared to PA 6.

Table 3.6: Initial selected unmodified polymers

	Density [kg/m^3]	Water absorption [%]	Yield stress [MPa]	Tensile modulus [GPa]	Cost index
PPE	1080	0.1	55	2.6	1.24 (1.61)*
ABS	1060	0.7	44	2.5	0.67
PA 6	1140	8,5 - 10	85	3.4	1.0
PA 6 conditioned	1140	8,5 - 10	55	1.8	1.0
PA 66	1150	8,5 - 10	90	3.3	1.11
PA 66 conditioned	1150	8,5 - 10	65	2	1.11
PC	1200	0.10	60	2.5	1.06 – 1.3
PEEK	1320	< 0.01	95	3.65	21.6
PPS	1350	0.05	65	3.7	5.1

 For all PA-types mechanical properties are given as dry and conditioned. All others are in dry condition.
* Price index for Norly (GTX) resins, a family of modified PPE

3.3.1   Selected Material

Nylons, polyester and high temperature resins, such as polyphenylene sulfide (PPS) and Polyamide-imide, possess high strength and stiffness. Nylons and polyester have high toughness and short term resistance, but lower environmental resistance. The high temperature resins have excellent dimensional stability, in addition to good long and short term heat resistance. However, the price is rather high and they generally have poor dimensional stability for injection molding. Syrenics, such as ABS and Styrene acrylonitrile (SAN), do not possess the same strength

and stiffness as Nylon and PPS, but are cheaper and have excellent dimensional accuracy in molding. Polycarbonate (PC) and other arylates have the same stiffness and strength as ABS, but better dimensional stability.

Unreinforced PEEK (polyether ether ketone) and PPS have a high price and density compared to the other polymers. BluEye communicated the need for a low price material, with a specific gravity close to 1.025, which made PEEK and PPS inapplicable for further evaluation. Within the limitations of the project, PC, ABS, PPE and PA proved to be the most appropriate polymers for mass production. To lower the risk for production errors, and long transport and manufacturing time, BluEye wanted an easy accessible material that could be manufactured without special equipment. PPE is not commercially used to the same extent as PC, ABS and PA and was excluded as a material choice. Four selected polymer resins are listed in Table 3.7, along with a few possible reinforcements and modifications. The selection was done together with designers at BluEye, and by using guidelines from Campo (2008c), Brigante (2014) and Hough and Dolbey (1998).

Table 3.7: Comparison of ABS, PA 66 and PPE

	Density [kg/m^3]	Water absorption [%]	Yield stress [MPa]	Tensile modulus [GPa]
ABS	1060	0.70	44	2.50
ABS 30% GF[1]	1280	0.12	100	7.58
PA 6	1140	8,5 - 10	85 (55)	3.4 (1.8)
PA 6 30% GF[3]	1360	6.3 - 6.9	185 (115)	9.5 (6.2)
PA 66	1150	8.5 - 10	90 (65)	3.3 (2)
PA 66 30% GF[2]	1360	5.2 - 5.8	190 (130)	10 (7.2)
PC	1200	0.10	60	2.50
PC 30% GF[4]	1430	0.26	131	8.62
PC/ABS 30% GF[5]	1340	0.10	117	6.21

PA types are listed with their saturated state in paranthesis. All other plastics are in dry condition.
GF = Glass filled, CF = Carbon filled, IM = Impact modified, MF = mineral filled
[1] Techmer ES HiFill® ABS GF30 30% Glass Filled
[2] BASF Ultramid® B3EG6 30% Glass Filled PA6
[3] BASF Ultramid® A3EG6 30% Glass Filled PA66
[4] Westlake Plastics Zelux® Polycarbonate 30% Glass Filled
[5] Techmer ES Plastiblend® PC/ABS FR 30% Glass Filled

Polyamide, or nylon, exhibits high strength, toughness and dynamic strength and have good processing properties. A disadvantage is relatively high water absorption, compared to PC and ABS. ABS is less expensive and has high impact strength and toughness. It also has a excellent surface finish, but lacks UV protection. PC have good impact resistance and dimensional stability and is easy injection molded (Platt (2003)). PA 66 30% glass filled (GF), PC/ABS 30% GF and ABS 30% GF were the three materials selected for application in the design and analysis phase. PA possess the most desirable properties and was used in the majority of the analyses. ABS was included to have a low price material to use, in case PA proved to possess too good properties. The PC/ABS blend possess beneficial properties from both ABS and PC. In case of a too weak ABS plastic, the ABS/PC blend has better properties than a pure ABS. In addition, the PC/ABS blend will absorb much less water than PA, and has a higher yield stress than ABS. In Figure 3.7 a polar plot are comparing some important properties of the three chosen plastics. Since PA absorbs a high amount of water it is displayed as both dry and conditioned.
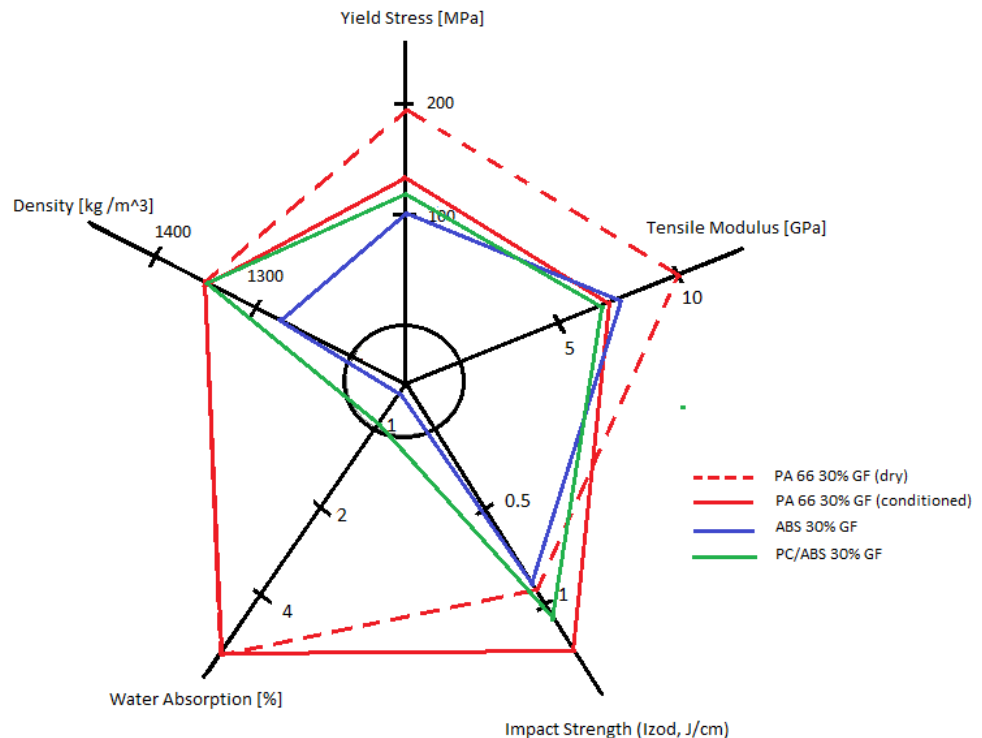

Figure 3.7: Polar plot of some properties of PPE 30%GF, ABS 30% GF and PA 66
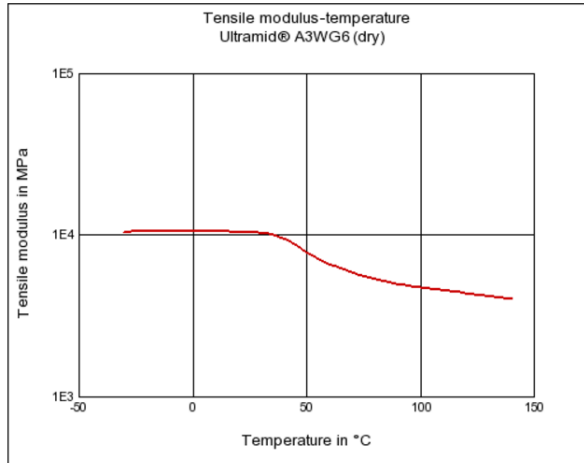
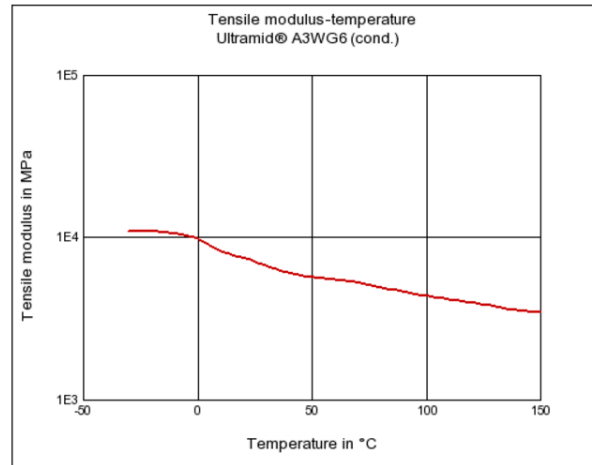Figure 3.8: Tensile modulus variation with temperature for dry PA 66 30%GF.

Figure 3.9: Tensile modulus variation with temperature for conditioned PA 66 30%GF.

Since polymers are temperature sensitive, and absorbs water, inclusion of all parameters that varies with the external environment and long term cycles, were required to ensure a good performance of the pressure vessel, under almost any condition. The material stress-strain curve describe well how the material behave, when loaded and elongated. Different temperatures will create new curves, and Figure 3.8 and Figure 3.9 shows how the tensile stress changes with the temperature for a dry and conditioned PA 66 30% GF. In dry condition the tensile modulus will not change in the temperature range for sea water. A fully conditioned polymer will however undergo large changes.

Figure 3.8 and 3.9 shows only the extremities of the temperature behavior for PA 66 30%GF. How the tensile (or yield) stress gradually reduce with increasing moisture content, can be seen in Figure 3.10. The PA 66 30%GF polymer used in the analyses is called A3EG6, and have a behavior as the A3EG5 and A3EG7 in the figure. When the PA 66 30% GF was used in analyses, both extremal conditions had to be included to indicate the materials best and worst performance.

Tensile stresses are drastically reduced with increasing moisture content, but will also reduce with increasing temperature, as Figure 3.11 and 3.12 shows. These considerations and extreme conditions had to be included in the analyses, to get a result that was compatible with all areas of application. Material properties at 23°C was used for the analyses, to get the most conservative results.
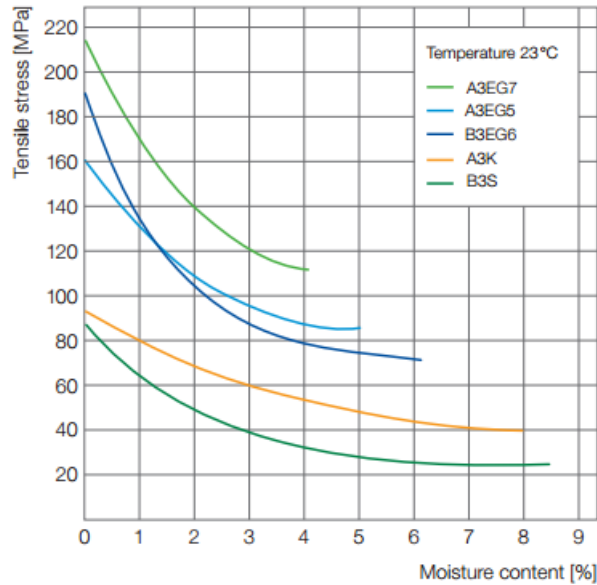
Fig. 4: Tensile stress (yield stress in the case of unreinforced grades) for Ultramid® as a function of moisture content at 23°C (ISO 527)

Figure 3.10: Tensile stress variation with moisture content for PA 66 30%GF
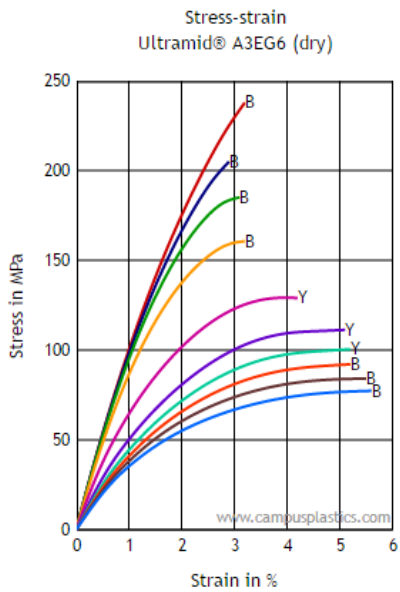
Figure 3.11: Stress strain curve for a dry polymer
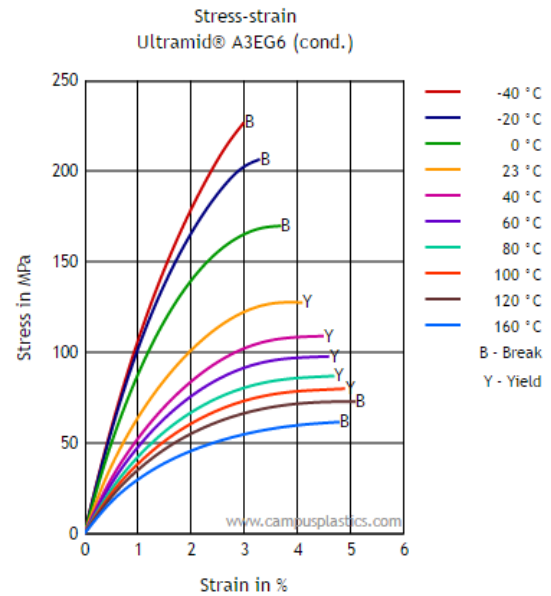
Figure 3.12: Stress strain curve for a conditioned polymer

Chapter 4

Design

Designing the pressure vessel for injection molding is just one part of the complete development process of the BluEye ROV. A new prototype requires a new product design and development process. The concept is firmly defined by BluEye for the previous prototypes, and this thesis mainly covers aspects within the product design phase.
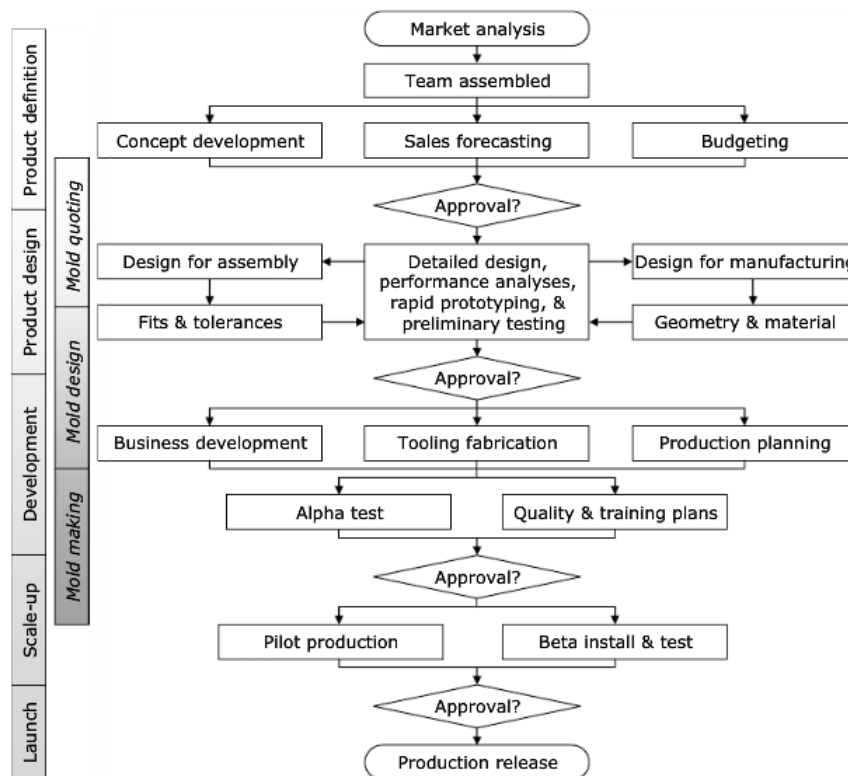
Figure 4.1: Product development process of an molded part(Kazmer (2007))

Figure 4.1 shows a typical design process for an injection molded plastic part. After the design phase, tooling and fabrication will be evaluated and the mold made ready for manufacturing. To have an efficient process the mold quoting should take place parallel to creating the design. Design of the mold is performed after the vehicle geometry are completed, to ensure compatibility with the geometry. Aspects from the development process, such as tooling and fabrication, are not covered to any extent in this master thesis. More detailed information about the mold design and tooling may be found in Bryce (1998), while Beaumont (2002) covers the injection molding manufacturing process. Scale-up and testing will not be covered in this thesis, but are required prior to the next prototype product release.

4.1   Concept

BluEye Robotics is currently (pr. 01.06.2016) in the test phase of the P1 prototype. A model of the P1 prototype is shown in Figure 4.2. This design was used as the basis for the design process performed in this thesis. The prototype is designed to give the user good control and flexibility to explore the ocean space. It has a total of four thrusters installed; one vertical, one lateral and two in the back.

Figure 4.2: Prototype of the BluEye Explorer P1 (early 2016)

Figure 4.2 displays the concept geometry without any of the thrusters installed. Two thrusters

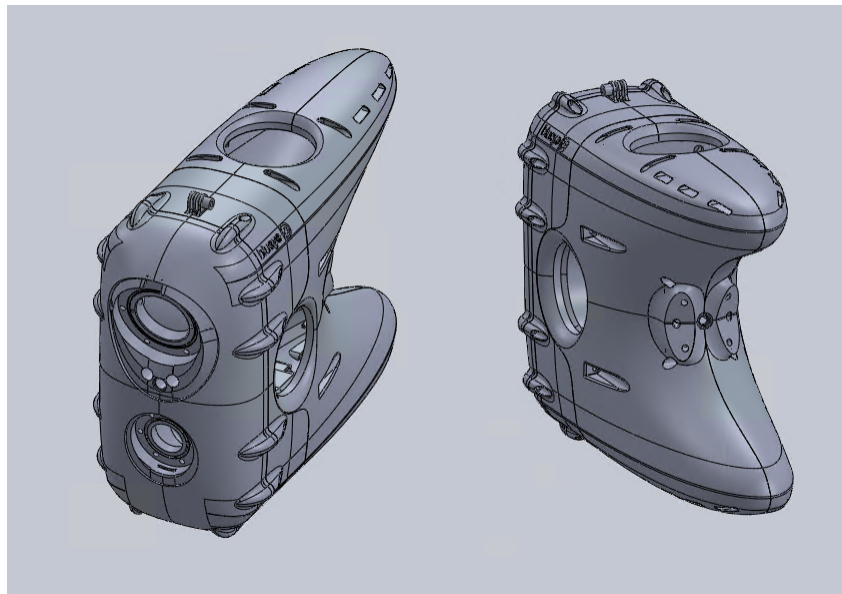are placesd inside the two, larger openings going through the middle section of the vehicle in lateral and vertical direction. In the rear end of the vehicle there are two fastening points for the two remaining thrusters. Camera and light sources will be placed inside the two openings in the front of the vehicle.

The concept of the P1 prototype is to have two separable compartments with different functions. The front compartment, called the pressure vessel, is entirely pressure resistant and should be able to withstand all forces down to 100 m depth. Inside the front compartment all the electric components are stored, as it is completely waterproof. Bolts are connecting the front and back of the pressure resistant front compartment. An o-seal is installed between the front and back part, by using machined tracks along the rims. The back compartment is not waterproof and, fills up with water when the vessel is submerged into the ocean. Therefore, no electronic equipment are stored in the rear part. The back part is used to connect the thrusters to the body and is required to ensure a hydrodynamic and stable body. A split-up of the front and back compartment are shown in Figure 4.3
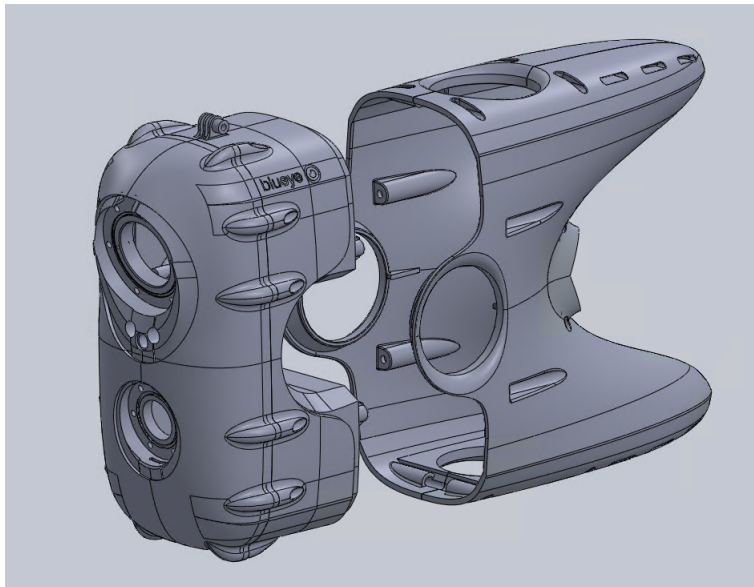

Figure 4.3: Separated back and front compartment of the BluEye Explorer P1 (early 2016)

The concept of the next generation prototype should be similar to the P1. Therefore, the modified prototype was designed with two separate compartments, where the pressure vessel was analyzed. The pressure vessel was designed in two parts, but without the bolted connections and tracks for an o-seal.

4.2 Aim of the Design Study

The P1 prototype, which was used as a basis in the design phase, have a uniform wall thickness of 15 mm, and is made of ABS plastic. Machining the P1 prototype proved to be expensive and was not a suitable manufacturing method for mass production. It was therefore desirable to change the production methodology to injection molding for the next prototype. The primary goal of the design study was to investigate the possibility for producing a pressure resistant vessel, with the present design, by injection molding. As the back compartment is not pressure resistant it does not require the same structural capacity as the front compartment. It was assumed that if it was possible to create a stiff and strong enough structure for the pressure vessel, the back compartment would not be problematic to manufacture and function optimally as well.

To prepare the pressure vessel for injection molding the vessel had to be re-designed. Tolerances and design principles discussed in Section 2.3 were applied for the new pressure vessel design. As discussed, injection molded parts has limitations related to the thickness, draft angles, undercuts, rib design and corner design, and this chapter discuss these design aspects and the related criteria. Maximum allowed thickness varied with the chosen plastic. For the selected plastics the current 15 mm hull thickness had to be reduced before the vessel was possible to mold. Modification of the pressure vessel had to be done so the thickness reduction do not compromise the structural integrity. The aim of the design study was to achieve the three design goals;

1. Modification of the current prototype, so that it was possible to produce it by injection molding

2. Achieve the same structural properties as the P1 prototype for the modified geometry

3. Discuss possible challenges and modifications needed before production, if pt. 1 and 2 were not possible to achieve

If the design goals were not possible to achieve the results had to be discussed, along with future design modifications and recommendations. This study will be used as a basis for future prototypes, as an indication of available materials, required thicknesses and rib structures.

4.3   Design for Injection Molding

Injection molding is one of the most popular processes for manufacturing thermoplastic products, and it is possible to manufacture very complex geometries with this production method. In the manufacturing process a thermoplastic, or thermoset, are fed into a heated barrel and mixed, before it is injected into the mold cavity. It further cools of and hardens. To avoid a faulty product, or a product not impossible to create a mold to, the design must be carefully adjusted to fit the requirements for injection molding. The pressure vessel was designed for optimal stiffness and strength, by use of a number of guidelines for achieving the best result. Some fundamental aspects of injection molding design, and challenges and constraints for the various parts of the design, are discussed in the below sections.

4.3.1   Wall thickness

Reducing the wall thickness will require less material, and result in cost savings. Thin sections will cool more easily and have a shorter cycle time, which allows a more parts to be produced per hour. In Kutz (2011), Rosato (2000) and Kazmer (2007) an uniform thickness is highly recommend wherever possible. Uniform thickness provides uniform molded-in stress distributions, uniform shrinkage and uniform filling patterns. In that case the melt does not have to be forced through varying sections which allows the mold cavity to fill more easily (Rucinski (2015)).

The wall thickness was used as a reference for many part adjustments, such as rib design and minimum corner radius. As the polymers have different molecular structures, the melt flow would also vary. To avoid improper filling of the mold cavity there are maximum and minimum thickness requirements for different polymers. Manufacturers may have slightly different limitations, but in Table 4.1 some conservative recommended wall thicknesses are listed for a selection of polymeric resins.

A fundamental issue with varying thicknesses is the cooling rate. Temperatures near the thin sections will be lower than for the thicker section, and a difference in temperature between sections may result in shrinkages. As shown in Figure 4.4, a result of this may be significant geometric distortions and creation of internal voids, warpage and local sink marks. This is a result of the high thermal expansion coefficient in plastics and excessive shrinkage for the thicker parts,

Table 4.1: Recommended wall thickness (3DSystems (2015), ProtoLabs (2016))

Resin	Recommended thickness [mm] (min - max)
Acrylonitrile butadiene styrene (ABS)	1.14 - 3.56
Acrylic	0.64 - 12.7
Long-fiber reinforced plastics	1.91 - 25.4
Nylon (PA)	0.76 - 2.92
Polycarbonate (PC)	1.02 - 3.81
Polyester (PET)	0.64 - 3.18
Polyethylene (PE)	0.76 - 5.08
Polyphenylene sulfide (PPS)	0.51 - 4.57
Polystyrene (PS)	0.89 - 3.81

even with extended cooling time. A thin and thick part connected to each other will cool off differently, and thin parts will shrink and harden faster than the thicker parts. When the thick section shrinks there will be a stress build up in the boundary between the two, and the thin part will harden and not yield as the thicker section yields. A result of this can be warping or twisted parts. It is not uncommon, for large differences in thickness, that severe cases of warping or cracking occur (Stratasys (2015)). In addition, warpage may also come from the mold temperature, cooling rate, injection pressure or packing.

Ribs, bosses and other intersecting walls may cause voids and shrinkage problems due to a slower solidifying rate for the thicker sections. This may result in an shrunken area in the nominal wall where they are attached. This effect can be minimized by putting restrictions on the rib design, which is further addressed in Chapter 4.3.2.

If an uniform thickness is not possible to obtain, the transition should be as gradually as possible. Figure 4.5 shows an illustration of some designs with different thickness distribution, and address favorable and erroneous designs. For thin to thick wall thickness, and other sharp transitions, the melt will be jetted from the thin section to the thick. This will result in reduced surface finish and replication. The thin section will have premature solidifications and it is difficult to have dimensional control of the thicker section of the plastic part. It is possible to improve the design by have a opposite flow direction or by having a thin-to-thick transition. By
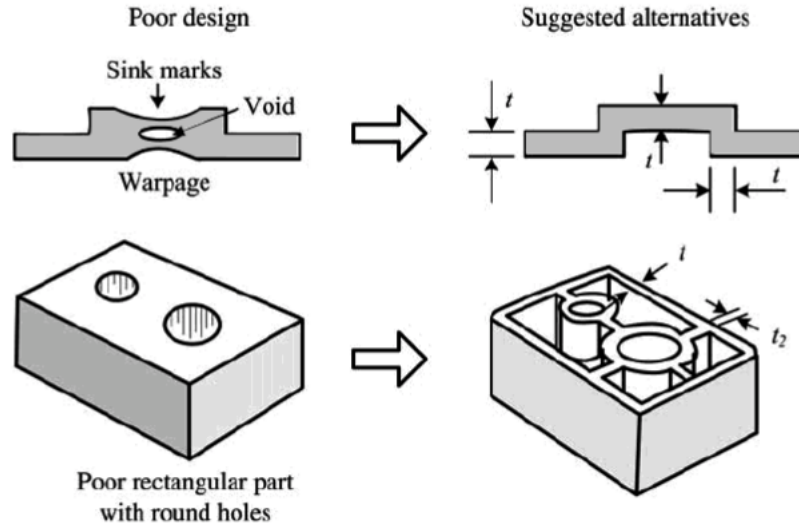
Figure 4.4: Voids, sink marks and warpage (Kutz (2011))

directly using the P1 prototype design, there were a number of surfaces with transitions from thin to thick. To avoid problems connected to transitions a simplified geometry was created. The simplified geometry was used when performing analyses, and had mostly smooth surfaces and few corrugations. This will be further covered in Chapter 4.5.
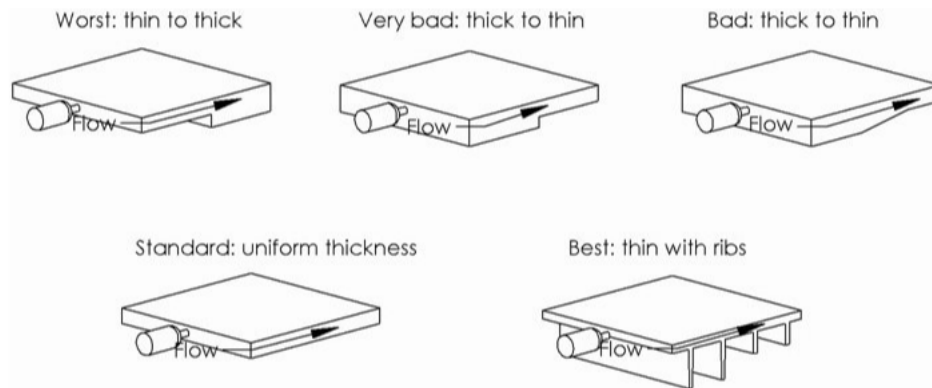
Figure 4.5: Wall thickness design (Kazmer (2007))

4.3.2   Rib Design

Bases or parts with different thickness than the nominal wall is a typical problem in injection molding. Injection molding enables, compared to other molding methods, the possibility of

adding vertical ribs to stiffen the structure. Ribs will increase the bending stiffness of the struc-
ture without adding any thickness. An example of how a thick base with thin side walls are
replaced with a stiffened, uniform part, are shown in Figure 4.6.

Thick base and thin side walls   *Rib with a uniform thickness*

Figure 4.6: Thickness and rib design (Kutz (2011))

Protruded parts of the mold should have a reduced thickness compared to the base wall to
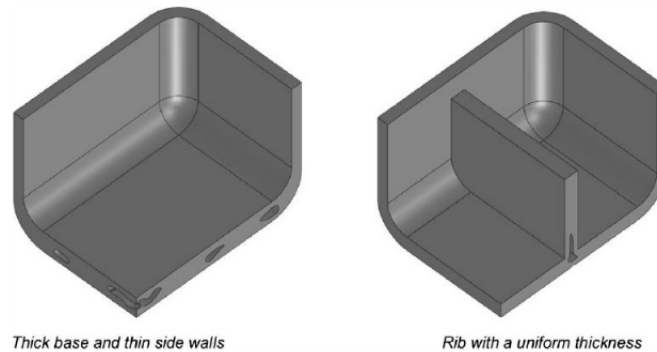avoid sinking and other thickness related problems. The thicker part will use more material and
have a longer cycle time than the ribbed part. Rib thicknesses at the base are only 70 % of the
flat part, but will have a stiffness equivalent to a 30% thicker part without ribs (Kazmer (2007)).
The bending stiffness follows from Equation 4.1, where the increase in bending stiffness, K_b,
depends on the rib geometry and the corresponding moment of inertia, I, and flexural modulus,
E.

$$K_b = E \times I \tag{4.1}$$

It was important that the ribs were oriented in such manner that they provided increasing
bending stiffness in the loading direction, as seen in Figure 4.7. The ribs had to be thinner at the
end to avoid sink marks and have a rounded base to avoid stress concentrations.

Designing ribs for injection molding required caution with respect to the thickness and tran-
sitions. As the material cools, thick ribs will tend to draw material away from the center of the
opposite wall. These problems would occur for ribs that was thicker than 70% of the wall thick-
ness, and the volume shrinkage will result in sinks on one rib-side. Materials with low shrinkage
may be designed with larger thickness. However, according to Zhou (2013) and Rucinski (2015) a
rib thickness of less than 75% of the wall thickness should be used for injection molding applica-
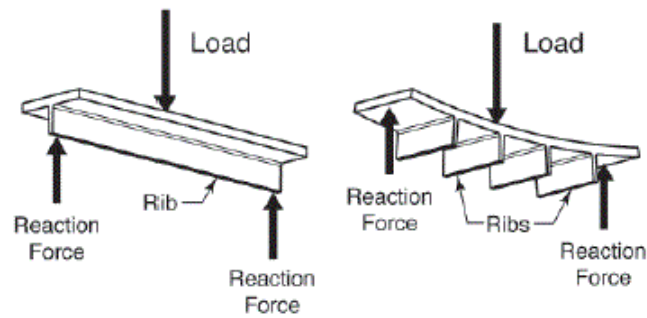
Figure 4.7: Rib orientation, load and reaction forces (Stratasys (2015))

tions. Limitations and recommendations for rib systems are closely connected to the nominal thickness. In Figure 4.8 some recommended proportions of ribs are given with respect to the nominal thickness. For crossing ribs the area of intersection will have a thickness larger than the recommended maximum. In those cases the need for removing material have to be evaluated to avoid excessive sinking. This is often done by coring or other means of material removal (Stratasys (2015)).
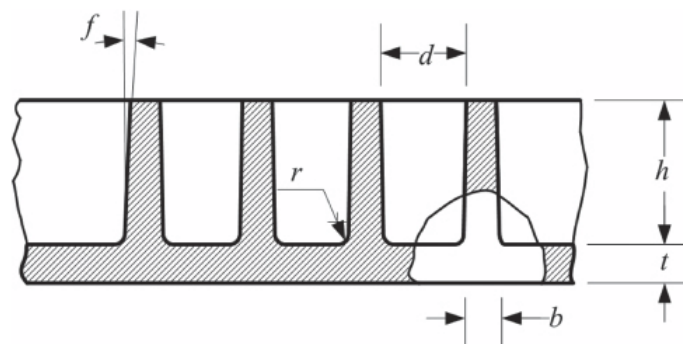
t = wall thickness
b = 0.5 to 0.75t
h = 3t maximum
    (if more stiffness
    is required, add
    additional ribs)

d = 2.5t minimum
r = 0.25t (radius corner)
f = ½ per side, minimum
    (draft angle)

Figure 4.8: Recommended proportions for ribs in thermoplastic moldings (Zhou (2013))

4.3.3 Corners

Sharp corners are often present to ensure mating between components, or due to esthetic design considerations. Areas with stress concentrations will depend on the geometry, and the P1 prototype had some areas with small corner radiuses that may lead to high stresses. Sharp corners give stress concentrations that can lead to part failure, which is crucial, especially for brittle materials. In addition, geometries with sharp corners will have lower torsional stiffness as one with rounded corners. Equation 4.2 shows how the stress concentration, K, depends on the radius, r, and the coefficient C. The C-coefficient depends on the geometry and material used (Kutz (2011)).

$$K = C \times \sqrt{\frac{1}{r}} \tag{4.2}$$

It is possible to see the behavior of the stress concentration factor from Figure 4.9. For R/T values less than 0.5 there is a significant increase in the K-factor, and it would therefore be beneficial to have a radius larger than 50% of the nominal thickness.
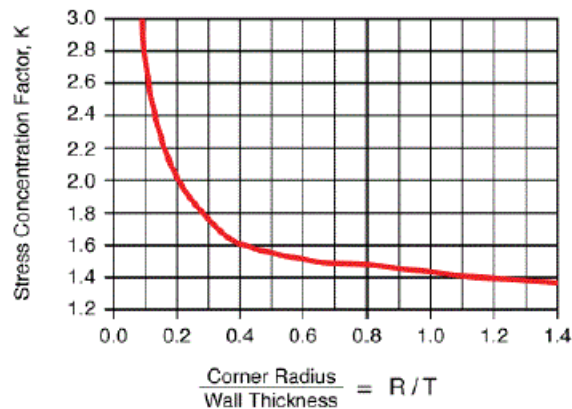
Figure 4.9: Stress concentration factor (Stratasys (2015))

Figure 4.10 shows some basic guidelines for corner designs. The inner radius was recommended to be at least 0.5 times the wall thickness, while the outer fillet radius should be 1.5 times the wall thickness. Larger fillets are preferred if possible and should be evaluated in accordance with the available tools. In injection molding the sharp corners will also be difficult to produce, and often require machining or other tools after molding.
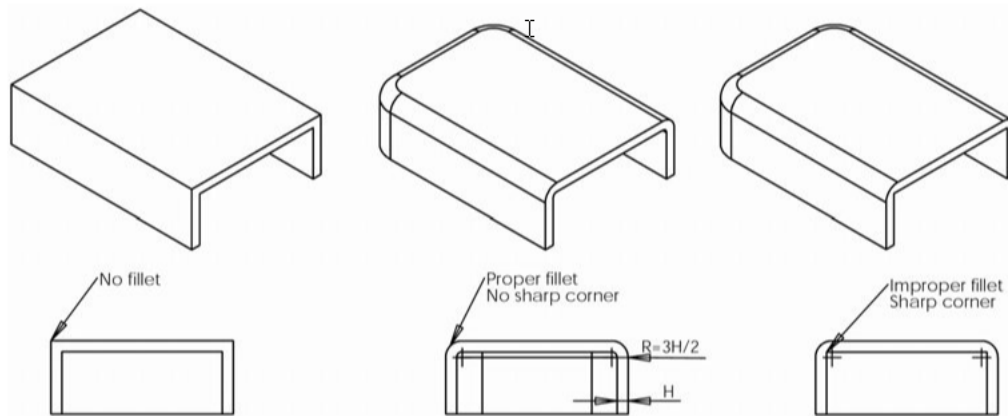
Figure 4.10: Guidelines for corner design (Kazmer (2007))

The corners will restrict the melt flow to the core insert, which result in variations in the shrinkage across the thickness. High stresses and rib buckling may also lead to unexpected brittle failure (Spoormaker (1995)). Nylon, acrylic and styrene are brittle materials and would exhibit less failure with internal fillets and rounded corners applied.

4.3.4 Bosses

Lack of lateral support on bosses and ribs will often result in brittle failure. An unsupported boss, as the one to the left in Figure 4.11, will most likely fracture. It would be necessary to introduce stiffening ribs, gussets and supports if bosses are introduced to the injection molded vessel, like the right model in Figure 4.11 shows. Generally all bosses, gussets and ribs should utilize a thickness of 60-70% of the nominal wall thickness (Kazmer (2007)).

Bosses are typically used to fasten components by using screws. Boss design will depend on where they are applied and what type of fasting mechanism that should be used. Ribs may be used to provide an elevated surface or support, at corners or middle sections. Draft angles depend on the chosen material and their shrinkage (Kutz (2011)). The boss radius should be a minimum of 25 % of the wall thickness and can be strengthened by ribs or gussets, or by connecting them to an adjacent wall (Stratasys (2015)). It is important that bosses are able to withstand the torque and pulling force when fastening the screws on the final assembly. A reason for the boss not to be oversized is the need for extended cycling times.

Unsupported boss   Boss with gussets and supporting ribs
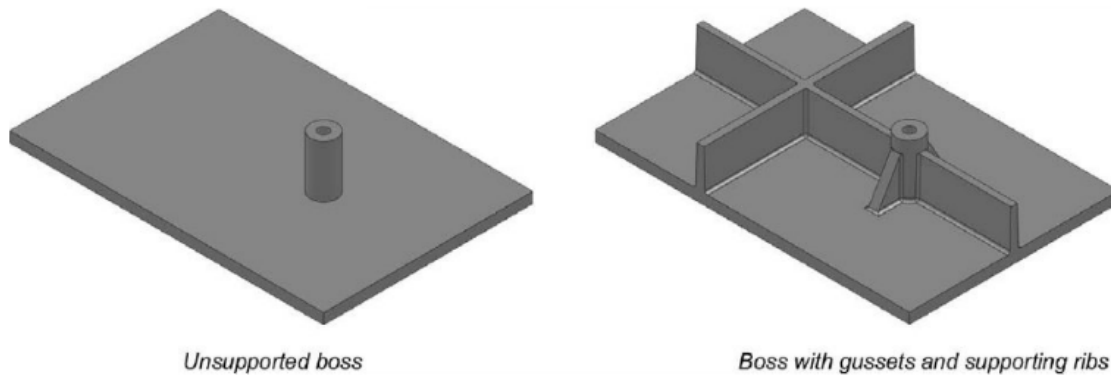
Figure 4.11: Structural support of a boss (Kutz (2011))

As the prototype have twelve machined bosses at the outside surface of the geometry. These bosses could eventually be placed on the inside of the pressure vessel to avoid the aesthetic problems, but they would require post processing of the injection molded part. Bosses similar to the current ones was not able to produce by injection molding. For the next prototype design, the connection method had to be re-evaluated, to avoid additional post processing and chance of damaging the part.

4.3.5   Undercuts and Draft Angle

In some parts of the pressure vessel there were undercuts. An undercut is a feature that will interfere with the mold injection, which include horizontal bosses and overhangs. When the part is pulled from its mold, the undercuts interfere with he pulling direction. Ribs at the side wall of the pressure vessel was categorized as undercuts. An example of some undercuts are shown in Figure 4.12.

Undercuts will prevent the part ejection or mold opening, but some undercuts are however added to simplify the post-molding assembly. Additional tooling and advanced molding techniques are required in the creation of undercuts. Machining are commonly used post molding, which would increase the complexity and the total cost of the mold. Additional mechanisms would also be needed to free the molded part. Improper use of these additional techniques may lead to damage of the mold and molded part. This topic is not covered further here, but is addressed in BayerMaterials (2014). It is desired to avoid undercuts, but it may not be feasible
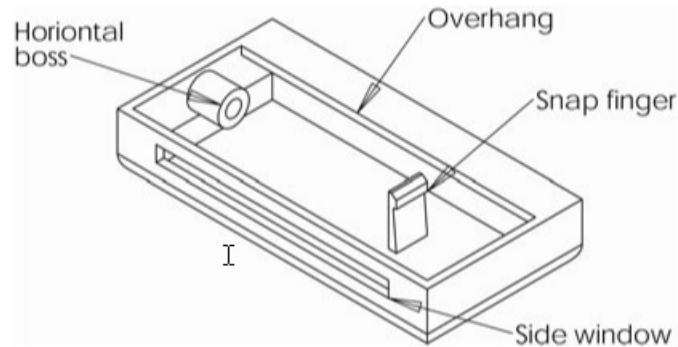
Figure 4.12: Typical undercuts (Kazmer (2007))

without compromising the functionality of the end product (Kazmer (2007)). The pressure vessel was designed to be fit for production, without the need for additional tools. The side walls was therefore designed with stiffeners in only one direction, to avoid presence of undercuts.
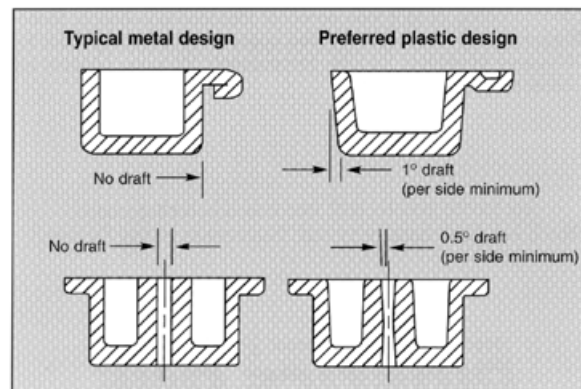
Figure 4.13: Draft angle recommendations for plastic design (PlasticsOne (2016))

Mold release requires a draft angle in the direction of the mold movement. Small draft angles may result in a poor mold release. As a result the molded part get dimensional variations and distortions. Male sections of the mold tends to shrink, and it is recommended to have a minimum draft angle of 1/2° for unreinforced polymers. Glass-reinforced resins has lower shrinking in and require a draft angle of 1.5° (Rosato (2000)). It is common that the parts shrink to cores when cooled, and the draft angle is required for the plastic part to be removed from the mold (PlasticsOne (2016)). This is a significant difference between metal and plastic designs, and an illustration of how the plastic draft angle is included for plastic parts is shown in Figure 4.13.

4.4 Work Process

The work process is the process of converting the prototype geometry to a part possible to injection mold. Inputs from the users, providers, design team and fabrication facilities were important factors when deciding geometry and functionality for the ROV. Design and analyses of the pressure vessel were done in cycles with multiple iterations, evaluations and re-designs. A visualization of the work process, based on the processes in Allmendinger (1990), are shown in Figure 4.14.
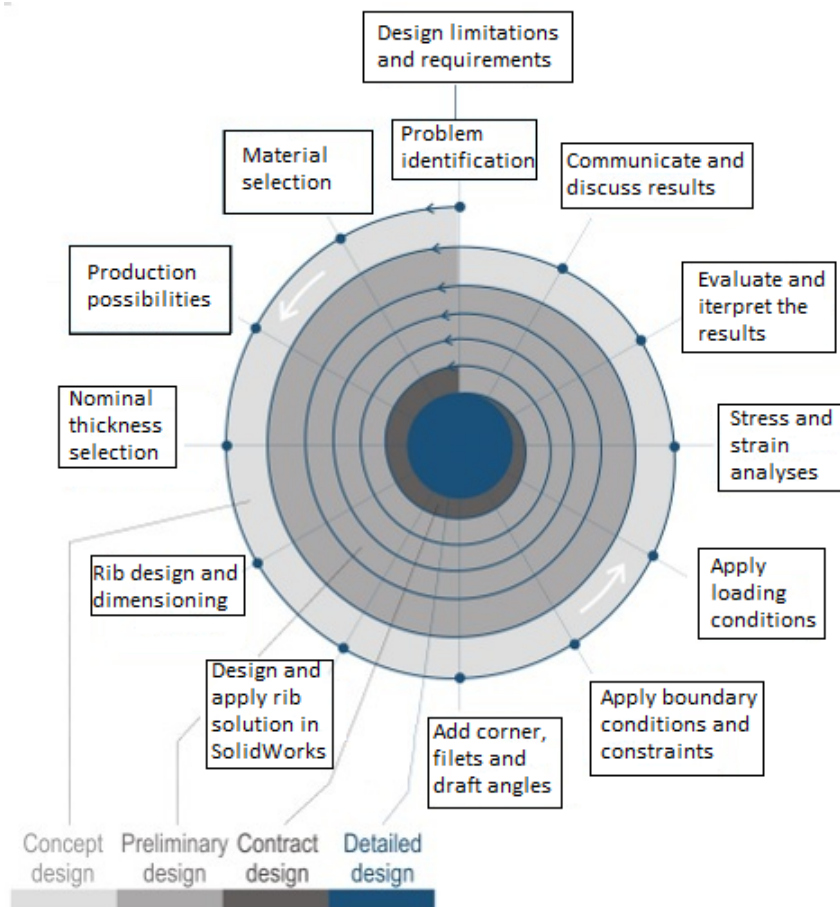
Figure 4.14: Design spiral from problem identification to results

Problem identification and enlightenment of design related problems were stated, after ensuring that all limitations and requirements were included. As a basis for any further design regulations a suitable plastic had to be selected. It had to be suitable for injection molding, and in addition have sufficient material properties. Further, the re-design of the vehicle was performed

accordingly to all limitations and requirements, before being analyzed. Results were then evaluated, based on the aim of the design study, and communicated to BluEye. The iteration process continued until the thesis time limit was reached, but could be repeated almost indefinite. Users opinions were gathered continuously by BluEye, but did not influence the design process in this thesis to any extent. Possibilities for making changes were documented and discussed openly. BluEye gave continuously feedback on the design, according to the objectives. Designer Rune Hansen from BluEye was the main contributor to new suggestions and design solutions in this thesis, along with professor Leira.

A flow chart of the design-loop is shown in Figure 4.15. The three blue squares to the left in the figure, circled with a green square, are the fundamental aspects of the design loop. Changing one of these would directly impact the physical and finite element (FE) model. Production methodology and prototype geometry was fixed in this work process, and the only changeable, fundamental aspect was material selection.
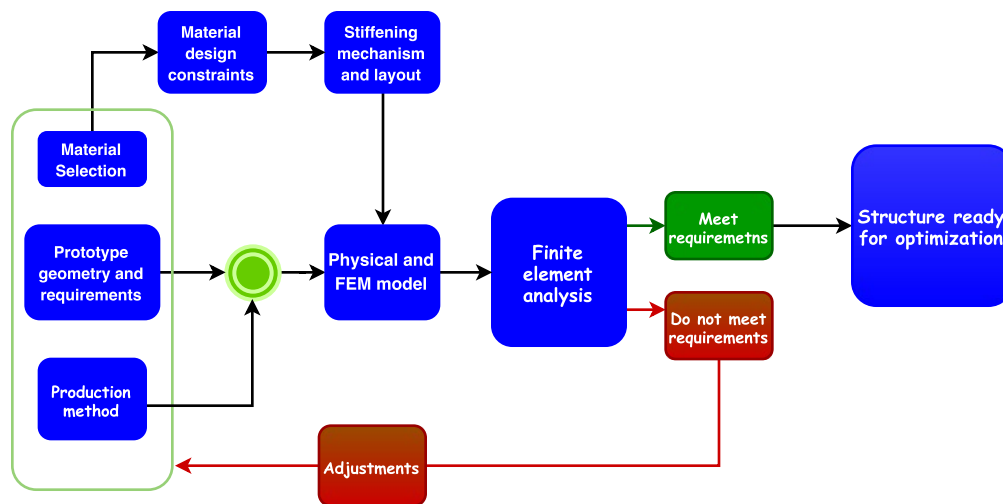
Figure 4.15: Design processes flow chart

If the hull structure proved to be within all given limitations and requirements, the design solution was proposed to BluEye and evaluated for further optimization. Adjustments were initially done for the design blocks closest to the FE-model, if the model did not meet the prescribed requirements. Stiffening mechanisms and layout were the first aspect that to be adjusted before redesigning the model. Material selection were done in a thorough selection process prior to the design phase, and was not changed if the element model did not meet its re-

quirements. The outcome of the work process should give clear indications of how the next generation prototype could be designed. That include what material that could be used and how ribbing structures and stiffening mechanisms that could be designed. In addition, the results should facilitate further implementations towards mass production. If the design goals were not met, it could be possible to obtain the best solution for the pressure vessel design, by following the flowchart and design spiral.

4.5   Model Simplification

A simplified geometry of the P1 prototype was designed in SolidWorks. The P1 prototype is a highly detailed model with a large number of faces and intersections, and directly apply the detailed geometry would lead to complications and an unnecessary time consuming analysis. Complex geometries often contain double or small edges and faces, which makes the meshing process difficult. Presence of geometrical singularities would possibly give large stress concentrations that, due to numerical errors, would exceed the general maximum stresses. Then it would not be possible to estimate the required thickness for the hull (Leira (2014)).

The simplified geometry was created with a smooth outer surface and few details, and was fully compatible with the manufacturing process for injection molding. Figure 4.16 shows the model simplification. This model was used for the iteration process and analyses. By simplifying the geometry it could be more directly compared with theory. A brief presentation of some theoretical aspects used in the design process are covered in Section 4.6.

4.6   Structural Design Aspects

As the pressure vessel required a high stiffness to withstand the pressure from the surroundings, the pressure vessel had to be designed for stiffness and shock absorption, with a strong connection between the front and back part. The front part of the pressure vessel was curved in two directions and had few sharp corners and edges. Curved sections were designed for the back part as well, both for structural considerations and to make room for the lateral thruster. The curved sections for the back part only had curvature in one direction. It was expected that
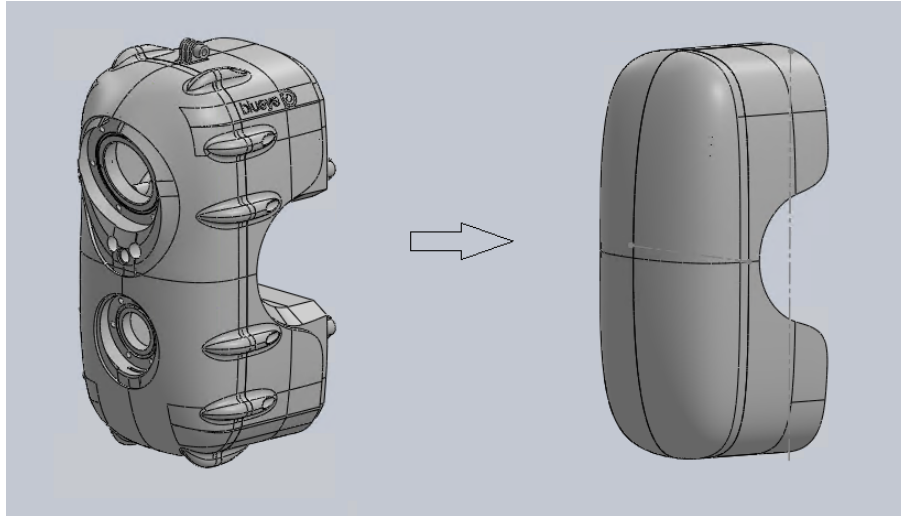
Figure 4.16: Model of the simplified geometry

none of the proposed plastics were able to keep its shape at high depths, due to its low material stiffness. Designing for high stiffness was the first, and most important, design aspect. Further, shock absorption and connections became important, but these were secondary considerations for meeting the objectives of the thesis.

4.6.1   Design Modifications for Increased Stiffness

Reducing the thickness lowered the structural capacity of the pressure vessel. The pressure vessel was mainly designed for stiffness, and some common methods for stiffen plastic structures that could be used, was;

- Corrugations, either bidirectional or straight

- Top / hat / doming sections

- Rib structures

Corrugation of the outer hull would increase the part stiffness, but was not expected to be sufficient for this application. Doming sections were applied in the global geometry, by curving the corners and making sections with rounded edges to increase the stiffness. Corrugations and additional hat and top sections were not further evaluated for stiffening the pressure vessel, as the outer surface had to be smooth, and the pressure vessel required a large strength to thickness

ratio. Ribs were used in the injection molding design, to increase the stiffness and improve the melt flow into the section. It is possible to increase the stiffness of the plastic up to 25 times, by only doubling the material weight of the structure (Zhou (2013)). Large displacements and stress concentrations would also be reduced by having one or multiply stiffeners. Introducing stiffeners would reduce the inner volume of the pressure vessel, which had the possibility to affect the arrangement of equipment and payload. In addition, to reduce the deflection of the hull, increased stiffness will benefit both the impact strength and creep strain. The stiffeners ability to provide stiffness in the pressure vessel depended on (Moan (2003));

- Stiffener geometry and the number of stiffeners present

- Distance between stiffeners

- Cross-sectional area of stiffeners

- Sectional modulus of stiffeners

The pressure vessel, subjected to pressure forces normal to all surfaces, was subjected to both tensile and flexural loads. Longitudinal compression loads, such as bending and shear, and transverse compression loads in the hull, were reduced by introducing ribs. As covered in Section 4.3.2, the deflection of a beam under flexural load is inversely proportional to the EI-module of the plate, which is the product of moment of inertia (I) and elasticity module (E). The largest moment of inertia would ideally be applied for sections with the highest bending loads.

To get a good indication of the pressure vessel capacity, the amount of stiffeners were maximized. Pressure forces were assumed uniform for the pressure vessel, and the moment of inertia was maximized to get full capacity utilization of the applied ribs. This was done by following design rules and regulations for injection molded parts, and maximizing the cross-sectional area of the stiffeners, while the distance between stiffeners were minimized. The stiffener shape remained the same throughout the design phase. All ribs were designed as flat and straight, and aligned with the pulling direction. This limited the possibilities for adjustments and new design solutions, but was done to avoid undercuts at the stiffeners.

4.6.2 Design for Shock Absorption

Designing for shock absorption is a complex procedure. The impact force will change with factors like impact area, load type and environmental conditions. A general guideline for shock absorption design is to deflect the load rather than providing a rigid impact. Selecting an impact modified material will increase the performance, by distributing the elongation from a shock impact and equalize the stress throughout the part (NilitPlastic (2015)). A varying thickness will perform better for shock loads, but is not preferred with respect to production. This is a somewhat contradictory design aspect for plastics. The best way of modifying the design without exceeding any design regulations was to avoid all stress concentrations, and provide rounded edges and bases for the ribs.

4.6.3 Design Against Buckling

Pressure forces will act simultaneously on all surfaces of the vessel, and axial forces will act normal to the plate thickness of the hull. If the axial forces have high magnitude they may cause bending of stiffeners or the hull plate itself. According to Leira (2014) the stiffeners applied in this design have three possible failure modes, as listed below;

1. Plate buckling between stiffeners

2. Buckling of stiffener and corresponding plate-flange between transverse frames

3. Global buckling of stiffeners and transverse frames

Buckling of stiffener and the corresponding plate flange, is the most critical buckling mode, and often the design criteria for buckling. For increasing deformations the plate capacity increases, since stresses are redistributed and more compressive stresses are carried by the plate relative to the stiffeners. Due to this deformation, the plate is weaker mid-way between stiffeners. This variable stress, over the width between stiffeners, are equal to the material yield stress over an *effective width*, which is determined by the slenderness of the stiffeners. Equation 4.3 shows the relation between this effective width and the slenderness and yield stress (from DNV-RP-C201: "Buckling Strength of Plated Structures", October 2008).

$$\frac{\sigma_{mean}}{\sigma_{yield}} = \frac{\frac{1}{b}\int_0^b \sigma \, dy}{\sigma_{yield}} = \frac{b_{effective}}{b} = \begin{cases} \frac{\lambda_p - 0.22}{\lambda_p^2} & \lambda_p \geq 1 \\ \\ 1 & \lambda_p \leq 1 \end{cases} \qquad (4.3)$$

Here, b is the distance between the stiffeners, while the $b_{effective}$ is the effective width. The slenderness, λ_p, is defined as in Equation 4.4, where t is the nominal thickness and E the elasticity module.

$$\lambda_p = 0.525 \times \frac{b}{t}\sqrt{\frac{\sigma_{yield}}{E}} \qquad (4.4)$$

By increasing the effective width the capacity of the plate-stiffener system and increased, as well as the resistance against buckling. Material properties, such as the E-module and yield strength were fixed for a selected plastic. The thickness and with between stiffeners were parameters controlled by the production methodology, but were also influenced by the material choice. The relation between yield stress and elasticity module was also considered when selecting the material, as it was desired to increase the effective width by reducing the slenderness.

4.7 Connections and Plastic Joining

The pressure vessel was designed in two parts, which would require two different molds. Connecting the two compartments could be done in many different ways, but was assumed to be perfect when analyzing the vessel. When selecting a bonding technique becomes relevant, in later stages of the project, the bonding must have sufficient strength to keep the hull together, as well as ensure no water penetration and provide a completely pressure resistant design. Two main groups of connectors are used for plastic joining, namely mechanical fastenings and solvent or adhesive welding. Selections have to be based on the final geometry design, and whether the hull should be able to disassemble and reassemble. Investment in additional tooling and equipment also needs to be considered.

Hot tooling is a widely used technique which requires heating and melting of the polymer to form a joint. Solvent welding and adhesive bonding are a bit weaker than the mechanical fastenings, and require a large amount of surface area to be effective. The created joint is of-

ten weaker than the base material, even if the adhesive itself is stronger than the base material (Troughton (2009b)). Melting of the material will remove imperfections, warps and sinks from the production process. Mechanical fastenings, such as snap fits, bots, screws or brackets are more commercially used, mostly due to its simplicity.

There are a number of other, more advanced, techniques for plastic joining which are not addressed here. Some applicable methods for connecting the pressure vessel are brief explained in the below sections, along with some design considerations and challenges.

4.7.1   Hot Tool Welding

There are many different techniques that uses hot tools to weld plastics together. Heated tool welding is a widely used technique for joining injection molded parts. In the process, the thermoplastic parts are melted by a plate that melts the interface between them. When the material is soft, it is brought together under pressure to form a weld. Melting of the material will also remove imperfections, warps and sinks from the production process (Troughton (2009a)).

Hot tool welding can be used to join parts down to a few centimeters, and is suitable for all of the selected thermoplastics. The selection of joint type depends on the part application. A few basic joints can be seen in Figure 4.7.1, along with the heated tool process.
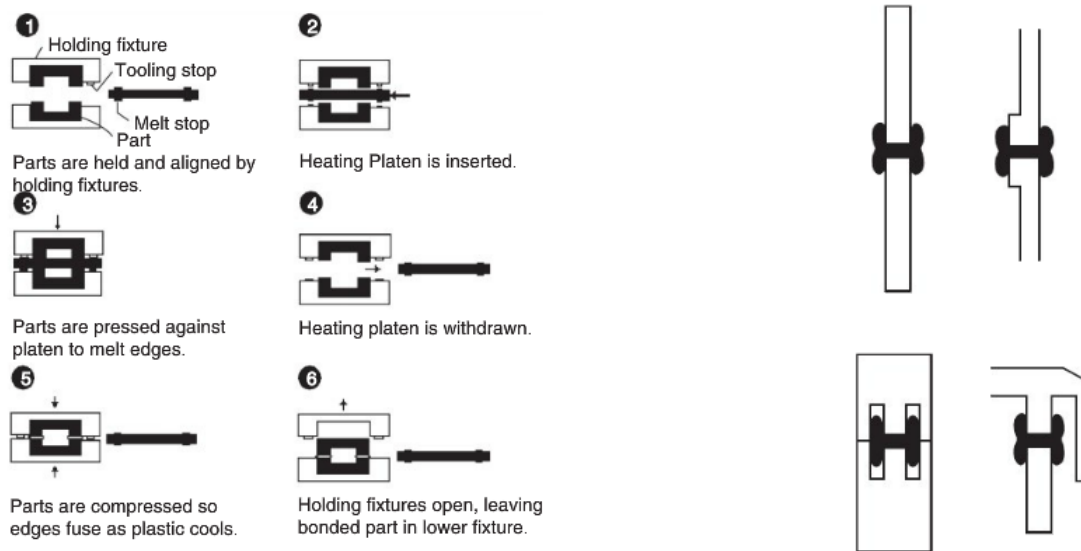

Figure 4.17: Heated tool process and selection of joints (Troughton (2009a))

Heated tool welding can be applied on both small and large parts, and the process can be easily automated since there are no foreign materials present. More manual options, such as extrusion welding and hot gas welding can be used for practically all types of shapes and sizes, but the result will depend on the skill of the operator.

Necessary equipment, welding speed, cycling times and how well the temperature are localized varies between the different hot tool welding techniques, and are important factors for choosing the best suited technique. The general benefits and disadvantages of hot tool welding are listed in Table 4.2. Manual techniques are not included since these techniques are not suitable for mass production.

Table 4.2: Advantages and disadvantages by hot tool welding

Advantages	Disadvantages
· Easily automated	· May create a weld flash
· High strength and lightweight welds	· High temperatures required
· Can join dissimilar materials	· High temperatures may
· Relatively low cost	degrade the material
· Cosmetic pleasing	· Part dimensions cannot
· Relatively low stress concentrations	always be reliably controlled

4.7.2 Solvent Welded Joints

Solvent welding and solvent cementing are two processes where a blend is used to create a joint between two thermoplastic parts. Post welding the material absorbs some of the solvent and get a soft and sticky surface., and after the welding process the glass temperature get reduced. In Table 4.3 shows some of the advantages and disadvantages by solvent welding.

There are different methods for applying solvents, but capillary bonding and soaking are the two most common methods. The solver is applied in the joint region of both substances, and the surfaces are pressed together and held in a place over several days. A long application time is needed for the solvent to be fully diffused out, and for the material to regain its normal properties. The joint strength will depend on temperature, application time and pressure (Troughton (2009b)). The capillary method is mainly used on small parts with fairly short binding lengths,

Table 4.3: Advantages and disadvantages by solvent welding

Advantages	Disadvantages
· Simple and inexpesive	· Limited to joining compatible polymers
· Lighter in weight	· Can reduce polymer strength around the weld
· No additional components	· Risk of solvent cracking
· Quick to assemble	· Reduce shrinking time
· Low stress concentrations	· Requires sufficient contact area

where the edges of the parts are dipped, or soaked, in a solvent tray before joined together. This method give a weaker joint than capillary boding, but follow the same design guidelines and recommendations. Figure 4.18 show some joint designs for solvent welding and solvent cementing.
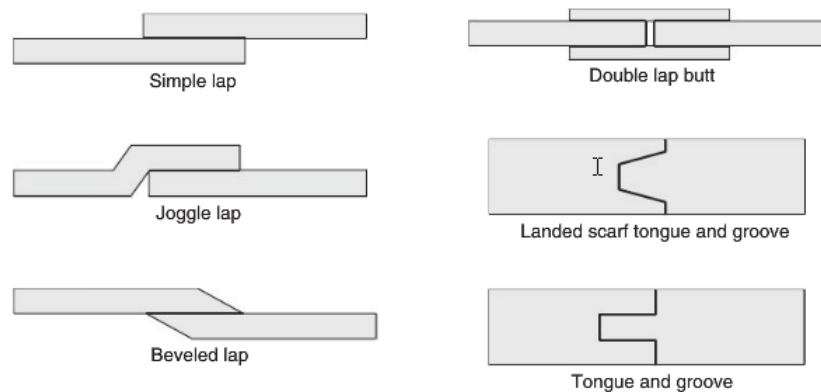
Figure 4.18: Joint design for solvent welding and solvent cementing (Troughton (2009b))

4.7.3 Adhesive Joints

Adhesive bonding is the most versatile type of all joining techniques and is somewhat similar to solvent welding. In the process an adhesive is applied to the adherent surfaces, to bond the compartments together. A significant difference from the solvent welding is the preparation of the surface before adhesion take place. Wetting, or cleaning, of the surface is required since liquids flatten out on the surface to bring a larger area of the liquid into contact wth the surface. The join is then created by holding the compartments together while the adhesive hardens

(Troughton (2009c)).

Table 4.4: Advantages and disadvantages by adhesive welded joints

Advantages	Disadvantages
· Economical, flexible and durable	· Require sufficient bond area
· Low stress concentrations	· Limited temperature range
· Tailored to application requirements	· Not possible to disassembly without
· Lightweight and easy process	destruction of bond or parts
· Does not require additional components	· Requires more surface area
· Provides bonding in irregular parts	· Difficult to inspect bond integrity

The main advantages and disadvantages of adhesive bonding are listed in Table 4.4. Mechanical properties of the weld will be important for the strength of the joint, and the adhesive joint design has a large dependency on the stresses in the compartment. If one is not aware of how the stress propagates and what shear forces the pressure vessel will encounter, stress cracking can occur from reaction between the polymer and particular adhesives. Some adhesives also become brittle at low temperatures, and experience degradation at high temperatures, so it is crucial to have a good overview of the area of application.
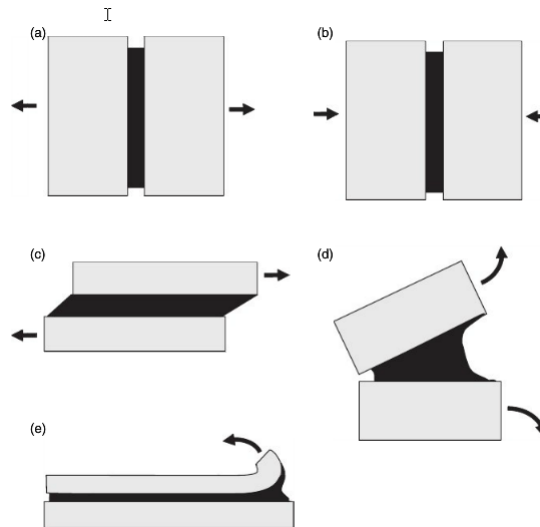

Figure 4.19: Stresses on the joints (Troughton (2009c))

Both the adhesive and solvent bondings are subjected to many different types of stresses through their lifetime. It is important that the joint stresses shown in Figure 4.19 are equally

distributed over the joint, to increase their ability to accommodate the loads. Especially cleavage and peeling should be avoided (Figure 4.19 d and e). Figure 4.20 shows a selection of good joint designs for adhesive bonding. Loads must be transferred through the layer and into the other component. The strength of the weld is closely connected to the bonding area. Double lap and straps (see Figure 4.20) will provide a larger bonding area, which again will result in a higher strength joint. Beveled laps and double butted laps would also be good solutions, as the joints must accommodate all types of loads between the parts.
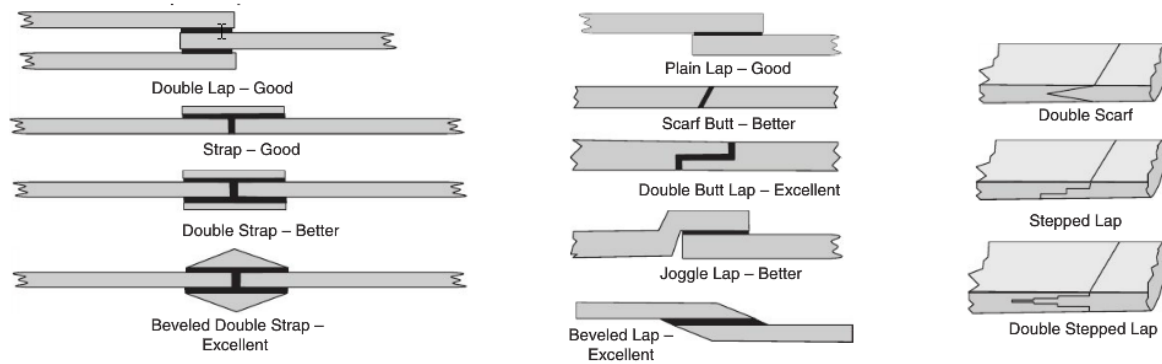
Figure 4.20: Selection of joint designs for adhesive welding (Troughton (2009c))

4.7.4   Mechanical Joints

Mechanical joining is the most common method due to its simplicity. BlueEye P1 has bolts that connect the two compartments, in addition to an o-seal to ensure a waterproof and pressure resistant hull. Many different types of mechanical joints may be used for such applications, both permanent and non-permanent selections.

When designing for mechanical joints it is important that the local stresses are kept at a reasonable level. Bolts have a broader application range than screws, which are mostly used for very strong plastics. It is necessary to have a high design and manufacturing precision to ensure a perfect connection between the two compartments. Then, the loads are transferred as friction at the surface, rather than shear forces through the bolts. Non-permanent fasteners can be disassembled and reassembled until the material fails. Usually the material is stronger in compression than for tensile stresses, and failure is often a result of tensile cracking or crazing

(Troughton (2009d)). A flat side of the screw head will provide more compressive stresses than conical screws, which produce more tensile stresses. An overview of the general pros and cons by using bolts and screws are shown in Table 4.5.

Table 4.5: Advantages and disadvantages by bolted joints

Advantages	Disadvantages
· Good impact resistance	· Stress concentrations
· Possible to assemble and deassemble	· Aestetic considerations
· Easy assembly	· Slightly more weight increase
· Require few tools	than other mechanims
· Easy to control joints	· Applicable for strong plastics

Chapter 5

Finite Element Analyze

Analyses were performed consecutively in SolidWorks and executed with basis in the finite element method. To ensure satisfactory results with sufficient accuracy the physical geometry had to be designed properly, and physical properties described along with the corresponding boundary conditions. A thorough formulation of the physical properties of the material, along with correct application of boundary conditions and loads, were the minimum requirements for performing buckling and static analyses (Belytschko (2014)). Obtaining, results from the structural analyses that could give indications of the pressure vessel performance, was the main objective for the analyses. After ribbing the structure, the two aspects of most interest was how the material and stiffened structure performed when subjected to pressure forces, and if buckling occurred before reaching the 100 m design depth. A perfect connection was applied between the back and front cover, before analyzing the pressure vessel in two steps:

1. Static analysis to locate maximum stresses and displacements

2. Buckling analysis to indicate the depth where buckling occur.

These two steps were done to find the most appropriate rib structure, based on its strength, stiffness and material utilization. Different rib patterns were applied in the first step and analyzed, before performing the second step. Buckling analyses were only performed for the best solution from the static analyses.

In additional, an analysis of the connections was performed. By removing the perfect connection between the compartments it was possible to address challenges connected to the join-

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ing, as previously discussed in Section 4.7. The most appropriate rib solution from the previous analyses were applied in this part of the analyze, and results could be used to indicate the loads present and how the deflection pattern would be.

5.1 Geometry and Material

A simplified geometry was designed in SolidWorks and used for all analyses. Figure 5.1 shows the model and how the internal rib pattern was designed. The rib pattern shown in Figure 5.1 was the basic rib pattern, but a number of additional rib solutions were applied on the front and back cover during the design and analyze process.
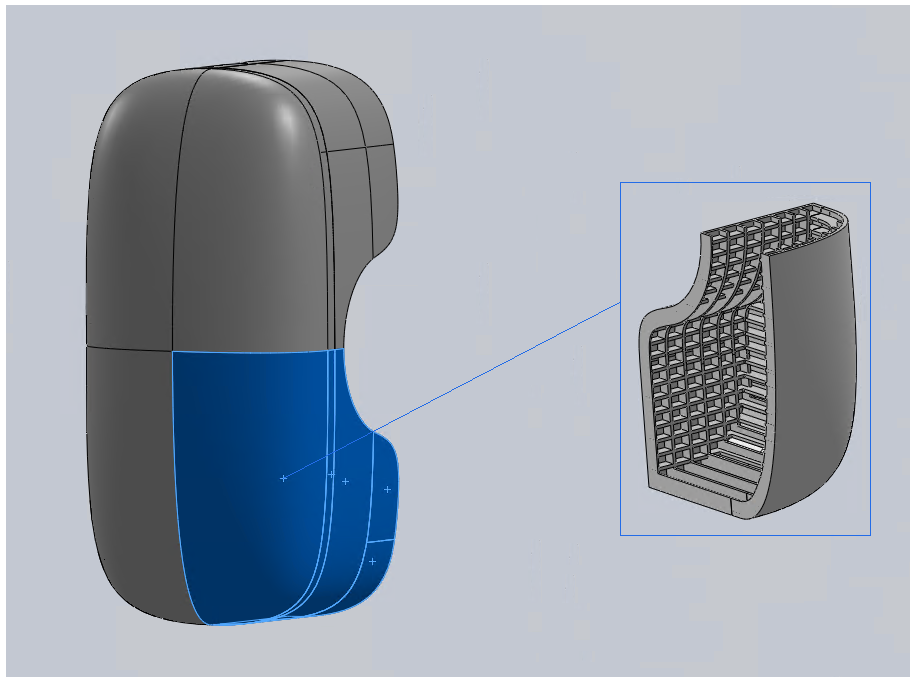
Figure 5.1: Simplified geometry and rib pattern

PA 66 30% GF was used as hull material, and the rib design used the recommended nominal thickness as the basis for stiffener dimensioning. Initial rib solutions maximized the thickness and number of stiffeners, while minimizing the distance between stiffeners. Table 5.1 shows some material properties of the three selected plastics, along with design limitations for stiffening solutions. The yield stress, or tensile stress for brittle materials, was used as an indication of material failure in the static analyses.

Table 5.1: Material properties and stiffener design restrictions for ABS 30% GF, PA 66 30% GF and PC/ABS 30% GF

	ABS 30% GF	PA 66 30% GF	PC/ABS 30% GF
Density [kg/m^3]	1280	1360	1340
Yield stress [MPa]	100	190 (130)	117
Tensile modulus [GPa]	7.58	10 (7.2)	6.21
Max. nominal thickness (t) [mm]	3.56	2.92	3.72
Max. rib thickness (0.75 x t) [mm]	2.74	2.19	2.79
Max. rib height (3 x t) [mm]	10.68	8.76	11.16
Min. rib distance[1] (2.5 x t) [mm]	8.9	7.3	9.3

[*] PA 66 is listed in saturated state in paranthesis. All other plastics are in dry condition.
[1] Minimum rib distance for the largest nominal thickness

5.2 Meshing

Meshing is the process of discretizing the model into a finite number of elements, and is an important aspect of the simulation. Too many elements will result unnecessary long solution time, and too few may lead to inaccurate results. The mesh size is selected to describe the plate bending and shear response with sufficient accuracy, thus describing the gradients of internal forces properly. How the mesh is generated depends on the geometry, and what specifications that are defined in SolidWorks. In addition, the generated mesh varies with the amount of connection points and global element size (SolidWorks(a) (2012)).

Inherent in determining stresses with finite element analysis is the discretization error. Obtaining sufficiently accurate stresses, are only possible when controlling the discretization error (Zienkiewicz (2000)). These errors are determined by convergence checks. It was essential to know if the stresses were diverging due to the presence of singularities in the mesh, before performing analyses. In the case of diverging stresses, it could not be possible to compare the hull strength with the corresponding stresses, when designing for structural integrity. It was necessary to conduct convergence tests on the FE-model, to confirm that a sufficient element discretization were being used. By analyzing several models with mesh refinements it was pos-

sible to find the mesh size where the results converged. This was done for the stresses or displacements. As stresses will in general converge more slowly than the displacements, it was not sufficient to only examine the displacement convergence (Ehlers (2013)).

Meshing was done by using curvature based mesh. Curvature based mesh created more elements in areas with high curvature, and checks for interference between bodies before meshing. This type of mesh supported compatible meshing between solid faces and touching, or partially touching, edges. Before meshing, bonding between elements had to be defined. This enabled evaluation and an easy access to all interferences. The global element size controlled the maximum and minimum size of each element, and reducing the mesh size gave a more detailed picture of the displacements and stress propagations.
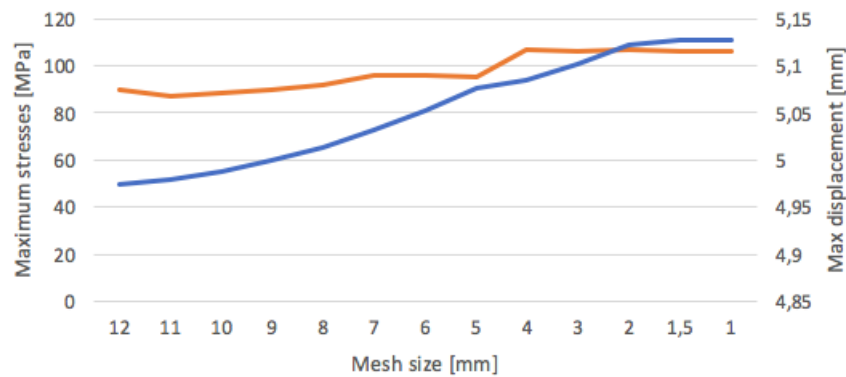

Figure 5.2: Mesh convergence for static analyses

Static and buckling analyses required separate convergence tests. Two mesh convergence tests were done to find the optimum connection between accuracy of the results and computational time. A graphic view of the convergence for the static analyses is shown in Figure 5.2, while Figure 5.3 shows the convergence for the buckling analyses. The blue line indicate the maximum stresses, while the orange show displacements. Convergence tests were performed by applying 0.2 MPa external pressure.

Large mesh sizes will result in fewer integration points and a stiffer behavior for each element. Decreasing the element size leads to more accurate calculations and stiffness matrices (Hughes (2014)). The displayed stresses and displacements varied with the mesh size, but reducing the mesh size below 2 mm only provided slightly more accurate results for the static analyses. Buckling analyses also indicated that a mesh size of 2 mm would not give more accu-
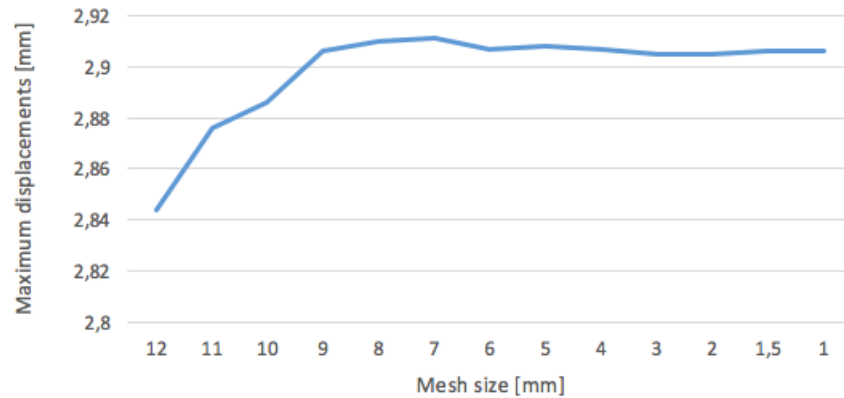
Figure 5.3: Mesh convergence for buckling analyses

racy in the result. Computational time did, however, increase significantly with reduced mesh size, and a mesh size of 2 mm was therefore chosen as sufficient for all analyses.

5.3 Elements

In SolidWorks Simulation the main element types are truss elements, frame line elements and shell elements. Shell elements with both displacement and rotational degrees of freedom were used for both types of analysis. The two types of shell elements, continuum and curved surface, require precise connection of all single and multilayer surfaces. Shell elements are used in geometries where the thickness is significantly smaller than the other dimensions. Membrane shells have only displacement degrees of freedom (DOF) and will not give an accurate result. Solid elements have a tetrahedra structure with four vertex nodes and six mid-edge nodes, and use linear or quadratic interpolation, depending on the number of nodes at the edge (Akin (2009)). Shell elements differ from the solids by only being represented by their mathematical surface geometry and thickness. Stresses at shell elements are only reported at the edges of the element, and are often opposite. The element then show the resultant of membrane and bending stresses, as shown in Figure 5.4.

Curvature based meshing supported all types of surfaces, both volume based and multi-threaded surface for multi-body parts and assemblies (SolidWorks(a) (2012)). Continuum shell elements was used in this analyses. The thickness was then determined from the element nodal
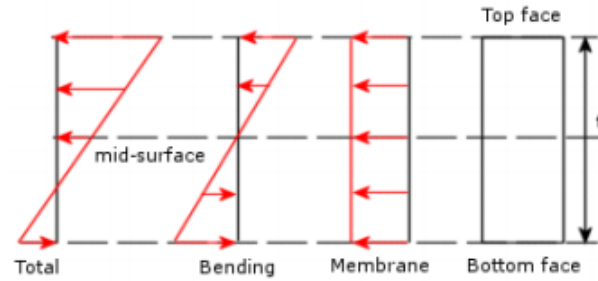
Figure 5.4: Shells in-plane stresses and transverse bending (Akin (2009))

geometry and the entire three-dimensional body was discretized. From a modeling point of view continuum shell elements look like three-dimensional continuum solids, but their kinematic and constitutive behavior is similar to conventional shell elements.

5.4 Boundary Conditions and Load Cases

Applying inaccurate boundary conditions may be a large source of error, and required a thorough evaluation prior to the analyses. The simulated vehicle is submerged in water and only affected by external, uniform pressure.

Figure 5.5: Z translation and symmetry boundary condition

The symmetry boundary condition was applied at the surfaces which had a similar part connected to it in full scale. To simulate a perfect connection between the compartments, and ensure that the model was not under-restrained, the pressure vessel was restricted against any

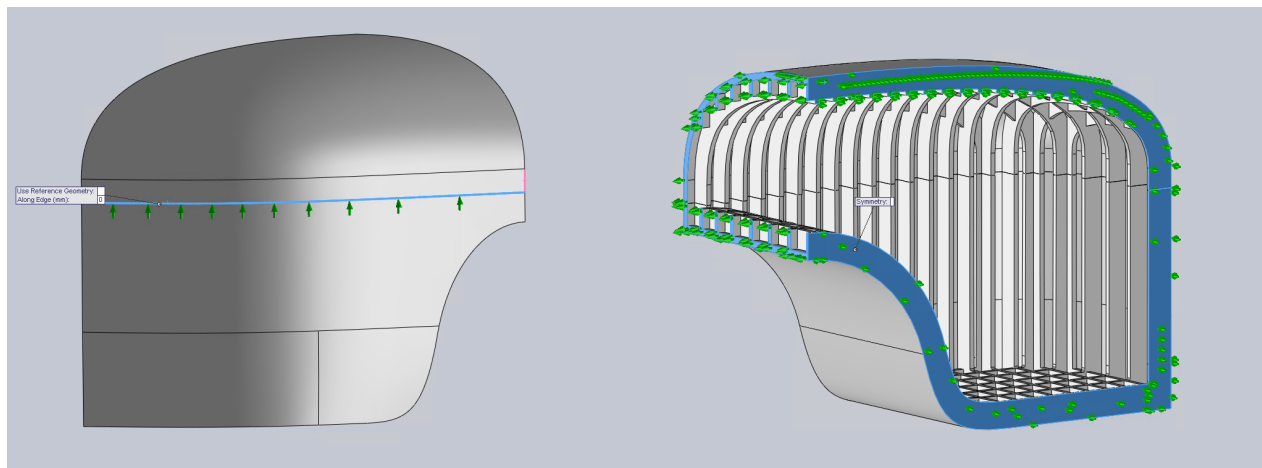z-translation in the intersecting boundary layer between the two compartments. Boundary conditions was applied as in Figure 5.5 when analyzing the pressure vessel.

Symmetry is an analyzing-tool that takes advantage of the model symmetry. This reduced computational time in stress and strain analyses. The option was only applicable for flat faces when the geometry, restraints, loads and material properties were symmetrical. Symmetry prevented the coincident face from moving in its normal direction. For shell elements the faces are also restricted against rotation about the other two orthogonal directions. It was important to note that the symmetry boundary condition could not be used for buckling analyses. Symmetry required that the structure with symmetrical restraints and loadings would respond symmetrically, which is not always the case for buckling analyses (SolidWorks(a) (2012)). Therefore a larger part of the model was used for buckling analyses, and no symmetry boundary condition was applied. Boundary conditions was otherwise similar to the quarter-model, to ensure the model not being under-restrained. The center face, which had the symmetry boundary condition in the static analyses, was restricted against x- and y-translation, in addition to rotations around all axis.

An uniform pressure force of 1.1 MPa was applied normal on all external surfaces, as seen in Figure 5.6. An additional 10% was added to the pressure at the design depth of 100 m depth, to account for variations in the hydrostatic pressure and ensure that the results were conservative.
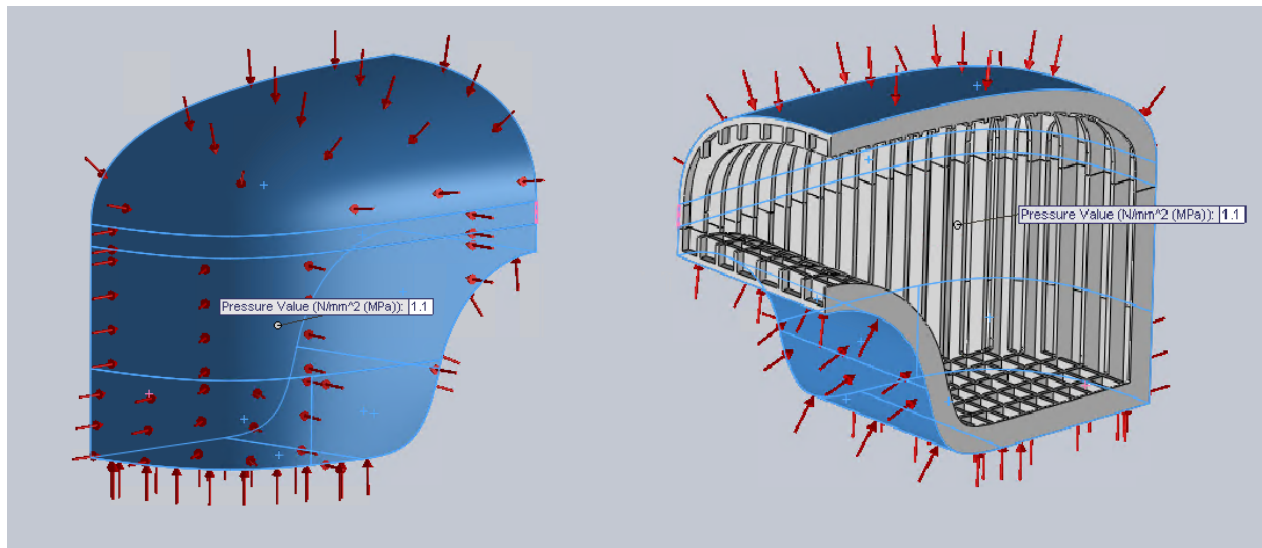

Figure 5.6: Applied pressure on outside and inside of the pressure vessel

5.5 Solution Method

Analyses in SolidWorks were performed with direct integration methods, which solved equations using exact numerical techniques. SolidWorks allows the user to select the solver method prior to running any analyses. Based on the displacement formulation of the finite element method, the component displacements, stresses and strains were calculated under external pressure forces.

Material failure and structural instability, also known as buckling, were the two leading categories of sudden failure in the pressure vessel. For elastic materials the yield stress was used as the failure limit, while for brittle materials the ultimate stress was used. This was due to the limited elongation for brittle materials. Buckling loads did not directly depend on the material strength, but rather the stiffness of the component. Loss of structural integrity and buckling may occur before the material yield or reaching the ultimate stress limit. These two failure modes were obtained by performing separate analyses, with different set up and solution methods.

5.5.1 Linear Stress Analysis

Static analyses were performed to give an indication of the structural capacity of the hull and an indication of when yield occur. Linear stress analyses in SolidWorks enabled validation of the pressure vessels performance, while creating the design. Static solvers can also be used for models with different material properties and mixed-mesh models, which could be the case if additional stiffening devises of another material is added. It was required that all surfaces were bonded and no components were penetrated.

All plastics were first evaluated in their linear elastic area, where the geometry would return to its original shape after the loads were removed. A static analysis calculates the unknown displacements using Equation 5.1. It was assumed that the the product loading was static, and displacements, δ, small enough to ignore the change in stiffness, K.

$$F = K \times \delta \tag{5.1}$$

Corresponding stresses were then calculated using the relation, and further compared to the maximum allowable stresses. If the design stresses were exceeded it was assumed that material

failure has occurred. This made the static solver suitable for the initial pressure vessel analyses. Generally, when the stresses have exceeded the material yield, or ultimate stress, the plastic will break or get elastic-plastic deformation. If the total behavior of the hull would be of interest, a non-linear analyze is required. In a non-linear analyze it is assumed that the deflections are large and stresses above the yield stress, and further assumptions and basic aspects of a non-linear analyze are covered in Section 5.5.3, which assumes that

5.5.2 Buckling Analysis

Analyses related to buckling becomes relevant when stiffened plated are subjected to forces normal to the plane, and is referring to sudden large displacements due to axial loads. The pressure vessel are subjected to pressure normal to every external surface, which creates axial stresses throughout the hull thickness. The buckling capacity depends on the component stiffness and not just material properties, as in static analyses (Leira (2014)). After satisfactory results from the static analyses were achieved, buckling analyses were performed to calculate the failure load of the slender pressure vessel under compression. Failure is characterized by loss of structural stiffness related to different modes. The pressure vessel had areas which were stiffened in two directions, and some with only one stiffened direction. Three characteristic buckling modes were present in the pressure vessel:

1. The plate buckled between the stiffeners. Known as local buckling

2. The plate buckled together with the stiffeners in one direction

3. Simultaneous buckling in both directions

Buckling analyses follows the eigenvalue solution in Equation 5.2, where K_f is the model geometric stiffness and λ_m the loading factor for mode m. The loading factor, λ_m, gives an multiplier-factor that indicates how much the external loads must be scaled for buckling to occur. A positive buckling factor above 1 indicates that buckling would not occur at the given load level.

$$\delta_m \times |K + \lambda_m \times K_f| = 0 \tag{5.2}$$

Sudden deformations will occur when the stored membrane stresses in the structure are converted into bending energy. Buckling is not a result of changed loads, but comes from the axial forces that works perpendicular to thin wall sections. These forces are altering the structural stiffness, and buckling occurs when the membrane forces reaches a critical point where the structure are incapable of supporting any incremental loads (SolidWorks(a) (2012)). Discretization errors in buckling analyses may overestimate the buckling load and give non-conservative results, and it is important to use the mesh sizes from the convergence tests in the buckling analysis.

The solver ran the analyses with the applied load, and returned a buckling loading factor (BLF). This factor gave an indication of how large portion of the current applied load that was necessary for buckling to occur. Buckling factors between 0 and 1 indicated that buckling had occurred before reaching the depth where the pressure was 1.1 MPa.

5.5.3 Non-Linear Analysis

Non-linear analyses becomes relevant if the most basic assumptions of a linear analysis is violated;

- Maximum stresses exceeds the yield stresses

- Deflections are large. That is, if the deflections are larger than the material thickness

- Dynamic effects, or the material experience creeping

- If stress stiffening or softening occurs

The hull stiffness will change as the model deforms. A non-linear analysis would analyze the model in time steps and use its possessed stiffness in each time step. Model geometry, restraints and material properties are deciding how the stiffness matrix, K, are updated at each step, as the model deforms. Materials will behave differently under varying operating conditions. Temperature, water absorption, time and strain are relevant with respect to the structural capacity. A firm definition of the stress-strain curve is required as a minimum in SolidWorks prior to any non-linear analysis.

Unfortunately, the student license available at NTNU did not allow for non-linear analysis of materials with an elastic-plastic behavior, as displayed in Figure 5.7. This put a limitation on the analyses. However, this design study were performed to investigate when the material yield. For the brittle materials used in the linear static analyses the material behavior is not non-linear, and there were small elongations before yield or break. Accuracy of the result may be slightly affected by not performing non-linear studies, but the linear static analyses gave a good indication of when the material was expected to yield.
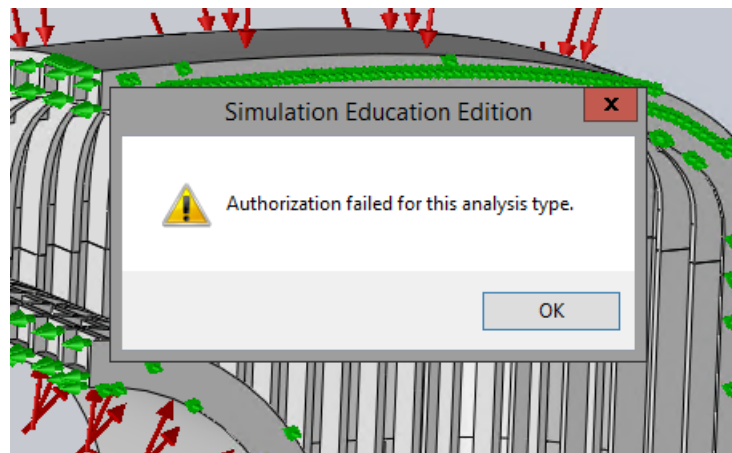

Figure 5.7: Error message when performing non-linear analysis with an elastic-plastic material

Chapter 6

Results

To investigate how pressure forces affected the pressure vessel, static and buckling analyses were performed using the software SolidWorks Simulation. The analysis methods used are presented in Chapter 5, while the model is described in Chapter 4. The three plastics selected in Chapter 3, PA66 30% GF, ABS 30% GF and PC / ABS 30% GF was used in the analyses. Ribs and stiffeners were designed with maximum capacity, to give an indication of the vessels structural performance.

The pressure vessel had two different compartments that was modeled and analyzed separately, before joined together as an assembly. Figure 6.1 shows the front compartment to the left and the back compartment to the right.

6.1   General Pressure Vessel Capacity

The first analyze was done for the front part of the pressure vessel, with ABS 30% GF. The front part was assumed to be the strongest compartment due to the large global radiuses and no prominent areas where stress concentrations could occur. Results from ABS 30% GF had too high stresses present, and proved to be unsatisfactory for the applying in the pressure vessel. ABS 30% GF was therefore not used for any further analyses. The ABS/PC 30% GF blend proved to be better, but also had stresses above yield and large displacements for relative low pressures. It was therefore decided to use PA66 30% GF, or Ultramid A3E6G, when performing analyses. Due to the high amount of absorbed water by the Ultramid A3E6G, all analyses were performed
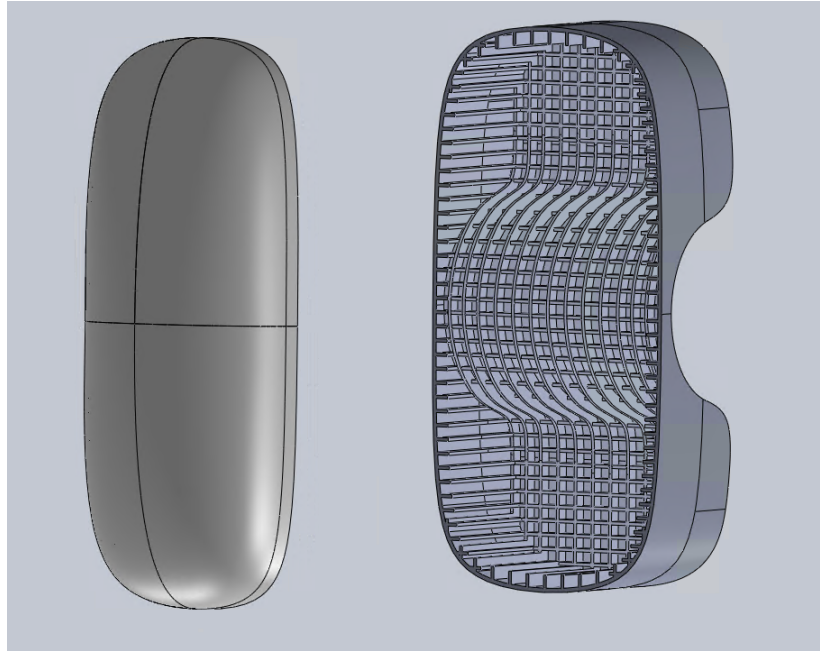
Figure 6.1: Front and back compartment of the pressure vessel

with both a dry and conditioned material. A conditioned polymer have a lower elasticity module and yield strength, as explained in Section 3.3.1. The plastic becomes fully saturated after several hours in the water, dependent on the material and reinforcement grade. As the battery capacity currently is limited to a few hours this would not be a concern for the vehicle at this point. For future applications the vehicle may have solutions that allows it to remain submerged for a longer amount of time. Thus, the structural analyses were performed for both a conditioned and dry Ultramid A3E6G, in order to indicate the vehicles best and poorest performance possible, with this plastic. Ultramid A3E6G will be referred to as Ultramid in the following sections.

Injection molded parts has recommendations with respect to thicknesses and corner radii, and maximum radius of the filets is limited to 0.25 times the nominal thickness (see Figure 4.8 in Section 4.3.2). Small filet radiuses and sharp corners, especially for the back compartment, led to large stress concentrations. Stress concentrations are further discussed in Section 6.2.1 and Section 6.2.2. Deformations were largest at the centre of both compartments, and were symmetric about the (x,z)-axis. Displacements are further covered in Section 6.2.3 and Section 6.2.4.

The different depths which the pressure vessel failed at, either in yield or in buckling, are listed in Table 6.1. Depths are an estimation based on the applied pressure, where 1 MPa corresponds to 100 m depth.

Table 6.1: Failure depths for PA 66 30% GF

	PA 66 30% GF Dry	PC/ABS 30% GF Conditioned
Static failure depth	36 m	24.5 m
Buckling failure depth	53 m	37 m

6.2   Static Analysis

Figures and results from the static analyses are displayed for PA 66 30% GF. The results indicated that the material might be strong enough to withstand the pressure at 100 m depth, but the displacements proved to be too high, with respect to the maximum volume reduction. Linear theory was used for the static analyses, where displacements was assumed small, and the stiffness of the model remained constant during the loading. The loading itself was applied instantly, rather than increasing for each time steps. Static, small displacement analyses were recommended as a first step in such simulation procedure. Non-linear (large displacement) analyses should have been performed if stresses were allowed to be close to, or exceed, the yield stress, strains allowed to exceed 4% material elongation or if there were noticeable mis-orientations of contact forces in the model (Simulation (2015)).

Both the front and back cover of the pressure vessel were analyzed separately, before connected in an assembly. The connection between the two compartments was assumed to be perfect, and strong enough to withstand the shear forces present.

6.2.1   Stresses for Conditioned PA 66 30%GF

Figure 6.2 shows the stress distribution in the front compartment for a conditioned PA 66 30%GF material. Maximum stresses were below the yield stress of 130 MPa, and there was no pronounced stress concentrations. The back part of the pressure vessel had stresses closer to the yield stress. The stiffeners were designed to its fully potential, and the stresses could not be reduced by modifying the stiffeners to any extent.

Distinct stress concentrations was present for the back compartment at the stiffener filets, as seen in Figure 6.3. These stress concentrations were present due to sharp corners and transitions, which came as a result of the outer geometry design. A well rounded geometry, like the
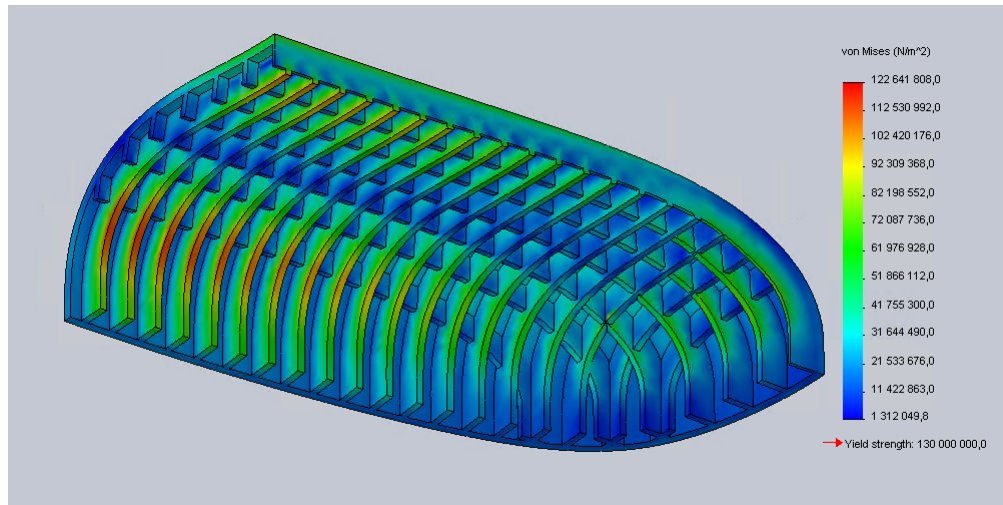
Figure 6.2: Stresses at front compartment. PA 66 30%GF Conditioned

front compartment, would distribute the stresses better. However, the compartment was designed to fit the thruster placements and was restricted against modification. Some of the high stresses also came from displacements of the walls. There were no horizontal stiffeners at the walls, due to production considerations, and the walls did not have sufficient support against horizontal displacements. To create horizontal stiffeners, additional tooling and equipment are needed.
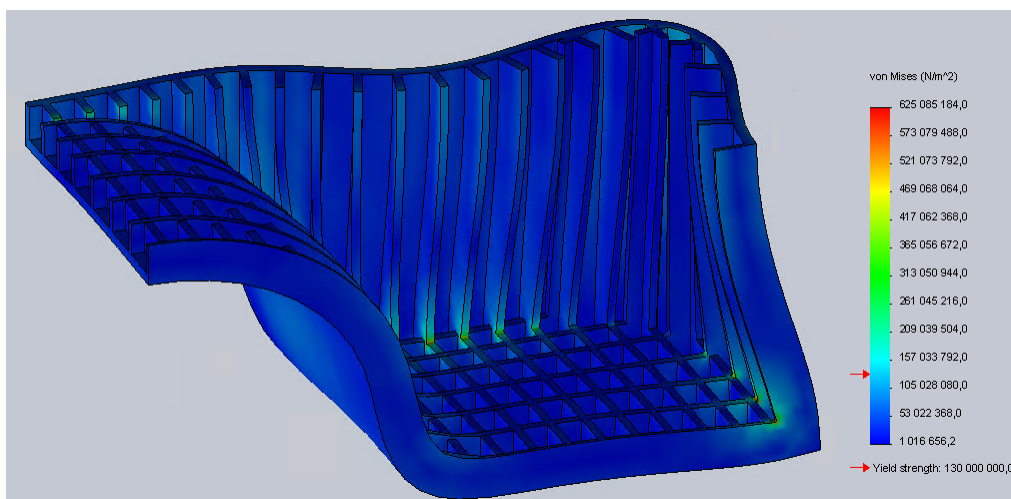

Figure 6.3: Back compartment stresses with 0.75 mm fillet radius

By increasing the fillet radius from 0.75 mm to 3 mm the maximum stresses were reduced

by 25%, as displayed in Figure 6.4. It is not recommended to apply such large filet radii, due to production considerations. Other solutions to decrease the stress concentration should be investigated.
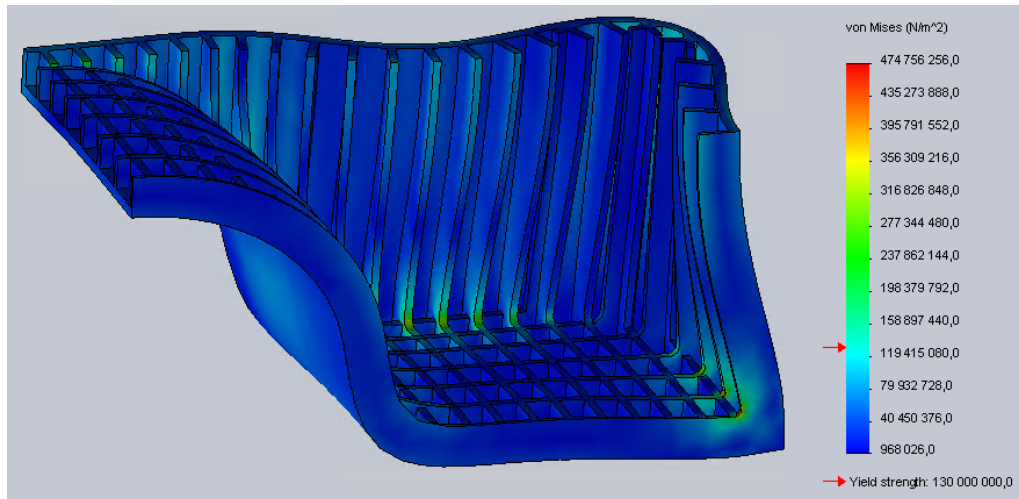

Figure 6.4: Back compartment stresses with 3 mm fillet radius

6.2.2 Stresses for Dry PA 66 30%GF

A dry Ultramid has a higher yield and tensile stress, and a noticeable higher elasticity module, than in conditioned state. Stress patterns were similar for the conditioned and dry polymer, but the magnitude was lower for the dry vessel. Higher elasticity module would result in smaller displacements, and less stress concentrations from the deflecting walls. By comparing the stresses in Figure 6.5 and Figure 6.6, the location of maximum stresses were the same in dry and conditioned state.

For a dry polymer the maximum stresses were 12% lower than in conditioned state at 110 m depth, but due to larger displacements for lower pressures, the yield and tensile stress limit were exceeded faster in conditioned state.

6.2.3 Displacements for Conditioned PA 66 30%GF

Displaying the displacements at 100 m depth would not be entirely correct by using small displacement solution. Small displacements solutions use linear material relationships to calculate
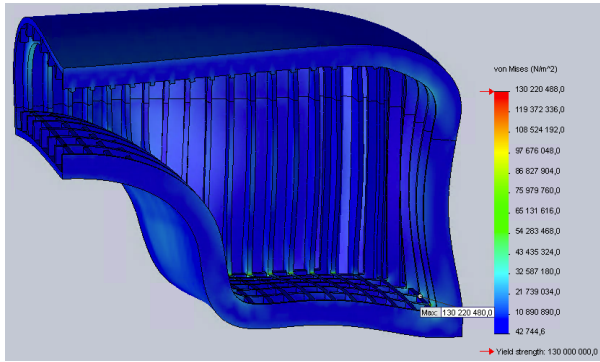
Figure 6.5: Conditioned assembly stress distribution at failure depth (24.5 m)
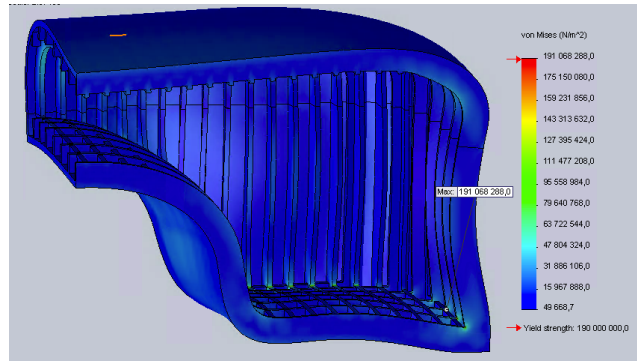
Figure 6.6: Dry assembly stress distribution at failure depth (36 m)

the displacements, and since the material yield stress was reached for 1.1 MPa external pressure, the results could only be used as an indication of the areas where there were significant displacements. Figure 6.7 shows the displacements at the failure depth for a conditioned Ultramid.
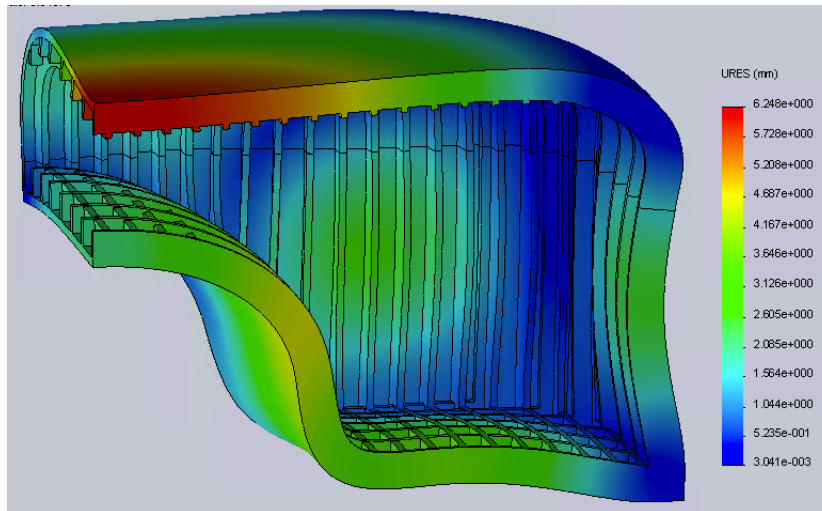
Figure 6.7: Conditioned assembly displacements at failure depth (24.5 m)

Maximum displacements were located at the middle section of the front compartment, due to the lack of structural support at that area. No horizontal stiffeners were applied at the side wall of the back compartment, which resulted in large displacements at those locations. Displacements for separate compartments at 100 m depth, can be seen in Figure 6.8 and Figure 6.9. As the displacement was much larger than the thickness of the hull, non-linear analyses may give a more correct displacement at 100 m depth.
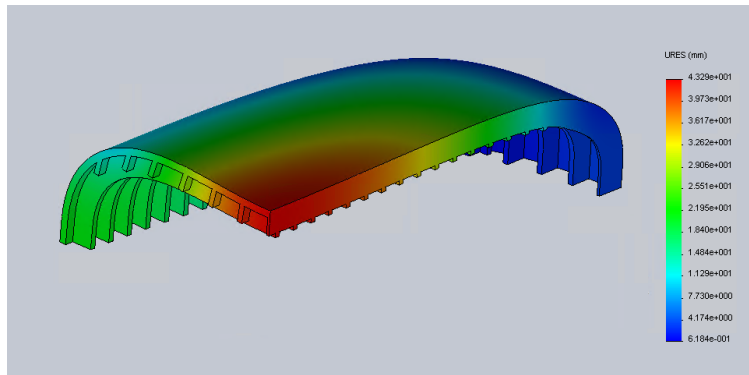
Figure 6.8: Front cover displacements for a conditioned PA 66 30%GF at 100 m depth
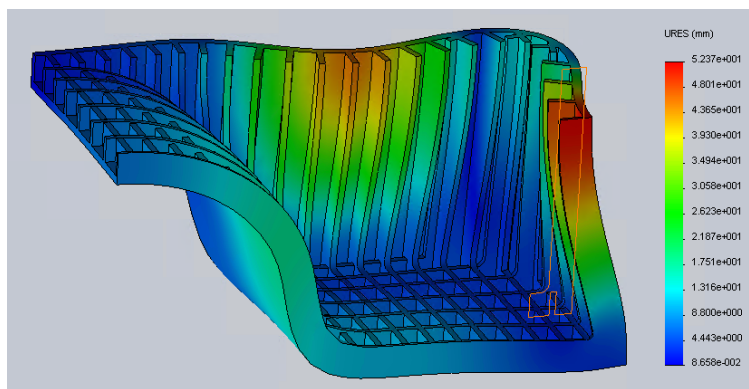
Figure 6.9: Back cover displacements for a conditioned PA 66 30%GF at 100 m depth

6.2.4   Displacements for Dry PA 66 30%GF

As for the conditioned polymer the displacements were large when high pressures were applied. The displacement pattern for the dry plastic was the same as for the conditioned, but generally smaller displacements were seen, due to the higher elasticity module. Analyses were performed to find the depths where yield stress was exceeded, which was before the design depth of 100 m. Figure 6.10 shows the displacements at 36 m, which is the failure depth for a dry Ultramid. At the failure depth for the dry polymer the maximum displacement only differed 0.3 mm from the conditioned polymer. This may indicate that the yield stress limit was reached when the pressure vessel was compressed to a certain point, where the stress concentrations became too large.
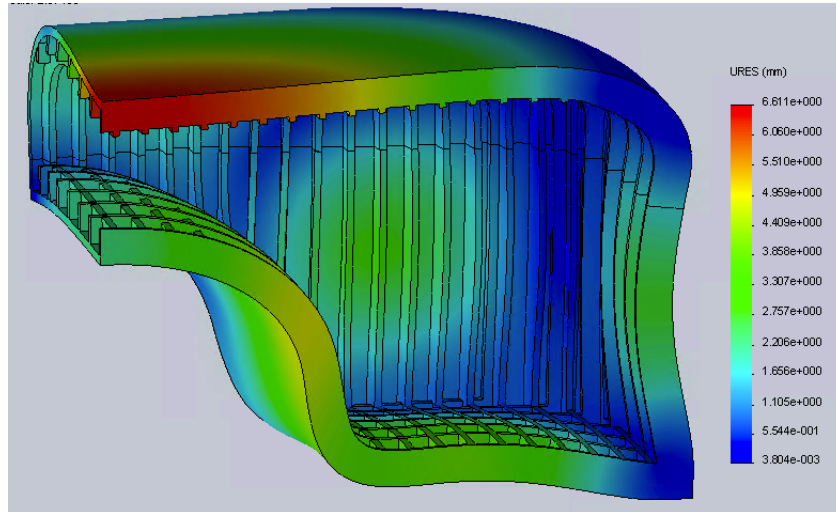
Figure 6.10: Dry assembly displacements at failure depth (36 m)

6.3   Buckling Analysis

Buckling analyses were performed for the pressure vessel to find the ratio between the applied pressure and when buckling occurred. Analyses returned a buckling safety factor, or buckling loading factor, which was used as an indicator to see how deep the vessel could operate before the plate, or stiffeners, buckled. An explanation of buckling related to the buckling safety factor (BSF) can be seen in Table 6.2. The vehicle was analyzed with a external pressure similar to 110 m depth..

Table 6.2: Buckling safety factor explanation (SolidWorks(b) (2016))

Safety factor (BSF)	Buckling Status	Notes
1 < BFS	Buckling not predicted	Applied loads are less than the estimated critical loads. Buckling is not expected.
0 < BFS < 1	Buckling predicted	The applied loads exceed the estimated critical loads. Buckling is expected.
BFS = 1	Buckling predicted	The applied loads are exactly equal to the estimated critical loads. Buckling is expected.
-1 < BFS < 0	Buckling not predicted	Buckling is predicted if you reverse all loads.

To find the buckling loads, the safety factor was multiplied by the applied pressure, which

was 1.1 MPa. For a conditioned PA 66 30%GF, the buckling analyses indicated that buckling would occur with a buckling load factor of 0.3365. Both the stiffeners and the plate was deflected in buckling mode 1, as shown in Figure 6.11. This load factor corresponded to an external pressure of 0.37 MPa, which is the pressure at 37 m depth.
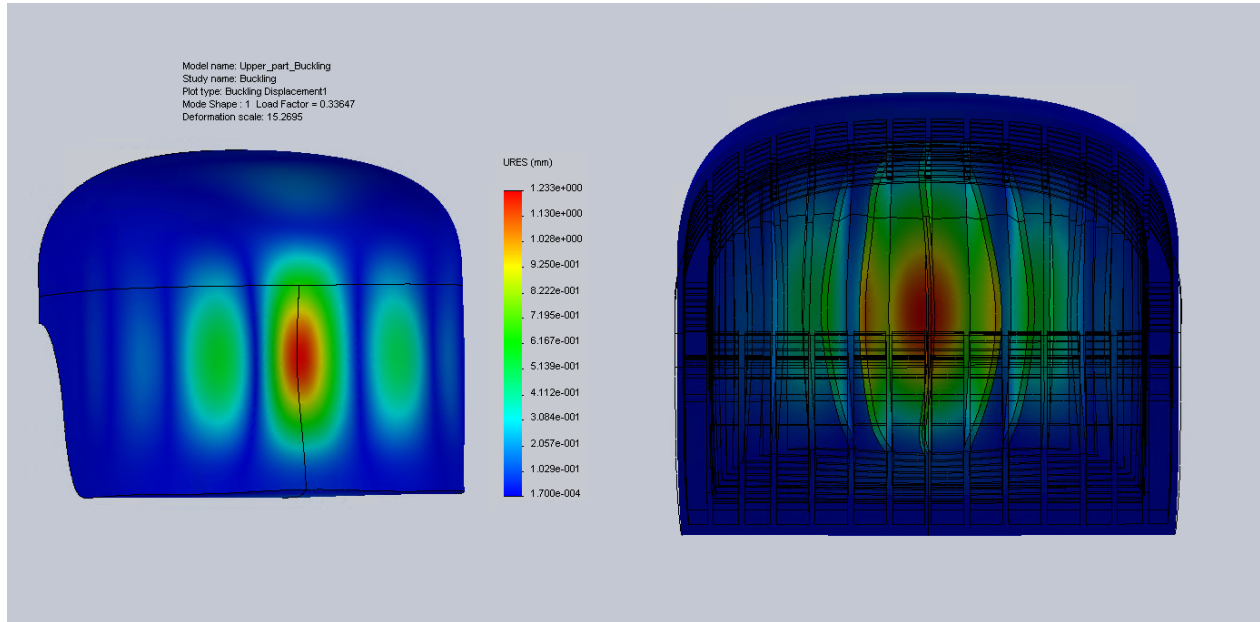
Figure 6.11: Result plot for buckling analysis of a conditioned PA 66 30% GF

A dry polymer, subjected to the same pressure, had the same failure mode, but a different loading factor. Ultramid in dry condition was more rigid and buckled for larger pressures than when fully conditioned. The buckling load factor was 0.4818 for a dry material, which equals the pressure at 53 m depth. From this, the results indicated that the model did not meet the required design criteria for buckling, neither for a dry or conditioned polymer.

6.4 Connections and plastic joining

The pressure hull is going to be produced in two parts, which require two different molds. Connection between the two compartments is an important aspect of the project, as it is required to be completely pressure resistant and waterproof. In the analyses, a perfect bonding was created between the compartments, as Figure 6.12 shows.

Significant forces will be transferred through this connection surface, especially with respect
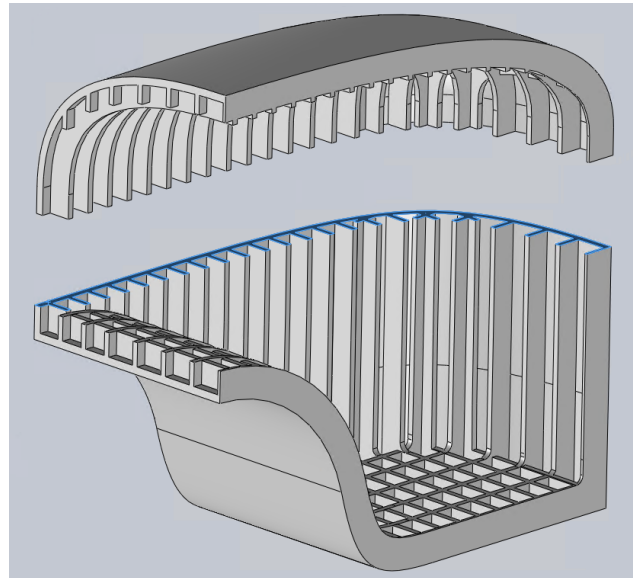
Figure 6.12: Surfaces that was perfectly bonded in the analyses

to handling of the vehicle and collisions. Shear forces in the connection area would bend the hull, as in Figure 6.13, and the resulted displacements have to be kept in place by the plastic joining.
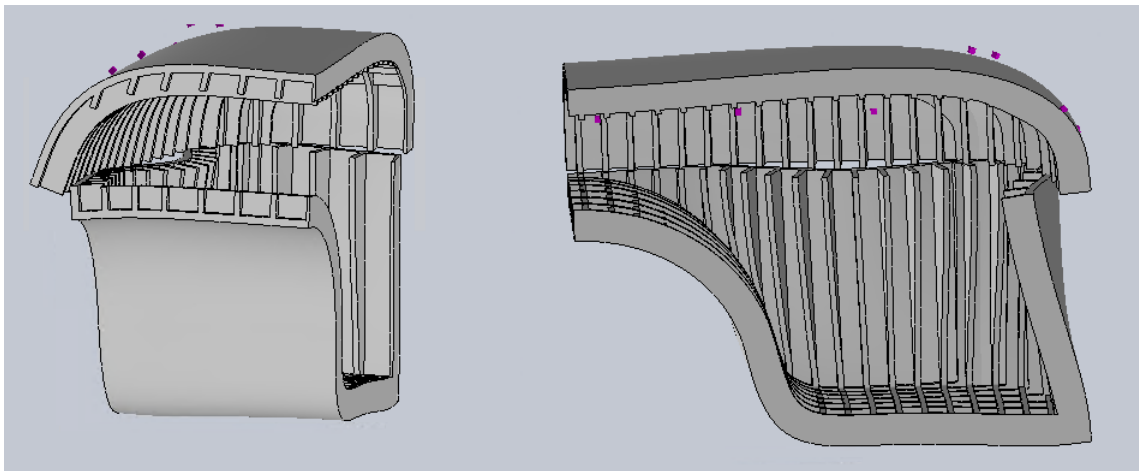
Figure 6.13: Displacements without any surface bonding applied

Prototype P1 was not produced by injection molding and had mechanical connections on the outside of the hull. These machined connections are difficult to produce by injection molding, as there are restrictions on the maximum recommended thickness, boss radii and undercuts. Inside the 15 mm thick prototype hull, there are tracks for the o-ring seal, which was essential for making the hull pressure resistant. An injection molded part with a 3 mm wall thickness

would not have sufficient surface area to create such tracks. If the current connection was a perfect adhesive or solvent bonding, it would generally have been a bit weaker than the base material, and the results from analyses would have been different. The challenges lies in joining the two compartments together without additional devices or including additional manufacturing procedures, by using some of the techniques described in Section 4.7.

Chapter 7

Summary and Further Work

7.1 Discussion

Selecting the most appropriate material for use in the pressure vessel is a process which have no absolute solution. There are a high amount of different types polymers, with different combinations of reinforcements, and their price and properties would vary between producers. Material data sheets often show the maximum and minimum values of many properties, and two PA 66 30% GF plastic batches, from two different producers, may have different properties. Values used in this thesis were the mid point of these max/min values, as it is expected that the plastic provider are well known and delivers high quality products. The PA 66 30% GF polymer used in the analyses proved not to be strong enough to meet the design requirements. This material was expected to perform better than it actually did. Selecting a material with better mechanical properties would most likely have increased the performance of the pressure vessel. However, the various materials would possess different productional limits and tolerances, and the benefit of selecting a stronger material has to be seen in connection with the effective thickness. If the pressure vessel is designed properly in accordance with restrictions for injection molding, the hull capacity would not automatically increase by selecting a stronger material. As the work process flow chart shows in Figure 4.15, this design and iteration process should be done all over again with an even stronger material, and with a new stiffener layout. A new material with better properties would most likely be more expensive, and it is recommended to change other aspects of the design before selecting a new material.

In this thesis there were limitations for the production methodology and global geometry. The model is simplified to such extent that most of the details are left out. When creating spacings to fit in camera and lighting in the front compartment, the deflection and stress pattern may show some changes compared to the performed analyses. When machining details, or adding material for design considerations, in the front compartment, the hull would stiffen and be better able to resist deformations.

The results obtained are conservative, but still, some fundamental design changes should be included in order for the global geometry to avoid large stress concentrations. Limitations connected to the production tolerances are conservative, but not absolute. Manufacturers and producers may give other recommendations, and there might be possibilities for increasing the limits to some extent. These tolerances have to be discussed with the production engineer at the manufacturing facility. This is the process of mold quoting, which is recommended to do before, and while, creating the injection mold design.

The connections between the front and back part were assumed perfect in all analyses. BluEye have had some challenges with gluing certain areas of the P1 prototype, and have expressed their skepticism towards glued solutions. Using a suitable adhesive bonding would create a strong bond between the two compartments. If BluEye does not desire other solutions than an o-seal and mechanical joints, the design process would be more challenging than for solvents or adhesive bonds. Creating tracks for the o-seal at the two connecting surfaces would require a larger surface area than the current 3 mm walls thickness. Bosses to fasten the bolts cannot be injection molded on the outside of the hull, but could be created after the molding process. However, since such bosses would require a thickness larger than the maximum recommended nominal thickness, the use would most likely lead to flaws, and reduced material strength. Using adhesive and solvent bondings would require additional connecting surfaces, e.g a joggled joint or an overlapping material. In that case, the connection area should most like be re-designed, to ensure a large enough surface area for the bonding to gain sufficient strength. It is then important that the adhesive bonding is compatible with the material itself, and that the design do not allow for undesired load states, such as cleaving and peeling loads.

This study was performed for a relatively brittle material with tensile strength as the design limit, and the ultimate stress was met after 3% strain. Until the yield stress is reached, the ma-

terial behavior is linear. The total deflections in the linear analyses was larger than the nominal thickness, which was one of the criteria where non-linear analyses should have been performed. By performing non-linear analyses the effect of increasing structural capacity from strain hardening is included. As the material deforms it will appear stronger, due to this strain hardening, and this effect is not included in a linear analysis. One of the objectives of the thesis was to investigate the possibility for operating the modified pressure vessel at 100 m depth. From the linear and buckling analyses, it could be stated that the results gave sufficient indications of the pressure vessels ability to operate at 100 m depth. The failure depths may change if the entire material model was included in a non-linear analyze. Hence, to investigate the current design further, non-linear analyses may be appropriate. However, the SolidWorks Student Editions has limitations on the ability to run non-linear analyses with a non-linear material model.

7.2 Conclusion

As BluEye is aiming to create a mass produced remotely operated vehicle for the consumer marked, that can be used everywhere in the world, a lot of challenges arise. Operating in different environments with different external conditions will have variable impact on the material performance and loading conditions. How the user handle the vehicle itself is a difficult aspect that need to be accounted for when designing the vehicle. To make the process cheaper, BluEye want to investigate the possibility for producing the ROV by using injection molding. Injection molding is a commercially well known method for plastic mass production, and will lower the production costs compared to machining, when manufacturing large quantities. However, the technique comes with a number of limitations and restrictions connected to material use and the design. The current prototype is created with a 15 mm thick ABS plastic hull, but most plastics that are injection molded requires a nominal thicknesses below 5 mm.

Selecting a suitable material for this purpose, is a comprehensive process that must be based upon many different selection criteria and decision gates. The selection process aimed towards a material suitable for production by injection molding, with appropriate physical properties and low raw material price. When designing a vehicle for a broad geographical range, the material should be evaluated based on how the properties varied with the environmental conditions.

As the temperatures ranged from 2°C and 32°C in the world oceans, the design should include all the material properties that varied with temperature. Thermoplastics are most commonly used in injection molding, and glass fiber reinforcements had a cost advantage over carbon fibers. Many different polymers are available, with an even larger range of possible reinforcements, and the main weighting criteria for the pressure vessel are density, thermal behavior, mechanical properties, water absorption and price. These criteria are not absolute, and change with the scope of the project. Polyamide 66 (PA) with 30% glass fibers was chosen as a suitable material for the pressure vessel. PA66 30%GF possess good structural properties, low price and were easy to manufacture. A disadvantage by selecting a polymer with oxygen groups was a relatively high amount of absorbed water. Due to this, the analyses had to be performed for both a dry and fully conditioned material.

Injection molding is a process where the polymer are heated before injected into a mold and kept in place over time, before cooling and ejected. Designing a part for injection molding require care in the dimensioning, and there are a number of design regulations that should be followed for the part to behave as designed. As the heated polymer are injected into a mold, it requires a smooth flow that fills all the corners and cavitation of the mold. If the design of the pressure vessel do not meet the requirements of injection molding it could lead to non-proper filling of the mold and imperfections in the produced part. Typical imperfections and flaws related to injection molding are sink holes, warping and deforming of the geometry, which could occur if the dimensions are too big, or improperly related to each other. A recommendation from plastic manufacturers are to have a nominal thickness of 2.92 mm for the PA 66 30%GF polymer. Stiffeners and girders are recommended to be designed in relation to the nominal thickness. There are also limitations for the filet radius, stiffener height and width, distance between stiffeners and how undercuts and bosses can be designed.

When the cooling process are finished the parts are pulled from the mold. To avoid use of additional, special equipment, all stiffeners should be aligned with the pulling direction. Following all design regulations are crucial in ensuring proper performance of the end product. Some aspects were fixed throughout the design phase, and could not be changed. The global geometry and production methodology was fixed, and only material selection, and the corresponding stiffener design, had an impact on the design of the ROV. This led to limitations for maximizing

the structural capacity, and results indicated that the total performance of the vehicle could be improved by doing small changes in the global geometry, before ribbing the hull.

The main goal was to create a vessel capable of operating down to 100 m depth. The selected polymer was more than four times as strong as the ABS plastic used for the prototype, but it proved to be too weak to meet all design requirements. Both the static and bucking analyses indicated that there were high stress concentrations present and that the displacements were too large for this to be an approved design. A completely dry PA66 30%GF were able to operate down to 36 m before the material yielded, while in fully conditioned state it was only are able to resist the hydrostatic pressure down to 24.5 m. As the maximum nominal thickness put restrictions on the stiffeners, there were no possibilities of adding more stiffness to the design. After performing all analyses with PA66 30%, both the static stress and displacement analyze results indicated that it was necessary to use a stronger material. Selecting a stronger material would increase the performance of the pressure vessel. It would be necessary to re-design the back compartment to avoid having large stress concentrations, since the radius of fillets were limited to 0.25 times the nominal thickness. Designing the geometry with larger global radiuses would be essential for distributing the stresses over a larger area. The design would also benefit from selecting a material that allows for a larger nominal thickness, but the benefit have to be seen in relation to the effective thickness and possible stiffening solutions. Stiffener spacing, thickness and height are directly connected to the nominal thickness, and a larger nominal thickness would not automatically give a higher hull strength. The front compartment of the pressure hull had a large global radius and the stresses was below the yield stress, even for a fully conditioned polymer. If the back compartment is designed with the same shape as the front compartment, it would still be significant displacements in the hull, and buckling would become a problem at 37 m and 53 m depth, in conditioned and dry state respectively. Since the buckling took place at the back compartment the vessel would buckle at different depths if the compartment was re-designed. Additional, internal stiffening devices, e.g. an aluminum ring in the center of the vehicle, would reduce the displacements but not help against buckling. As the buckling occurs at the sides which are not stiffened in both direction, there might be necessary to invest in additional equipment to produce horizontal stiffeners. This will be a benefit versus investment evaluation BluEye have to do, before creating the next prototype design.

7.3   Recommendations for Future Work

The current pressure vessel have a good looking design, with great functionality. When looking for possibilities for producing the vehicle by injection molding, some aspects are recommended to be re-evaluated. Some aspects of this thesis may be addressed and changed relatively quickly, while other must be seen as more long term objectives. A number of challenges occurred during the process, and especially three objectives are important to address when designing for the next prototype, namely;

1. Find connection solutions suitable for the pressure vessel

2. Changing the pressure vessel design to:

    - Reduce stress concentrations

    - Reduce maximum displacements

    - Design for the selected connection method

3. Evaluate the need for a stronger and more stable material, and the need for additional manufacturing tools

Connections will be a challenge for the next generation BluEye ROV, especially if it is desired to have the possibility to open the vehicle. Adhesive bonding are a strong and applicable technique, but does not allow for disassembly and assembly, without performing a new boding process. Having a small wall thickness makes it challenging to add an o-seal to the connection surfaces, and even more challenging to add mounts for bolts or screws. It is recommended to have additional material on both sides of the connection area, to increase the adhesive bonding area and ensure sufficient strength in the connection. Before designing any new model, a selection of connection method should be done. In that way the design changes, due to the connection solution, may be implemented early in the design process. This will avoid problems to related to connecting the parts in a later design stage.

Results from the analyses gave clear indications that the pressure vessel could not be directly modified and stiffened for injection molding, and perform as the machined prototype. To increase the capacity of the injection molded vessel it would be necessary to change the design,

especially the back compartment. From the analyses results it could be that it is critical to have large global radiuses in the curved sections of the vessel, in order to avoid stress concentrations. This could be done by either rounding off the edges of the current design, or creating a completely different geometry. Rounding the corners may reduce the stresses below the yield limit, but these modifications will not be adequate to prevent buckling in the side walls. The pressure vessel design would likely benefit from reducing the total length of hull. A possible solution is to have two smaller compartments, or using internal reinforcements to stiffen the geometry.

Selecting a new material with a more stable performance and smaller water absorption would also benefit the end product. Compared to the PA66 30%GF, a less water absorbing material will not add additional weight to the vehicle, and neither change its mechanical properties to such extent, when conditioned. Different reinforcement grades and filler types would also benefit the performance of the polymer. It might be required to increase the raw material cost budget, in order to get the desired material characteristics and hull performance. Based on the material selection process, a recommended order to evaluate a new material is:

1. Manufacturing and production considerations

2. Mechanical properties and possible reinforcements

3. Price and economics

4. Thermal properties and water absorption

5. Density

Appendix A

Acronyms

ABS Acrylonitrile Butadiene Syrene

AUV Autonomous Underwater Vehicle

BC Boundary Condition

BLF Buckling Loading Factor

BPF British Polymer Foundation

CAD Computer Aided Design

CAE Computer Aided Engineering

CF Carbon Filled

CLTE Coefficient of Linear Thermal Expansion

DOF Degrees of Freedom

ETFE Ethylene Tetrafluoroethylene

FEA Finite Element Analyze

FEM Finite Element method

FEP Fluorinated Ethylene Propylene

GF Glass Filled

HDPE High Density Polyethylene

HDT Heat Deflection Temperature

IM Injection Molding

IMT Institutt for Marin Teknikk

ISO International Organization for Standardization

LDPE Low Density Polyethylene

NOK Norwegian Kroner

NTNU Norwegian University of Science and Technology

PA Polyamide

PC Polycarbonate

PE Polyethylene

PEEK Polyether Ether Ketone

PET Polyethylene Terephthalate

PETP Polyethylene Terephthalate

PMMA Poly(methyl methacrylate)

PO Polyoxymethylene

PP Polypropylene

PPE Poly(p-phenylene ether)

PPO Poly(p-phenylene oxide)

PS Polystyrene

PSU Polysulfone

PVC Polyvinyl Chloride

ROV Remotely Operated Vehicle

SAN Styrene Acrylonitrile

UV Ultra Violet

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