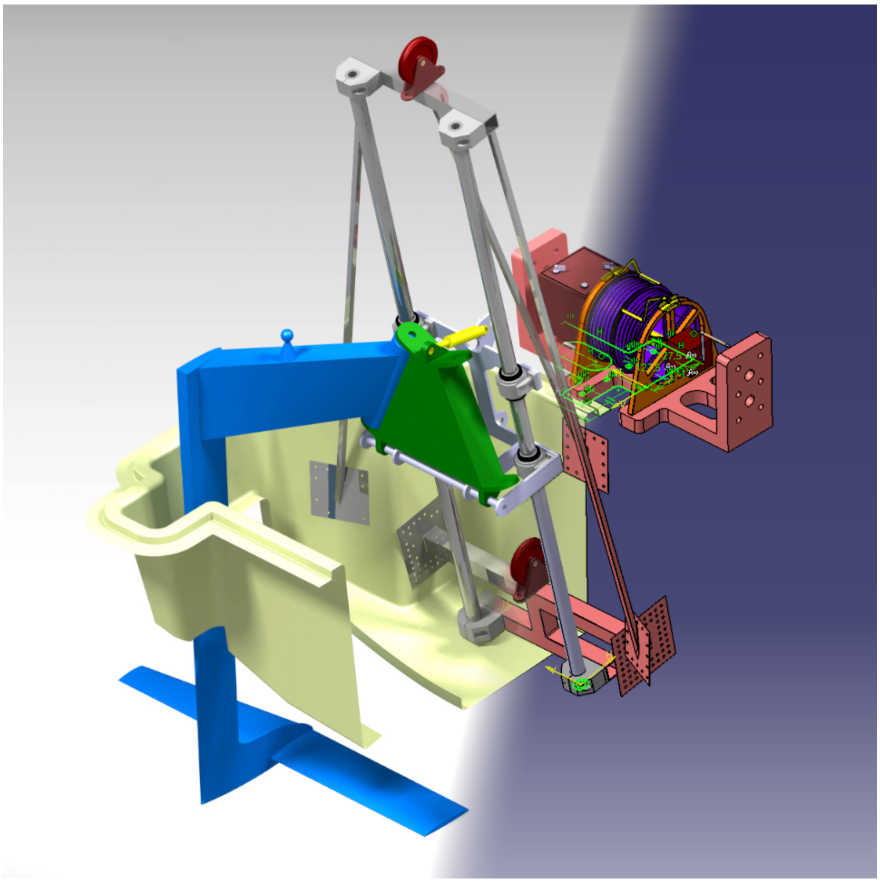

Hydrofoil deployment mechanism for rescue boat

Lightweight and cost-efficient solution to deploy a hydrofoil from a mid-sized boat.

Master's thesis in Product development

FREDRIK BYSTRÖM

DEPARTMENT OF INDUSTRIAL AND MATERIALS SCIENCE

CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2023
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Master's thesis 2023

Product development of a hydrofoil deployment mechanism

A lightweight and cost-efficient solution aimed to be used in the Swedish Sea Rescue Society's next-gen electric rescue boats.

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Department of Industrial and Materials Science
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Supervisor: Professor Johan Malmqvist, Industrial and Materials Science
Examiner: Professor Johan Malmqvist, Industrial and Materials Science

Master's Thesis 2023
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Cover: A rendering of the developed hydrofoil deployment mechanism concept. All parts designed by the student except for: shaft block and linear bearing units (Ewellix 2, n.d.), blue hydrofoil and parts of green hub from Mantaray (A. Sahlin, personal communication, February 8, 2022) and pale-yellow moon pool from SSPA (M. Wikander, personal communication, February 8, 2022).

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Abstract

Product development of a hydrofoil deployment mechanism
A lightweight and cost-efficient solution aimed to be used in the Swedish Sea Rescue Society's next-gen electric rescue boats.
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Hydrofoiling boats are being introduced into commercial usage throughout the world. Being a relatively new phenomena (relative to the industries' more established non-foiling boats) optimal ways of handling and controlling hydrofoils have not yet been discovered or established. This thesis aims to investigate a possible solution to the handling (restricted to managing lift and drag forces generated by a hydrofoil) and controlling of hydrofoils (restricted to deploying, retracting and controlling trim). The context in which this has been performed is on a moderately sized 2.6 ton and 8 meter long rescue boat from the Swedish Sea Rescue Society, aimed to be used in lakes and coastal areas in Sweden.

Possible solutions have been investigated by following a typical new product development process between the phases of establishing requirements up until early detail design. Results include proof-of-concept CAD models that have been evaluated using FEA simulations. Findings suggest that a winch-based solution using HMPE-fiber synthetic rope is optimal as a means to lower overall weight. Furthermore, a support structure that allows for vertical linear movement whilst being exposed to lift- and drag forces from the hydrofoil is needed. Future work would entail proceeding with detail design with a focus on design-for-manufacturing and performing further stress- and weight optimizations.

Keywords: Hydrofoil, Engineering design, Swedish Sea Rescue Society, Boat, Seafaring vessel, Mechanics, Winch

Sammanfattning

Bärplansbåtar håller i skrivande stund på att introduceras till kommerciellt bruk runtom i världen. Då bärplan är ett relativt nytt fenomen (relativt till industrins mer etablerade icke-bärplansbåtar) så har optimala sätt att hantera och kontrollera bärplan inte ännu upptäckts eller etablerats. Det här examensarbetet syftar till att undersöka möjliga lösningar till hantering (avgränsat till att hantera lyft- och dragkrafter från bärplanet) och kontrollering (avgränsat till nedsänkning, upphöjning och trimstyrning) av bärplan. Sammanhanget för en sådan mekanism är att användas på en måttligt stor båt på 2,6 ton och 8 meter från Svenska Sjöräddningssällskapet, ämnad att användas på sjöar och i kustområden.

Möjliga lösningar har undersökts genom att följa en typisk produktutvecklings process mellan faserna ”etablera kravspecifikation” upp till tidig ”detaljdesign”. Arbetet inkluderar konceptvaliderade CAD modeller vars design har blivit utvärderad med finita elementmetodssimuleringar. Resultatet tyder på att en vinschbaserad lösning med lättvikts HMPE-fiber syntetiskt rep är optimalt. Dessutom behövs en stödjande struktur som tillåter vertikal linjär rörelse samtidigt som den motstår belastning från bärplanets lyft- och dragkraft. Framtida arbete skulle innebära en fortsättning på detaljdesignen med ett fokus på anpassningar till produktion och ytterligare spännings- och viktoptimeringar.

Nyckelord: Bärplan, Ingenjörsdesign, Svenska Sjöräddningssällskapet, Båt, Fartyg, Mekanik, Vinsch

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Alexander Sahlin - *Mantaray - Co-worker*
for your help during HDM meetings

Fredrik Byström, Gothenburg, September 2022

Terminology and abbreviations

Bow	The front part of a boat.
CFRP	Abbreviation for carbon fiber reinforced polymer.
Chunk	Product development methodology to refer to a system within the product. A chunk consists of several modules.
Client	SSRS, the industrial contact who requested this thesis.
Deployment	When the hydrofoil is deployed into the water from the hull, i.e. moving downwards.
FoS	Abbreviation for factor of safety.
HDM	Abbreviation of hydrofoil deployment mechanism.
Hydrofoil	A wing-system that is submerged into water which pushes the entire (or most of) the boat's hull above water-level, reducing or completely eliminating the water-drag in favor of less deleterious air-drag.
Module	A system within a chunk. For example, within the chunk ”frame” we have the module ”bracing”.
Moon Pool	The chunk SSPA is providing as an interface unit between the HDM and the rest of the ship's hull.
OEM	Abbreviation for original equipment manufacturer. Refers to company that designs and sells components to be used within another product, e.g. SKF with their bearings is an OEM to Volvo Trucks.
Portside	The left side of the vessel's nominal traveling direction.
Retraction	When the hydrofoil is rectracted from the water into the hull, i.e. moving upwards.
SSRS	Swedish Sea Rescue Society – Commonly known in Swedish as ”Sjöräddningssällskapet”.
Starboard	The right side of the vessel's nominal traveling direction.
Stern	The back part of a boat.
Struts	The two vertical beams that connect the hydrofoil's wings to the hydrofoil's horizontal beams.
Wing	The horizontally-aligned plane at the bottom of the hydrofoil that's submerged in water.
X8	Development name for the new hydrofoiling boat that is in development for SSRS.

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1
Introduction

This chapter will present a background to the thesis project, presenting the concept of hydrofoiling and the specific boat the hydrofoil deployment mechanism will be implemented in. Additionally, the thesis aim, research questions, delimitations, problem analysis and thesis outline will be presented.

1.1 Background

The Swedish Sea Rescue Society, hereby abbreviated as SSRS, is a non-profit organization consisting of roughly 2 400 volunteers, 260 rescue vessels, 74 rescue stations and 143 000 members, and are at the time of writing involved in over 90% of all sea rescues in Sweden (Svenska Sjöräddningssällskapet, n.d.). SSRS are as of the thesis publication date in the process of designing a testing platform for a new generation of rescue boats named ”X8” (personal communication, F. Falkman, January 25, 2022). A testing platform entails a prototype boat which primary purpose is testing if the developed design works as intended, and with actually serving as a rescue boat being a secondary purpose. A theoretical maximum number of X8 boats to be manufactured equals that of the total number of SSRS rescue stations (74 stations).

The new generation X8 boats will have a fully electric, battery-driven powertrain, where the weight of such batteries will negatively affect the boats’ range. This is countered by having the boats ride on two hydrofoils, a frontal main hydrofoil that carries the majority of the boat weight, and a rear hydrofoil that acts as a stabilizing hydrofoil and houses the boat’s motor (i.e. the boat’s motor is *inside* of the rear hydrofoil). However, since the boats are aimed at rescue operations they need to be able to run aground, something which is not compatible with permanent hydrofoil fixtures underneath the hull. There is a need to retract and deploy both hydrofoils into and from the hull, where the design of the mechanism to deploy and retract the frontal hydrofoil is the basis for this master’s thesis. This device is henceforth called ”hydrofoil deployment mechanism” and is hereby abbreviated as HDM.

1.1.1 Hydrofoiling boats in general

A hydrofoiling vessel (including boats, surfboards and more) is one where the majority or entirety of the vessel’s hull that’s normally wetted is lifted up over the surface of the water during motion. The purpose of this is to replace the deleterious

Figure 1.1: Left: A hydrofoil surfboard (Fewings, 2020). **Right:** A large boat riding on hydrofoils (Tong, 2020).

drag effects from water with the relatively favorable drag effects from air. This is achieved by having some sort of winged profile (comparable to that of a wing on an airplane) beneath the surface of the water. This winged profile generates lift which pushes the vessel up above water. Some vessels have a hydrofoil that's permanently fixed underneath the hull, while some vessels have a hydrofoil that can be deployed and retracted into and out of the water. See figure 1.1 for examples of hydrofoiling vessels.

1.1.2 X8 boat's involved parties

The project to develop the X8 boat is officially called ELINN, which is an abbreviation for ELectric INNovation. There are several major actors involved in the project, their names and roles are listed below.

- Aston Harald – Main manufacturer of the X8 hull.
- Chalmers University of Technology – Primarily designing the mechatronics aspect of controlling the stabilizing action of the rear hydrofoil, motor and front hydrofoil.
- Mantaray – Design of the front hydrofoil.
- Micropower – Design of the battery system.
- Sigma – Design of the hydrofoil deployment mechanism (HDM) for the front hydrofoil.
  - Thesis student (myself) – Overall HDM design.
  - Sigma Embedded Engineering – A branch of Sigma, acting in collaboration with the myself, that is responsible for everything electrical related to the HDM.
  - Sigma Energy and Marine – A branch of Sigma that acts as the official company supervisors.

- SSPA – Hull design, rear hydrofoil and propeller design, rear hydrofoil deployment mechanism design.
- SSRS – Client, user, secondary project lead and coordinator.
- Zparq – Primary project lead and coordinator. Drivetrain design.

1.1.3 Closely related thesis work

From SSRS's point of view, this master's thesis can be regarded as a follow-up to Lundin's & Eriksson's master's thesis named *Concept development and design of retractable hydrofoil systems* (2021). From my point of view, the thesis is regarded as a source of information, but not as preceding work. I.e., my master's thesis is not a continuation of Lundin's & Eriksson's thesis but rather a completely separate one.

1.2 Aim

The aim of the this master's thesis is to perform a product develop process for a hydrofoil deployment mechanism for SSRS' next generation X8 rescue boat in accordance to SSRS' demands and wishes. The development process started at the establish requirements phase, which entails gathering information to determine and formulate requirements and wishes stakeholders have on a HDM in this context. The development process ended at an early detail design phase, which entails that a final concept has been chosen and further elaborated upon, but that the design is not to be considered complete or ready-to-manufacture.

1.3 Research questions & deliverables

Based on the aims, more hands-on research questions were to answer the following:

- RQ1 – What requirements does the operational profile of SSRS' activities, as well as the surrounding chunks, set on an HDM?
- RQ2 – How should an HDM best be driven (i.e. actuator to provide movement)?
- RQ3 – How should an HDM be designed to withstand the forces applied to it?

More concrete examples of what the master's thesis produced is listed below as deliverables:

- Requirement specification.
- Design priorities (goal weighting list).
- Concept catalog for different chunks.
- Final HDM concept.

- Initial material selection.
- Strength simulations using CAE software.
- Future work plan.

1.4 Delimitations

The following delimitations have been placed on the project's scope:

- How the HDM is controlled by the user, i.e. the human-machine interface.
- Electrical aspects of the HDM, for example voltage of motors, has only been regarded in minimal amounts. These aspects are otherwise the responsibility of Sigma Embedded Engineering.
- Financial aspects in terms of production ramp-up, i.e. cost analysis, has only been loosely done in regards to a *single* prototype boat.
- The thesis has been delimited to the deployment mechanism and any necessary support functions that emerge as a result of the HDM. Already existing chunks, e.g. the hydrofoil and the hull, has been investigated but not developed upon.
- The produced design's adherence to any maritime (or other) standards, rules or regulations has not been regarded.

1.5 Problem analysis

The starting point for this thesis was conducting design work for a product that's in its entirety (the entire X8 boat) in an early to mid phase of development. The design of the HDM was however completely untouched at the start of this thesis. As such, a major part of the design work entailed investigating the problem area and translating the soft customer and user statements into hard engineering requirements. After such a prerequisite step had been done, the actual design work could commence.

This master's thesis entailed being a part of a real engineering product development project with multiple coordinating companies and strict internal deadlines. Whilst such strict deadlines exist (such as design freeze for the hull, etc), they are all past the point in which I am finished with the thesis. Thus they are not discussed any further in this report.

Having been part of a real engineering project entailed another challenge, namely that progress is expected to be made in a timely fashion towards a result that is practically implementable. This is often at odds with the thesis goals of choices being thorough and theoretically motivated. In practice I developed two versions of an HDM in parallel, one thesis version and one industrial version. The reader is not expected to notice this duplicity in this thesis report until possibly towards the end of the project where industrial influences into the HDM project intensify greatly.

1.6 Thesis outline

This thesis is split up into three different stages: Pre-study, concept design, and early detail design. The methodology used in each of these stages is first explained in chapter 2. Results are then presented, split up into the three aforementioned stages. Following the results, I present future work for implementing the proposed design, a discussion regarding the thesis work, a conclusion and finally all appendices.

This thesis consists of the following chapters:

1. **Introduction** - Elaborates on the thesis' background along with its aim, research questions and delimitations.
2. **Methodology** - Presents the methodology used in the thesis throughout its three major stages: Pre-study, concept design and detail design.
3. **Pre-study results** - Further elaborating on the background of the thesis, presenting theory relevant to this project, calculations, and summarizing all pre-study results.
4. **Concept design results** - Mapping of existing solutions related to an HDM, an initial materials study, generation of concepts and choice of final concept.
5. **Detail design results** - Presentation of final design, evaluations via FEA simulations, presenting material choice and evaluation of the design's fulfillment of requirements.
6. **Future work** - Highlighting key areas for future work should this project be continued.
7. **Discussion** - Discussing different aspects of the development process used in this thesis, as well as highlighting possible uncertainties and errors present in the thesis work.
8. **Conclusion** - Brief summary of results and answers to research questions.

2

Method

This master's thesis entails conducting a product development process from post-opportunity identification up until early detailed design. Figure 2.1 visualizes timing and how each step correlates to each other whilst the remainder of this chapter goes into the details and motivations of each method. The visualization should be regarded as merely a nominal description of the overarching method. The reality of the project is less rigid and more fluid (for example steps being done in other orders or repeated out-of-order).

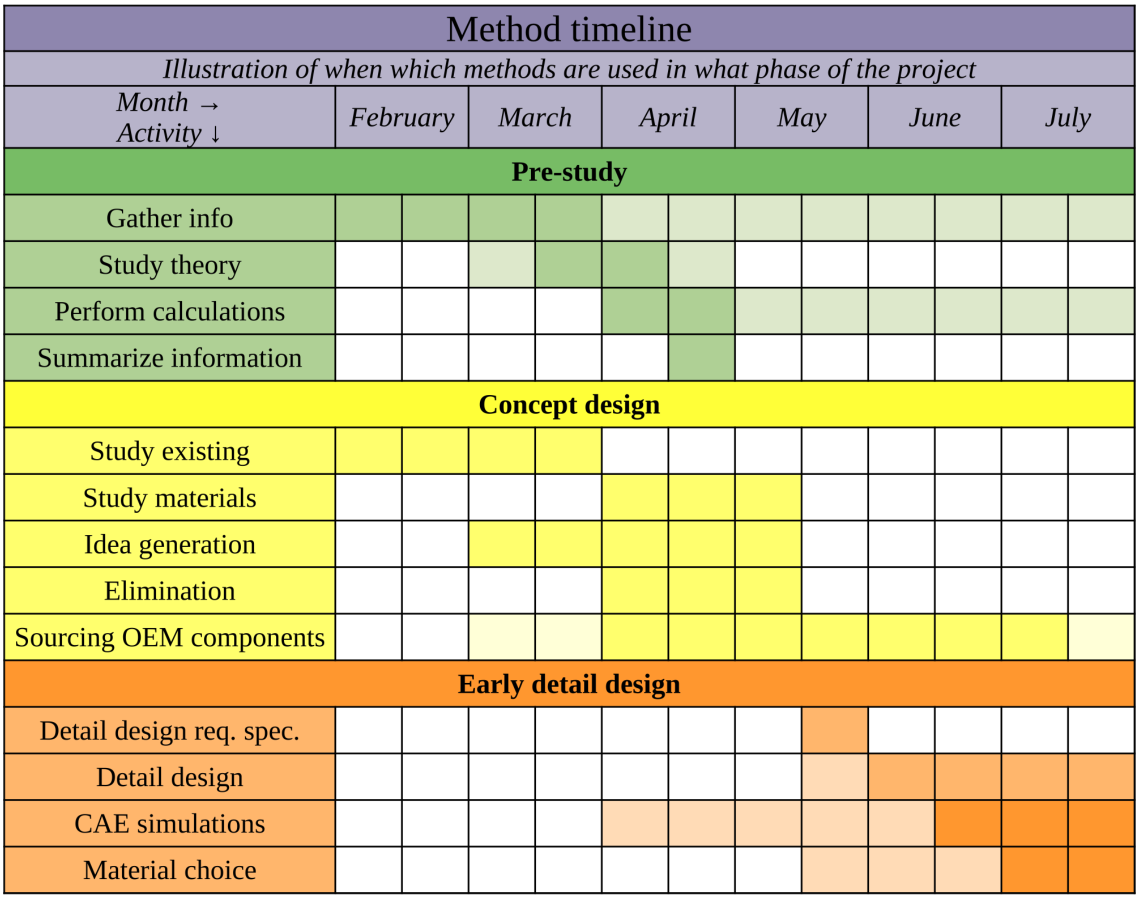


Figure 2.1: An illustration of roughly when in the project methods were applied. Note that lightly colored blocks indicate that *some* work with the method was done during that period.

2.1 Pre-study

A pre-study was performed to further define the problem area and get an insight into the field. As the problem area was so loosely defined, I worked in collaboration with the client to determine what the actual goals of the project are and how to best reach them. Additionally, since I am a novice in the field of hydrofoiling boats, I had to first get an insight into the unspoken aspects of said field as to minimize the risk of knowledge gaps negatively affecting the project.

2.1.1 Gather scenario-specific information

To further understand the specific scenario of the SSRS boat I gathered information in numerous ways. The primary source of information was personal contact in the form of interviews as well as meetings with the involved parties (primarily SSRS). Further major sources of information included the master's thesis from Lundin & Eriksson (2021) and previous internal documentation regarding other aspects from the ELINN project. Gathering information via meetings with involved parties continued throughout the entirety of the project.

Interviews were firstly conducted in an open format to make up for the lack of prior knowledge in the subject area. Additional interviews were subsequently performed in a semi-structured format to gather information about specific topics not covered in documentation or open interviews. Informal interview then took place throughout the remainder of the project in the form of bi-weekly meeting with all parties somehow directly affected by or involved in the HDM design: SSRS, Sigma Energy and Marine, Sigma Embedded Engineering, Mantaray and SSPA.

2.1.2 Study theory

Any theory closely related to HDM development that would affect design parameters in a major way was studied. Fundamental theory in product development and mechanics is not included in this list. Major theory and corresponding sources include:

- Drag forces in fluids – *Fluid Mechanics* by Frank White (2016).
- Submergence factor – *Hydrofoils: Design, Build, Fly* by Ray Vellinga (2009) and personal communication with Docent Arash Eslamdoost from Chalmers University of Technology (March 18, 2022).
- Slamming – Magnus Wikander from SSPA and Alexander Sahlin from Mantaray (personal communication, February 21, 2022).

2.1.3 Perform calculations

Rough calculations were performed as part of the pre-study to act as guidance for the rest of the development. As the main parameters affecting forces (the hydrofoil and speed of the vessel) were finalized enough to provide approximate numbers at the thesis' inception, calculations could be done and used in defining a requirement specification for the project. Main calculations performed were:

- Drag forces in fluids.
- Torque.
- Submergence factor (with respect to fatigue).

To account for the rapidly changing variables of early product development phases, these calculations were performed using Python scripts to easily and quickly repeat calculations using different parameters.

2.1.4 Summarize information

As a finalizing step in the pre-study, all gathered and defined information was summarized in various different models. These models were used as guidance and quick references throughout the development process. Major models includes:

- Concept-phase requirement specification[1].
- Function list.
- Goal definition.
- System architecture diagram.

2.2 Concept design

This section will present methods used in the concept design phase of the project.

2.2.1 Study existing solutions

To act as design-inspiration, and possibly as chunks or modules with the HDM, similar solutions in different contexts were examined. Areas examined include:

- Hydrofoil solutions
- Lifting and lowering solutions
- Structural geometries
- Actuator types
- Linear motion supports (e.g. rails)
- Locking mechanisms

[1]Will be expanded upon later with a detail design requirement specification.

2.2.2 Study materials

In order to expand the solution space with respects to non-conventional materials, and to restrict the solution space to suitable materials, a mapping and investigation of materials was performed. I regarded the study of materials as important to perform early in the project, since the definition of materials defines the available manufacturing methods, which in turn defines available design shapes and geometries. This materials study was primarily performed using the GRANTA Selector software (Version 21.2.0; 2021) and secondarily by discussing with professional engineers from involved parties.

2.2.3 Generate ideas

Two types of methods for generating ideas were performed in this project: Internal idea generation where a design was created from scratch, and external idea generation where existing solutions (such as motors and linear bearings) were pieced together to fit a larger context. I aimed to re-use as many existing solutions as possible and to only design from scratch that for which no existing solutions could be sourced.

Generation of ideas was first performed on an all-encompassing level with high abstraction, and then with a lower abstraction level but within specific chunks or modules. Compiling chunks and modules into an entire HDM was done via a morphological matrix or informally.

As an additional source of input, involved companies occasionally contributed with suggestions for designs of chunks or modules within the HDM.

2.2.4 Concept elimination and selection

Continual elimination of concepts was done using the following methods:

- **Meeting with stakeholders** - Discussions were held with stakeholders such as SSRS (the client), industrial supervisors or representatives from OEMs to determine the suitability (or lack thereof) of concepts.
- **Evaluation matrices** - I used evaluation matrices to eliminate and rank different concepts throughout the thesis. These evaluation matrices were designed by myself and are largely based on a Pugh matrix (Ulrich et al., 2020). These matrices are elaborated upon in chapter 4.3.4.
- **Investigation of specific components** - Since I was reliant on the usage of externally sourced components and modules to produce an industry-implementable design, the availability (or lack thereof) of such was a determining factor in the elimination of concepts.

When the concept design phase was concluded, I had produced a final concept which was further elaborated upon during the subsequent detail design phase.

2.3 Detail design

This section will present the methods used in the project's final phase: the detail design phase.

2.3.1 Formulate detail design requirement specification

Detailed design was initiated by defining a more detailed requirement specification of the chosen concept. As the abstraction level reduced with a chosen concept, the previously defined concept-phase requirement specification was expanded upon to include requirements that arose as a consequence of the chosen solution. This document was used to ensure that the detailed design fulfilled requirements and to lower the risk of oversight.

2.3.2 Overall detail design methodology

The detailed design was performed in three different ways (in chronological order):

1. System-level design, as defined by Ulrich et al. (2020), was used to define a product architecture. This was done mainly by using existing commercially available components in a designed architecture. In other words, this method essentially entailed piecing together chunks using commercially available products.
2. External companies presenting designs with exact components to be used within a specific module. This methodology had to be included in the project due to the thesis doubling as industrial work expecting to produce tangible results. These designs are clearly marked in the thesis as being external companies suggestions.
3. Component-independent design where I designed a solution regardless of the availability of parts. In this case, I used existing ideas of solutions (for example the concept behind a drum brake) without locking the design to the availability of specific parts.

I initiated the design procedure with a high focus on (1) coordinating design, as specification and design of components from the ground-up was regarded as unrealistic with the given time-frame. However, I experienced major setbacks using method (1) and (2) in terms of time being spent sourcing components, contacting companies, and switching between concepts. For that reason, I finalized some of the chunks using method (3).

Strength simulations: I performed strength simulations and displacement simulations on many of the designed components. Although no components are presented as a final design suggestion, these simulations were performed to validate the concepts are feasible to be regarded as proof of concept. Simulations were performed using the ANSYS suite (version 21.1.0; 2021).

2.3.3 Choose material

Suitable materials for designed components were investigated using three different methods:

- Theoretical study using the database GRANTA Selector as a source (Version 21.2.0; 2021).
- Industrial study using commercial companies dealing in materials as sources.
- Personal communication with involved companies as sources.

2.3.4 Evaluate detail design requirement specification

An evaluation of the developed concept was performed based on the detail design requirement specification. Since the design is at an early detail design phase (and not complete) the evaluation is regarded to be preliminary and an estimation.

3

Pre-study results

This chapter will present further details on the X8 boat, its hydrofoil and its usage. It will then go on to present theory relevant to this project and performed calculations. Finally, this chapter will present a summary of results in the form of a requirement specification, function list and goal weighting list.

3.1 Gather scenario-specific information

This section will present further details on the context of which the boat and HDM will be used in and the state of development for the rest of the boat at the time of the interviews. All information presented is sourced from interviews and meetings with SSRS or other involved parties (personal communication, January 25, 2022 to June 23, 2022).

Figure 3.1 shows an early concept of the to-be designed boat, hereby referred to as X8. It is aimed to be a rescue boat to be used in all bodies of water (except icy water) within the territories of Sweden. The types of rescue missions X8 will be used in is everything from towing broken-down vessels to life-or-death ambulance scenarios. A vast majority of rescue operations occur during summertime with sunny and calm weather, as this is when most people are out on their boats. SSRS aims for X8 to fulfill 95% of all SSRS assignments, where the unfulfilled 5% includes planned non-emergency towing and very-far-away rescue missions.

The dimensions of X8 are planned to be roughly 8 [m] length, 3 [m] width and with a weight of 2.6 [tons]. The structural part of the hull will be in a carbon fiber sandwich material. It is a semi-covered boat, i.e. there is no place on the boat that is both roofed and walled off. It is aimed to be replacing the previous ”Gunnel Larsson” boat of nearly identical dimensions and weight. SSRS has two other main rescue boats in other classes, one completely uncovered boat with a length of 3 [m] and one fully-covered boat with a length of 13 [m]. Further dimensions of the boat can be found in appendix B, where the reader is advised to take note of the two battery backs next to the hydrofoil, both towards the bow and the stern, which greatly constrains the available design space for the HDM and a retracted hydrofoil.

The X8's hull's underside has two mechanism protruding from it. At the stern is a smaller steerable hydrofoil and torpedo motor package which will provide the propulsion and a majority of the steering. In the middle of the hull is a larger

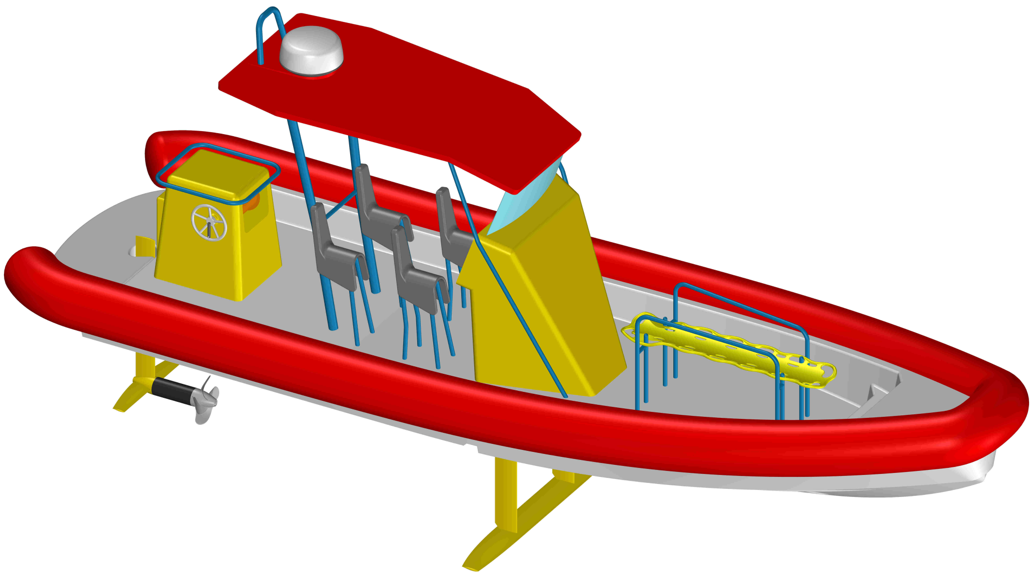


Figure 3.1: An early concept drawing of X8 (F. Falkman, personal communication, February 2, 2022).

hydrofoil that will uphold >80% of the boat's weight under flight (this thesis only includes work with the deployment of the larger hydrofoil). **The larger hydrofoil will always generate lift when the boat is in motion relative to the water** with a *nominal* lifting force of roughly 26 000 [N]. Both hydrofoils are constructed in carbon fiber.

When the boat is traveling fast enough with both hydrofoils deployed, the lift generated from them will lift the entire boat out of the water and enable the boat's entire hull to rise above water-level, i.e. **the entire boat's weight will rest on the hydrofoils**. Rough geometry specifications of how big the allotted space is for the HDM (hydrofoil deployment mechanism for the larger hydrofoil) can be seen in figure 3.2, where the HDM must be contained between points A – F with a port - starboard width of roughly 2.5 [m]. **The cutout in point A is a literal hole in the hull, i.e. sea water will come in contact with any equipment placed in its vicinity.**

The X8 is a fully electric boat with batteries as a power cell. It has a sprint range of 15 nautical miles and top speed of 35 [knots]. The boat will begin to fly (hull lifted above water level) at speeds above 17.5 [knots]. SSRS aims for both hydrofoils to be deployable at speeds between 0 – 8 [knots]. The deployment must also be automated (i.e. using some sort of motor and not being hand-powered).

The boat is aimed to be in use for a total of 25 years. Maintenance is done by laymen on-site as much as possible, with a professional renovation in a professional workshop only aimed to be done after 12.5 years. Maintenance is aimed to be conduced as needed in addition to planned yearly maintenance sessions between seasons.

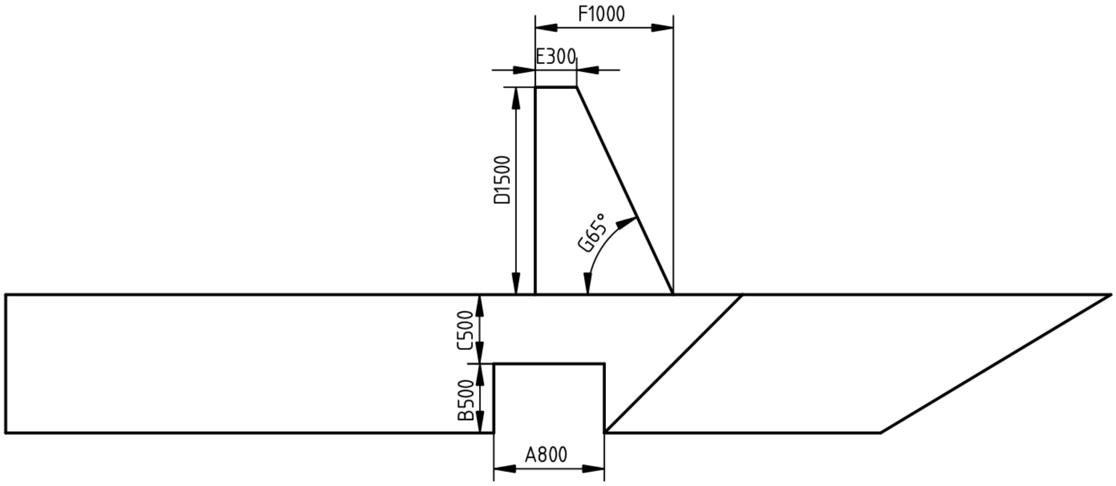


Figure 3.2: Rough geometry specifications of allotted volume for HDM.

3.1.1 Hydrofoil explanation and terminology

The hydrofoil to be used in the X8 boat is a specific hydrofoil model produced by the company Mantaray. Figure 3.3 shows a mock-up illustration of the hydrofoil, as well as what different parts of the hydrofoil are called and their purpose.

- Arms: Acts as supporting levers to provide rigidity and connect axles to struts.
- Axles: The only allowed connection point between the hydrofoil and the rest of the HDM. The axle rotate slightly in their sockets.
- Struts: Creates distance between the axles and the wing, allowing the wing to be sufficiently submerged under water.
- Wing: Source of lift for the hydrofoil.
- Chord length: The length of the wing in the direction of travel.

The hydrofoil will be attached to the rest of the HDM (and in turn the rest of the boat) via a "hub". The hub is a custom-molded component that is made to fit onto the axles of the hydrofoil. This to-be-designed **hub is the only interface that's allowed to be touched by an HDM**, i.e. the HDM cannot grab or come into contact with any part of the hydrofoil directly but must go via the hub.

Important to note is that the entire hydrofoil will be in movement during usage. I.e. the wing will twist, causing the strut to twist, causing the arms to make the axles rotate slightly. This movement is intentional and is a part of the stabilizing action of the hydrofoil. This is the main reason behind why the only allowed interface between the hydrofoil and the HDM is at the hydrofoil's axles.

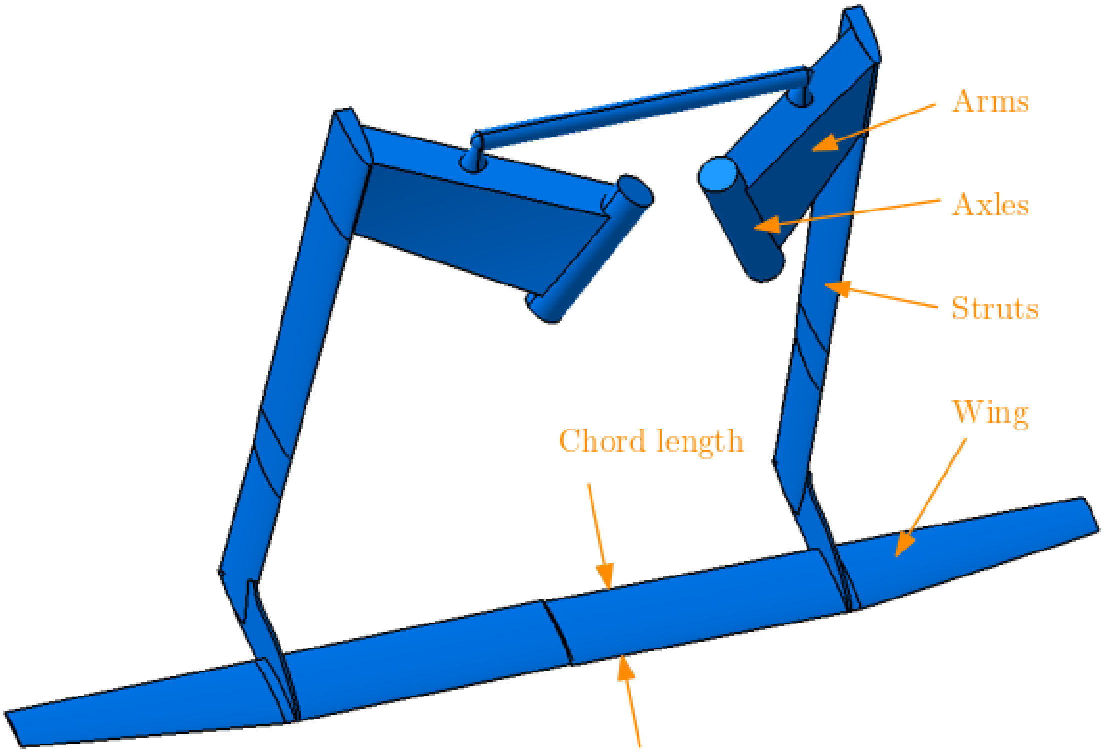


Figure 3.3: What different parts of the hydrofoil are called.

3.2 Theory

This section will present theory regarding drag forces, the so-called submergence factor when using hydrofoils, and a phenomena known as slamming.

3.2.1 Drag forces in fluids

Solids submerged in fluids, where there is relative motion between the solid and the fluid, experience drag forces (White, 2016). These magnitude of these drag forces can be estimated using the equation:

$$F_D = 0.5 C_D \rho v^2 A$$

where:

F_D $[N]$ - drag force	C_D [] - drag coefficient
ρ $[kg/m^3]$ - density	v $[m/s]$ - relative velocity
A $[m^2]$ - area	

In this case the area used in calculations is the planform area, i.e. the projected area parallel to the velocity vector (i.e. the "flat side" of the wing).

3.2.2 Submergence factor

The lift-generating capabilities of a hydrofoil wing will be negatively affected by close proximity to the water's surface (Vellinga, 2009; A. Eslamdoost (personal

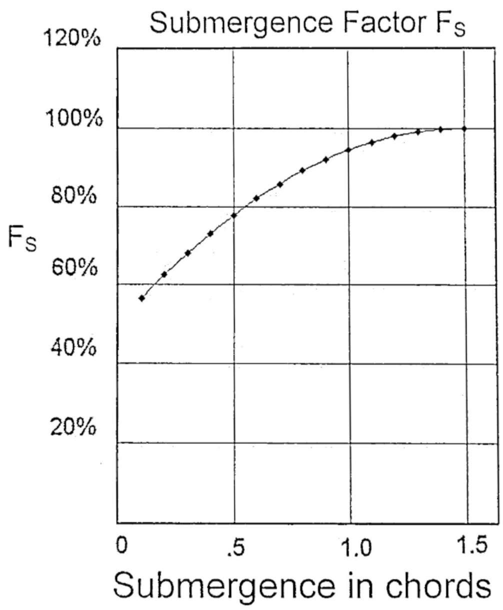


Figure 3.4: Function of a wing's lifting force in relation to its submergence factor (Vellinga, 2009).

communication, March 18, 2022). The area in which lift is negatively affected is measured by the ratio of the distance between the water's surface to the wing, and the wing's chord length. This ratio is called the submergence factor. In other words, a wing with a longer chord length needs to be submerged further to ensure maximum lifting capacity.

The lift force can be defined as a function of chord length and submergence mathematically with the equation (Vellinga, 2009):

$$F_L = 1 - 0.222 \left(\frac{1.5 * chord - submergence}{chord} \right)^2 \ [N]$$

where the equation is only valid between the submergence factors of roughly 0.2 – 1.5. At submergence factors of 0.2, the wing is down to roughly 60% of its lifting capabilities, whilst at submergence factors of 1.5 or greater the wing has achieved practically 100% of its lifting capacities. See figure 3.4 for a visualization of the equation. Vellinga (2009) states that the underlying reasons for a decrease in lift force is a combination of air being sucked in from the atmosphere to the low-pressure area above the wing, as well as there being less mass of water above the wing (negatively effecting Newton's third law of motion) the closer the wing is to the surface of the water.

The effects of submergence factor affects this development project in terms of potential fatigue problems in wavy water as it may entail a rapidly varying lift force from the hydrofoil. Should the wing of the hydrofoil be at a shallow depth at wavy conditions and high speeds, the lift force from the wing will vary at a high frequency. See figure 3.5 for an illustration.

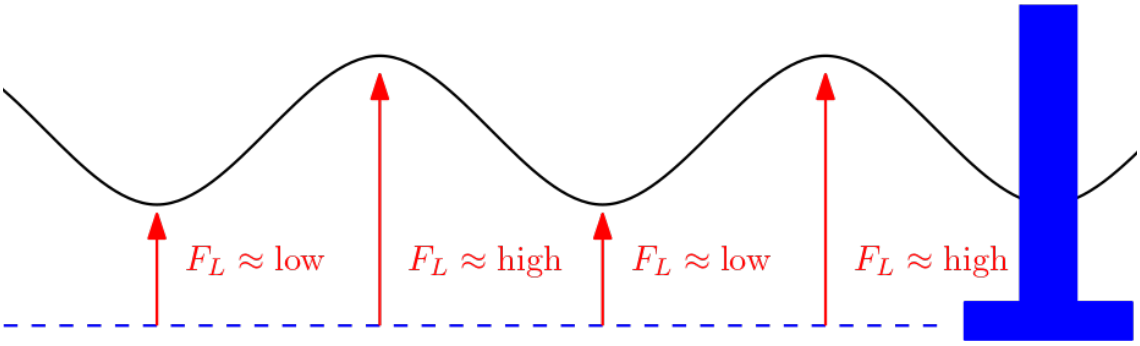


Figure 3.5: Illustration of varying lift force (red) depending on wing (blue) submergence depth underneath the water's surface (black). The force vector magnitudes are exaggerated for illustrative purposes.

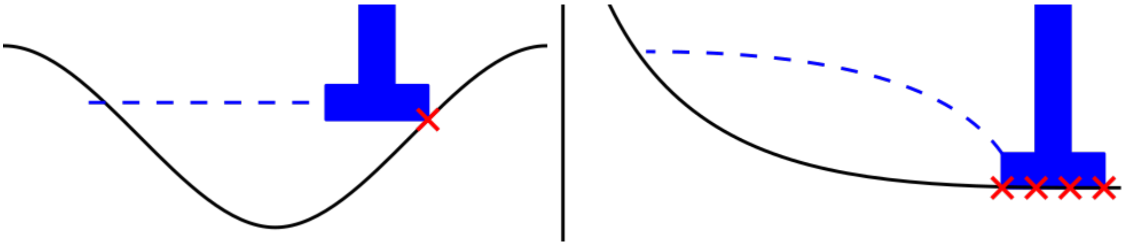


Figure 3.6: Left: Illustration of a more likely but not deleterious slamming. **Right:** Illustration of deleterious but very unlikely slamming.

3.2.3 Slamming

A phenomena known as slamming can occur during usage of hydrofoils (M. Wikander, A. Sahlin, personal communication, February 22, 2022). This involves the wing of the hydrofoil and the trough of a wave (the lowest part two waves, i.e. the ''valley" between two waves). The scenario where this phenomena has a risk of occurring is when the boat is traveling at higher speeds during wavy conditions. Slamming is when the hydrofoil wing completely exits the water between a wave's peak and trough, and then subsequently slams against the surface tension of the water. This slamming action is akin to an impact and transfers loads higher than nominal to the hydrofoil and any connected supporting structure.

Slamming can only generate a relevantly large force when the impact area of the water is parallel to the wing's underside. This is highly unlikely to occur for the following reasons. For slamming to occur at all the wing has to leave the water completely, something which will only rarely happen during wavy conditions and not at all during flat-water conditions. For the impact area of the water to be parallel to the wing's underside, conditions have to be akin to flat-water conditions, i.e. no waves. In other words, the only deleterious slamming scenario is when slamming cannot occur (possibly with the exception of travelling perpendicular to the wake of another larger seafaring vessel). See figure 3.6 for illustrations. Whilst slamming can in theory happen, recommendations were made by SSPA and Mantaray (M. Wikander, A. Sahlin, personal communication, February 22, 2022) to disregard any slamming effects in the development process.

3.3 Calculations

This section will present the main calculations performed during the pre-study phase of the project. This includes calculations on drag forces the hydrofoil experiences, the generated torque from the aforementioned drag forces, and finally initial calculations regarding fatigue via submergence factor.

3.3.1 Drag forces

Drag forces were calculated to get an estimation of what magnitude of force the HDM has to withstand in the direction opposite of the boat's traveling direction. This was performed using the equations presented in chapter 3.2.1 and a Python script, see appendix C.

Reynolds number: In order to determine the flow state (turbulent or laminar), numbers for a worst case scenario was used in the calculation of the Reynold's number ($Re = \rho UL/\mu$). Worst case scenario in this case refers to making it as far away as possible from fully turbulent (as this is when aforementioned equations stop applying). The following numbers were used.

The viscosity of the water was chosen at 0[°C] to evaluate a worst case scenario. Viscosities for water at different temperatures are (White, 2016):

$$\mu_{H2O,20^\circ C} \approx 0.001003\ [Ns/m^2]$$
$$\mu_{H2O,10^\circ C} \approx 0.001307\ [Ns/m^2]$$
$$\mu_{H2O,0^\circ C} \approx 0.001788\ [Ns/m^2]$$

The density of water is roughly constant regardless of temperature:

$$\rho_{H2O} \approx 1000\ [kg/m^3]$$

Speeds assessed were 8 knots (maximum speed of the boat during HDM activation) and 35 knots (planned top speed of X8).

$$U_{8[knots]} \approx 4.12\ [m/s] \quad U_{35[knots]} \approx 18.01\ [m/s]$$

The shortest part of the hydrofoil (strut) relative to the direction of the fluid's motion:

$$L \approx 0.21\ [m]$$

This produced a result of:

$$Re_{8,strut} = \frac{\rho UL}{\mu} = \frac{1000 * 4.12 * 0.21}{0.001788} \approx 483\ 893 \gg 3500 = Re_{turbulent}$$

Calculations show that the flow is very clearly turbulent even in the most conservative scenario with a low speed, short foil length and cold water.

Drag forces: In order to calculate the drag forces ($F_D = 0.5 C_D \rho v^2 A$) several variables must first be determined.

There are different parts of the hydrofoil which have different drag coefficients. The drag coefficient for the wing differs at different speeds and are according to A. Sahlin (personal communication, February 9, 2022):

$$C_{D,wing,8[knots]} \approx 0.035 \quad ; \quad C_{D,wing,35[knots]} \approx 0.005$$

The drag coefficient of the struts were estimated according to NASA's numbers (n.d.) for an airfoil which the struts' shape moderately closely resembles.

$$C_{D,struts} \approx 0.05 \quad \text{(regardless of velocity)}$$

Two different speeds were evaluated. Note that $F_{D,struts}$ encompasses the drag force for both struts.

8 knots - The highest speed in which the HDM will perform a deployment or retraction. At these speeds the boat is not flying and the hydrofoil is completely submerged.

$$F_{D,total,8} = 472\ [N] \quad ; \quad F_{D,wing,8} = 235\ [N] \quad ; \quad F_{D,struts,8} = 237\ [N]$$

35 knots – The highest speed in which the boat will be traveling. In these speeds the boat will be flying with the wing experiencing a submergence of roughly 0.3 [m].

$$F_{D,total,35} = 1455\ [N] \quad ; \quad F_{D,wing,35} = 642\ [N] \quad ; \quad F_{D,struts,35} = 813\ [N]$$

These drag force numbers have been confirmed by Mantaray to be accurate.

3.3.2 Torque

A drag force created by the wing will, due to the lever between the struts and the top of the hydrofoil, generate a torque that will have to be counteracted by the HDM. As the length of the lever can reasonably be estimated to be equal to the length of the struts, calculations regarding torque can already at the stage be determined. Torque was calculated using the equation:

$$T \approx F_D H\ [Nm]$$

where:

$T\ [Nm]$ - torque measured from the top of the hydrofoil
$F\ [N]$ - drag force from the water
$H\ [m]$ - height of the struts

An overwhelming majority of the drag force will be opposite the boat's travel direction, and as such only the drag force with the lever of the struts will have to be assessed. Torque is calculated at max speeds and conservatively by placing all drag forces at the wing (thus maximizing the forces' lever).

$$T \approx F_D H \approx 813 * 1.3 \approx 1060\ [Nm]$$

3.3.3 Submergence factor

Lifting force as a function of the submergence factor was calculated using the equation:

$$F_L = 1 - 0.222 \left(\frac{1.5 * chord - submergence}{chord} \right)^2 \quad [N]$$

Using this equation, a ΔF could be calculated using the aforementioned equation at different submergence depths (representing the wing's differing submergence in wavy conditions). A frequency of that ΔF could be calculated by assessing a ratio between the boat's speed and wave length. See appendix D.

Calculations show that ΔF is only relevantly significant when wave height is approaching the submergence depth of the wing. The frequency of ΔF is low from a high frequency fatigue point of view. Since numbers regarding submergence depth and wave characteristics in areas of operation are either undecided or unknown at the time of thesis writing, I cannot draw any conclusions of the relevance of submergence factor fatigue.

3.4 Summarized information

This section will present the most important information gained during the pre-study phase in the form of a concept-phase requirement specification, a function list and a goal weighting list.

3.4.1 Concept-phase requirement specification

A concept-phase general requirement specification (one that is not solution-specific) is presented in its fullest in appendix E. The requirement specification has its requirements categorized in the following groups:

- *General* – Uncategorized requirements.
- *Geometry* – The HDM is allotted a specific limited volume on the boat.
- *Mechanical strength* – The HDM must withstand any forces applied to the system, primarily the lifting-force generated by the hydrofoil and the self-weight of the boat.
- *Use-environment* – The HDM must be fully functional in a marine environment, primarily being able to handle salt-water.
- *Movement* – The HDM must be able to move in certain dimensions, primarily vertical movement to deploy and retract.
- *Maintenance* – The HDM must facilitate minor maintenance by laymen and major maintenance by professionals.

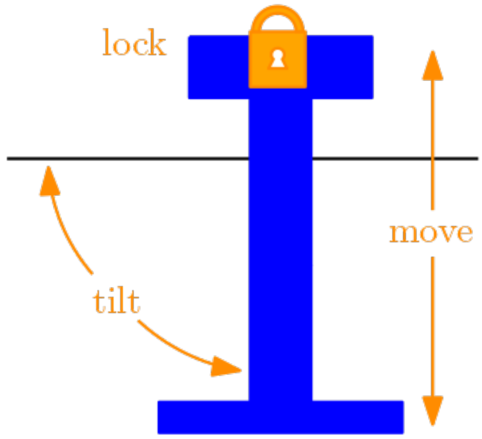


Figure 3.7: Major functions the HDM must perform.

3.4.2 Function list

The HDM has a number of functions which must be fulfilled and are listed below. Illustrations of each function can be seen in figure 3.7.

1. Provide vertical movement in both directions. This movement will have to be done both when the boat is stationary (hydrofoil is generating no lift) and when the boat is moving (hydrofoil is generating lift upwards).

2. Provide trim in both directions, i.e. angle the entire hydrofoil towards either the bow or the stern with a rotation axle at the top of the hydrofoil.

3. Lock hydrofoil when the hydrofoil is completely deployed (in the water) or completely retracted (away from the water). Whilst the hydrofoil is locked in its deployed position, it must also withstand the lifting force generated foil at top speed.

3.4.3 Goal weighting list

SSRS and myself compiled factors aimed to be optimized in the boat's design. These are general guidelines for the entire boat and applies to the HDM as well. They are presented in order of decreasing importance below.

1. *Dependability* – Dependability is a combination of the aspects *reliability* (time between failures) and *maintainability* (how easily a system is maintained), where the factor of primary importance in this case is reliability. Due to the safety-critical nature in which these boats could operate in (e.g. acting as an ambulance in a medical emergency), an untimely failure of the HDM could entail loss of life.

2. *Mass* – Boats are conventionally quite lenient on weight demands (M. Wikander, personal communication, April 27, 2022), but that is not the case with hydrofoiling boats. The lifting capacity of a hydrofoil wing is far more limited than that of a conventional boat's hull, and thus the mass of all systems on-board ought to be minimized to increase the beneficial mass.

   (a) *Center of gravity* – In addition to the magnitude of the mass, the placement of the mass is also of importance. A low center of gravity is sought after to increase the boat's stability. Whilst the mass placement is important, it is not as important as its magnitude.

3. *Maintenance cost and effort* – The boat will primarily be maintained by laymen on site, without the usage of significant workshop equipment. Major renovations are only planned to be performed once at the midway point (12.5 years) of the boat's total lifespan (25 years) at a proper workshop by professionals. Due to the desired maintenance routine of the boats, systems must be designed in a way to minimize both the amount and the complexity of necessary maintenance.

4. *Initial purchasing cost* – Whilst SSRS naturally wishes to lower purchasing cost as much as possible, they would rather have a slightly more lightweight and easily maintained system at the expense of initial purchasing cost.

3.4.4 System architecture diagram

An HDM in this context consists of different chunks which interacts with each other. Figure 3.8 illustrates these chunks and their interactions at a high abstraction level, as well as which chunks are included in the thesis scope.

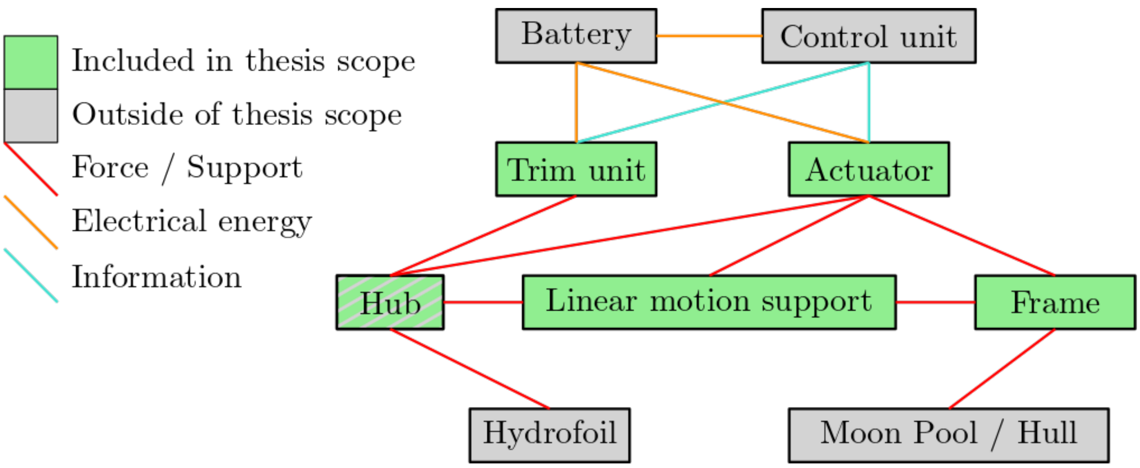


Figure 3.8: An overview of the system architecture of the HDM.

The process flow can be described as such: The captain uses the control unit to activate the trim function or the actuator, the latter entailing a retraction or deployment of the hydrofoil. Powered by the battery, the trim unit and actuator apply forces to the hub which in turn redirects these to the hydrofoil. The hub and actuator are supported by the linear motion supports (e.g. shafts that guide the movement of the hub) and the frame, i.e. these provide reaction forces that hold everything in place. All forces and reaction forces go through the frame which is itself supported by the hull, more specifically the Moon Pool.

4

Concept design results

This chapter will present all results from steps taken during the concept design phase. This phase is initialized with a study of existing solutions to different functions within the HDM, and an initial study of unconventional materials. Concepts are then presented with regards to different chunks within the HDM. The chapter is concluded with a presentation and motivation of the final concept choice.

4.1 Study existing solutions

A study of existing solutions to different functions was performed as a first step to the concept generation phase. Study of existing solutions would allow myself to integrate and gain inspiration from solutions for usage in the developed HDM. Patents were found on Espacenet and Google patents whilst commercial solutions were found by performing general internet searches.

4.1.1 Hydrofoil solutions

There are numerous deployable hydrofoil solutions to examine via patents. Relevant patents that I managed to find, see figure 4.1, were however not directly applicable.

1. *Retractable hydrofoils for marine vehicles* (Ulgen, 2008) – Presents a solution where a hydrofoil wing is deployed by a linear actuator pushing on a lever that's connected to a hinge on the wing. Whilst not directly applicable for the Mantaray hydrofoil, it demonstrates that a non-vertically aligned linear actuator can be used to achieve a deploying motion.

2. *Retractable Power Drive Surfboard for Wave Foils* (Derrah, 2020) – A manually operated mechanism that allows users to deploy a power drive via a simple hinge. Not applicable for the Mantaray hydrofoil as it needs to be lowered whilst in motion (where this solution would cause massive drag) and since there are battery packs both next to the hydrofoil towards both the bow and the stern (not allowing the foil to be stored there when retracted).

3. *Retractable hydrofoil on vessel* (Kearney, 2020) – A solution where several U-shaped hydrofoils are lowered into the water via some sort of mechanism. Externally mounted components (outside of the hull) would not be advised as it would introduce additional drag in addition to what's already existing in

the Mantaray hydrofoil.

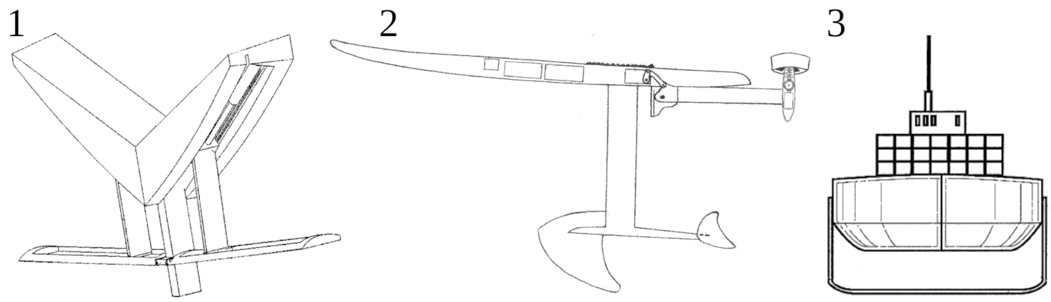


Figure 4.1: Left: Retractable hydrofoils for marine vehicles (Ulgen, 2008). **Center:** Retractable Power Drive Surfboard for Wave Foils (Derrah, 2020). **Right:** Retractable hydrofoil on vessel (Kearney, 2020).

Commercial solutions that deploy hydrofoils are available to investigate but difficult to get exact details on. The Candela C7 hydrofoiling boat can be regarded as a close similarity to X8. The boat has roughly equivalent length and its hydrofoil is deployed in a vertical motion from inside the hull, however its weight is only roughly half of that of the X8 (Candela, n.d.). The Candela C7 deploys its hydrofoil using a rack and pinion system (Motor Boat & Yachting, 2021). This solution has the rack attached to the struts of the hydrofoil, meaning that using this solution directly would not be possible as the X8's hydrofoil's only interface to the HDM is the hydrofoil's axles.

4.1.2 Lifting and lowering solutions

Commercial solutions to lift and lower heavy loads are abundant, although a vast majority of them are not directly mirroring the load case of X8. X8's hydrofoil will be pushing vertically upwards whilst having to be moved vertically downwards, i.e. opposite the load case of most normal lifting scenarios. This entails that the designed solution cannot rely on the lowering (nor the lifting) action being reliant on gravity, where the mechanism simply ''releases" and allows the object to fall downwards. What this means for the design project is that few solutions found could be applied directly.

Found commercial solutions that may be of use or inspiration in the design process are listed below. To summarize, lifting heavy loads linearly where the actuator is moderately small in volume is primarily done using hydraulics. The client has explicitly stated that hydraulics and pneumatics are not desired solutions due to the relative complexity and number of components (filters, tubes, pumps, etc) compared to that of fully electrical solutions.

Forklifts: Forklifts' main function is to raise and lower a heavy load, roughly in the same stroke length as the X8's hydrofoil. Following a quick study on numerous companies' product lineup, I conclude that forklifts primarily use hydraulics, secondarily pneumatics, as their actuators. Acting as an intermediate coupling between the hydraulic actuator and the forks (where the load is placed) is typically a roller chain (BigRentz, 2019). This allows for the force to the actuator to stay in-line with the hydraulic piston and for the force to be transferred without the fork being attached directly to the piston head. See figure 4.2.

Cranes: Cranes lift loads vertically, similar to forklifts, but at a far greater weight and distance. An overwhelming majority of cranes found use multi-stage (a.k.a. telescopic) hydraulics to move the boom-part of the crane. A hydraulic system is usually placed inside of the boom, which in turn also is of a multi-stage construction (allowing for compact geometries in its retracted state). An overwhelming majority of cranes use some sort of hoist-pulley system to achieve the strictly vertical motion of the load without moving the boom. These hoists can be powered with a purely electrical motor or by a hydraulic motor. I theorize that the chosen cross-section profile of large hollow tubes for the boom is an intentional design decision to provide bending stiffness (as opposed to using small tubes or solids) in a simple way. See figure 4.2

Figure 4.2: Left: A forklift using a roller chain (Pixabay 2, 2017). **Right:** A crane using a multi-stage boom arm (Maedausa, 2017).

Deep drilling: Deep drilling can be referring to either deep vertical depths (ocean or mountain drilling) or drilling where the length of the drill hole is disproportionately long compared to the hole's diameter. This was briefly investigated to determine how such mechanisms prevented buckling. In both cases designs seem to utilize bushings or plain bearings along axially loaded solids (such as drill bits) to act as bracing and provide rigidity against buckling.

4.1.3 Structural geometries

At this stage of the thesis, I theorized that the designed solution would most likely entail some sort of structure that needs to withstand vertical loads whilst being supported in areas that are not in-line with the vertical load. Figure 4.3 exemplifies this by illustrating a tower (left) where the support structure is directly underneath the load, compared to a bridge (right) where the support structure is not directly underneath the load.

Such structures can be compared to that of bridges, see figure 4.4, which are designed to withstand heavy vertical downward loads whilst having fixed supports that are

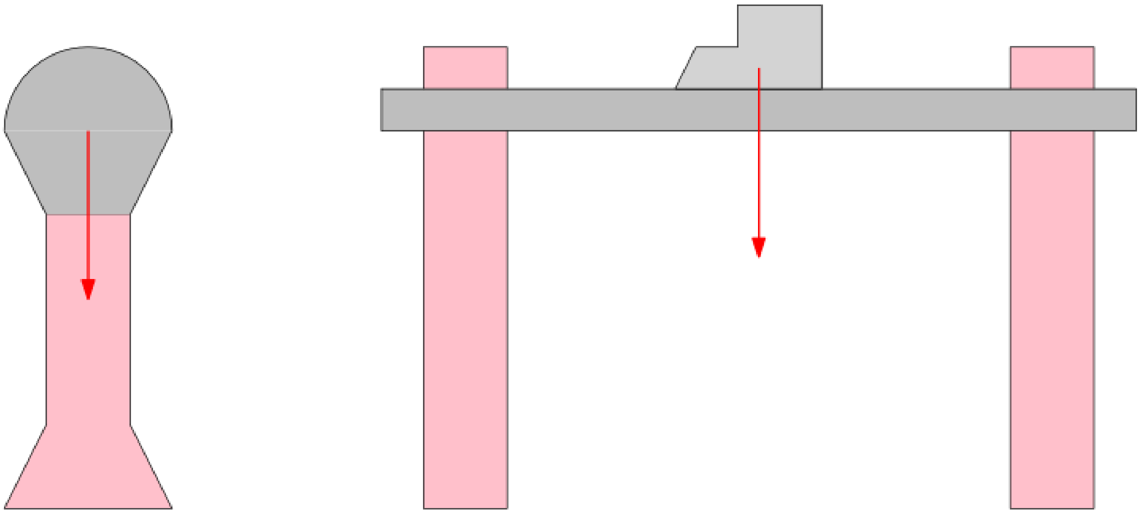


Figure 4.3: Gray: A load pulled downwards by gravitation. **Pink:** Support structures withstanding the downward force.

not completely in-line with the load. Whilst not completely applicable, as bridges are designed to withstand vertical loads in one direction (gravitational force of its own structure *and* applied weights), design inspiration can nonetheless be sought in bridges. To summarize, I learned be aware of the mechanical strength of arches, the tensile strength of cables, and the locking rigidity of trusses and possibly apply this knowledge in an HDM design.

Figure 4.4: An arch bridge (Geograph, n.d.), cable-suspended bridge (Pixabay 1, 2017) and a truss bridge (Science Stock Photos, n.d.).

4.1.4 Actuator types

Linear actuators examined in the pre-study phase in this thesis can be broadly categorized into fluid actuators and electric actuators.

Fluid actuators: Fluid actuators be powered by both air (pneumatic) and oil (hydraulic), but to simplify for the reader this thesis is focusing on hydraulics. There are many types of hydraulic actuators that provide linear motion and force in one or two directions. Overall these can be categorized in two types of hydraulics, ones where the stroke is roughly half the length of the entire envelope size (single stage), and ones where the stroke is longer than the envelop size (multi-stage or telescopic), see figure 4.5.

On the whole, hydraulics can be regarded as superior in terms of sheer output force compared to most other actuators. As a negative, they are heavier and more complex (in terms of total number of required components) than alternatives. Additionally, the potential power of hydraulics may even be regarded as over-kill for this application. Finally, I wish to remind the reader that the client have explicitly stated that they do not desire a fluid actuator solution due to their complexity.

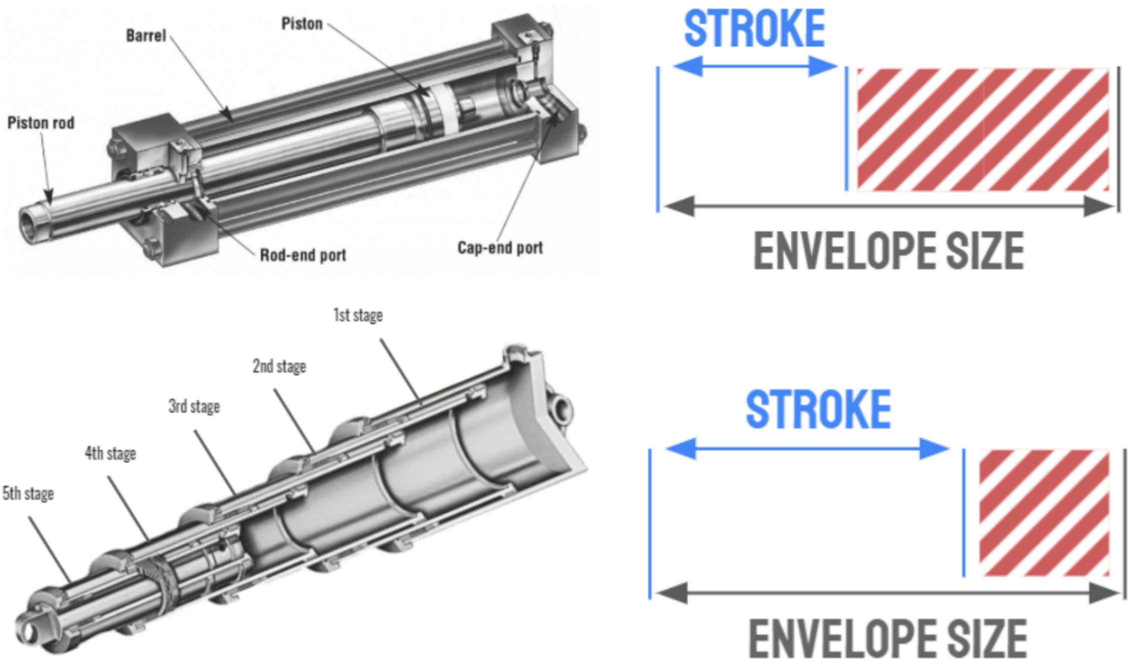


Figure 4.5: Fluid actuators showing their stroke (length they provide movement) to envelop size (total length whilst expanded) ratio. **Top:** Single-stage fluid actuator. **Bottom:** Multi-stage fluid actuator. (Lundin & Eriksson, 2021)

Lead screws: Lead screws achieve linear motion by translating a rotational movement to linear via the helical threads of a screw, see figure 4.6. This linear motion can either manifest as the entire screw moving linearly or as a stationary screw with a nut traveling along the screw length. Lead screws can be broadly categorized into either ball screws (where the nut is a ball bearing mechanism) or acme screw (where the nut is a plain threaded nut), see figure. Ball screws have lower friction between the nut and the screw, at the cost of higher complexity and sensitivity (particularly to salt water), whilst acme screws have higher simplicity and a potential for self-locking via friction (at the cost of a higher base friction requiring more power to drive the screw). Both nut and screw can be produced in a variety of materials (for example aluminum screw, or brass or plastic nut) with steel being the most common material for screws.

Figure 4.6: Left: A cross section of a ball screw (Misumiusa, 2016). **Right:** An acme screw (CNC3D, n.d.).

4.1.5 Linear motion supports

In addition to the HDM requiring linear motion, it is very likely to also include some form of linear motion support (to raise structural stability). The most relevant linear motion supports in this project have been linear plain bearings, linear ball bearings and rail guides.

Linear plain bearing: Linear plain bearings are a very simple construction consisting of a cylinder (similar to that of a bushing in appearance) that slide along a smooth shaft. Plain bearings are attached to external components via a housing, see figure 4.7, and have a relatively high axial friction against its shaft. They are superior in terms of cost, maximum allowed radial force, water and general debris resistance, and maintenance. Whilst shafts are overwhelmingly either steel or aluminum, the bearing are most often produced in a plastic or softer metallic material (e.g. brass).

Linear ball bearings: Linear ball bearings (see figure 4.7) are used with equivalent shafts to that of plain bearings, but instead house a construction with ball or roller bearings inside of it, greatly reducing the bearing's friction along the shaft. Whilst being more expensive than plain counterparts, linear ball bearings are still a low-cost alternative to linear motion. Relative to other alternatives, their weaknesses are a lower maximum radial force, lower water and general debris resistance and sometimes higher maintenance demands (if the chosen ball bearing requires lubrication). Whilst the water resistance is lower than that of plain bearings, the simple round shape of the shaft allows for wipers to be installed into the bearing housing to repel as much moisture as possible from entering the housing.

Rail guides: Rail guides (see figure 4.8) can be regarded as a continuation (complexity-wise) of linear ball bearings. Rail guides consist of a small unit traveling along a custom-shaped track that matches the dimensions of the unit. The unit houses ball or roller bearings inside of it to facilitate low friction along its track. Rail guides are superior to linear ball bearings (and to some extent plain bearings) in terms radial force and also have the capability of withstanding torque in more dimensions than plain or ball bearings (which would merely rotate along its directional axis). As

Figure 4.7: Left: A linear plain bearing (Ewellix 3, n.d.). **Center:** A linear ball bearing (Ewellix 4, n.d.). **Right:** A linear ball bearing unit (Ewellix 5, n.d.).

a negative, rail guides are significantly more expensive than both linear plain and linear ball bearings, are equally sensitive to water and debris as ball bearings but with less effective wipers compared to ball bearings. As a side-note, there are commercially available low-precision and low-force rail guides with a majority plastic elements that are completely compatible with water (even sea water). These have been disregarded in the remainder of the project due to their low maximum radial force capacity.

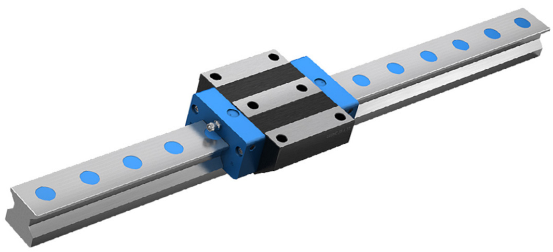


Figure 4.8: A rail linear unit on a rail guide (Ewellix 6, n.d.).

4.1.6 Locking mechanisms

Electromagnetic passive brakes: Electromagnetic *passive* brakes entails that the locking mechanism is active without any electricity flowing through the brake (with ''active" being the opposite). An example of such a brake would be a simple drum brake, see figure 4.9. In this design, brake pads are pressed against a rotating axle using springs and a electromechanical actuator, which when activated releases the brake pads from the axle.

Locking latch: A locking latch entails the placement of a supported smaller object (usually a small rod or bar) to prevent movement of a larger object, see figure 4.9. These vary from very simple manual locking logs to automated variants in numerous sizes. Automated electromagnetic locking latches are very commonly used in the context of locking doors, but I did not manage to find any commercial even close to being solutions capable of holding the weight of the entire X8 boat.

Figure 4.9: Left: An illustration of a drum brake (Mägi, et al. 1, 2017). **Center:** A manual locking latch (Wallmart, n.d.). **Right:** An automated locking latch (Made-in-China, n.md.).

4.1.7 Conclusion

The *main* findings from the study of existing solutions were:

- How bridges use different geometries and components to withstand high vertical loads whilst not placing the supports directly underneath the load.
- A mapping of different ways industry uses to generate high force and motion in a straight linear direction (e.g. lead screws and winches).
- A mapping of different ways industry uses to facilitate linear motion along a certain axis whilst being constrained in others (e.g. linear bearings).

4.2 Study materials

Suitable materials for structural geometries (components later defined as *wagon* and the frame's *bracing*) were investigated and is presented in this chapter. These materials studies were performed to open up the design space as much as possible by considering less conventional structural materials. For this reason, conventional materials such as metal alloys was excluded in the concept level study. Material families that were excluded from the start were glasses, non-technical ceramics and technical ceramics due to their brittle nature and weakness against tensile loads (making them highly unsuitable for this application). The study used GRANTA Selector as a source (Version 21.2.0; 2021).

Exploration of materials was done with a specific strength (yield limit divided by density) to cost graph, see figure 4.10, and investigating promising alternatives within different material families. Promising alternatives are those with as high yield strength per density as possible, with an as low price per mass as possible. Out of the seemingly promising material families: woods, elastomers, polymers, composites, and foams, only some were deemed suitable. These include the materials (along with suitable manufacturing methods for low batch numbers):

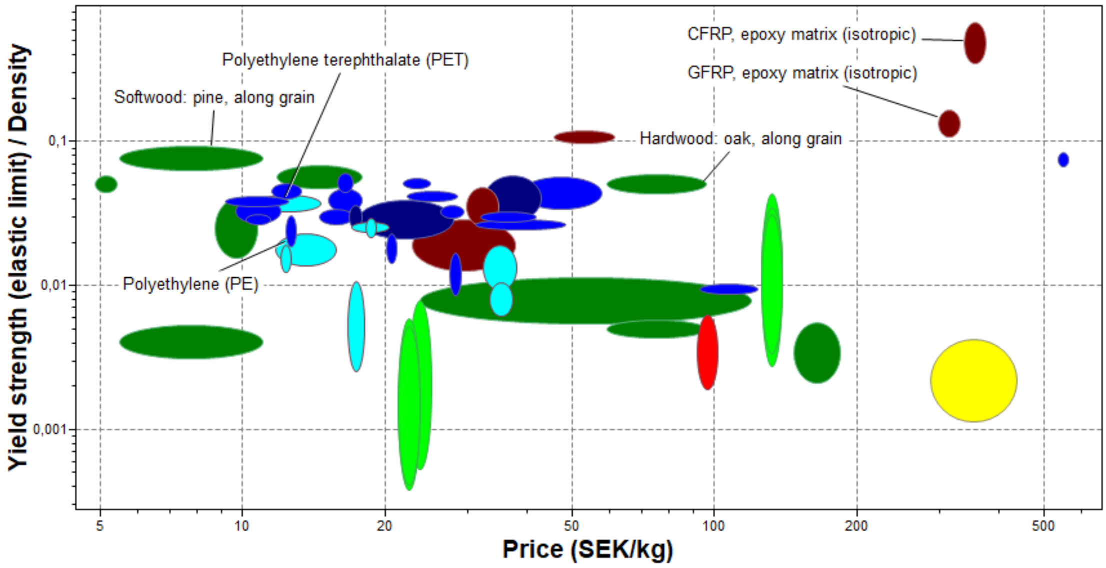


Figure 4.10: A mapping of common materials on a specific strength to cost chart. Materials are more promising further towards the top left of the chart.

- Oak (manual shaping using sawing, drilling, gluing and more).
- PET 45% glass fiber (forging or using standard components and joining through gluing or using fasteners).
- Polyester cast (casting and joining through gluing or using fasteners).
- Glass fiber reinforced polymer (GFRP) (manual molding and joining through gluing or using fasteners).
- Carbon fiber reinforced polymer (CFRP) (manual molding and joining through gluing or using fasteners).

In short, these materials were deemed suitable due to their stiffness (Young's modulus) being high enough, their specific strength compared to their cost being high enough, as well as them having manufacturing methods that enable the production of a frame-like form in low batch numbers. More comprehensive material number presentations can be found in appendix F. I would like to remind the reader that these materials are excluding metal alloys.

4.3 Concepts

This section is divided into general concept generation for the entire HDM and specific chunks within the HDM, which also includes the involvement of external companies' designs. Concept ideas are summarized in a morphological matrix. Afterwards, evaluation and motivation for elimination is presented, with a final concept being presented.

4.3.1 Early concept ideas

Early concept ideas involved viewing the chunks (or the entire HDM) at a high abstraction level. Whilst a moderately wide variety of solutions were generated, only a few of them are actually realizable. Even though the thesis assignment is rather loosely defined, the design limitations for a fully electrical automated solution and the geometrical limitations (total height as well as bow - stern space) entails that the solution space is deceptively limited. I would like to highlight four different concepts, see figure 4.11

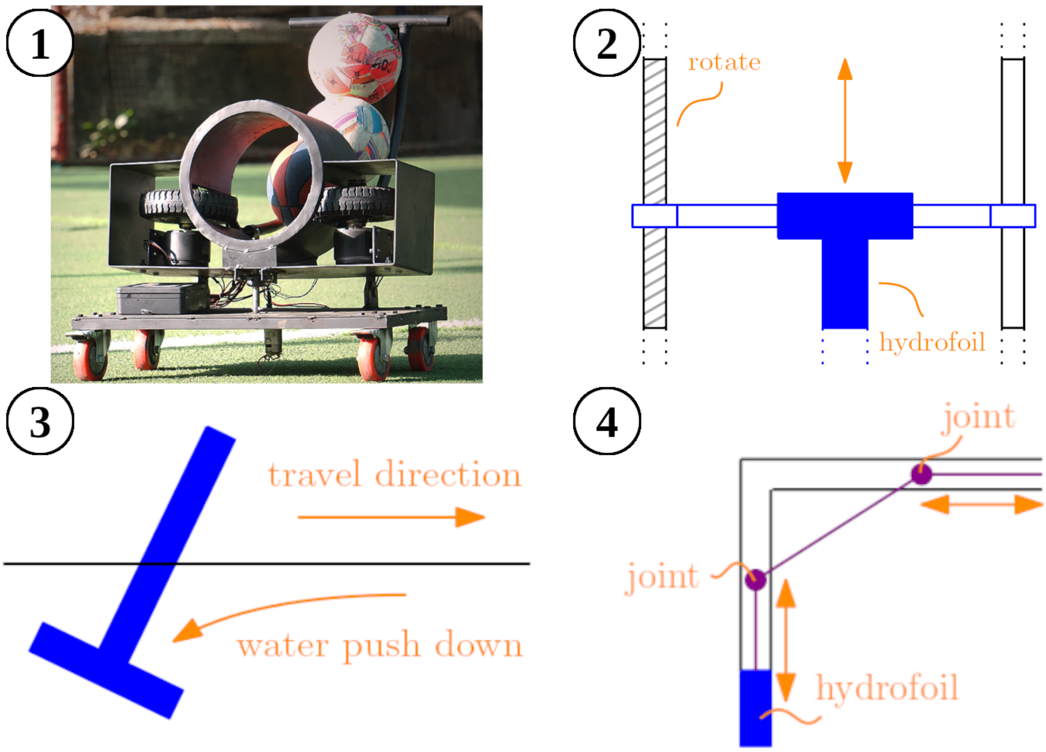


Figure 4.11: Early HDM concepts illustrating a **1:** thread / wheel system (Nevon Projects, n.d.), **2:** actuator not in line, **3:** trim foil self pulling, **4:** and lever concept.

1) Thread / wheel system: Squeezing the struts, or any vertical component, between wheels and raising and lowering by friction. This was discarded due to the struts not being available to touch (as doing so would interfere with their stabilizing movement) as well as the uncertainty of using friction as a method of transferring forces (since friction changes depending on factors like lubrication and material state).

2) Actuator not in line: A lead screw solution that raises and lowers the HDM whilst attaching it to two frame points, one on the lead screw and one on a linear motion support. The purpose of this particular design was to lower the amount of supporting shafts (for linear motion support) and thus lowering total weight. An alternate version of this was explored further in the project.

3) Trim foil self pulling: A solution that trims the entirety of the foil downwards as to change the vertical direction of the lift force from upwards to downwards (i.e. the foil pulls itself down). In this concept, the hydrofoil pulls itself down via inverted lift during motion and by mere gravity during stand-still. The hydrofoil is connected to a winch which would control the descent as well as pull the hydrofoil up to a retracted position.

Trimming the foil in such a way that it pulls itself down would necessitate a larger hole in the bottom of the hull. SSRS and SSPA found that the optimal geometry with respect to as a small hole in the hull as possible is a horizontally aligned wing that's lowered at a roughly $-10°$ angle (with a vertical reference line), see figure 4.12. As such, any solution that deviated from this geometry was disallowed.

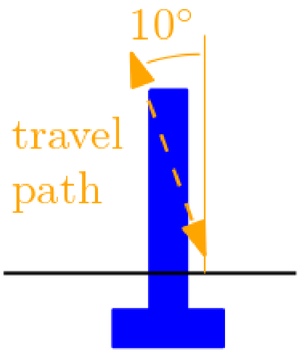
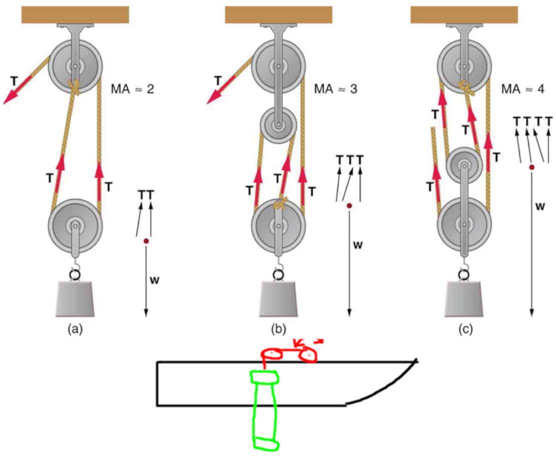


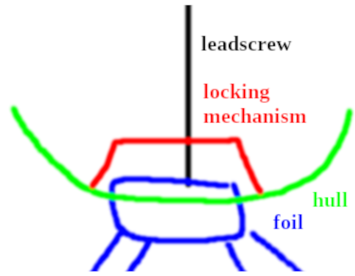
Figure 4.12: The hydrofoil's travel path is not perfectly vertical.

4) Lever concept: A lever mechanism that translates linear motion at an angle to linear vertical motion of the entire hydrofoil. An alternate version of this concept was explored further in the project.

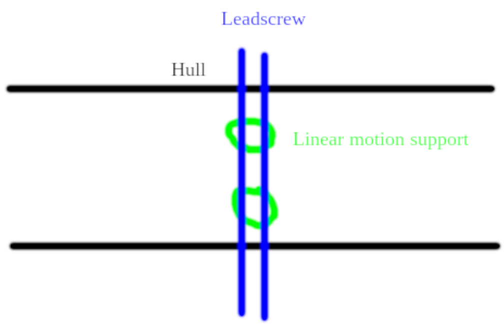
Early concept catalog: An excerpt of the full concept catalog of early concept ideas can be seen in figure 4.13.



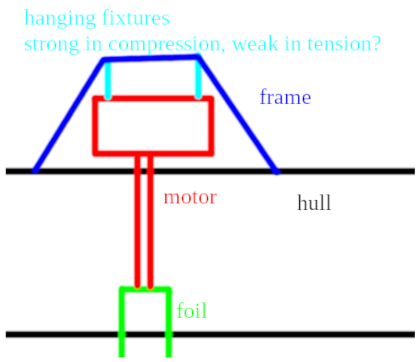
Pulley system with either chains or steel cables. Not necessarily for the force, but rather for the space and weight savings. (Urone & Hinrichs, 2012)



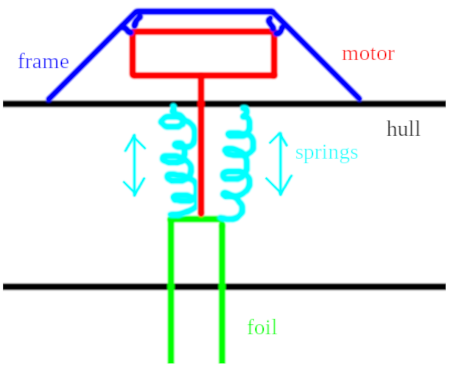
Thin leadscrew with locking mechanism. The thinness of the leadscrew is okay from a buckling perspective since it'll be locked by another mechanism. If no other support is added, it'll "only" have to withstand lifting the weight of the foil. But this solution is only acceptable if we don't want to be able to retract the foil whilst it's still flying (otherwise the leadscrew will have to lift the entire hull without buckling).



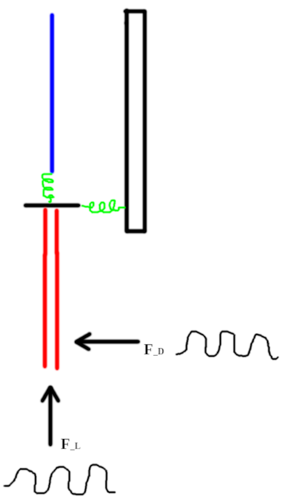
Leadscrew with linear motion support in places that are reasonable to prevent buckling. How will such a support handle the non-smooth and moving geometry of a screw?



Leadscrew motor hanging loosely to the frame. Weight of motor helps push hydrofoil down. Attach motor on frame instead of on hydrofoil to help bring electronics away from water.



Springs that are compressed when pulling up and that decompress to help push the foil down.



Springs to counter fatigue? Both from drag force and lifting force. Unsure how big fatigue amplitudes are.

Figure 4.13: An excerpt of early concepts.

4.3.2 Mid concept ideas

Mid concept ideas revolved heavily around the geometry of the frame and linear motion support. For simplicity's sake, the HDM in this phase is assumed to be driven by some form of lead screw with rail units as linear motion supports.

The mid concept phase was initiated with generative design using ANSYS Topology Optimization (version 21.1.0; 2021) where a solid block of material, roughly resembling the shape of the allowed design volume, was optimized by ANSYS to achieve as low of a weight as possible (i.e. remove material). See figure 4.14 for an illustration. Optimization was performed by fixing the surrounding frame area to the hull and applying a large vertical load and a slight horizontal load (to account for any drag forces and forces that may occur when the boat is leaning). The ANSYS algorithm deemed it most suitable to generate a geometry roughly resembling four struts, two vertical and two leaning, attached to a top place (where the force load was applied).

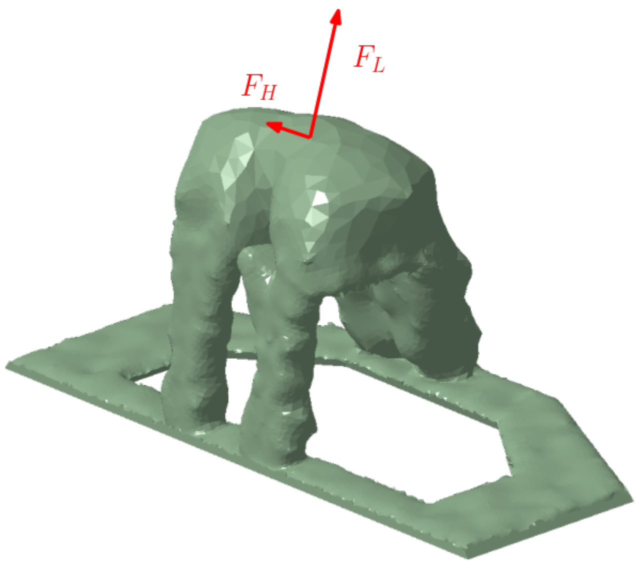


Figure 4.14: A computer-generated shape of a potential frame design made to withstand applied loads. F_L represents a vertical lifting force from the hydrofoil and F_H represents a smaller horizontal force that occurs whilst the boat is leaning.

I would like to highlight five different concepts, see figure 4.15. On all concepts the gray can be regarded as a frame and linear motion supports, the blue can be regarded as a lead screw and the red can be regarded as the connection to the HDM. In all concepts the boat's travel direction can be assumed to be towards the upper right corner of the image.

Foxtrot: A relatively simple design with a linear support towards the bow and one towards starboard, as well as a frame support towards the stern. This design's main benefit is the reduction in weight due to a relatively low number of supports. Seeing as the drag forces and potential portside - starboard forces when the boat is leaning

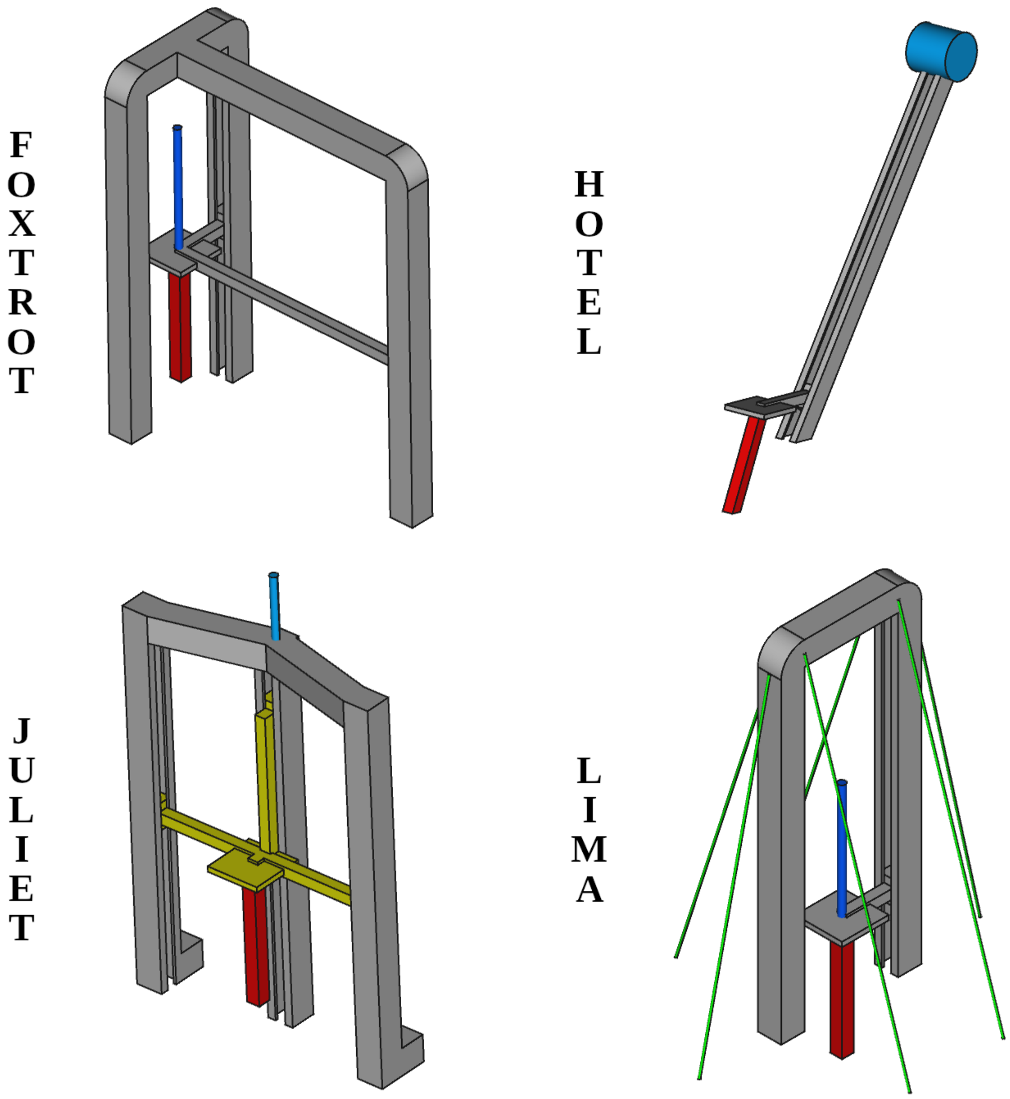


Figure 4.15: 3D models of early frame concepts.

are low comparatively to the vertical force, one bow and one starboard arm could be more than enough to counter those loads.

Hotel: A continuation of the trim foil self pulling concept from the early concept ideas.

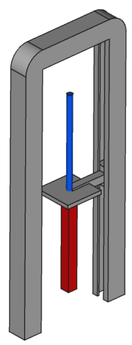
Juliet: An optimized frame variant of foxtrot with respect to the length of the levers. The portside - starboard and stern - bow levers are shortened to save weight and the bottom - topside lever is lengthened to lower the radial forces (originating from torque) of the bow railing unit. The lengthened aspect was later discarded due to it requiring too much vertical space on the boat.

Lima: A concept demonstrating the use of wires for structural stability (inspired by studying cable-suspended bridges), as a weight-saving alternative to solid (or hollow) beams.

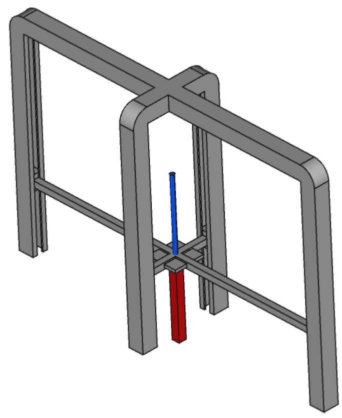
Oscar: See appendix H. A continuation of the lever concept from early concept ideas. This concept was discarded due to it requiring a lot of bow- and stern space neighboring the hydrofoil (which is occupied by batteries) in addition to presumably requiring an unreasonably strong actuator and beams to overcome the leverage force of the system.

Mid concept catalog: An excerpt of the full catalog of mid concept ideas can be seen in figure 4.16. All shown frame concepts are based on a leadscrew actuator-solution and rail for linear-motion-solution (for simplicity as the frame is in focus). The blue component represents the leadscrew and red component represents the hydrofoil.

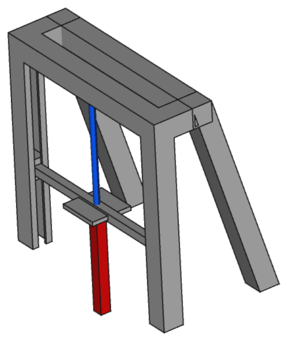
Mike: A dome-shaped frame inspired by the shape of arch-bridges, see figure 4.16. Whilst an interesting idea, it increases complexity and is perhaps more suited for structures built with materials that have high compressive but low tensile strength. Such materials (ceramics) have been excluded from this thesis due to them being unsuitable because of their brittle nature (see chapter 4.2).



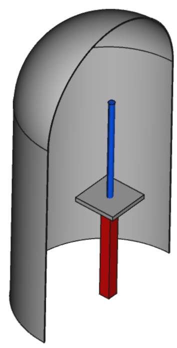
Epsilon: If using a leadscrew to lock, the leadscrew would also be under torque. Might be bad for buckling. It might be so that the majority of the moment is taken up by the rail assembly, if that part is more stiff than the leadscrew. Only requires one arm on the front because the rail would lock regardless.



Golf: Same as epsilon but arm on both sides and front and back. Requires less sturdy railing components.



Kilo: Same as Golf but built into the cockpit.



Mike: Sectional view. Using the geometry of a dome to achieve strength whilst minimizing material usage (and thus weight).

Figure 4.16: An excerpt of concepts focusing on frame geometry variants.

4.3.3 Trim concept ideas

I would like to highlight two different concepts in regards to trim, see figure 4.17. In the figure the red is the hydrofoil, the green is a connecting hub between the hydrofoil and the rest of the HDM, and the orange-yellow is some form of electric motor.

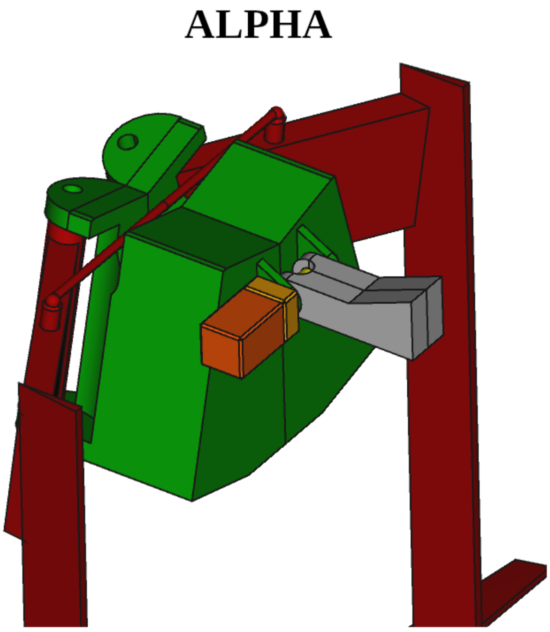


Figure 4.17: Trim concepts alpha and charlie.

Alpha: A concept idea where a rotational motor and gearbox is attached to the hub and the HDM, transferring rotation to the hub which in turn trims the hydrofoil. Such a motor and gearbox would be encased in some fort of water proof box or be of a very high IP rating. This concept was discarded due to its design being more aimed towards large scale trim or rotations.

Charlie: A concept idea where a short linear actuator is attached to the hub and the rest of the HDM in such a way that that an expanding or contracting linear actuator would rotate the hub and thus trim the hydrofoil. This trim concept was chosen as a similar solution is established to work in a marine environment (to trim flaps on boats (Watski, n.d.)).

4.3.4 Linear motion concept ideas

Different combinations of geometric placements and linear motion unit types were generated using simple sketches. A full catalog of linear motion concepts can be found in figure 4.18. All linear bearings are assigned the same symbol as they can all take up the same *type* of loads. A "dovetail" can be described as a combination of a linear plain bearing and a rail, i.e. it has high friction but a high maximum load as well as being able to withstand loads in more torque directions than linear bearings.

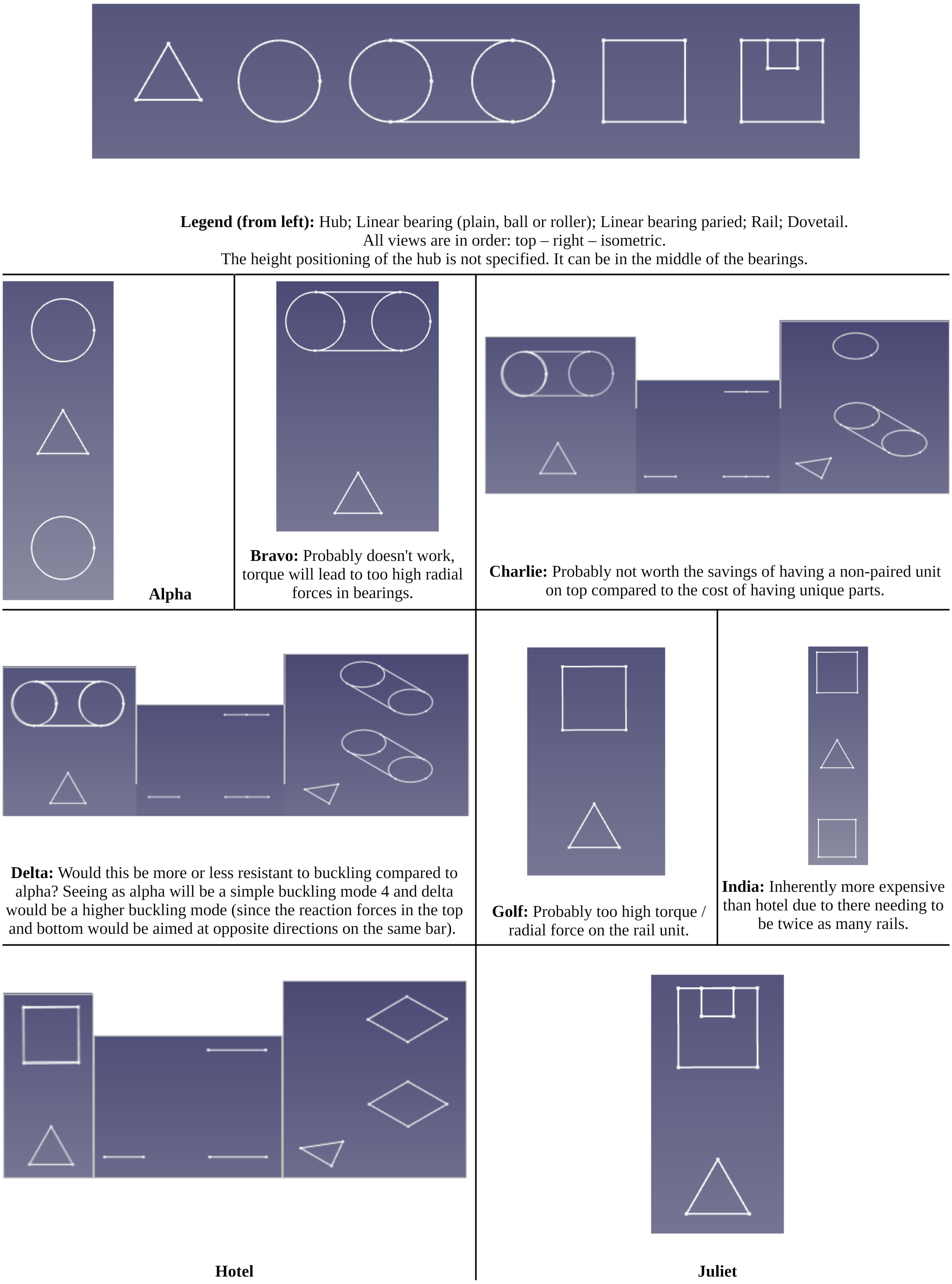


Figure 4.18: Concept catalog for linear motion support. Catalog shows a top-down and sometimes isometric view of the placement of linear motion supports in relation to the hub (i.e. where the hydrofoil is attached).

Costs for different unit types were sourced from Ewellix (personal communication, S. Uggla, March 17, 2022). On an overall level the costs of the linear units are negligibly cheap but the cost of the shafts or rails are not. Units which prices were sourced, with codes corresponding to Ewellix item numbers, can be seen below. Note that the thesis excludes any pricing due to confidentiality reasons.

1. Ball unit – LUCR 20 D
2. Plain unit – LUCR 20 PA
3. Shaft – LJMR 20x2000
4. Rail ball unit – LLTHC 20 A T0 P5
5. Rail – LLTHR 20 2000 P5

With torque originating from the drag of the hydrofoil roughly calculated, additional calculations for the radial forces and consequential friction in linear motion supports can be performed. Using an estimated lever (to counteract torque generated from the hydrofoil) that seems reasonable to fit in an HDM design, calculations for radial forces in linear units were performed. Given a wing to hub lever of 1.3 [m], a drag force of 813 [N], two upper bearing and two lower bearings all with a hub to bearing lever of 0.1 [m], calculations were made (see figure 4.19). The calculated radial force is in the ballpark of around 2000 [N]. Such a radial force would with plain linear bearings generate a friction of roughly 400 [N] per bearing, something that I have regarded to be negligibly small compared to other vertical forces.

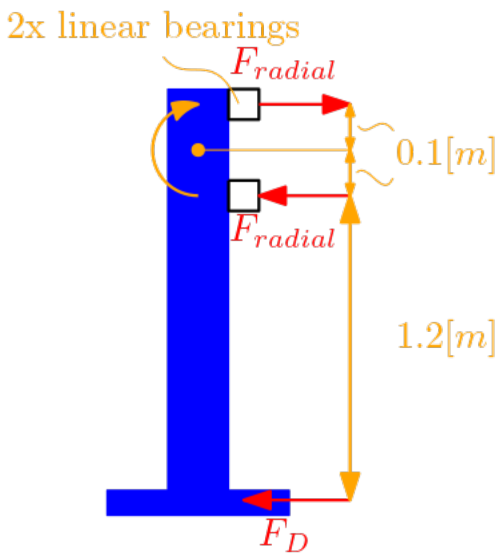


Figure 4.19: An illustration of how the drag force with its lever will generate radial forces in the linear bearings.

Evaluation of generated concepts were done using a custom-made evaluation matrix. The matrix contains evaluation criteria (on its rows) which I estimate the fulfillment of for each concept (on its columns) on a scale from 1 – 5; the top score is 5 and its reference is equal to the highest performing concept (i.e. the best concept will score a 5 and the worst concept will score a 1). The criteria were evaluated numerically where possible (e.g. cost, friction coefficient, etc) and otherwise estimated. See figure 4.20 for evaluation matrix. The evaluation criteria used are elaborated upon in the list below. Note that weight is not evaluated as all concepts' weights are relatively similar.

- Water resistance: How sensitive the equipment is to water. This includes the effectiveness of wipers.
  - Dovetail slides was rated 2 due to lubrication possibly being washed away by water.
  - Rails were rated 3 due to their wipers and scrapers being less effective than that of linear ball bearings (source personal contact Ewellix 2022-03-14).
- Cost: Estimated for the units, the bars/railings/tracks, and the total number of units / bars. Maintenance was not included in the cost.
- Overall volume: All bars / rails on one side 5, opposite sides 3.
- Friction: In regards to the numbers gained from Lundin & Eriksson (2021).
- Design flexibility: If the design can be altered by switching components at a later stage. Only possible for plain vs ball bearings.
- Radial strength: Numbers gained from Lundin & Eriksson (2021).
- Unique components: If there are multiple types of units. Accessories not accounted for. Fewer unique components is regarded as better.
- Torque starboard: Torque strength if the torque vector is aimed towards starboard. Concepts with multiple units (e.g. delta) were rated higher than ones with fewer (e.g. alpha) since the resulting radial force will be distributed among more units. Concepts with one illustration have their units regarded as not being on the same altitude.
- Torque bow / stern: Torque strength if the torque vector is aimed towards bow or stern. Concepts with one illustration have their units regarded as not being on the same altitude.
- Torque topside / below: Torque strength if the torque vector is aimed towards topside or below. Concepts with one illustration have their units regarded as not being on the same altitude.

The evaluation shows that the most promising concepts are those with simple linear bearing units (plain or ball) that are of the same type (as to lower total number of unique components). Additionally, linear units must be placed as such to best prevent any generated torque from the hydrofoil.

Even though the alpha concepts scored the highest, the final HDM concept is using concept delta for a solution to provide linear motion support. This is due to delta having twice as many linear bearings, entailing that any radial loads experienced

Evaluation matrix

Linear motion support Ranked from **1** (worst) – **5** (best) and graded with red (unacceptable), orange (borderline) or green (acceptable).

Concepts→ Criteria ↓	Weight	Linear alpha plain	Linear alpha ball	Linear charlie plain	Linear charlie ball	Linear delta plain	Linear delta ball	Linear echo plain	Linear echo ball	Linear foxtrot plain	Linear foxtrot ball	Linear golf	Linear hotel	Linear india	Linear juliet
Water resistance	2	5	4	5	4	5	4	3	3	3	3	3	3	3	2
Cost	4	5	5	4	4	4	4	2	2	2	2	3	2	2	3
Overall volume, of frame design.	1	3	3	5	5	5	5	3	3	3	3	5	5	3	5
Friction	1	1	4	1	4	1	4	3	4	3	4	5	5	5	3
Design flexibility Possibility of later design changes.	1	5	5	5	5	5	5	5	5	5	5	1	1	1	1
Radial strength	2	3	2	3	2	3	2	4	3	4	3	4	4	4	5
Unique components	2	4	4	4	4	4	4	3	3	3	3	4	4	4	5
Torque starboard	2	3	3	3	3	4	4	4	4	3	3	2	4	4	5
Torque bow / stern	1	4	4	4	4	4	4	4	4	4	4	2	4	4	5
Torque topside / below	1	5	5	3	3	3	3	5	5	5	5	1	2	5	4
Sum points		**68**	**67**	**64**	**63**	**66**	**65**	**56**	**55**	**54**	**53**	**52**	**55**	**56**	**64**
Worst color & amount of worst color		**1**	**1**	**2**	**2**	**2**	**2**	**3**	**3**	**3**	**3**	**3**	**4**	**3**	**1**

Figure 4.20: Evaluation matrix used to guide eliminations regarding linear motion supports.

by the bearings will be halved. Choosing delta allows the HDM to be compatible with a wider range of linear bearing components (as it is not limited to only be compatible with bearings that have a high radial-load-tolerance).

4.3.5 Lock

Concepts around locking were investigated. Locking refers to holding the hydrofoil in place in an either retracted (up in the boat) or deployed (down in the water) position. Note that whilst the hydrofoil is deployed, it will have to withstand an equivalent weight of roughly 2.6 tons.

I had great difficulties in finding commercially available locking solutions for such a high load, let alone for a marine environment. The only two promising solutions were to either use the built-in locking mechanism of an actuator or to design an own custom locking mechanism. Using existing smaller-scaled locking solutions as inspiration, I designed a locking mechanism from scratch.

Linear bearing concepts: Concepts were generated using a short-stroke actuator to push a rod into a locking-slot via a linear bearing, as can be seen in figure 4.21. These concepts were deemed unpromising by myself as they had potential problems with torque (#1), problems with requiring a far-too-large plain bearing of roughly 80 [mm] to handle the applied load (#2), or problems with completely extruding and inserting the locking rod from the plain bearings (#3).

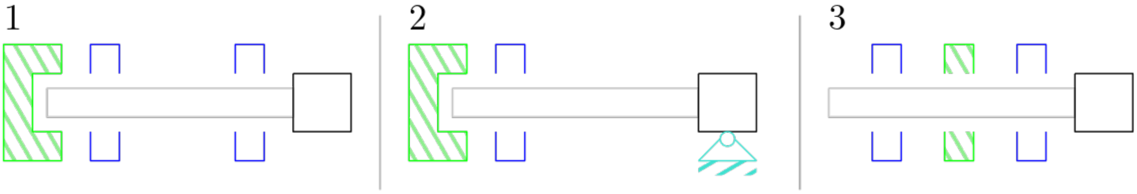
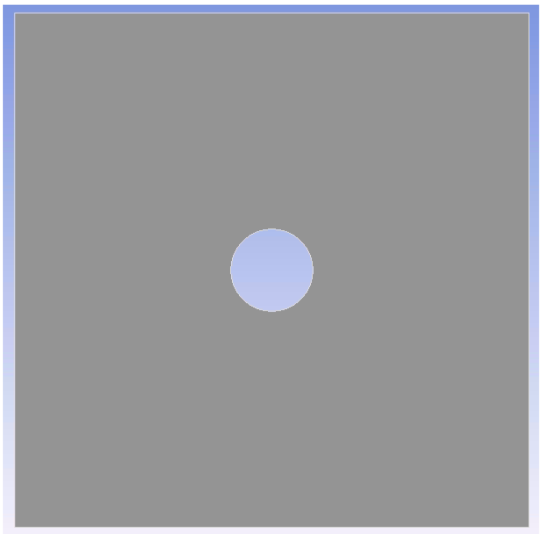


Figure 4.21: Images of locking concepts using linear bearing(s) as support. **Gray:** Axially moving rod. **Green:** Slot for rod to be inserted into. **Blue:** Linear bearings. **Black:** Actuator.

Hole without bearings concept: A stress analysis of designing a similar locking solution without any plain bearings was performed via ANSYS (version 21.1.0; 2021), see figure 4.22. The analysis was performed to determine how large would a slot's diameter have to be to have a factor of safety = 3 against a 26 000 [N] bearing load. Simulations were performed on a hole in a very large plate in 2D with a thickness of 20 [mm]. The bottom edge of the plate, see figure 33, was fixed and the hole was assigned a vertical bearing load. Simulations showed that required diameter would be 25 [mm], i.e. several times smaller than with a plain bearing. Further geometrical simulations and numbers can be found in appendix I.



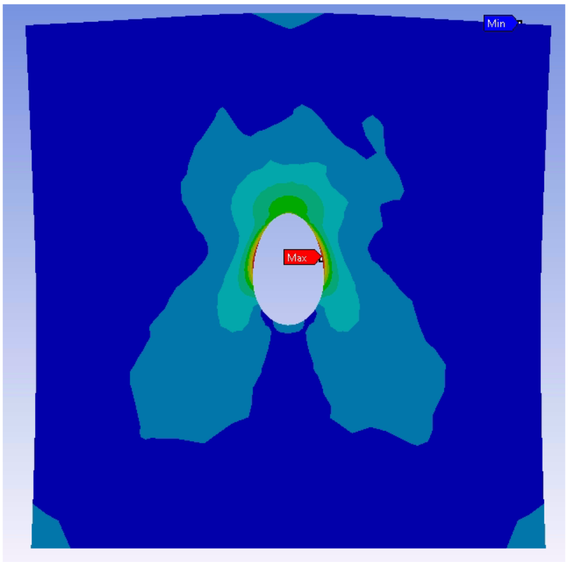


Figure 4.22: Left: Geometry of plate with hole. **Right:** FoS simulations on a hole that's radially loaded.

Inserting and extracting such a bar without any bearing would entail overcoming frictional forces, thus a brief investigation regarding frictional coefficients between different materials and their generated frictional force in this scenario was performed, see table 4.1. Frictional coefficients were sourced from the Engineering ToolBox (2004).

Material 1 **Material 2**	**Static dry** **Friction**	**Static lubed** **Friction**	**Sliding dry** **Friction**	**Sliding lubed** **Friction**
Hard steel Hard steel	0.78 20280	0.11 (oil) 2860	0.42 10920	0.029 (oil) 754
Mild steel Copper	0.53 13780	-	0.36 9360	0.18 (oil) 4680
Steel Brass	0.51 13260	0.19 4940	0.44 11440	

Table 4.1: Different frictional coefficients (μ) between different materials, lubrications and movement conditions.

A lubricated solution would produce a low enough friction to be overcome by a reasonably sized actuator, but such a lubrication would most likely be washed away from any splashing sea water. Additionally, even though a certain material combination would entail lower friction, it would also entail lower strength than a pure steel combination. An alternative to using different metal alloys could be to infuse the surface of a metal with a polymer to act as a long-lasting lubricant (SKF, n.d.).

Bar solution: To attempt to counteract any bearing problems with linear bearing concepts and any frictional problems with hole without bearings concept, I developed a bar concept, see figure 4.23. This concept entails an attachment somewhere on the hydrofoil (orange) being lowered next to similarly shaped attachments on the hull (blue). An actuator (gray) then pushes a locking bar (purple) into all attachments thereby locking the hydrofoil attachment to the hull attachments.

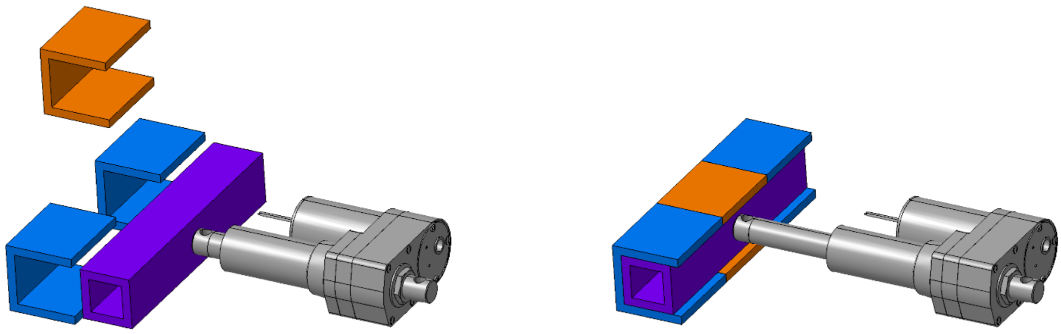


Figure 4.23: Locking - Bar solution. **Orange:** Hydrofoil hub. **Blue:** Fixture attached to the hull. **Purple:** Locking bar. **Grey:** Actuator CAHB-20 (Ewellix 1, n.d.). **Left:** Unlocked. **Right:** Hydrofoil is lowered and is locked into place.

To solve eventual frictional problems, all contact surfaces could be lined with SKF strips (SKF, n.d.). These are PTFE composite strips that are attached with adhesive to a metal. They require no lubrication and lower the frictional coefficient to between 0.03 - 0.25. They are highly resistant to sea water and can support a maximum load of 250 [N/mm^2] (which would add up to 1 250 000 with the presented geometry). Such a solution would produce frictional forces between 780 - 6500 [N] which are reasonable to overcome. The total weight for such a bar-locking solution is roughly estimated to be around 15 [kg] (with the Ewellix actuator CAHB-20 weighing 5.5 [kg] (Ewellix 1, n.d.)).

It is unclear at the time of thesis publication if an external locking mechanism or a locking mechanism built into the actuator will be used.

4.3.6 Actuator - Lead screws

Different geometries for lead screws, and corresponding linear motion support shafts, were briefly investigated, see figure 4.24. Left - right in the figure can either depict bow - stern or portside - starboard, in this simple case it is of no consequence.

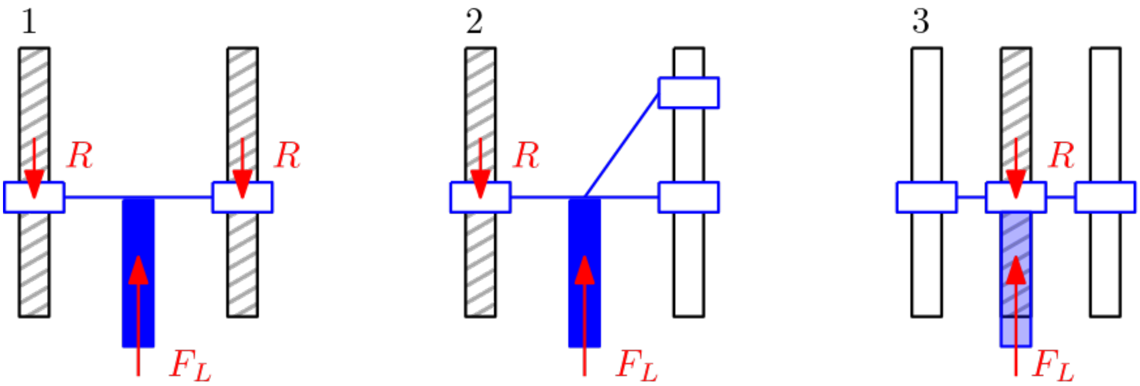


Figure 4.24: White: Friction-free shaft. **Grey:** Lead screw. **Blue:** Hydrofoil. Different lead screw constellations.

1. Two smaller lead screws and corresponding motors. Torque from the hydrofoil lift is counteracted by both lead screws taking up an equal load.
2. One lead screw and one linear motion support. Since the support on the shaft would travel more freely than that of the lead screw, it would generate unwanted torque in the lead screw. This is countered by the actuator and hydrofoil interface being designed as a branch, such that it converts the generated torque into radial loads on the linear support shaft.
3. One lead screw and two linear bearings. Torque is not generated from hydrofoil lifting force.

Out of these three combinations, the third option is regarded most promising for its simplicity and robustness, as well as its geometry allowing it to withstand additional force vectors that may occur when the boat is leaning.

Rack and pinion system: I regarded a rack and pinion system to be similar enough to a lead screw as to not investigate it further; the similarities mainly revolving around that both the rack and a lead screw with a traveling screw require a solid piece of metal to be traveling linearly to provide linear motion to the hydrofoil (equivalent weight, stiffness, geometry and buckling). Furthermore, a rack and pinion system cannot be installed in the struts similarly to the commercially available Candela C7 as the hydrofoil struts in the X8 design will twist and be in motion during usage.

4.3.7 Actuator - Winches and hoists

Winches and hoists generate linear movement by attaching a rope (or chain) from its drum to the object which it is wishes to move. Rotating the drum pulls the rope which in turn pulls the object. The direction of linear movement can be controlled by running the rope via pulleys. A hoist is a type of winch that is designed to raise and lower heavy loads vertically, for example in a ceiling-mounted setup or as a crane, and it achieves this by including mechanical braking in its design (as to lower loads slowly). A winch is generally designed to only pull and hold in one direction (for example via a ratchet brake) and completely release in the other (for example via a clutch). For simplicity's sake, all variants of winches or hoists are now called ‘’winches” in the rest of the report.

Achieving linear motion in this design scenario with winches is possible but not so straight-forward, namely because a rope can only support tensile loads and not compressive loads. There are three main concepts briefly investigated regarding winches, see figure 4.25.

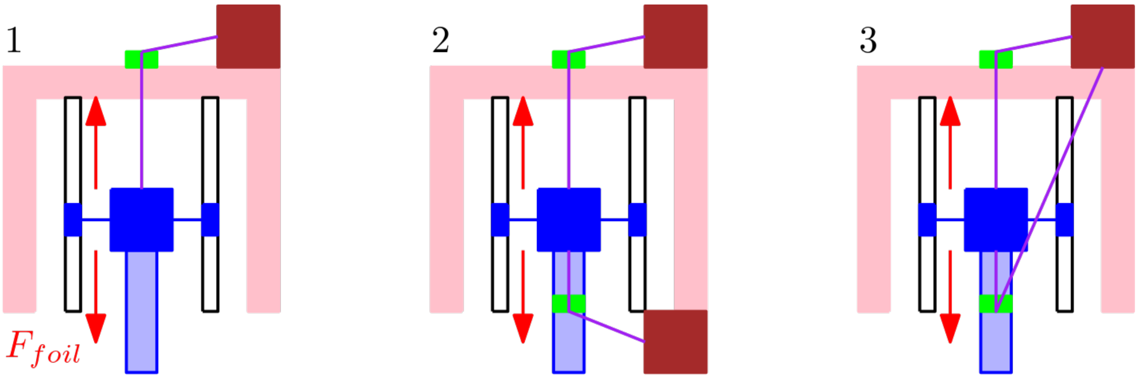


Figure 4.25: Pink: Support structure. **White:** Shafts. **Blue:** Hydrofoil. **Brown:** Winch. **Green:** Pulley block. **Purple:** Rope. All unmarked forces are rope forces generated by a winch.

1. A winch that is connected to the top of the hydrofoil and some sort of framed structure (that's in turn connected to the hull). The winch would then brake as the hydrofoil pulls itself down (if trimmed) during deployment and pull the hydrofoil up during retraction. This concept was not investigated further due to a self-pulling hydrofoil via trim was disallowed by the client (see concept 3 in chapter 4.3.1).
2. Two winches that are connected to two sides respectively of the hydrofoil via pulleys. This allows one winch to pull the foil downwards and another winch to pull the foil upwards. Using internal locking within the winches, the winches can also hold the hydrofoil in place during its deployed or retracted position. This concept was not investigated further due to the redundancy of winch units increasing weight and cost.
3. A winch that is connected to two sides of the top of the hydrofoil via pulleys. This allows the winch to pull the foil in two directions by rotating the drum either clockwise or counter-clockwise. Using internal locking within the winch, the winch can also hold the hydrofoil in place during its deployed or retracted position. This concept was explored further in the project.

4.3.8 Actuator evaluation

An evaluation on actuators was performed to guide focus in the project. Actuator types I deemed relevant to use in this project were: lead screw (traveling nut), winch, multi-stage hydraulics and multi-stage pneumatics. An evaluation matrix was used to rank actuator types against each other. See figure 4.26 for the evaluation matrix. I found additional reasons to disregard fluid actuators other than total number of unique components required, and found that a winch is superior in theory to a lead screw. Both winches and lead screws were investigated further in this project.

Evaluation matrix

Motor

Ranked from **1** (worst) – **5** (best) and graded with **red** (unacceptable), **orange** (borderline) or **green** (acceptable).

Concepts → **Criteria ↓**	**Weight**	**Lead screw** **(traveling nut)**	**Winch**	**Multi-stage hydraulics**	**Multi-stage pneumatics**
Water resistance	3	3	5	2	2
Weight	3	3	5	1	2
Torque robustness	2	3	5	2	2
Total number of components	2	4	5	1	2
Cost	2	3	5	1	2
Overall volume, of frame design.	1	4	5	3	3
Power	1	4	3	5	2
Sum points		**46**	**68**	**25**	**29**
Worst color & amount of worst color				**4**	**1**

Figure 4.26: Evaluation matrix of actuators. Summation of points are calculated by multiplying a ranking with a weighting (where the weigh indicates the importance of the criteria).

4.3.9 Avoid collision

One of the wishes (not demands) from the client is for the HDM to retract very quickly in emergency situations to avoid collisions, for example if there is heavy debris in the water. Such a retraction is estimated to be used very sparingly, and as such it would be deemed acceptable for such a retraction to cause minor damage to the vessel if major damage can be avoided. I generated concepts to fulfill this wish, some of which are presented in figure 4.27.

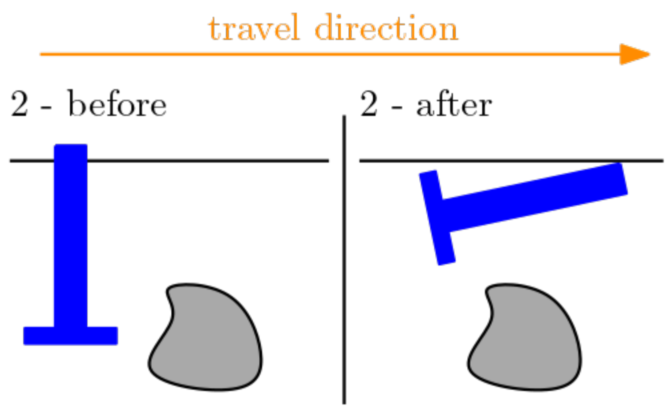


Figure 4.27: Collision avoidance concept 2 visualized. Grey is an obstacle in the water.

1. Intentionally cause a crack in the hydrofoil struts, for example via a spring-loaded ax mechanism. Such a crack would control the location of the failure causing the hydrofoil to rupture, and thus prevent more costly damage to the hull. This concept was not investigated further in this project due to its high cost of activation.

2. Release the trim of the hydrofoil causing it to fall back due to drag and take up less vertical space (and hopefully avoid collision). Such a solution would risk minor damage to the hydrofoil and the hull should the hydrofoil slam hard into the hull. It would also not guarantee the hydrofoil avoiding a potential collision as the hydrofoil is still submerged in water. Finally, it would cause heavy loads for a short period of time as the wing in such a position would cause massive drag forces to occur. This concept was not investigated further in this project due to the aforementioned reasons.

3. By somehow disconnecting the motor, braking and locking mechanism, allow the hydrofoil to quickly shoot into the hull simply by the forces applied to it. I would like to remind the reader that the hydrofoil is always exposed to a heavy upwards force whilst in motion, so removing any type of locking mechanism (whist still maintaining vessel speed) would entail that the hydrofoil would shoot up into the hull. Such a design is most easily implemented with a winch solution as activating it can be achieved by removing any internal winch-brakes or to merely pull the clutch.

4.4 Morphological matrix

An early concept-stage morphological matrix was created, but one that acts more as a visualization of possible concepts as opposed to being a tool to generate further concept combinations. I discovered that the HDM can be approached as a very modular system, with different variants of chunks (such as frame or actuator) being interchangeable without it grossly affecting others. For that reason, performing combinations of chunks or modules to generate entire concepts is not as meaningful, but time is better spent evaluating alternatives within each chunk. The morphological matrix can be found in appendix J.

A future morphological matrix containing all functions and solutions is not included in the thesis due to the overwhelming size such a matrix entails.

4.5 Evaluation of late-stage concept designs

Preface: At this point in the project I was focusing on the main actuator for the HDM and had filtered solutions down to either lead screw or winch.

Late-stage concept designs primarily revolve around the usage of specific components and models from OEMs. Realizing concepts via availability of parts was at this stage of the project regarded as a deciding factor in choosing an appropriate concept to pursue.

From the client's perspective the design of an HDM must be completed regardless if it comes from my master's thesis or from another external company. For that reason, my design partner Sigma Embedded Engineering shouldered the responsibility of outsourcing possible actuator solutions from external companies. Solution types outsourced would be lead screw variants whilst I would pursue winch-type solutions.

In this section two different conceptual solutions are presented and are evaluated against each other.

4.5.1 Company A

A major company in linear motion, company A, was contacted as a possible design partner for the HDM actuator. Despite lengthy coordinating meetings and plenty of calendar time, the contacted company was not able to produce a solution that would sufficiently solve the requirements for an HDM actuator. The two major problems the company struggled with was producing a solution that was able to handle the salt water from a marine environment, and to keep the overall footprint of the solution down to the required volume. Company A attempted to design an HDM primarily using lead screws as actuators.

4.5.2 Company B

A company producing winches. This company was contacted and the applicability of their winches was discussed. The relevance of this company is presenter further along in this chapter.

4.5.3 Company Servomech

Another major company in linear motion, Servomech, was contacted as a possible design partner for the HDM actuator. I generated a concept (by piecing together catalog components) in parallel to the company designing their own. Both concepts turned out to be very similar which increased the confidence that such a solution is the optimal available given Servomech's components. See figure 4.28 for a visualization of the proposed solution.

The designed actuator solution entails:

- Lead screw with a traveling nut of the acme screw type. I.e, the traveling nut is not a low friction ball bearing but rather a high friction threaded nut.
- Lead screw is attached to a jack which in turn is attached to an electric motor. I.e., the jack translates the horizontally aligned electrical motor's rotation into vertically aligned rotation of the lead screw.
- Protective bellows are attached to both sides of the traveling nut, ensuring that minimal or no water gets in contact with the lead screw or the jack's internal parts. These bellows expand and contract as the screw is traveling along the length of the lead screw.

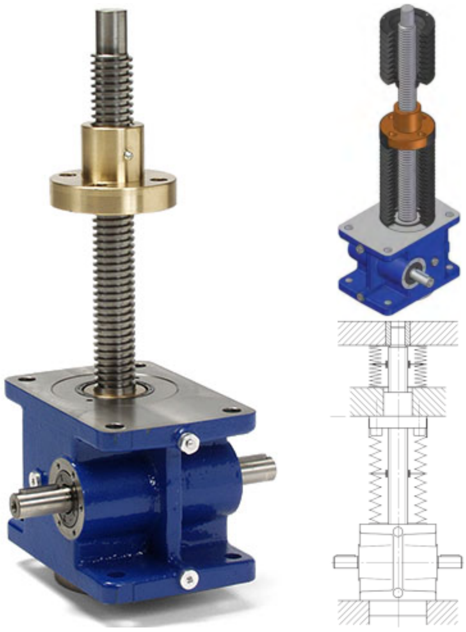


Figure 4.28: The key module within Servomech's solution. A jack transforming a rotational input (see grey axle for attachment point) into linear motion by means of a traveling nut on an acme screw. Bellows attached to the top and bottom of the nut ensures corrosion resistance. (Servomech, n.d.)

- The lead screw is produced in stainless steel, to act as a safety mechanism against corrosion should the bellow fail.
- The nut is produced in brass, to lower friction with the lead screw and to act as a safety mechanism against corrosion should the bellow fail.
- The self-locking properties of the acme screw, in addition to brakes in the motor, would enable the locking mechanism to be integrated into the actuator solution.

In summary, Servomech's proposed solution is a marine-tolerant actuator solution with an integrated braking system.

4.5.4 CWA - Concept winch alpha

I designed a winch-based concept with the internal name CWA (concept winch alpha). This concept primarily uses the winch-related components from company B and linear motion support components from company A. See figure 4.29 for a visualization of CWA.

- CWA uses a bi-directional automated winch (with a manual override) with two outgoing ropes to provide linear motion in two directions (given clockwise and counter-clockwise rotation of the drum). These outgoing ropes are connected to the top and bottom of the wagon via pulleys.
- The wagon rides on two shafts with a total of four linear bearing units, the

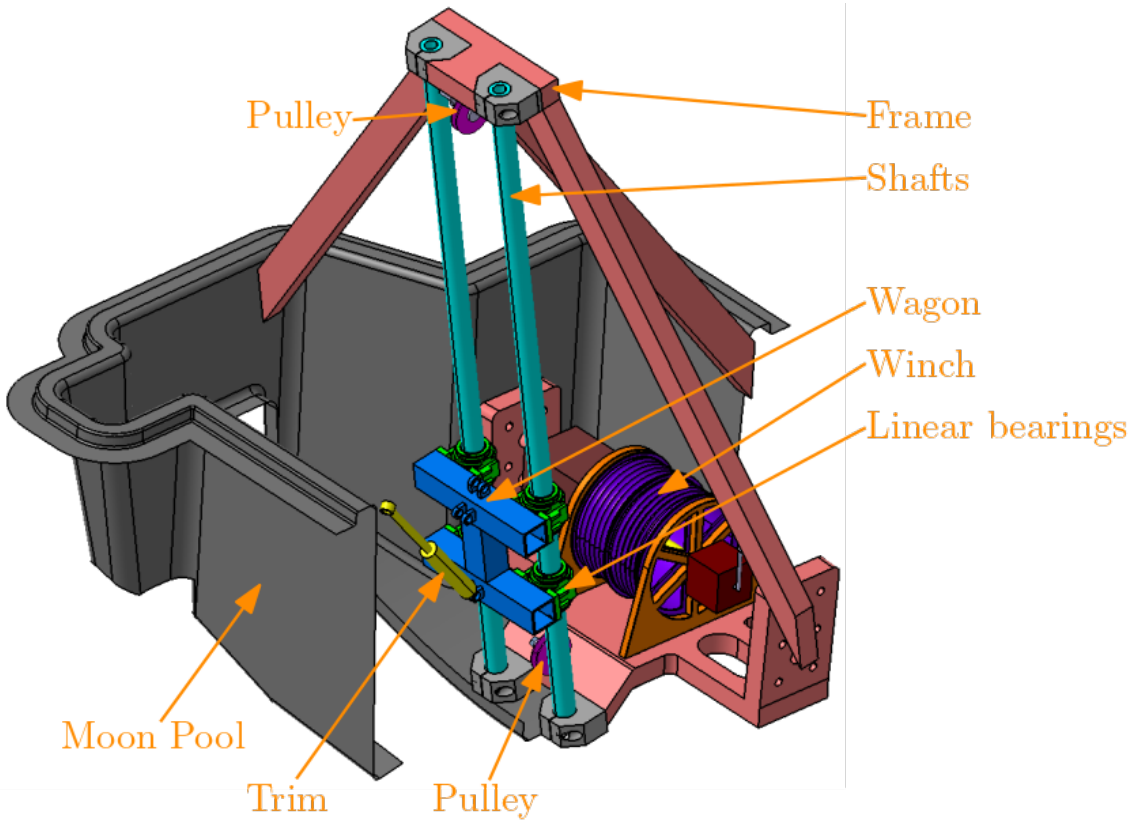


Figure 4.29: Concept winch alpha. The Moon Pool (gray) is viewed in a sectional view as to not obscure the CWA. The hub or hydrofoil is not included in the image, but would be attached to the trim (yellow) and the wagon (blue). Ropes are not included in the image.

geometry of which is to counter torque from drag forces as well as any potential lateral forces that occur should the boat start to lean.

- A short actuator controlling trim is mounted between the wagon and the hub. The attachment point between the hub and the wagon (as well as the actuator and the wagon) is hinged to allow for rotation.
- The locking mechanism for this solution is integrated into the winch braking system.
- The rope intended to be used for this solution is synthetic rope, to reduce weight.

4.5.5 Winch vs lead screw

Myself and involved parties are in agreement that both the lead screw and CWA concepts would manage to produce a successful design. However, both are not as likely to be equally successful. A qualitative evaluation between the concepts was made with respects to weight, center of mass, price, maintenance, and collision avoidance. See figure 4.30 for a summary of the concept comparison.

Winch vs lead screw	Green = Win | Orange = Draw Red = Lose.	
Solutions → Criteria ↓	Lead screw (traveling nut)	Winch
Weight	Red	Green
Center of mass	Red	Green
Price	Red	Green
Maintenance	Orange	Orange
Collision avoidance	Red	Green

Figure 4.30: A lead screw and a winch with a synthetic rope compared

- *Weight:* Main difference is that a steel screw weighs more than a synthetic rope. Both otherwise include equivalently large electrical motors and gearboxes.
- *Center of mass:* Minor difference. The steel screw raises the center of mass. The winch with its ropes allows for more flexible placement of the motor.
- *Price:* Servomech's invoice vs Company B's invoice.
- *Maintenance:* Failed to identify and suitable winner.
- *Collision avoidance:* A winch-based solution allows for quick retraction by merely unlocking the drum and allowing the lift-force of the foil to hastily retract itself into the hull. Such a solution would be impossible with a lead screw without some sort of re-design to the nut or the ends of the lead screw.

The comparison shows that the winch is superior in most if not all aspects. It is lighter, has a better (lower) center of mass, it is cheaper to purchase, and integrating collision avoidance is easier. For that reason, the concept moving into detailed design is a winch-based concept.

4.5.6 Final concept decision

I experienced major set backs during late stage concept design, as sourcing components, contacting companies and finding proper information took extensive amounts of time. Additionally, a meeting with an engineer from company B revealed that their winches (used in CWA) are not compatible with the scenario for various reasons, see appendix K for a full list of reasons for incompatibility. Note that the incompatibility is in regards to company B's winches, not winches in general.

For the aforementioned reasons I decided to perform the detailed design by first designing a theoretical winch, regardless of currently available components, to establish a baseline winch solution that would work in theory. The design would then be realized by future work by sourcing components that most closely match that of the theoretical winch.

5

Early detail design results

I will first present a more thorough requirement specification and then an overview of the HDM design in the state of which it was during the thesis conclusion. To clarify, the design of none of the chunks can be considered to in its final state. Each chunk of the HDM is the elaborated upon in subsequent sections.

5.1 Detail design requirement specification

As a specific concept solution has been chosen, a detail design requirement specification was formulated. This is based on the concept-phase requirement specification but has additional requirement added stemming from the chosen concept solution. See appendix L. The detail design requirement specification has been split into different sections, first listing the four global goals for the HDM, global requirements that relate to the entirety of the HDM, and requirements sectioned off by specific parts of the HDM. In general, the main dimensioning factor can be regarded to be the lift force generated by the hydrofoil, since a significant portion of all requirements either stem directly or indirectly from that parameter.

5.2 HDM overview

Figure 5.1 presents the HDM in its current state. See appendix M for additional images. See subsequent sections for images of each chunk. Figure 5.2 shows an overview of the main functions of each chunk and an approximate physical interaction between chunks.

The HDM can be regarded as a refinement of the CWA presented in chapter 4.5.4. A winch unit (now placed outside of the Moon Pool for available space reasons) acts as an actuator with two outgoing ropes that attach to two attachment points on the wagon via an upper and lower pulley. The wagon gets pulled along the shafts by use of linear bearings. Attached to the wagon is the hub and a trim unit, and attached to the hub is the hydrofoil. The loads are transferred from the hydrofoil to the hull or Moon Pool via the winch mount, an upper frame bracing and a lower frame bracing.

I would like to remind the reader that the hydrofoil needs to travel vertically at a roughly 10° angle with its wing still being level horizontally. I.e. the hydrofoil needs

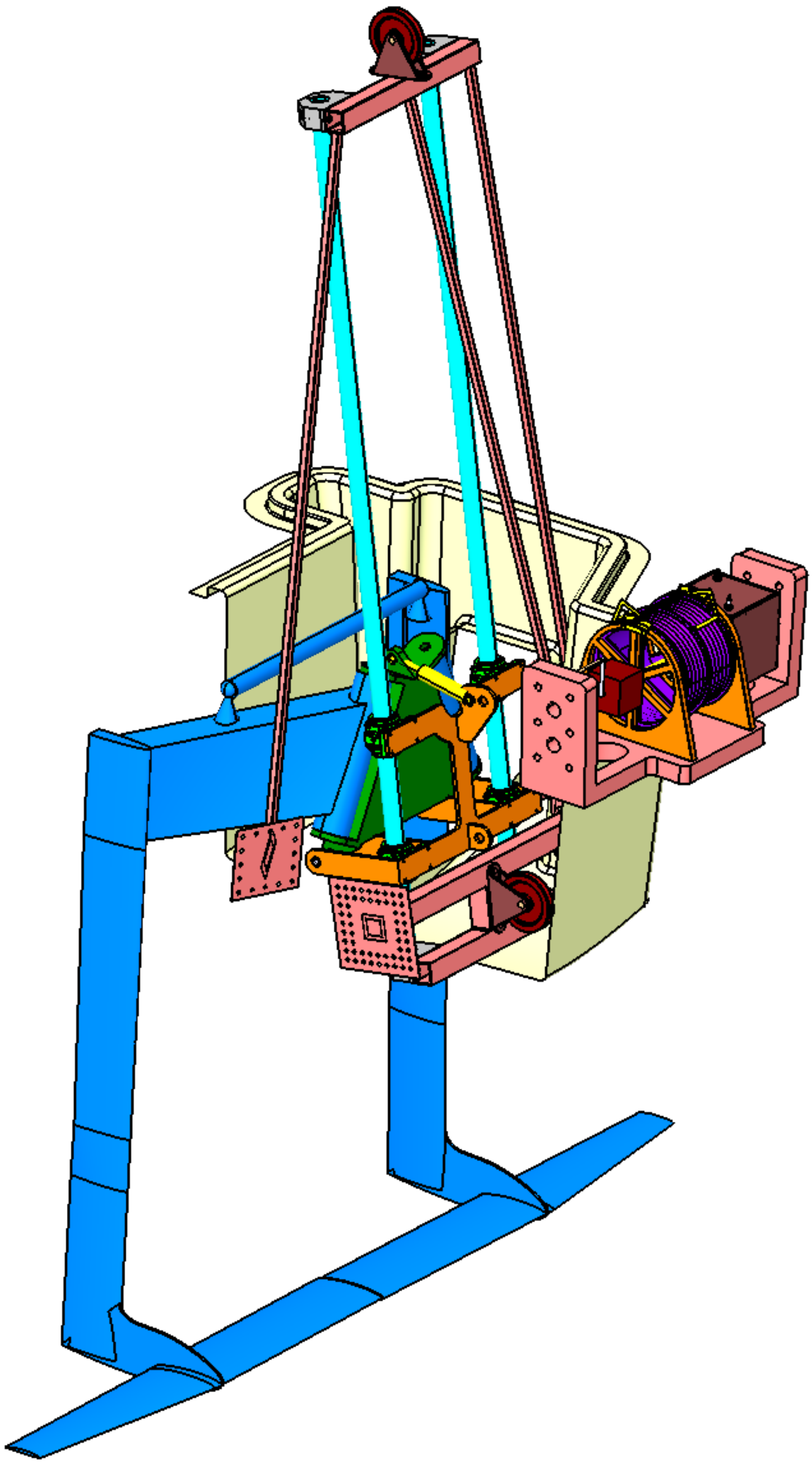


Figure 5.1: An overview of the HDM. **Pink:** Frame. **Orange:** Wagon. **Green:** Hub. **Blue:** Hydrofoil. **Multi-colored:** Cuwb (custom winch bravo). **Teal:** Shafts. **Yellow:** Trim unit.

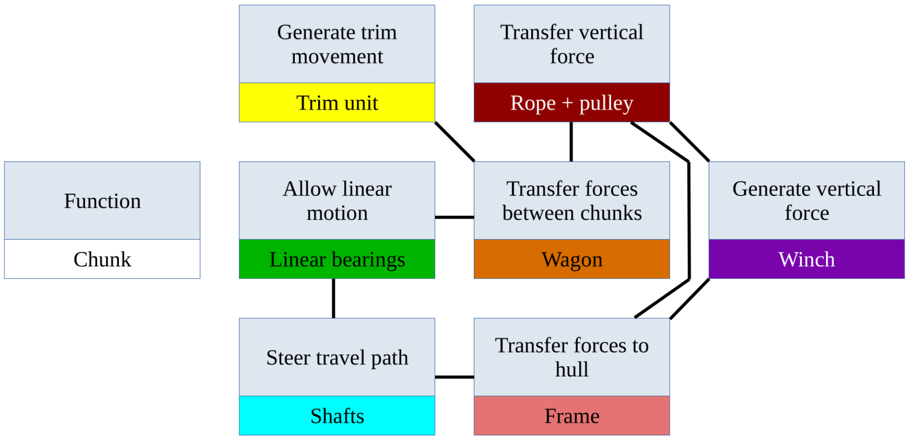


Figure 5.2: Main functions of the HDM and the chunks that fulfill them. Black lines indicate physical interfaces between chunks.

to be straight but travel in a not-perfectly-vertical travel-path. This requirement entails that the entire HDM is tilting slightly to account for such a travel-path.

5.3 Moon Pool

Interfaces with chunks: Frame

The Moon Pool (see figure 5.3) is not strictly a part of the HDM, but is its primary attachment point to the remainder of the hull. The Moon Pool has a partially predetermined shape; the outer surfaces and bottom cut-out cannot be altered to match the HDM design, but inner surfaces can be strengthened and braced to best suit the HDM.

During the vast majority of the thesis design I worked with the following restrictions: Any interface between the HDM and Moon Pool must be fastened via bolting and the only interfacing surfaces allowed are the vertical walls (and not the bottom surface). At the end of the project, new demands emerged to instead fasten HDM components by molding them in carbon fiber directly into any of the Moon Pool's surfaces (these possibilities were however not explored during the thesis).

5.4 Hub

Interfaces with chunks: Hydrofoil, trim, wagon

The hub, see figure 5.3, serves as an interface point between the hydrofoil and the rest of the HDM. More precisely, the hub has the following interfaces: axles of the hydrofoil, trim, wagon. The geometry of the hub is modeled after the axles of the hydrofoil, and attachment points (abstractly visualized as loops) are placed on the underside for the wagon interface and topside for the trim interface. The hub will be constructed in a CFRP material.

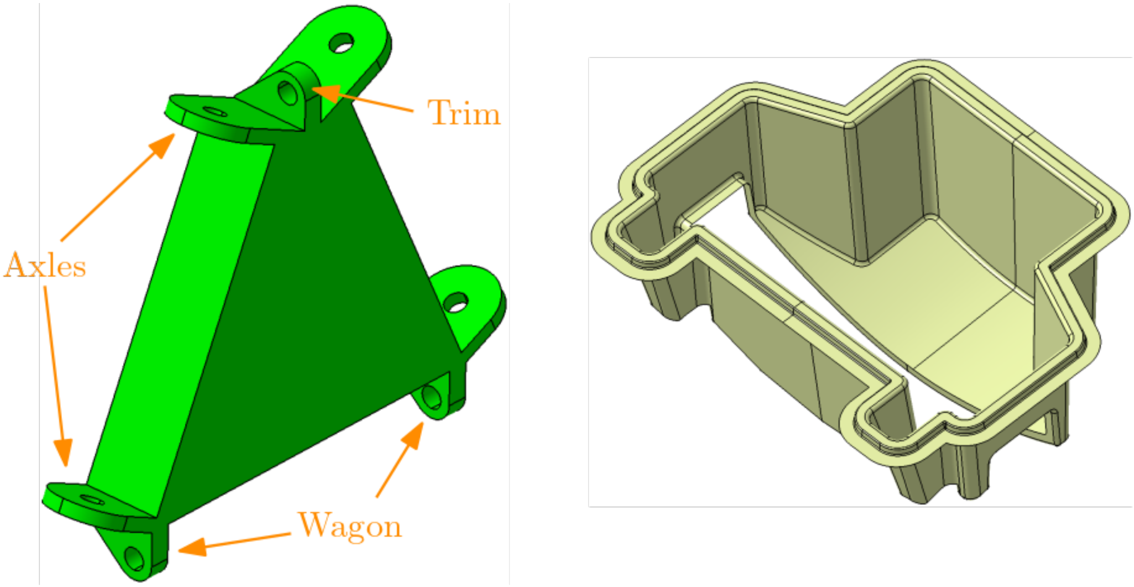


Figure 5.3: Left: The hub and its interface regions. **Right:** A full view of the Moon Pool, note the cutout in the bottom to make room for the hydrofoil's wing.

5.5 Wagon

Interfaces with: Cuwb, frame, hub, trim

The wagon consists of the following elements (see figure 5.4): 4 linear bearings, an interface for the trim, as well as a top and bottom interface for the rope coming from cuwb.

The trim interface is on its own plate to facilitate a wider range of trim units. For example, should a shorter trim be chosen, the trim-plate can be shortened to place it in an appropriate placement relative to the hub. The hub interface is as such to allow for some distance between the top of the hub and other parts of the HDM (the shafts). This is necessary to account for the previously mentioned horizontal alignment of the hydrofoil whilst traveling on a not-perfectly-vertical travel-path.

The wagon's components is theorized to be manufactured using drilling for the holes, and either water-cutting or sawing for the profile of the plate. Given the required thickness of the plates, shaping via water-cutting seems more likely than sawing. Each component would then be attached to one another through welding.

ANSYS Simulations: *Note – These simulations were performed in an earlier iteration of the wagon with a slightly different geometry.* Loads applied to the wagon are visualized in a free body diagram in appendix N. Figure 5.4 shows simulation results in terms of factor of safety (against yielding) whilst the wagon is constructed in Duplex S32550 stainless steel.

Simulations show that the general design has promise yet warrants further iteration. Should stress concentrations near bearing unit attachment points and sharp corners be optimized, the FoS (factor of safety) is high enough to be satisfactory

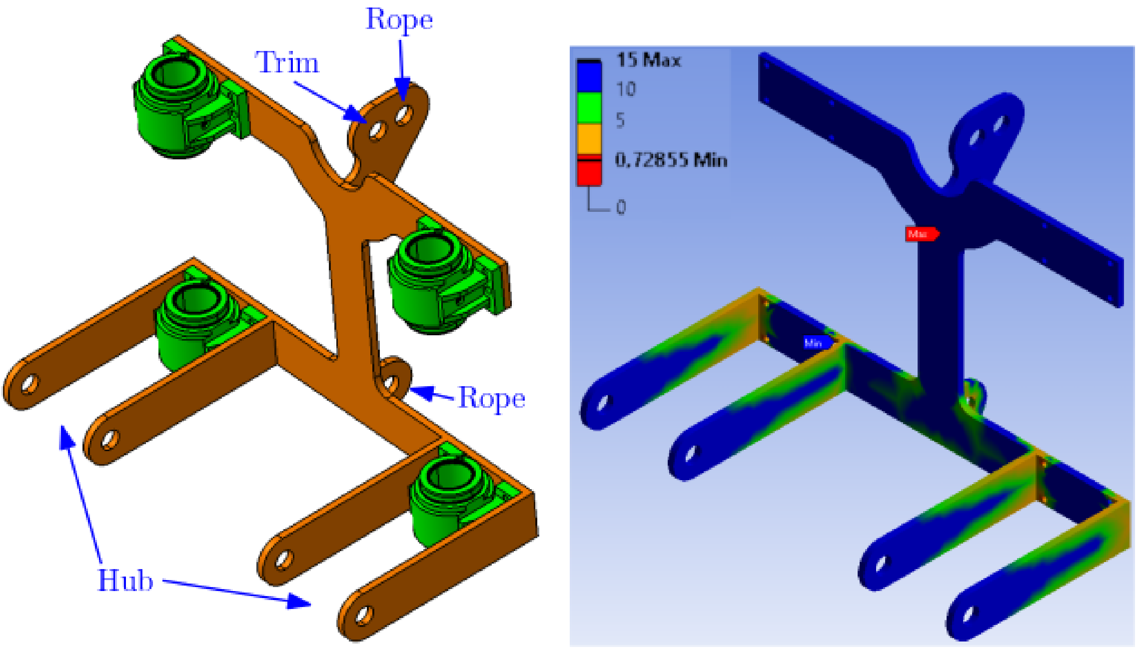


Figure 5.4: Left: Different parts of the Wagon chunk. **Right:** Strength simulations done in regards to FoS, see legend for which color indicates which FoS. Simulation done in ANSYS (version 21.1.0; 2021).

and even high enough to remove material to reduce weight in some areas. Since the current design is completely unoptimized regarding stress concentrations, the design has potential for improvement. Displacement or angle-change of the hydrofoil via deformation of the wagon is negligible. Simulations done with Al-5456O aluminum show that stresses are far too high for the wagon to be constructed in that material with roughly the given shape. The weight of the wagon in steel is roughly 14 [kg].

5.5.1 Linear movement

The linear units attached to the hub are chosen to be linear plain bearings (as opposed to linear roller bearings). This is due to plain bearing being able to handle a marine environment better than roller bearings (in terms of corrosion and debris), and that a lower frictional force is not necessary in this case (since it is negligible compared to the lifting force of the hydrofoil). A promising component might be the LUCR 30 PA from Ewellix (Ewellix 2, n.d.), capable of handling previously calculated radial forces with a high FoS without necessitating an impractically large shaft diameter.

5.6 Trim

The trim unit is a linear actuator that essentially expands or contracts. Since it's attached to the wagon (which cannot move horizontally once it is locked into place) and the upper part of the hub (which is hinged at its lower part), expanding or contracting the linear actuator will entail tilting the entire hub which in turn produces a trimming effect on the hydrofoil.

The most important factor on the trim mechanism is not the effect, but rather the output force and its locking power. I.e. the speed of movement is not important, but the trim needs to be able to withstand and counteract forces that stem from the torque generated from the hydrofoil. Figure 5.5 shows a free body diagram of forces and levers in the context of producing trim. Calculations show that the trim

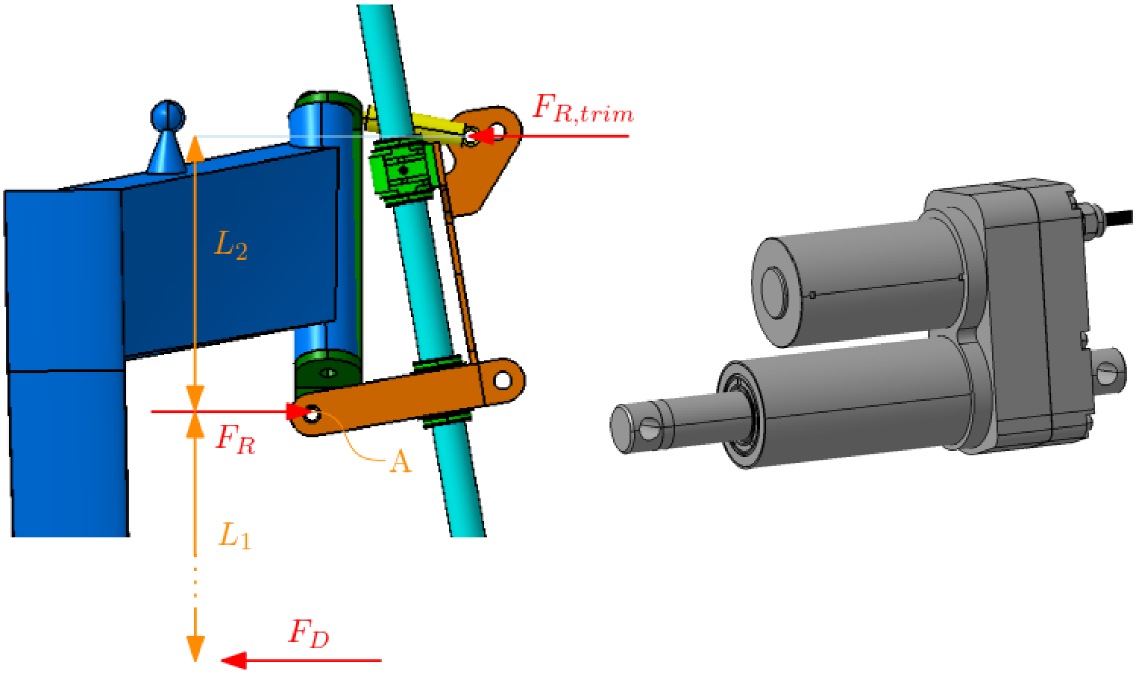


Figure 5.5: Left: A free body diagram of the torque that drag forces will generate that will lead to axial forces in the trim unit. **Right:** Ewellix CAHB-22-A4E.

mechanism needs to overcome and hold an axial force of at least 4 380 [N]. Given the current positioning and wagon design, to achieve a 1° trim the trim mechanism needs to expand or contract roughly 8 [mm].

A promising component might be CAHB-22-x4E from Ewellix (Ewellix 1, n.d.). It has a maximum load of 10 000 [N], stroke of 48 [mm], is IP69K and IP66M rated, salt-water spray tested according to standard ISO 9227:2012 – 250 hours, and weighs less than 4.8 [kg] (4.8 [kg] for the 200 [mm] stroke model).

5.7 Cuwb - Custom winch bravo

Interfaces with: Frame, wagon

The designed winch solution (internal name cuwb) consists of several different modules. These are presented in figure 5.6 and elaborated upon in this section. Cuwb can be viewed as an evolution of cwa with the overarching design idea from cwa still applying. I would like to remind the reader that the presented design is theoretical and not based on specific commercially available components.

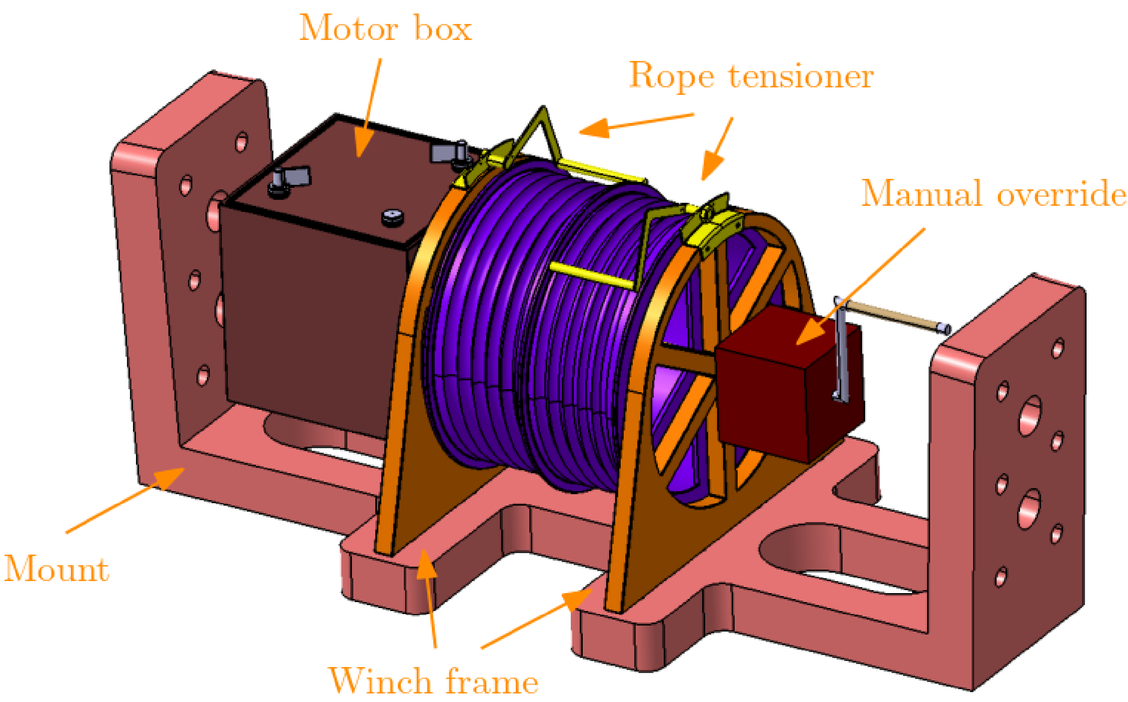


Figure 5.6: A visualization of some of cuwb's modules.

Design of cuwb modules has been done in stages since the design of one would affect the design of others. Roughly speaking, the design hierarchy and order has been: external load → rope diameter → drum dimensions → drivetrain dimensions.

Rope: Synthetic (polymer) rope will be used due to its vastly higher specific strength compared to steel, something that aids in lowering weight. The type of synthetic rope will be based on Dyneema, which is a polyethylene-based material with a proven track-record of being utilized in marine environments (DSM, n.d.). The specific strength $[Nm/kg]$ of Dyneema fiber compared to steel depends on application, however the order of magnitude is roughly 15 times higher than steel (DSM, n.d.), roughly 30 times higher than stainless steel (see appendix O), and roughly 8 times higher than steel in a rope configuration (DSR, n.d.).

Dimensions – Rope: Data surrounding Dyneema ropes have been sourced from DSR, specifically looking into the DSR SuperMax 78 synthetic fiber (DSR, n.d.) due to its marine environment tolerance, high specific weight (i.e. it floats in water) and high abrasion resistance (i.e. wear-and-tear when winding on drums and running along pulleys). Dimensioning of the rope was performed by studying the catalog of available ropes. To account for a very high factor of safety, the SuperMax 78 á ⌀14 [mm] giving a breaking strength of 215 800 [N] was chosen.

Dimensions – Drum: The diameter ratio between a drum and synthetic rope is recommended to be at least 20:1 (R. Hovgaard, personal communication, May 3, 2022). A diameter of ⌀280 [mm] was chosen. For safety reasons, the number of wraps of a rope on a drum should never be lower than 5 wraps (company B, personal communication, April 28, 2022). For simplicity's sake, all safety wraps and the entire stroke (length of which the hydrofoil travels vertically) is placed on the bottom layer. Including some margins, a drum length of 100 [mm] is chosen. Keep in mind that the drum used is a split drum, meaning that the total length of the drum in the winch is roughly 200 [mm].

Dimensions – Pulley: The diameter ratio between a pulley and synthetic rope is recommended to be at least 10:1 (R. Hovgaard, personal communication, May 3, 2022). A diameter of ∅140 [mm] was chosen.

Dimensions – Grooves: The diameter ratio between grooves (in pulleys or grooved drums) and synthetic ropes is recommended to be at least 1.1:1 (R. Hovgaard, personal communication, May 3, 2022). A groove diameter of ∅16 [mm] was chosen. A groove depth of half of that, 8 [mm], was chosen.

Guide system vs grooves: Cuwb does not use a guide system (i.e. a system that guides the rope to a certain part of the drum[1]) but instead relies on grooves on the drum to guide the rope along the length of the drum. This was chosen to lower total number of unique parts and lower overall volume of the winch. Additionally, the entire rope length being able to fit on a single layer entails that grooves can be used to guide the entire stroke length.

Rope tensioner: To ensure a snug fit and even distribution of friction along the entire length of the rope, a rope tensioner is included for both ropes of the drum. The rope tensioner is attached to the frame and applies pressure by the usage of a spring.

Specifications – Drivetrain: Assuming a drum diameter of ∅280 [mm], the force on the rope would transfer a torque on the drum axle of roughly 16 000 [N] * 0.14 [m] = 2240 [Nm]. The rope needs to travel 1.8 [m] in 20 [s], i.e. a speed of 0.09 [m/s]. A drum radius of ∅140 [mm] would necessitate an angular speed of 0.643 [rad/s] or 6.139 [RPM]. I.e., the drivetrain has to overcome a torque of 2240 [Nm] at a speed of at least 6.139 [RPM]. At the current design-state, the drivetrain is viewed as a black box containing some sort of electrical motor and gearbox combination.

$$\omega = \frac{v}{r} = \frac{0.09}{0.140} = 0.643\ [rad/s]$$

$$v = \frac{2\pi}{60} * r * RPM\ [m/s] \Leftrightarrow RPM = \frac{60v}{2\pi r} = \frac{60 * 0.09}{2\pi * 0.140} = 6.139\ [RPM]$$

Braking: Winches are designed to transfer torque in the same rotational direction as the rotational direction of the driving axle. I.e., if the drum is rotating clockwise, the winch is designed to deliver torque clockwise. During a braking scenario, the opposite would instead apply (e.g. the drum rotates counter-clockwise and torque is applied clockwise to slow down rotation). A winch capable of doing this, producing torque opposite of rotational direction, is called a hoist. This braking torque is commonly applied using a mechanical brake, but can in theory be applied by activating the electric motor in the opposite direction to that of the axle's rotational direction. It is unknown at the time of thesis publication time how cuwb would solve braking.

[1]Example: `https://en.strongwinch.com.tr/winches-guide-system`

Motor box: As the HDM will be used in a marine environment, the drivetrain would in theory have to have a high IP rating and resist a marine environment. Setting these two demands on electrical motors and gearboxes disallows the usage of an overwhelming majority of components. To counteract that, the drivetrain has instead been placed inside of a sealed box. The box's only outputs is a driving axle and cables for power and steering through watertight openings.

The motor box is openable for servicing via two latches and has a water-tight gasket between the lid and a lip on the inner walls of the box. A hermetically sealed box might risk taking damage during temperature changes, as that would cause a pressure difference between the inside and outside of the box (A. Berg, personal communication, June, 2022). To counteract such damage, the box has been outfitted with a vent plug that allows air to pass through but no liquids. Gaskets have been to seal latches and the vent plug.

Clutches: Cuwb requires a clutch between the manual winch - drum to enable manual winching. It also requires a clutch between the motor box - drum to ease manual winching not having to turn a passive electrical motor, in addition to allowing for a fast rotation of the drum in case of a collision avoidance scenario. Clutches can roughly be divided into those aimed to be used during axial rotation (i.e. when the engine or axle is spinning) and those aimed to be used during stationary conditions. Clutches used in the former scenario are commonly based on some sort of friction-based solution to allow a gradual transmission of force between two axles with different angular rotational speeds. The HDM would not require this and only require to be activated during stationary conditions.

The chosen clutch type is a dog and spline clutch (see figure 5.7) as such a design allows for a high torque with minimal volume, and due to its design simplicity. The primary drawback of a dog and spline clutch, namely that clutch engagement is not possible during high rotational speeds, is not a concern in the HDM scenario. It is unknown at the time of thesis publication how any of the clutches will be activated or where exactly they would be placed.

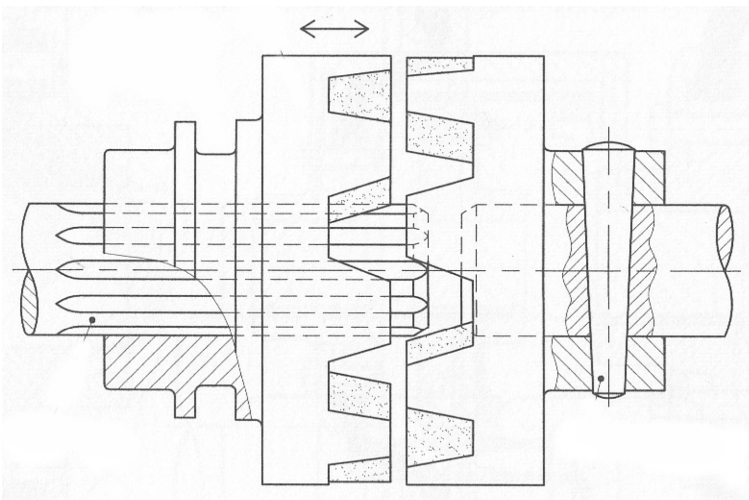


Figure 5.7: A dog and spline clutch (Mägi, et al. 2, 2017).

Manual override: Should the drivetrain of the HDM fail, the crew of the X8 stills needs to be able to raise and lower the hydrofoil in emergency situations. This functionality is fulfilled by attaching a manually operated winch on the drum, opposite the side of the motor box. Usage of such an override is done by disengaging the motor box clutch, engaging the manual winch clutch, and using a hand-powered crank and gearbox to turn the drum. Manual operation would only be suitable and safe when the boat is stationary, entailing that the manual winch would only need to counteract the weight of the hydrofoil and any attached accessories. The manual winch would be sealed from the marine environment similarly to that of the motor box. Further details of the manual winch are unknown at the time of thesis publication.

Confirming calculations using Liebherr's design manual: As a safety precaution, many aspects of the design of the winch has been confirmed by following the design manual of winch-manufacturer Liebherr (Liebherr, n.d.). The usage of this design manual has lead to me concluding that no major mistakes have been made during the design of cuwb.

Collision avoidance: It is unknown the time of thesis publication how cuwb would fulfill the functionality of collision avoidance.

Mount: The current design of the mount is a legacy version from then the winch was located inside of the Moon Pool. Due to its outdated nature compared to the rest of the HDM design, it will not be elaborated upon in this report.

Winch frame: The winch frame's purpose is to transfer forces applied to the drum (from the rope) to some sort of attachment point to the hull. The shape is roughly dimensioned after the drum. Material has been removed to lower weight.

ANSYS simulations – Winch frame: Simulations were performed to evaluate the winch frame design. The entire underside of the frame was given a fixed support. A bearing load of 22 500 [N] vertical and 13 000 [N] horizontal was applied to the center axle slot.

See figure 5.8. Simulations show that the frame has good potential to be manufactured in Al-5456O if stress concentrations can be optimized enough to raise the minimum observed FoS of 1.4. A possible hindrance of using aluminum is the driving axle size. Should the axle size not be able to be increased to lower the stress concentration in the frame, the design might not be able to be realized in aluminum. A possible work-around would be to maintain the axle size, increase the slot hole and make up the difference with bushings. The frame experienced negligible amounts of deformation. The frame in Al-5456O weighs 2.85 [kg].

See figure 5.8. Simulations show that the frame has great potential to be manufactured in Duplex S32550 stainless steel (with potential for weight savings) as the minimum FoS observed is 5.5, even without optimizing stress concentrations. The frame experienced negligible amounts of deformation. The frame in S32550 weighs 8.39 [kg].

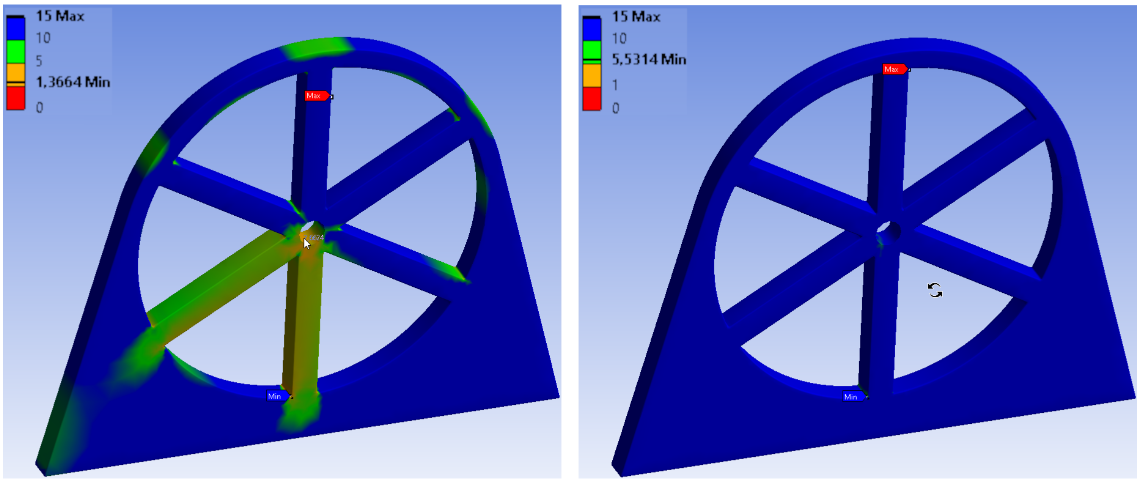


Figure 5.8: Left: FoS simulations in aluminum. **Right:** FoS simulations in stainless steel. Simulations done in ANSYS (version 21.1.0; 2021).

ANSYS simulations – Drum: Simulations were performed to evaluate the drum design, see figure 5.9. The drum is given a fixed support on both ends of the axle slot and given a load of 26 000 [N] in an arbitrary direction (due to it's rotational symmetry) on half of a groove's revolution's surface.

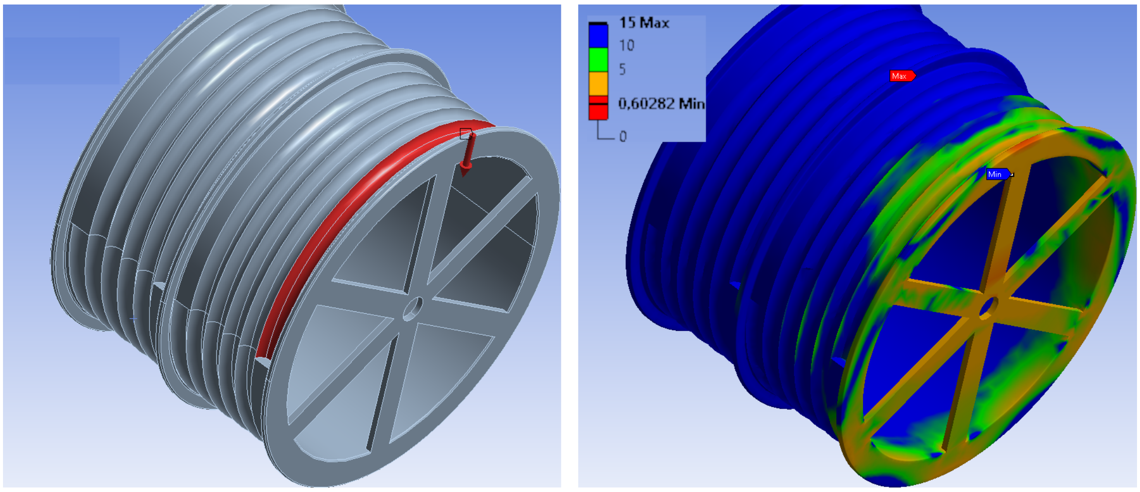


Figure 5.9: Left: Placement of the force on the drum. **Right:** FoS simulations in aluminum. Simulations done in ANSYS (version 21.1.0; 2021).

See figure 5.9. Simulations show that the drum has moderate potential to be manufactured in Al-5456O. The low FoS in the drum's spokes can be remedied by increasing their cross-sectional area, but the low FoS in the grooves might not be fixable since they need to hold a certain shape. The drum experiences negligible amount of deformation and weighs 4.74 [kg].

Further simulations show that the drum has great potential to be manufactured in S32550, following some minor adjustments. The drum experiences negligible deformation and weighs 13.93 [kg].

5.8 Frame

Interfaces with: Cuwb, Moon Pool

The frame chunk consists of two different modules: the bracings and the shafts, see figure 5.10. The frame chunk was designed under the initial constraints, namely that components had to be bolted to the Moon Pool (see chapter 5.3).

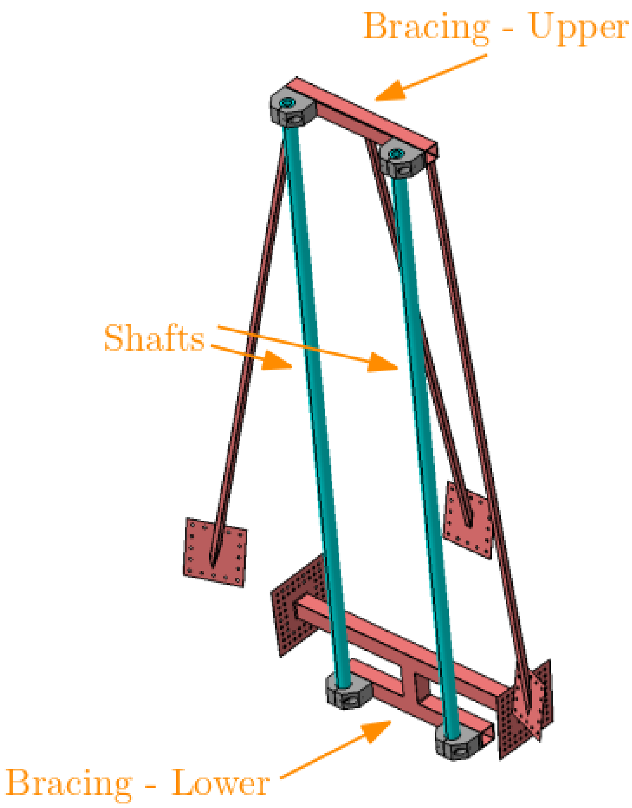


Figure 5.10: An overview of the frame's modules. Note that Bracing - Upper and Bracing - Lower are not *directly* attached to each other.

5.8.1 Bracing

The bracing consists of an upper and lower part joined together by the shafts. All bars of the bracing consists of square tubes. All bars are attached to other bars or plates by welding. All shaft blocks are attached to the bars by bolting.

Bracing – Upper: The top bar of the upper bracing serves as an attachment point for the upper shaft blocks and upper pulley. This attachment point is supported by three bars, one bow-facing and two side-facing, that are attached to the vertical walls of the Moon Pool by bolting. The Moon Pool interface is many-holes-in-plates to distribute the load over a larger area on the Moon Pool.

Bracing – Lower: The lower bracing consists of two bars: one to serve as an attachment point to the Moon Pool and another to serve as an attachment point for the lower shaft blocks. These are split among two bars to allow the shaft blocks to be as low as possible (to reduce overall HDM height) and to give enough space for interface plates on the Moon Pool walls. Future iterations may join these into one curved bar.

ANSYS simulations: See figure5.11. Simulations were performed in different load scenarios, the ones presented in this report will be when the hydrofoil is deployed and when the hydrofoil is completely retracted and the boat is leaning 30°. All plates were given a fixed support condition and loads are applied radially to the shafts in different locations that correspond to the wagon's position in deployed and retracted scenarios. All components are assigned the stainless steel alloy AISI316.

Simulations show that the design has great potential in a retracted position. The lowest FoS of 6.5 is at one of the side bars of the upper bracing and irrelevant amounts of displacement. This simulation shows that the upper bracing could afford to be structurally weakened in favor of weight savings.

Simulations show that the design has moderate potential to be realized in a deployed position with a lowest FoS of 0.69, nominal FoS of 2.3 and a FoS of 1.0 towards stress concentrations. These numbers are exaggerated since the geometry is simplified for the sake of meshing with non-existing radii on all corners, entailing that stress concentrations will be higher than in a real scenario. Simulations also show a maximum displacement of 3.5 [mm] in the shafts, which is considered completely acceptable given that this is during usage when the hydrofoil is stationary and locked into place.

The lowest FoS are predominantly on the lower bar connecting the frame to the Moon Pool, however the current design of the brace allows for the bars to easily be scaled up for increased strength. An alternate design improvement would be adding a support towards the bow-side of the Moon Pool wall to prevent displacement of the lower bars in the bow-direction, possibly greatly reducing stress.

5.8.2 Shafts

The shafts are metal tubes in either a stainless steel or aluminum alloy. Its outer diameter is set to match the linear bearing units (currently ∅50 [mm]). Its thickness is kept flexible as an easily adjustable parameter to affect frame strength and weight. During simulations the shafts have been allotted a wall thickness of 10 [mm].

5.9 Material choice

This section presents the material choice for the primary material within HDM. This is initiated by eliminating materials brought up during the concept phase, evaluating different metal types, and making a decision on a specific steel alloy. Finally, steel aluminum and carbon fiber are evaluated against each other in a comparison regarding weight.

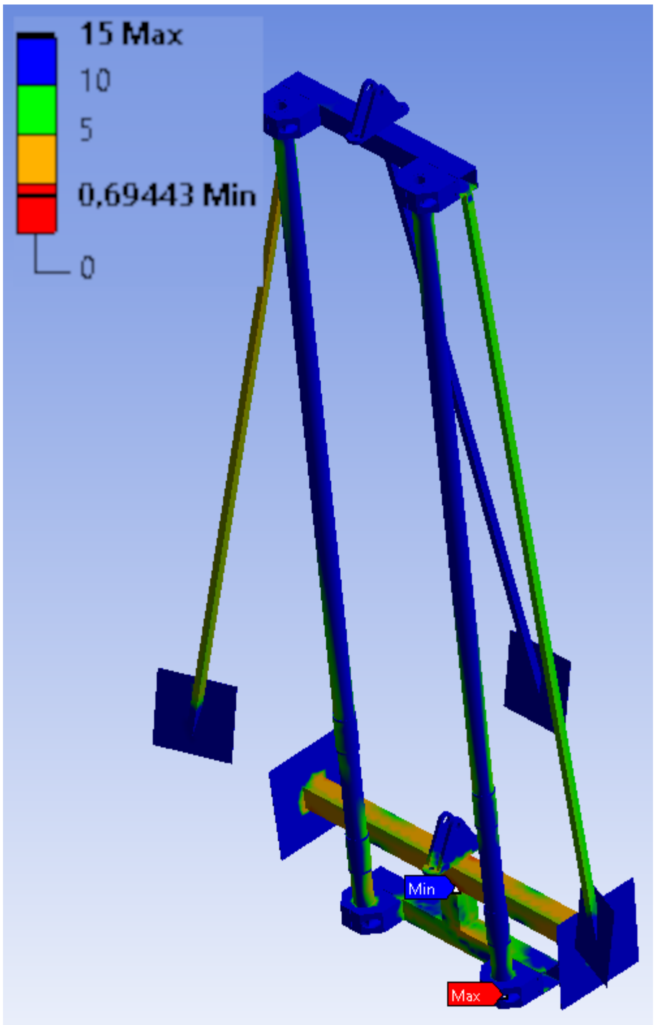


Figure 5.11: FoS simulations on the frame chunk whilst assigned the material AISI316 stainless steel. Simulations done in ANSYS (version 21.1.0; 2021).

5.9.1 Material for which components?

Different components within the HDM are suited to be manufactured in different materials. E.g. the shafts within the frame are always most suited to be manufactured in a metallic material, whilst the hub may for legacy reasons only be available to be manufactured in CFRP. The material choice in this section refers to the majority of the structural components of the HDM, i.e. the components within the frame and the wagon (with the exception of the linear bearings in the wagon). It is unknown at the time of thesis publication *exactly* which components will be manufactured in which material.

5.9.2 Material concept elimination

There are many materials which would be suitable to construct different components with. All promising materials listed in chapter 4.2 would be feasible to use to create the desired shapes capable of holding the desired loads (with perhaps some exceptions for the materials with lower yield limits). The material properties themselves are not enough to pick the optimal material, other factors must be considered as well.

The knowledge and industrial infrastructure surrounding different materials (especially those aimed to be used in a marine environment) is more limited than the materials whose properties alone would warrant usage. For example, whilst hardwood is capable of handling a marine environment, is an environmentally sustainable material, easy to form in low-volume products and has a moderately high specific strength; the knowledge for working with such a material in this specific use case (high-strength structure with a moderately complex shape in a marine environment) simply does not exist in any notable amount in the marine industry. This conclusion was made by myself after a lack of success attempting to find such competence to interview, and also after numerous recommendations by industry contacts to avoid using wood as a material (due to a lack of marine-industry competence).

To maximize the likelihood of the design being able to be realized, I was advised by the client and involved parties (A. Berg et al., personal communication, June 15, 2022) to pursue the following material groups: aluminum alloys and stainless steel alloys. To get a more encompassing study on metal alloys (as they were excluded completely in the previous concept level material study), I chose to also investigate titanium alloys.

5.9.3 Alloy elimination

Aluminum alloys, stainless steel alloys, and titanium alloys are relevant as they are most commonly regarded as structural materials in relatively large geometries. Whilst other metal alloys (such as magnesium, nickel, bronze, brass, and more) are theoretically suitable (based on their specific strength, price and corrosion-resistance), their relative rarity entails the design team would face difficulties in realizing such a design (for example sourcing geometries in a cost-effective manner, finding competence that can weld in unique-alloys, etc.).

See figure 5.12. Aluminum alloys and stainless steel are somewhat equivalent in regards to specific strength, but most stainless steel (especially Duplex) sees a price increase of roughly 50% - 100% in comparison to aluminum alloys. Titanium is superior in regards to specific strength, roughly by a factor of 2.

Titanium has superior specific strength and is impervious to corrosion in the given environment, but it is simultaneously roughly 10x more expensive than aluminum and stainless steel. Manufacturability is more favorable with aluminum and stainless alloys compared to that of titanium, see appendix P. Finally, company contacts discourage the pursuit of titanium alloys (A. Berg, personal communication, June 15, 2022) for the sake of practicality and cost. Titanium parts are usually custom-made (vastly increasing price) and joining by welding would require specialists to perform (once again increasing price). Due to the aforementioned reasons, I will not pursue titanium alloys as a material choice.

Multiple sources (Yari, 2021; A. Sahlin, personal communication, June 15, 2022)

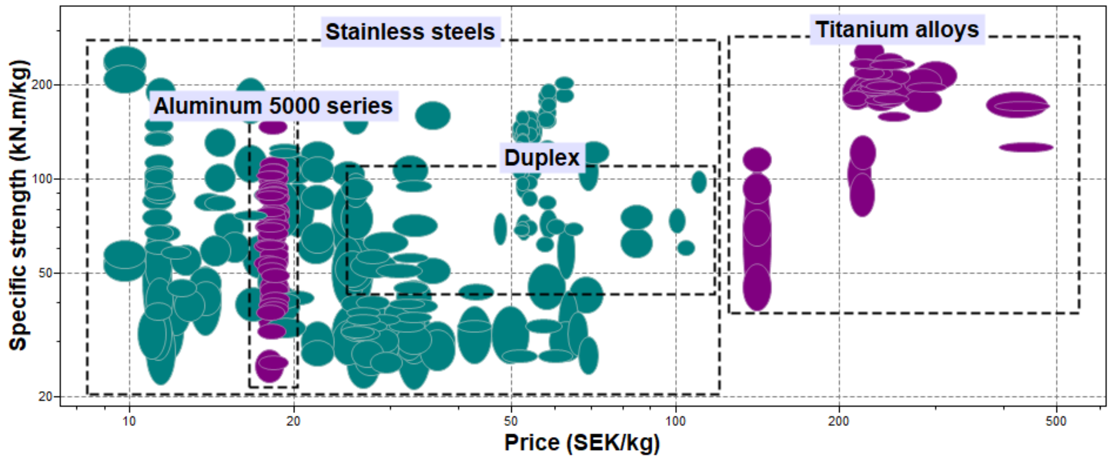


Figure 5.12: A graph of specific strength vs price showing stainless steels, Duplex (a high-end variant of stainless steel), aluminum 5000 series and titanium alloys. (GRANTA Selector, Version 21.2.0; 2021)

claim that aluminum alloys suffer significant galvanic corrosion when in contact with carbon fiber, of which the boat's hull and the hydrofoil is manufactured with. The only realistic and low-maintenance solution to this corrosion problem if using aluminum would be to electrically isolate all aluminum components from the carbon fiber, and add sacrificial anodes as a back-up. Electrically isolating all metallic components is regarded as difficult to ensure 100% success on, and for that reason I chose to not pursue aluminum alloys further.

I was recommended (A. Berg, personal communication, June 15, 2022) to downgrade the quality of stainless steel from Duplex to more conventional stainless steel types (for example 316). The reasoning for this is that the corrosion resistance properties of Duplex is overkill for the application and that regular more budget-friendly variants of stainless would handle salt-water and carbon fiber. For context, Duplex is often used in severely corrosion hostile environment (chemicals) or for components where *absolute* corrosion resistance is critical.

Conclusion: I will pursue some sort of stainless steel variant.

5.9.4 Stainless steel variants

Theoretical choice: I concluded that there are four stainless steel alloys, see table 5.1, that are most suitable for the use case. The following have a material cost of less than 15 SEK / kg, excellent salt water durability and superior specific strength compared to other alloys, see appendix Q. Note that AISI 410 or AISI 431 are also known as EN 1.4006 or EN 1.4057 respectively. There is a critical drawback with these materials however, namely that they are not stocked by Swedish steel retailers[2].

[2]All comments of retailers stocking certain materials are with respects to material availability at the time of writing.

Stainless steels	ASTM CA-15, cast, tempered 315°	ASTM CA-40, cast, tempered 315°	AISI 410, hard temper	AISI 431 tempered at 260°
Price [kr / kg]	10.6 – 11.9	10.6 – 11.9	10.6 – 11.9	13.7 – 15.5
Density [kg / m³]	7560 – 7660	7560 – 7660	7650 – 7850	7700 – 7900
Young's modulus [GPa]	195 – 205	195 – 205	190 – 220	190 – 220
Yield limit [MPa]	985 – 1090	1080 – 1200	1000 – 1100	925 – 1140
Specific strength [kNm / kg]	129 – 143	142 – 158	129 – 142	119 – 146
Elongation [% strain]	5 – 9	1 – 2	12 – 18	12 – 20
Fatigue strength At 10^7 cycles [MPa]	500 – 572	530 – 613	374 – 398	494 – 572
Toughness [kJ / m²]	6.67 – 13.2	5.96 – 12.1	10.7 – 16.7	5.8 – 15.4
Thermal expansion Coefficient [µstrain / °C]	9 – 11	9 – 11	9 – 11	9 – 11
Galvanic potential [V]	(-0.28) - (-0.2)	(-0.27) - (-0.19)	(-0.28) - (-0.2)	(-0.22) - (-0.14)
Magnetic	Yes	Yes	Yes	Yes
Salt water durability	Excellent	Excellent	Excellent	Excellent
Similar steels	GX12Cr13 to UNI 3161	CAF 420	410-16 AC-DC	AL Tech Stainless type 341

Table 5.1: Relevant material properties of different relevant stainless steels. (GRANTA Selector, Version 21.2.0; 2021)

Industrial choice: To ensure that the material choice is actually implementable, a separate study was conducted using solely industrial sources. The most common stainless steel for stock shapes found in Swedish retailers are (Stena Stål, n.d.; Smålands Stål, n.d.; Tibnor, n.d.):

- AISI 304 (a.k.a. EN 1.4301 / EN 1.4037)
- AISI 316L (a.k.a. EN. 1.4401 / EN 1.4404 / EN 1.4436 / SAE 316 / UNS 31600)

These stainless steels are significantly more expensive than the theoretical choice (by a factor 2 – 5), but they are actually available to to be purchased.

Sources (Valbruna Nordic 1 (n.d.)) mention AISI 304 to be unsuitable for a marine environment, but mention no such drawbacks for AISI 316 (Valbruna Nordic 2 (n.d.)). Furthermore, RM Import AB (n.d.) confirms AISI 316 to be suitable for a marine environment. Swedish steel retailers Tibnor (n.d.) and Smålands Stål (n.d.) stock numerous profiles in AISI 316. Due to aforementioned sources, A. Berg's recommendation (personal communication, June 15, 2022) and product availability, AISI 316 is chosen as the primary material choice. More specifically, AISI 316L was chosen due to heavy sections being welded does not require post-weld annealing to retain their corrosion resistance (Madras Engineering Works, n.d.).

See appendix R for an overview of these stainless steels' material properties.

5.9.5 Steel, aluminum and carbon fiber comparison

During the final parts of the thesis, industry contacts introduced a new demand requiring the usage of aluminum alloys and carbon fiber composites in the design. Using industrial sources (Continental Steel & Tube Company, 2020; Metal Boat Kits, n.d.) aluminum 5083-H321 was chosen for its (relative) marine suitability and retained strength post-weld. Welding is prioritized due to it being expected to be the primary joining method. Due to confidentiality reasons I cannot elaborate on why a specifically woven and prepreg CFRP material is chosen. Table 5.2 shows a comparison of relevant material properties of the three main material choices.

Materials	CFRP woven prepreg	AISI 316L	Aluminum 5083-H321
Price [kr / kg]	468 – 519	30.8 – 35.6	17.3 – 19.4
Density [kg / m³]	1540 – 1610	7870 – 8070	2640 – 2670
Young's modulus at 23°C [GPa]	62.7 – 68.7	195	70.0 – 73.6
Yield limit at 23°C [MPa]	627 – 910	226	193 – 221
Specific strength [kNm / kg]	398 – 579	21.3 – 38.9	72.6 – 83.2
Elongation [% strain]	0.98 – 1.41	30 – 50	12 – 14
Fatigue strength At 10^7 cycles [MPa]	345 – 592	256 – 299	160 – 166
Toughness [kJ / m²]	22.2 – 38.0	14.5 – 25.8	10.4 – 18.7
Thermal expansion Coefficient [µstrain / °C]	5.51 – 29.3	15 – 18	23.6 – 24.8
Galvanic potential [V]	0.14 – 0.22	(-0.18) - (-0.1)	(-0.83) - (-0.75)
Magnetic	No	No	No
Salt water durability	Excellent	Excellent	Acceptable

Table 5.2: Relevant material properties of the most suitable type of CFRP, stainless steel 316, and aluminum 5000-series. (GRANTA Selector, Version 21.2.0; 2021)

A weight estimation was performed where different components were assigned different materials. AISI 316L was used as a baseline as strength simulations has been performed to ensure the designed geometries are all at least relatively promising. The theoretical weight of such components in aluminum or CFRP were calculated using the ratio of specific strength of both respective materials to that of AISI 316L. Some components must be produced in an either metallic or CFRP material and were omitted from material changes. In short, the following weights were produced for the entire HDM, see appendix S:

- As many components as possible in AISI 316L: 127 [kg].
- As many components as possible in 5083-H321: 65 [kg].
- As many components as possible in CFRP and the rest in AISI 316L: 85 [kg].
- As many components as possible in CFRP and the rest in 5083-H321: 56 [kg].

It is unknown at the time of thesis publication what material choice, or combination of material choices, will be implemented in the final design.

5.10 Requirement specification evaluation

A preliminary evaluation based on the previously formulated detail design requirement specification was performed. Evaluations were performed by estimating if the designed solution would likely fulfill requirements or not once finalized based on the current concept. This is the only way to evaluate the design as it is not yet finished. See appendix T.

The HDM concept performs well overall in regards to a requirement specification evaluation. Most requirements are expected to be fulfilled. I wish to comment on the following unevaluated requirements (which were unevaluated due to no or only minor work being done in that area):

- (R12) Galvanic corrosion between the chosen primary metal and the carbon fiber of the hull, as well as any involved sacrificial anodes, need to be investigated further.
- (R3, R7, R8) Maintenance cost and frequency needs to be investigated further.
- (W7, W9) A manual override system is not yet well-enough developed to be evaluated.
- (H) Most requirements relating to the hub have not been evaluated due to the hub still being in a early design state.

I wish to comment on the following borderline fulfilled requirements:

- (R1) The mass of the HDM is uncertain if it will exceed the marginal required value of 100 [kg]. This is primarily due to the uncertainty of which drivetrain solution will be used. The drivetrain solution used in weight evaluations has been done with a complete winch that's geared down with a pulley solution (see appendix S for exact components). Should that solution *not* be satisfactory, there is a risk that the HDM will exceed 100 [kg] in weight.
- (R2) Initial purchasing cost will presumably be lower than the required 100 000 [SEK] if looking at components only, but additional investigations need to be done in regards to how much manufacturing (welding, etc) will add to the initial purchasing cost.
- (W2, W4) Requirements for the winch to pull rope at certain speeds involve Wattage and are left as uncertain as they are slightly outside of the scope for this master's thesis.
- (W19) Available space within the HDM is relatively constricted and for that reason it is uncertain if a 10:1 diameter ratio between pulley and rope can be achieved, *especially* if a pulley gearing system will be used.
- (F3) The structural strength of Bracing - Lower might need significant shape redesign as it is uncertain if such an overall shape (long bar) will be able to ever achieve low enough stresses at stress concentrations (since aspects like increasing tube thickness will not affect stress concentrations). It may be

necessary to include an additional attachment point between Bracing - Lower and the bottom surface of the Moon Pool.

The distribution of requirement evaluations is as follows (I would like to remind the reader that these are merely estimations if the final design will fulfill these requirements or not):

- 34 - Requirement optimally or marginally fulfilled.
- 7 - Requirement borderline to be fulfilled or not.
- 1 - Requirement unfulfilled.
- 14 - Requirement unevaluated.

6

Future work

This chapter presents areas for further work to be performed in order for this project to be implemented in industry. Since the project was cut-off in the middle of detail design there is plenty of future work to be performed, all of which will not be elaborated upon in this chapter. Apart from finishing the design (in particular a locking mechanism), the following points ought to be considered.

Requirement fulfillment evaluation: Several requirements in the requirement specification were left unevaluated. These must be investigated further. Additionally, several requirements were deemed to be borderline fulfilled, and these must also be investigated further.

Risk analysis: Due to the context of which the boat will be used, a risk analysis is crucial to perform. The most critical risks to identify are those that make the boat completely unable to fly on its hydrofoil. This is because the estimated response time and total range of the boat is completely dependent on the boat's hydrofoils, meaning that a failure could entail the boat not being able to reach an emergency in a timely fashion. By comparison, an important but less critical risk would be the breakdown of the electrically powered drivetrain, something that would lengthen deployment time but still enable the boat to reach its destination (due to the manual winch backup). A risk analysis could be performed using a Failure Mode and Effects Analysis or a Fault Tree Analysis (Lindstedt, 2006).

Dependability: Since the boat only has a single planned service performed by professionals, the dependability (in particular the maintainability) of the boat needs to be assessed and maximized. Reliability would in practice presumably be assessed by using OEM specifications and statements. Custom-made components should receive more thorough strength simulations, including fatigue analysis. Dependability should be maximized by creating a maintenance guideline for users to follow (e.g. ”clean shafts four times a year”, ”replace winch rope after 1000 hours usage”, etcetera). Maintainability should be ensured by allowing disassembly of HDM, to facilitate repair. Particular focus should be put on the replace-ability of wear-and-tear components such as linear bearings and rope.

Fatigue: Fatigue should be investigated further and simulated on. Further details on the hydrofoil use case and ocean environment should be studied to judge the relevance of fatigue via submergence factor. Since the X8 is aimed to be a testing platform for possible future boats, the X8 ought to be equipped with sensors measuring vibrations the boat is exposed to. Real-life measurements are arguably the most appropriate method to investigate fatigue, as I had difficulty finding *any* expertise that could provide a method in which to reliably estimate fatigue loads that the HDM system could be exposed to.

Creep: Dyneema ropes will exhibit some levels of creep during usage, especially if the hydrofoil is locked into deployed position via the winch (DSR, n.d.). Investigations should be performed regarding the magnitude of such creep and any potential negative effects (if any).

F = ma: The X8 is controlled by an automated system aiming to keep the ship's hull away from the water line. This is achieved by rapidly adjusting the steering of the stern hydrofoil and torpedo motor to adjust the boat's vertical travel. Should the boat be in a scenario where the system rapidly raises the boat, the HDM will experience loads higher than that of the boat's nominal weight due to Newton's second law of motion. Investigations should be performed on how high these loads are and if they are covered by the set safety factor of 2 (marginally acceptable), or up to and beyond 3 (optimally acceptable).

Production implications: The design has only some levels of design for manufacturing taken into account, but further work needs to be done. For example, welding two plates will produce a weld-radius along the edges. This radius will in practice lower available space to place other components (for example the lower linear bearing units on the wagon). Different metal alloys also react differently to being welded. To ensure future simulations and calculations are accurate, further investigation should be performed on how production methods (such as welding) affect material properties.

Re-investigate actuator types: Due to a major miscommunication regarding loads (see chapter 7.1), actuator types may have been unjustifiably eliminated as valid actuator types. For that reason, future work could be done to investigate if any actuator types more suitable than winches have been unjustifiably eliminated.

7

Discussion

Discussions surrounding this thesis are categorized into two groups: general reflections on the process used during the project, and comments made on uncertain areas or errors discovered in the thesis.

7.1 Process reflection

Reflection on the process is presented by focusing on different subjects with a short reflection attached to each subject.

Overall process: Looking at the big picture, I managed to more or less follow a traditional product development process. I am satisfied with this process and believe it is a very suitable method to tackle product development projects like these, when the starting point is very early in development. The biggest deviations from the standardized process has been the relative lack of use of formal elimination methods (such as an elimination matrix, Pugh matrix or a Kesselring matrix ((Lindstedt, 2006))), as well as the relatively early introduction of existing commercial products. Both of these points are elaborated upon further in this section.

Informal elimination: A majority of the concept elimination done in this project been done informally, i.e. not without any underlying matrix guiding the decision. This elimination method started when I was working with sourcing components during the concept generation phase, and realized that components (and their corresponding concept) were immediately obvious to be unsuitable due to specific reasons. For example, a winch might not have been marine compatible, or a lead screw might have been too heavy. Going through a formal elimination process for the hundreds of components sources would have taken a monumental amount of time. Furthermore, many decisions were left to the more experienced supervisors or left decided by the client, where a decision on their part would overrule any decision made by a formal elimination method regardless.

Modular or integrated: I had difficulties in deciding whether to adopt a modular or integrated approach (Ulrich, 2020) when working with the HDM. A modular approach would ease development in the early phase (since designs are rapidly changing) but I learned that even when trying to adopt modularity, the different chunks' interfaces still generate a domino-effect when making a change somewhere. I am unsure of how to handle the approach differently and theorize that my issues might simply stem from the project being too challenging for myself as a master's thesis student.

A possible solution to the aforementioned problem could have been to spend significant time defining suitable chunks and defining interfaces between the chunks. That might have allowed design iterations to be performed within chunks with only minimal effects to other chunks. Doing so would however take up a significant portion of the work-hours spent on this thesis.

Environmental aspects: Environmental aspects are difficult to include in this project as many aspects relating to environment (material choice, energy usage, etc) are either decided by clients or primarily dictated by other systems of the boat. Overall the greatest environmental impact relating to the design of the HDM is presumably to aid in lowering the boats negative environmental contributions during its use-phase. This is done by lowering the boat's mass, something that already is a primary goal. Due to the aforementioned difficulties of including environmental aspects, weight savings already being a primary goal, and the low estimated volume of produced boats: environmental aspects were not worked upon in this master's thesis.

Time on OEMs and sourcing components: I am of the opinion that too much time was spent sourcing components and contacting OEMs too early in the project. The reasons for doing this in the first place is to ensure implementability, since designing every component from the ground-up would never be realized in the industrial project that was running alongside the thesis. However, time spent sourcing components would have been better spent more thoroughly investigating concepts in general, so work finding sourced components would not be discarded when a concept was eliminated. Further worsening this problem was difficulties communicating with OEMs. Roughly ¾ of all inquiries sent to companies were left unanswered. This entailed not being able to use a large portion of sourced components since I could not confirm their applicability (for example maximum loads not stated in specifications, etcetera).

Industry affecting academic thesis: The outcome of this thesis is not regarded as a purely academic study; it is also regarded as an industrial contribution to the design of the X8 boat. In other words, the client is relying on the thesis producing usable design work within a specified time-deadline to be implemented in the live development project. As such, the methodology used in this report is more heavily weighted towards a practical result-driven approach to the detriment of a more academically-driven approach. The most notable example of this is the relatively early introduction of commercially available components (to show progress and feasibility) highly influencing the outcome of the development process. A consequence of this has been spending too much time on sourcing components for concepts that would eventually be eliminated regardless.

Missed opportunity: Trim foil self pulling: In chapter 4.3.1 I present a conceptual solution using gravity and the hydrofoil's own lifting force (but reversed during deployment) to create vertical motion. This concept was disallowed by stakeholders due to it deviating from the hydrofoil's pre-determined travel path and thus being believed to require a larger hole in the hull (which is undesirable). In hindsight, it would have been very advisable to perform further investigations on *how* an altered travel path would increase the size of the hole in the hull (if it were to increase it at all). Should the size difference be negligible (or even favorable) this concept could be regarded as far superior as its actuators would only need to produce a small fraction of the force necessary compared to the current hydrofoil travel path. This would entail lower cost, lower energy requirements and most importantly significantly lower weight.

Major miscommunication regarding loads: Due to a miscommunication, I (and my co-workers) learned three weeks prior to the thesis' conclusion that one of the forces had been wrongly defined. The force the HDM has to overcome during deployment had been designated to be 16 000 [N] when in reality it was around 6 500 [N]. The miscommunication occurred when the load (defined by one of the project's involved companies) was given with a factor of safety included whilst it was interpreted to be a load without a FoS. This entails that I and my co-workers had been working towards finding a solution that could generate a force of roughly 32 000 – 48 000 [N] (16 000 [N] * a FoS between 2 – 3) when in reality the solution only had to generate a force of 13 000 – 19 500 [N] (6 500 [N] * a FoS between 2 – 3). In essence, I and my co-workers had been working with two layers of safety factor. The difference between an actuator generating roughly 40 000 [N] and one generating roughly 15 000 [N] is huge when it comes to the availability of components and there is a significant risk that an actuator type has been eliminated for being regarded as too weak.

7.2 Uncertainties and errors

Uncertainties and errors are discussed by presenting in which areas uncertainties (i.e. possible undiscovered errors) may lie, as well as commenting on actual errors discovered by myself (or external parties).

Not fully explored solution space: Whilst I have explored a moderately broad range of solutions for different functions, the complete solution space has not been fully explored. For example a rack and pinion system could be a possible actuator, as could a self-contained multi-phase hydraulic system (self-contained lowering the perceived number of unique components and maintenance effort). Whilst both the primary winch-based solution and the secondary lead screw solution are promising, it is not *certain* that they are the most suitable actuator types for this scenario.

Manufacturing affecting material properties: The latest iteration of FEA simulations have been performed without any respects to how manufacturing (shaping, joining, etc) will affect material properties. For example, a curved profile that has been stamped will behave differently under load than one that has been sawed. Another example would be how material properties are affected near weld points. Manufacturing that affects material properties positively (e.g. cold-forming raising stiffness) is obviously not problematic, but effects lowering yield limits or fatigue limits is a concern. This potential lowering has hopefully been offset by the design around stress concentrations not yet having been optimized (to lower stress in the material around said stress concentrations). Nonetheless not taking manufacturing into account has resulted in a somewhat uncertain FEA simulation result.

Winch grooves and rope angles: Dynamica ropes strongly advise against the usage of grooves in a winch drum if using synthetic ropes as they would wear out very rapidly (R. Hovgaard, personal communication, May 3, 2022). Dynamica ropes instead recommend the usage of a guide system, an addition that they mean would presumably be more cost effective than the expensive manufacturing method of adding grooves to a drum. The winch design ought to be altered to follow Dynamica ropes advice. Furthermore, Dynamica ropes recommend against any lateral angles exceeding 2° when ropes are exiting pulleys or drums. This angular requirement has not been assessed.

Winches are complex: I am not confident that the proposed design have taken *all* aspects of winch design into account, despite being double-checked with Liebherr's design manual. Sources like *Design criteria for multilayer wound winch drums following lightweight design principles* (Dietz, 2004) indicate that additional aspects may warrant investigation that I have not considered. An example would be ensuring non-rotational symmetric loading and deformation. Although, the designed winch is not pulling 300 000 [N] like of that in the research paper, and the winch is currently not in a production-ready design as of yet, additional winch-design manuals could still warrant investigation.

Load scenario missed: I realized that I has missed to consider a certain drag-force scenario, see chapter 3.3.1. I was previously under the assumption that maximum drag force would occur at maximum speeds, that might however not be true. Whilst the boat is at maximum speed, only a part of the hydrofoil is submerged under water, entailing that only a part of the struts are exposed to drag. Whilst the boat is just on the verge of flying, 17.5 [knots], the entire wing and almost entirety of both struts at exposed to drag. Assuming a $C_{D,wing,17.5}$, the generated drag would exceed that of the drag at 35 [knots].

- $F_{D,total} = 2253\ [N]$ compared to $F_{D,total,35} = 1455\ [N]$
- $F_{D,wing} = 1123\ [N]$
- $F_{D,struts} = 1130\ [N]$

I has been dimensioning and simulating the structure's strength at loads experienced at 35 [knots]. In other words, the structure may be under-dimensioned in some respects.

Submergence depth uncertain: I discovered prior to concluding thesis work that something is amiss regarding the nominal submergence depth of the hydrofoil, and consequently the resulting drag force. Mantaray confirmed calculated drag forces numbers at a nominal submergence of 0.3 [m] to be accurate, but in a meeting a few months later stated that the nominal submergence is 0.7 [m]. It is uncertain if the nominal submergence is 0.3 [m] or 0.7 [m]. Should the nominal submergence be 0.7 [m], the resulting drag force at 35 [knots] will be increased from 1455 [N] to

2538 [N]. This increase would invalidate many results for simulations of the HDM strength (as they were performed with a 1455 [N] drag force).

Material properties uncertainties: I realized that the primary source of material properties, GRANTA Selector (Version 21.2.0; 2021), can assign vastly different material properties compared to industrial companies. Aspects like yield limit, price and salt-water durability are not always mirrored between GRANTA Selector and industrial sources. Due to this I am uncertain which source to trust, and am concerned that appropriate materials have been eliminated due to them being unfairly represented in a certain source.

8 Conclusion

This chapter will summarize answers for the thesis research questions and summarize the main concept developed during this project.

8.1 Research questions answered

This thesis aimed to answer three research questions, the answers of which will be presented in this chapter.

- RQ1 – What requirements does the operational profile of SSRS' activities, as well as the surrounding chunks, set on an HDM?

The operational profile of SSRS mainly sets demands on high levels of dependability, lowering mass and lowering cost.

The surrounding chunks set relatively few requirements on the design of an HDM. Seeing as it is a largely self-contained system, the requirements are mainly that is fits geometrically within an allotted space and attaches appropriately to given interfaces.

Additional requirements can be viewed in chapter 5.1.

A requirement outside of the thesis scope worth mentioning is the compatibility between the HDM's electrical equipment and the boat's on-board power supply. E.g., making sure the electrical motor and trim work with the batteries.

- RQ2 – How should an HDM best be driven (i.e. actuator to provide movement)?

An HDM (in this context) is best driven using a winch-based system. The primary reason for this is the importance of weight when designing for a hydrofoiling boat. In order to provide linear motion, something must be transmitting force to the hydrofoil. This something must have components that are at least the same length as the stroke length. Compared to the numerous solutions using steel for these components, a synthetic polymer rope offers vast weight savings.

- RQ3 – How should an HDM be designed to withstand the forces applied to it?

How structural integrity of an HDM is achieved depends on the chosen design (e.g. chosen actuator) and geometric constraints of the HDM (e.g. Moon Pool interface requirements). An example of achieving this is having load-bearing components consist of a system of bars that transfer the lifting force from the hydrofoil to the remainder of the hull. In this scenario, structural integrity is achieved by ensuring stability and strength in different load cases (retracted and deployed). This strength is achieved using materials with a high specific strength (to lower mass). Given a certain material, the geometry of the HDM ought to be in a sweet spot that minimizes mass and maximizes safety factor. This can in practice entail designing components with deceptively small cross-sectional areas (such as the frame bracing) or with areas of components having material removed from them (such as the winch frame).

8.2 HDM realized

Development of an HDM aimed to be used in SSRS X8 boat consists of:

- A winch-based solution for an actuator, as a means of lowering mass.
- Linear movement is provided by linear plain bearings, since marine compatibility of plain bearings is more important than the low friction of roller bearings.
- A frame consisting of multiple bars to transfer the lifting force from the hydrofoil to the remainder of the hull, via the Moon Pool.
- Stainless steel components to eliminate the risk of galvanic corrosion between the carbon fiber hull and the alloy, and also to ensure that there is competence in the marine industry to manufacture and develop future iterations of an HDM.

This HDM design is supported by the following deliverables, also produced during this thesis:

- Description overview of the to-be-designed X8 boat.
- Calculations surrounding approximate drag forces and possible fatigue problems to be expected in different speeds.
- Function list.
- Requirement specification.
- Concept catalogue for systems within an HDM.
- Results from a qualitative method of eliminating concepts.
- CAD models of HDM design.
- Strength simulations of HDM design.
- Material recommendation.
- Requirement specification evaluation.

8.3 Final conclusions

Overall, this thesis has shown that there are a multitude of ways to realize an HDM. There are different actuators that could be used (e.g. winch, leadscrew, and more), many different materials (e.g. stainless steel, aluminum with electric isolation, hardwood), and the HDM functionalities can be implemented in many different variants (e.g. integrating locking in actuator or having a self-contained locking system).

Seeing as there are so many possibilities to realize the HDM design, the key question to answer is not if a design *can* be made, but rather which design is the most *optimal*.

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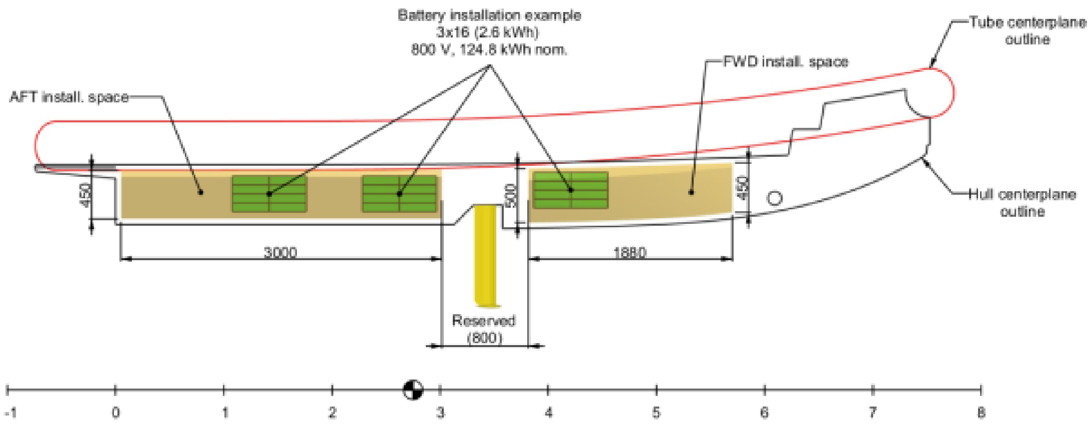
A

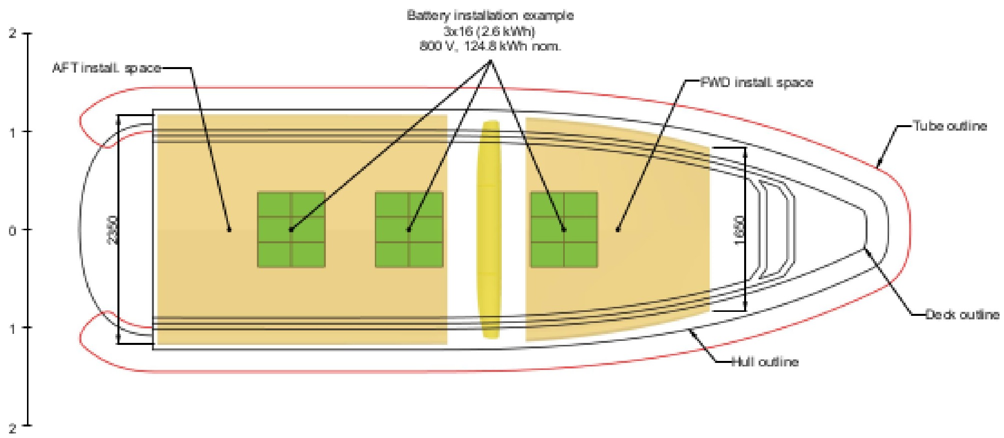
Appendix

B

X8 early dimensions

Source: F. Falkman, personal communication, February 2, 2022)





C
Python code - Drag forces

```python
from tkinter import *


class Frames(object):

    def __init__(self):
        pass

    def main_frame(self, root):
        root.title("Drag force calculations")

        Label(root, text="rho_water? [kg/m³]").grid(row=0, column=0)
        self.rho_water = Entry(root, textvariable=StringVar())
        self.rho_water.insert(0, "1000")
        self.rho_water.grid(row=0, column=1)

        Label(root, text="U? [m/s] (8 knots = 4.12 [m/s]; 35 knots = 18.01 [m/s])").grid(row=2, column=0)
        self.U_i = Entry(root, textvariable=StringVar())
        self.U_i.insert(0, "4.12")
        self.U_i.grid(row=2, column=1)

        Label(root, text="C_D,wing? [-] (8 knots: 0.035; 35 knots: 0.005)").grid(row=3, column=0)
        self.C_D_wing_i = Entry(root, textvariable=StringVar())
        self.C_D_wing_i.insert(0, "0.035")
        self.C_D_wing_i.grid(row=3, column=1)

        Label(root, text="C_D,struts? [-] (based on typical NASA airfoil)").grid(row=4, column=0)
        self.C_D_struts_i = Entry(root, textvariable=StringVar())
        self.C_D_struts_i.insert(0, "0.05")
        self.C_D_struts_i.grid(row=4, column=1)

        Label(root, text="A_wing? [m²]").grid(row=5, column=0)
        self.A_wing_i = Entry(root, textvariable=StringVar())
        self.A_wing_i.insert(0, "0.792")
        self.A_wing_i.grid(row=5, column=1)

        Label(root, text="Submergence depth foil? [m] (0 < S < 1.67; for wetted area of struts)").grid(row=6, column=0)
        self.submergence_i = Entry(root, textvariable=StringVar())
```

```
        self.submergence_i.insert(0, "1.67")
        self.submergence_i.grid(row=6, column=1)

        Button(root, text="Submit", command=self.result_frame).grid(row=7,
        ↪   column=0)
        Button(root, text="Exit", command=root.destroy).grid(row=7, column=1)

    def result_frame(self):
        # Gather input
        rho_water = float(self.rho_water.get())
        # mu_water = float(self.mu_water.get())
        U = float(self.U_i.get())
        C_D_wing = float(self.C_D_wing_i.get())
        C_D_struts = float(self.C_D_struts_i.get())
        A_wing = float(self.A_wing_i.get())
        submergence = float(self.submergence_i.get())

        # Perform calculations
        A_struts = submergence * 0.167
        F_D_wing = round(0.5 * C_D_wing * rho_water * U**2 * A_wing)
        F_D_struts = round(2 * 0.5 * C_D_struts * rho_water * U**2 * A_struts)  #
        ↪   for both struts
        F_D = round(F_D_wing + F_D_struts)

        # Generate output window
        result = Toplevel()
        result.title("Results")
        result.geometry("300x100")

        Label(result, text=f"F_D_wing = {F_D_wing} [N]").pack()
        Label(result, text=f"F_D_struts = {F_D_struts} [N]").pack()
        Label(result, text=f"F_D = {F_D} [N]").pack()

        Button(result, text="Exit", command=result.destroy).pack()


root = Tk()
app = Frames()
app.main_frame(root)
root.mainloop()
```

D

Python code - Submergence factor

```
import tkinter as tk
from tkinter import simpledialog
from tkinter import ttk


# Functions
def submit():
    # Gather inputs
    depth_nom = float(input_depth.get())
    wave_height = float(input_wave_height.get())
    wave_length = float(input_wave_length.get())
    C_L = float(input_C_L.get())

    # Perform calculations
    f = round(V / wave_length, 4)  # [Hz] frequency of trough exposure to foil
    sub = depth_nom - wave_height  # [m] submergence, depth of foil relative to
    ↪  water surface
    F_S = 1 - 0.222 * ((1.5 * chord - sub) / chord)**2  # [-] submergence factor
    F_Lmax = round(0.5 * C_L * rho * V**2 * A)  # [N] maximum lift force with no
    ↪  submergence factor
    F_Lmin = round(0.5 * C_L * rho * V**2 * A * F_S)  # [N] minimum lift force
    ↪  with submergence factor
    F_Delta = F_Lmax - F_Lmin  # [N]

    print(F_S)

    # Generate output window
    output_window = tk.Tk()
    output_window.geometry("500x200")
    output_window.title("Results")

    tk.Label(output_window, text=f"f = {f} [Hz] - Frequency").grid(row=0,
    ↪  column=0)
    tk.Label(output_window, text=f"F_max = {F_Lmax} [N] - Lift force
    ↪  maximum").grid(row=1, column=0)
    tk.Label(output_window, text=f"F_min = {F_Lmin} [N] - Lift force
    ↪  minimum").grid(row=2, column=0)
    tk.Label(output_window, text=f"F_Delta = {F_Delta} [N] - Lift force
    ↪  difference").grid(row=3, column=0)


# Defining constants - Everything in SI-unit
```

```
A = 0.425  # [m²] surface area, vertical projection
chord = 0.230  # [m] chord
rho = 1025  # [kg/m³] density
V = 18.0  # [m/s] foil velocity at 35 knots, equating 65 km/h

# Input variables
input_window = tk.Tk()
input_window.geometry("500x200")
input_window.title("Submergence factor fatigue calculation")

tk.Label(input_window, text="Nominal depth of foil?").grid(row=0, column=0)
input_depth = tk.Entry(input_window)
input_depth.grid(row=0, column=1)

tk.Label(input_window, text="Average wave height?").grid(row=1, column=0)
input_wave_height = tk.Entry(input_window)
input_wave_height.grid(row=1, column=1)

tk.Label(input_window, text="Average wave length?").grid(row=2, column=0)
input_wave_length = tk.Entry(input_window)
input_wave_length.grid(row=2, column=1)

tk.Label(input_window, text="C_L?").grid(row=3, column=0)
input_C_L = tk.Entry(input_window)
input_C_L.insert(0, "0.37")
input_C_L.grid(row=3, column=1)

tk.Label(input_window, text="NOTE: Equations only valid in chord depths between
↪  0.2 - 1.5.").grid(row=4, column=0)

tk.Button(input_window, text="Submit", command=submit).grid(row=5, column=0)

input_window.mainloop()
```

E

Concept-phase requirement specification

Requirement specification

No.	Requirement	D/W 1-5	Unit	Value	Evaluation	Reference
G	**General**					
G1	Operating volume Should be less noisy than heavy wind (to ensure crewmates can talk whilst foil is deploying).	W4	dB	<95	OEM specification	SSRS
G2	Mass	D	kg	<100	CAD measurements	SSRS
G3	Purchasing cost	D	SEK	<100 000	OEM specification	SSRS
G4	Yearly maintenance cost	D	SEK	<10 000	OEM specification + cost estimations	SSRS
G5	Electrically driven Root energy source must be compatible with boat's main drive train battery.	D	Y/N	Y	OEM specification	Micropower
G6	Depth monitoring Users can monitor the depth of the hydrofoil at any given time, including during deployment or retraction. Outside the scope of this project.	W3			-	SSRS
Geo	**Geometry**					
Geo1	Deck width	D	mm	<1000	CAD measurements	SSPA
Geo2	Deck length	D	mm	<800	CAD measurements	SSPA
Geo3	Deck height	D	mm	<1500	CAD measurements	SSPA
Geo4	Allow foil rotation	D	°	±15	CAD measurements	Mantaray
Geo5	Allow axle rotation	D	°	±15	CAD measurements	Mantaray
Me	**Mechanical strength**					
Me1	Withstand frontal drag force Top speed of 35 knots.	D	N	>813	FEA simulations	Student
Me2	Withstand static weight Main foil must carry 80-90% of the 2600 kg boat mass.	D	kg	>2600	FEA simulations	Student
Me3	Withstand lateral forces Mantaray says these can occur.	D	N	>4000	FEA simulations	Student
Me4	Maintain trim angle during drag forces	D	°	0.1	FEA simulations	Mantaray
Env	**Use-environment**					
Env1	Withstand sea-water	D	Y/N	IP67	OEM specification	SSRS
Env2	Compatibility with carbon fiber	D	Y/N	Y	GRANTA Selector	Student
Env3	Withstand corrosion Sea water in close proximity to carbon fiber.	D	Y/N	Y	GRANTA Selector	Student
Env4	Operating air temperature Biggest issue is ice build-up.	D	°C	-10 ↔ 70	OEM specification FEA simulations	SSRS

M	**Movement**					
M1	Lift foil automated – Mass > 40kg probably required in case it needs to lift in dry-dock.	W5	kg	>40	OEM specification	SSRS
M2	Deployment force The lift force from the foil when traveling at max deployment speed.	D	N	16000	OEM specification FEA simulations	Student
M3	Foil movement angle During deployment and retraction, the foil moves along a line that's 78° from the hull plane.	D	°	78	CAD measurements	SSPA
M4	Lift foil automated – Time	W4	s	<20	Calculations	SSRS
M5	Lower foil automated – Time	W4	s	<20	Calculations	SSRS
M6	Lift foil automated during drag Extra forces might occur due to friction (pinching).	D	knots	>8	FEA simulations	SSRS
M7	Lower foil automated during drag Extra forces due to foil lift as well as possible extra forces due to friction (pinching).	D	knots	>8	FEA simulations	SSRS
M8	Lift foil manually – Mass	D	kg	>40	Calculations	SSRS
M9	Lift foil manually – Time	D	s	<180	Calculations	Student
M10	Lower foil manually – Time	D	s	<180	Calculations	Student
M11	Change trim	W2	°	±5	CAD measurements	Mantaray
M12	Change trim speed	W4	s	>10	Calculations	SSRS
M13	Allow foil depth	D	mm	2000	CAD measurements	Mantaray
M14	Allow multiple foil depths	W1	mm	300↔2000	Design	SSRS
M15	Avoid collision Foil would somehow avoid collision with objects if detected. Emergency use.	W1	Y/N	Y	Design	SSRS
Ma	**Maintenance**					
Ma1	Facilitate maintenance by laypeople	D	Y/N	Y	Subjective evaluation	SSRS
Ma2	Facilitate repair by professionals	D	Y/N	Y	Subjective evaluation	SSRS
Ma3	Time between minor maintenance	D	year	<0.1	OEM specification + lifespan calculations	SSRS
Ma4	Time between major maintenance	D	year	<1	OEM specification + lifespan calculations	SSRS
Ma5	Time between renovation	D	year	<12.5	OEM specification	SSRS
Ma6	Lifespan	D	year	>25	OEM specification	SSRS
Ma7	Hydrofoil change possible	D	Y/N	Y	-	SSRS

F

Initial material investigation

Using GRANTA Selector (Version 21.2.0; 2021) to look at material database level 2 within the composites, polymers and elastomers. The following materials were not included due to:

1. Glasses, non-technical ceramics and technical ceramics: their brittle nature combined with a cyclic load and a purely tensile load make them highly unsuitable for this component.
2. Metal and alloys: Will do a separate analysis of metals later.

A chart was made using specific strength to cost to determine which materials are most suited. In order, suitable material types that stood out were:

1. Softwood
2. Bamboo
3. Polyurethane (elastomer)
4. Polypropylene (PP)
5. Polyvinylchloride (PVC)
6. Al / Si carbide composite
7. CFRP
8. Polyester
9. Starch-based thermoplastics
10. Hardwood
11. GFRP
12. Rigid polymer foam

I.e. in material families we have:

1. Woods
2. Elastomers
3. Polymers
4. Composites
5. Foams

In regards to wood, it seems like hardwoods are traditionally used for the exterior / load bearing parts of the ship (source). Some woods (such as cedar apparently) have natural chemicals which prevents rot. Bamboo is not explored further due to the limited geometry of non-composite woods and being relative newcomers in western engineering making reliable information and local manufacturing more difficult (relative to e.g. oak).

The most promising materials and their most relevant properties are listed below. All values are approximate.

Legend:

- C $[SEK/kg]$ - Price
- ρ $[kg/m^3]$ - Density
- E $[GPa]$ - Young's modulus
- σ_y $[MPa]$ - Yield strength
- R $[\mu\Omega cm]$ - Resistivity
- *Rec* - Recyclable
- *Down* - Downcyclable
- *Bio* - Biodegradable
- *Ren* - Renewable

Woods – Oak – Quercus Robur		
C - 20	ρ - 700	E - 14
σ_y - 58	R - 6e13	*Rec* - No
Down - Yes	*Bio* - Yes	*Ren* - Yes

Elastomers - Hard rubber - Ebonite		
C - 28	ρ - 1015	E - 1.5
σ_y - 70	R - 1e21	*Rec* - No
Down - Yes	*Bio* - No	*Ren* - Yes (probably)

Polymers – Thermoplastics – PET 45% glass fiber		
C - 15	ρ - 1700	E - 14
σ_y - 160	R - 3e20	*Rec* - No
Down - Yes	*Bio* - No	*Ren* - No

Polymers – Thermosets – Polyester cast		
C - 17	ρ - 1040	E - 3
σ_y - 38	R - 3e18	*Rec* - No
Down - Yes	*Bio* - No	*Ren* - No

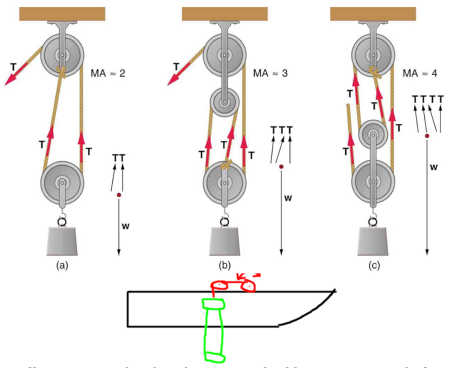
GFRP		
C - 310	ρ - 1800	E - 21
σ_y - 250	R - 2.5e21	*Rec* - No
Down - Yes	*Bio* - No	*Ren* - No

CFRP		
C - 350	ρ - 1500	E - 100
σ_y - 750	R - 5e5	*Rec* - No
Down - Yes	*Bio* - No	*Ren* - No

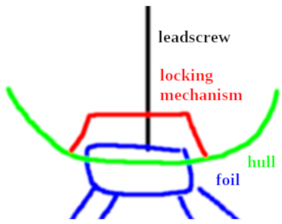
Rigid polymer foam (HD)		
C - 140	ρ - 300	E - 0.4
σ_y - 6	R - 1e18	*Rec* - No
Down - Yes	*Bio* - No	*Ren* - No

G

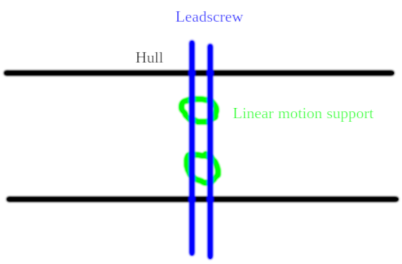
Early concept catalog



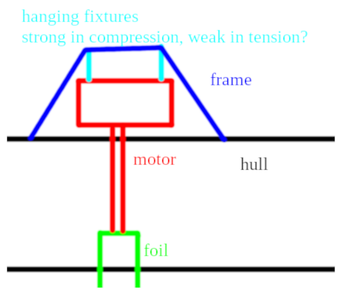
Pulley system with either chains or steel cables. Not necessarily for the force, but rather for the space and weight savings. https://philschatz.com/physics-book/resources/Figure_10_05_04a.jpg



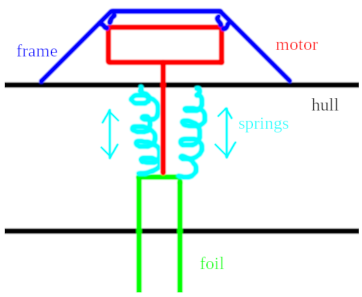
Thin leadscrew with locking mechanism. The thinness of the leadscrew is okay from a buckling perspective since it'll be locked by another mechanism. If no other support is added, it'll "only" have to withstand lifting the weight of the foil. But this solution is only acceptable if we don't want to be able to retract the foil whilst it's still flying (otherwise the leadscrew will have to lift the entire hull without buckling).



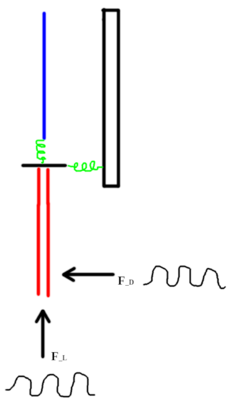
Leadscrew with linear motion support in places that are reasonable to prevent buckling. How will such a support with with the non-smooth and moving geometry of a screw?



Leadscrew motor hanging loosely to the frame. Weight of motor helps push hydrofoil down. Attach motor on frame instead of on hydrofoil to help bring electronics away from water.



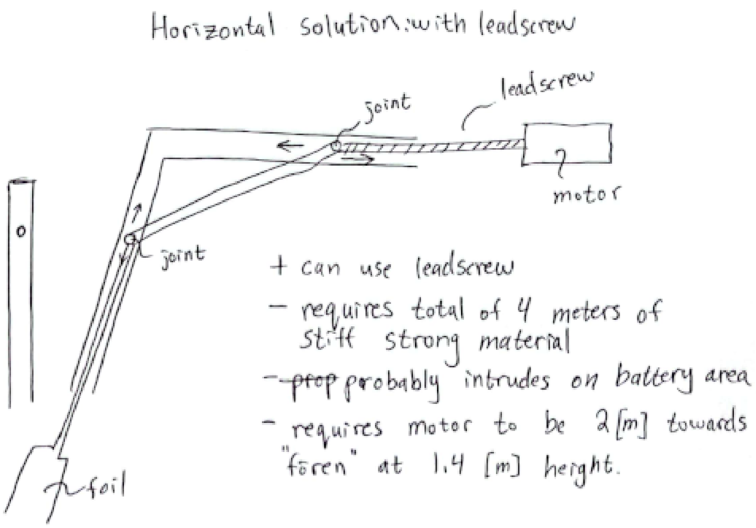
Springs that are compressed when pulling up and that decompress to push the foil down

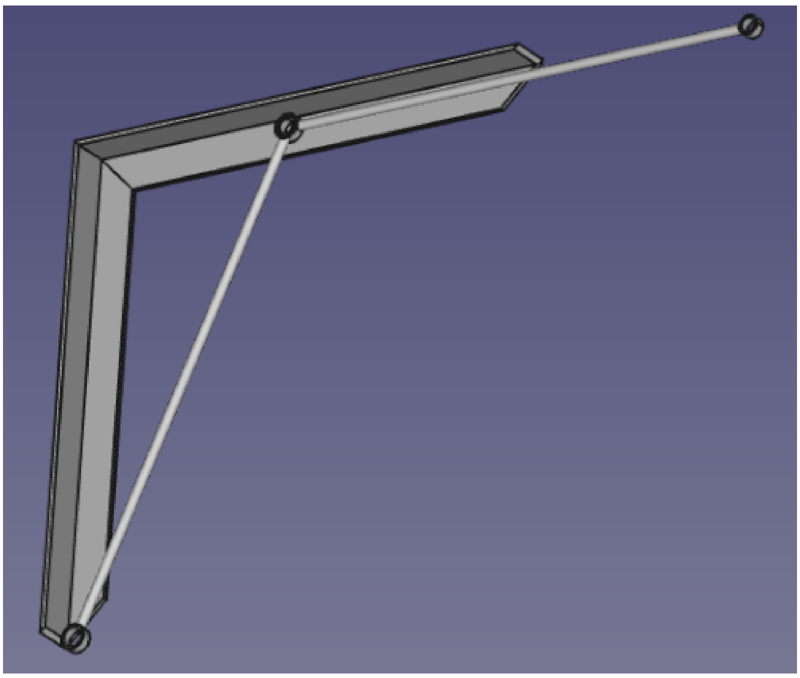


Springs to counter fatigue? Both from drag force and lifting force. Unsure how big amplitudes are.

H

Frame concept oscar





I

Lock - Hole ANSYS simulation numbers

Simulations done in ANSYS (version 21.1.0; 2021).

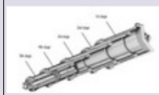
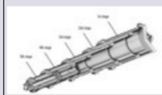
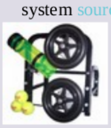
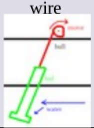
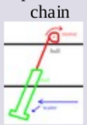
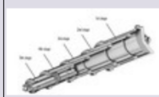
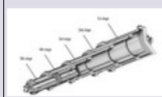
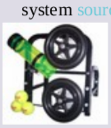
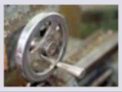
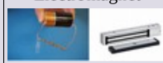
Name	P3 - DS_Diameter	P4 - Surface Body Thickness	P5 - Equivalent Stress Maximum	P6 - Safety Factor Minimum
Units	m	m	Pa	
DP 0 (Current)	0,08	0,02	2,7483E+07	9,0964
DP 1	0,06	0,02	3,4637E+07	7,2177
DP 2	0,05	0,02	4,0116E+07	6,2319
DP 3	0,04	0,02	4,8924E+07	5,11
DP 4	0,03	0,02	6,3816E+07	3,9175
DP 5	0,02	0,02	9,3626E+07	2,6702
DP 6	0,01	0,02	1,8293E+08	1,3666
DP 7	0,025	0,02	7,5589E+07	3,3074

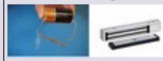
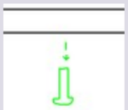
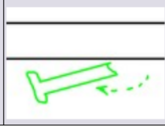
J

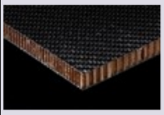
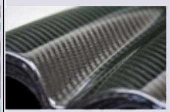
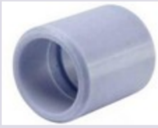
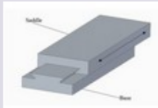
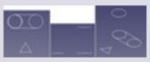
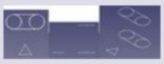
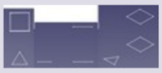
Morphological matrix

Morphological matrix

Solutions are named <function number> - <S + solution number>

No.	Solutions → Functions ↓	1 / 6	2 / 7	3 / 8	4 / 9	5 / 10
	Movement					
M1	Provide upwards motion (automated)	Leadscrew source 	Winch wire source 	Winch chain source 	Mutli-stage hydraulics source 	Multi-stage pneumatics source 
		Gear rack source 	Lever mechanism source 	Wheel / thread system source 		
M2	Provide downwards motion (automated)	Leadscrew source 	Foil pull + winch wire 	Foil pull + winch chain 	Mutli-stage hydraulics source 	Multi-stage pneumatics source 
		Gear rack source 	Lever mechanism source 	Wheel / thread system source 	Expanding springs source 	
M3	Provide upwards motion (manual)	Manual override for M1 source 	Manual crank wire / chain source 			
M4	Provide downwards motion (manual)	Manual override for M2 source 	Manual crank wire / chain + gravity source 			
M5	Lock foil (retracted)	Dampener (electromagnetic?) 	Automated screw 	Electromagnet 	Locking lug spring inserted source 	Electromagnetic passive break source 
		Clamp 	Latch 	Leadscrew thread lock source 		

M6	Lock foil (deployed)	Dampener (electromagnetic?) 	Automated screw 	Electromagnet 	Locking lug spring inserted source 	Electromagnetic passive break source 
		Clamp 	Latch 	Leadscrew thread lock source 		
M7	(Lock foil in user-chosen depth)	Locking pin spring inserted source 	Electromagnetic passive break source 	Leadscrew thread lock source 		
M8	Resist fatigue	Dampener 	Electromagnetic dampener 	Springs 		
	(Trim)					
T1	(Allow trim movement)	Plain bearing source 	Roller bearing source 			
T2	(Provide trim movement)	Manual crank source 	Electric motor geared to axle source 	Electric motor + pulley wire source 	Electric motor + pulley chain source 	Leadscrew source 
T3	(Lock trim)	Locking lug spring inserted source 	Electromagnetic passive break source 	Leadscrew thread lock source 		
	Other					
O1	Avoid collision Make foil somehow avoid collision with objects if detected. Emergency use.	Foil crack source 	Drop foil (+wire?) 	Expanding springs source 	Release trim 	Disconnect break & motor source 

	Transfer foil forces to hull					
FΣ1	Structure geometry Image sources	Arch solid 	Arch truss 	Beam 	Solid + arch 	Box girder 
		Cable 	Suspension 	Cantilever 	Plate girder 	Truss 
FΣ2	Structure material	Sandwich source 	Steel source 	Carbon fiber source 	Polymer source 	Aluminum alloy source 
		Foam source 				
F3	Allow vertical motion Image sources	Plain linear bearing 	Linear ball bearing 	Dovetail slide 	Rail guide 	Track-roller guide 
F5	Allow vertical motion geometry	Linear alpha 	Linear charlie 	Linear delta 	Linear echo 	Linear foxtrot 
		Linear golf 	Linear hotel 	Linear india 	Linear juliet 	

K

Company B incompatability reasons

Reasons for company B's products incompatabilities with the usage scenario (Company B, personal communication April 28, 2022):

- Saltwater - A droplet och mist of saltwater in the wrong place (in some crevace or motor windings) will cause corrosion. Saltwater on the surface of the drum is okay. Company B recommends putting it inside of a water-sealed box.
- Braking is done via a ratchet brake. Implementing locking via a ratchet brake will lead to creep over time where the locking will suddenly fail without warning.
- Winch drums are not designed to have two output ropes.
- It's a winch and not a hoist, meaning that it can only provide torque in one direction and locking in the other direction (ratchet brake). In other words, it's impossible to "brake" the hydrofoil during retraction when the boat is in motion and the hydrofoil is generating lift.
- Ratchet brake under the current load will need to be replaced once a again (early estimation).
- Company B means that a synthetic rope cannot be used in the design scenario with their products due to it wearing out too quickly. Company B recommends steel rope instead.

L

Detail design requirement specification

System-level requirement specification

No.	Requirement	D/W 1-5	Unit	Value marginal	Value optimal	Evaluation	Reference	Notes
G	**Global goals**							**All modules combined.**
G1	Raise dependability – Focus on reliability.					OEM statements and subjective evaluation.	SSRS	
G2	Lower mass – Considerations to center of gravity.					OEM specifications and CAD measurements.	SSRS	
G3	Lower maintenance cost – Considerations to expertise and time required.					OEM statements and subjective evaluation.	SSRS	
G4	Lower initial purchasing cost.					OEM statements.	SSRS	
R	**Global requirements**							**All modules combined.**
R1	Mass	D	kg	<100	<50	OEM specifications + CAD measurements	SSRS	
R2	Initial purchasing cost	W5	SEK	<100 000		OEM invoice	SSRS	Purchasing cost limit depends on maintenance costs.
R3	Yearly maintenance cost	D	SEK	<10 000		OEM statements + cost estimations	SSRS	
R4	Chunk width	D	m	<0.9		CAD measurements	SSPA	Port ↔ Starboard
R5	Chunk length	D	m	<0.6		CAD measurements	SSPA	Bow ↔ Stern
R6	Chunk height	D	m	<0.7		CAD measurements	SSPA	
R7	Time between minor maintenance	D	year	<1		OEM statements	SSRS	
R8	Time between major maintenance	D	year	<12.5		OEM statements + lifespan calculations	SSRS	
R9	Minor maintenance by laypeople possible	W5	Y/N	Y		Subjective evaluation	SSRS	
R10	Major renovation by professionals possible	D	Y/N	Y		Subjective evaluation	SSRS	
R11	Water-proof	D	IP	IP54	IP67	OEM specifications + Subjective evaluation	SSRS	Custom-built boxes without IP ratings could facilitate usage of lower IP-rated equipment.
R12	Marine environment	D	Y/N	Y		OEM statements	SSRS	Saltwater.
R13	Voltage	D	V	1 ↔ 800	12 ; 24 ; 48	OEM specifications	Micropower	Any voltage is possible but 12, 24 or 48 is highly preferable.
R14	Operating temperature	D	°C	-10 ↔ 70		OEM specifications	SSRS	Coldest air temperature without ice on water, hottest air temperature in enclosed area with sun shining.
W	**Winch**							**Winch unit & winch accessories.**
W1	Brake downwards line (automated)	D	N	>32 000	>48 000	OEM specifications	Mantaray	Motion in same direction as force, hence braking required. 16 000 [N] nominal load * 2 or 3 factor of safety.
W2	Pull upwards line (automated)	D	m/s	>0.06	>0.09	OEM specifications	SSRS	1.8 [m] in 20 or 30 seconds.
W3	Pull downwards line (automated)	D	N	>32 000	>48 000	OEM specifications	Mantaray	16 000 [N] nominal load * 2 or 3 factor of safety.
W4	Pull downwards line (automated)	D	m/s	>0.06	>0.09	OEM specifications	SSRS	1.8 [m] in 20 or 30 seconds.
W5	Manual winch force input	D	N	<150	<50	Lever + frictional losses calculations	Student	Reasonable demands for human exertion.
W6	Pull upwards line (manual)	D	N	>4 000	>6 000	OEM specifications	Student	Weight of hydrofoil + accessories 200 [kg] * 2 or 3 factor of safety. Strength of winch mechanism.
W7	Pull upwards line (manual)	D	m/s	>0.01	>0.015	Assumptions	Student	1.8 [m] in 120 or 180 seconds. Reasonable demands for human exertion.
W8	Brake upwards line (manual)	D	N	>4 000	>6 000	OEM specifications	Student	Motion in same direction as force, hence braking required. Weight of hydrofoil + accessories 200 [kg] * 2 or 3 factor of safety. Strength of winch mechanism.
W9	Pull downwards line (manual)	D	m/s	>0.01	>0.015	Assumptions	Student	1.8 [m] in 120 or 180 seconds. Reasonable demands for human exertion.
W10	Lock foil (retracted)	D	N	>4 000	>6 000	OEM specifications	Student	Weight of hydrofoil + accessories 200 [kg] * 2 or 3 factor of safety.
W11	Lock foil (deployed)	D	N	>52 000	>78 000	OEM specifications	SSPA	Lift force of 26 000 [N] * 2 or 3 factor of safety.
W12	Lock foil in user-chosen depth	W1	Y/N	Y		Assumptions	SSRS	Trivial implementation with winch locking solution.
W13	Tension line	D	Y/N	Y		Design	Student	Ensure proper feeding into drum by tensioning line onto drum.

W14	Emergency foil retraction	W4	m/s	>0.6	>3.6	Calculations	Student	Calculations of lift force boat has in air vs own-weight + clutch retraction time.
W15	Positional awareness	D	Y/N	Y		Sigma Embedded evaluates	Sigma Embedded Engineering	Only end-point necessary. Primarily Sigma Embedded's responsibility.
W16	Measured volumed near crewmates	W4	dB	<95	60 ↔ 40	OEM specifications	SSRS	Should be less noisy than heavy wind to ensure crewmates can talk whilst foil is deploying. But not so quiet as not to be heard when activated.
W17	Rope operating strength	D	N	>52 000	>78 000	OEM specifications	Student	Lift force of 26 000 [N] * 2 or 3 factor of safety.
W18	Drum ↔ Rope diameter ratio	D	-	>20:1		CAD measurements	Dynamica ropes	Dynamica ropes states synthetic Dyneema needs this.
W19	Pulley ↔ Rope diameter ratio	D	-	>10:1		CAD measurements	Dynamica ropes	Dynamica ropes states synthetic Dyneema needs this.
W20	Groove ↔ Rope diameter ratio	D	-	>1.1:1		CAD measurements	Dynamica ropes	Dynamica ropes states synthetic Dyneema needs this.
W21	Number of output ropes	D	-	2		Design	Student	The winch needs to pull a load in two directions.
W22	Number of output directions	D	-	2		Design	Student	The winch needs to pull a load in two directions.
W23	Structural strength	D	N	>52 000	>78 000	FEA simulations	Student	The winch and its attachment points to the HDM frame need to be able to withstand the rope force of 26 000 [N] * 2 or 3 factor of safety.

T	**Trim**							**Trim unit & trim accessories.**
T1	Allow trim-unit movement	D	Y/N	Y		Design	SSRS	Friction calculations.
T2	Provide trim movement	D	N	>8 760	>13 140	Calculations + OEM specifications	Student	Axial force to counter torque from drag 4 380 * 2 or 3 factor of safety.
T3	Provide trim movement	D	°/s	<0.5	<1.5	OEM specifications	Mantaray	Trim unit axial speed + lever → rotational speed.
T4	Lock trim	D	N	>8 760	>13 140	Calculations + OEM specifications	Student	Axial force to counter torque from drag 4 380 * 2 or 3 factor of safety.

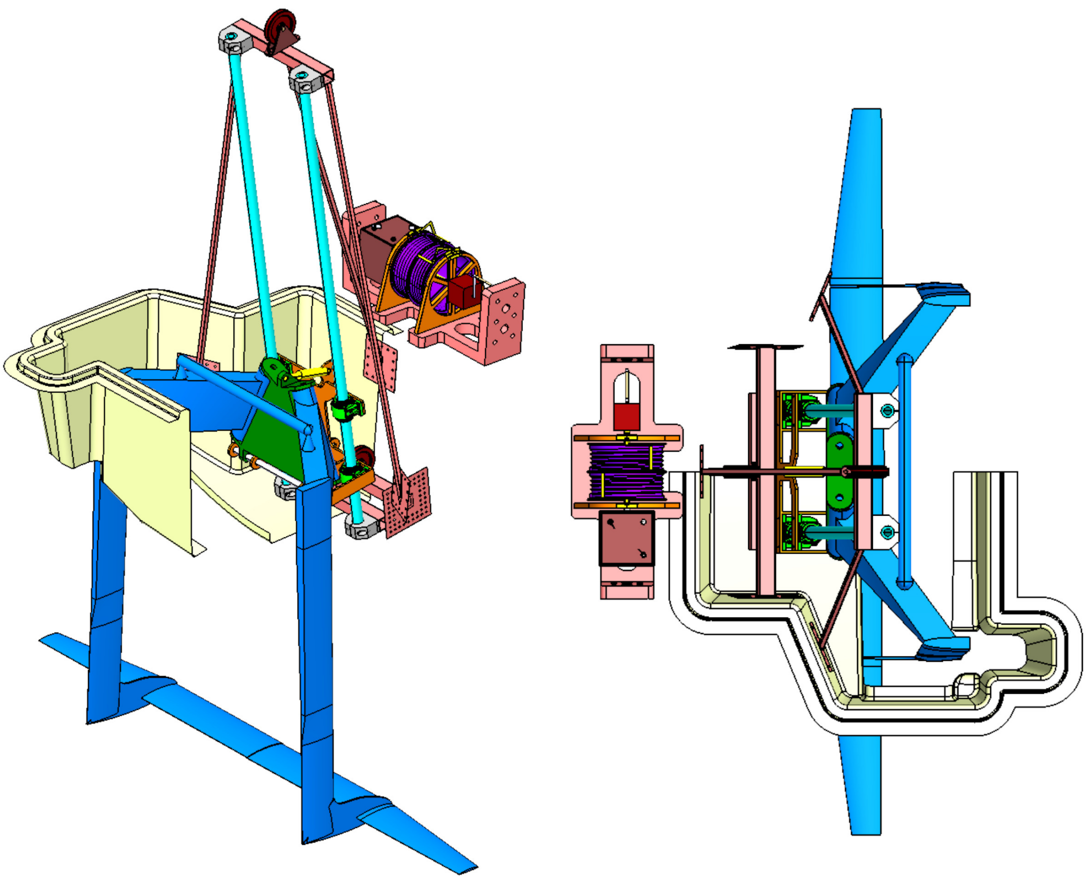
Wa	**Wagon**							**Wagon & linear bearings / rail unit, excluding shafts / rails.**
Wa1	Withstand radial force	R	N	>4 000	>6 000	Calculations + OEM specifications	Student	Radial force in linear bearings of 2000 * 2 or 3 factor of safety.
Wa2	Hold trim unit	R	N	>8 760	>13 140	Calculations + OEM specifications	Student	Axial force to counter torque from drag 4 380 * 2 or 3 factor of safety.
Wa3	Hold upwards line	R	N	>4 000	>6 000	FEA simulations	Student	Weight of hydrofoil + accessories 200 [kg] * 2 or 3 factor of safety.
Wa4	Hold downwards line	R	N	>52 000	>78 000	FEA simulations	Student	Lift force of 26 000 [N] * 2 or 3 factor of safety.
Wa5	Hold hub	R	N	>52 000	>78 000	FEA simulations	Student	Lift force of 26 000 [N] * 2 or 3 factor of safety.

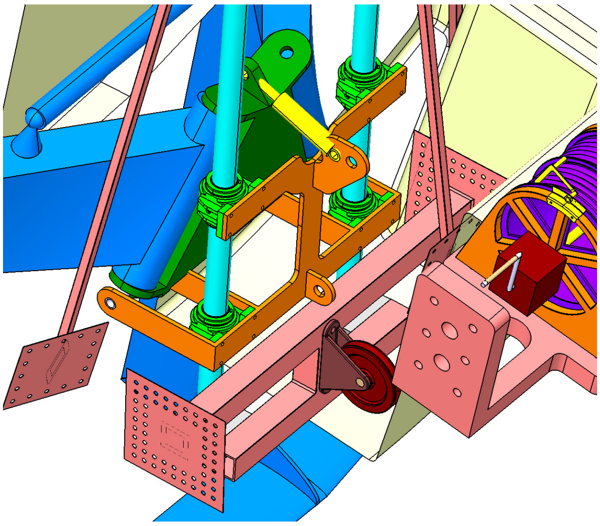
H	**Hub**							
H1	Hold foil	R	Pa	Yield limit * 2 FoS	Yield limit * 3 FoS	FEA simulations	Student	Yield limit of the hub's material.
H2	Allow foil axle rotation	R	Y/N	Y		Design	Mantaray	Copy Mantaray's 0.6 scale version.
H3	Hold trim unit	R	N	>8 760	>13 140	Calculations + OEM specifications	Student	Axial force to counter torque from drag 4 380 * 2 or 3 factor of safety.
H4	Hold wagon	R	N	>52 000	>78 000	FEA simulations	Student	Lift force of 26 000 [N] * 2 or 3 factor of safety.
H5	Allow hub rotation	R	N	>26 000	>39 000	Calculations + OEM specifications	Student	Radial force of bearings: Lift force of 26 000 * ½ (two bearings) * 2 or 2 factor of safety.
H6	Hydrofoil change possible	W3	Y/N	Y		Assessment	SSRS	Design capable of non-destructive disassembly.

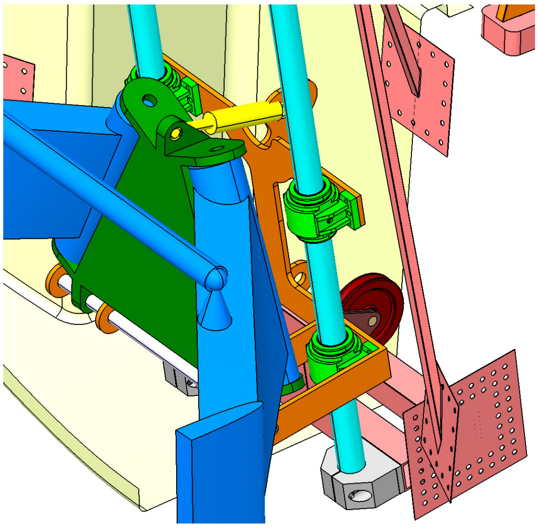
F	**Frame**							**Frame & shafts / rails**
F1	Counter drag-torque	R	Nm	>3 600	>5 400	FEA simulations	Student	Drag from the main foil causing torque in the starboard → port vector. Torque of roughly 1 800 [Nm] * 2 or 3 factor of safety.
F2	Counter twisting-torque	R	Nm	>2 000	>3 000	FEA simulations	Student	Uneven drag on wing tips causing torque in topside ↔ below vector. Rough estimations of 1 000 [Nm] * 2 or 3 factor of safety.
F3	Transfer axial upwards forces	R	N	>52 000	>78 000	FEA simulations	Student	Lift force of 26 000 [N] * 2 or 3 factor of safety.
F4	Counter leaning	R	N	>6 000	>9 000	FEA simulations	Student	Lateral forces of roughly 3 000 * 2 or 3 factor of safety.
F5	Foil travel angle	R	°	12		CAD measurements	SSPA	Foil travel angle to fit into hull cutout.

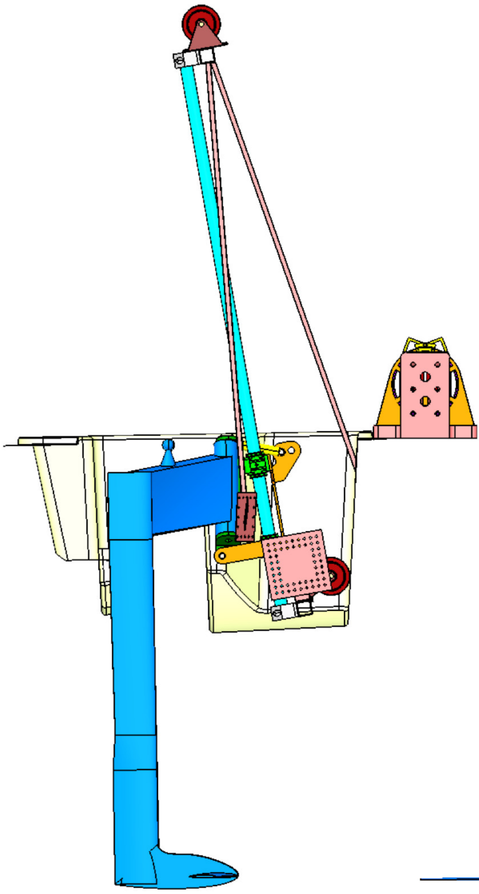
M

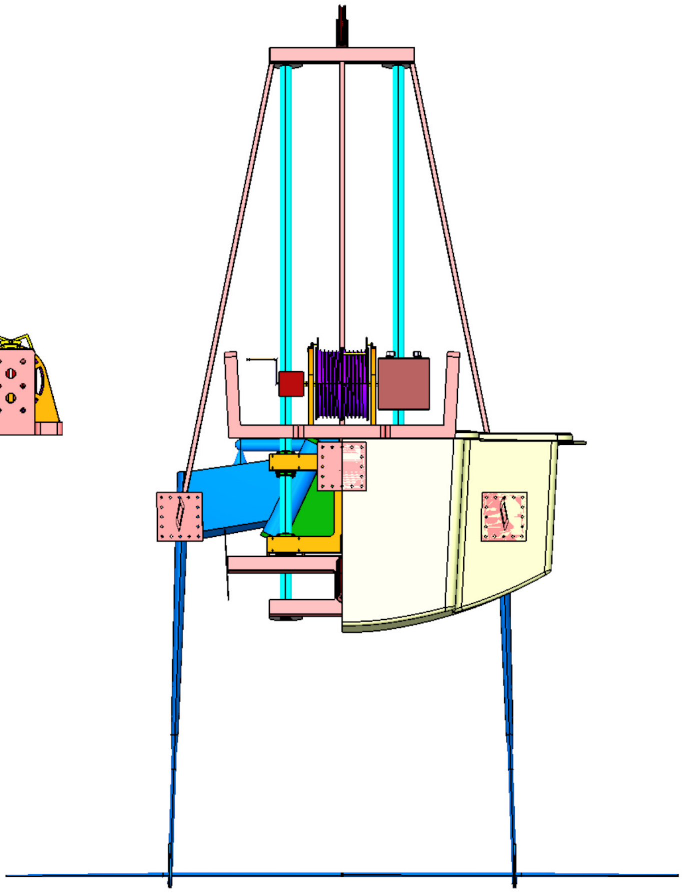
HDM overview - Additional images





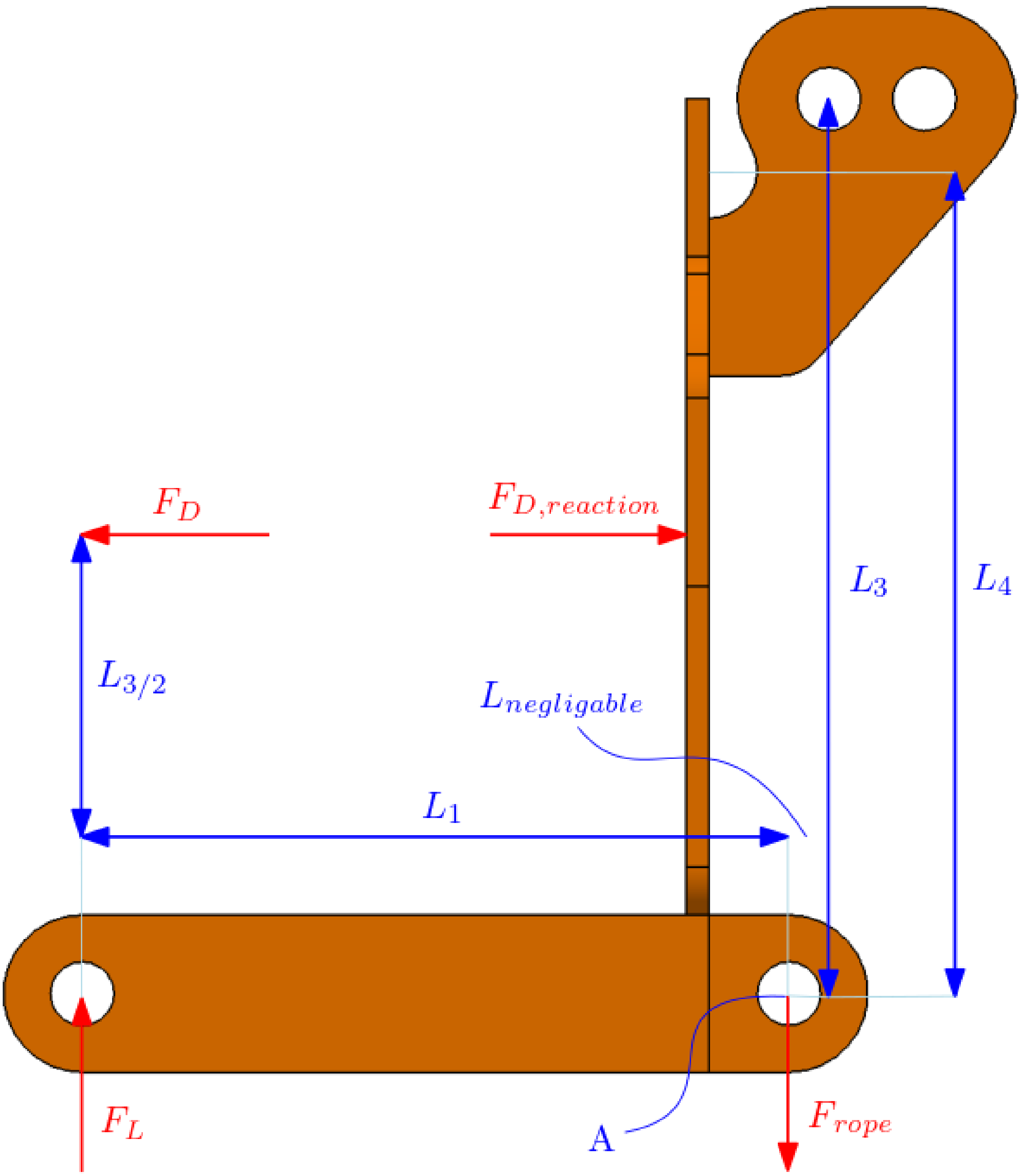






N

Wagon - Free body diagram



- F_{lift} (lifting-force from the hydrofoil) is assumed to exist at the lower flanges only since the trim-interface is hinged. A reaction force from the lower rope (locking the wagon into place) exists at the rope's lower attachment point.
- It is assumed that the F_{drag} force will be equally distributed between the lower and upper hub's interface, and a consequential reaction force equally distributed between all four linear bearings.

O

Rope specific strength calculations

Synthetic – SuperMax 78 (source) If ρ_s is specific gravity relative to water, then the density is:

$$\rho = \rho_s * \rho_{water} = 0.97 * 997 = 967.09 \ [kg/m^3]$$

The yield strength as a function of the maximum allowed force is:

$$\sigma_y = \frac{F_y}{A} = \frac{8829 * 10^3}{\pi 0.12^2/4} = 780.66 * 10^6$$

The specific strength is:

$$\sigma_y/\rho = 807300$$

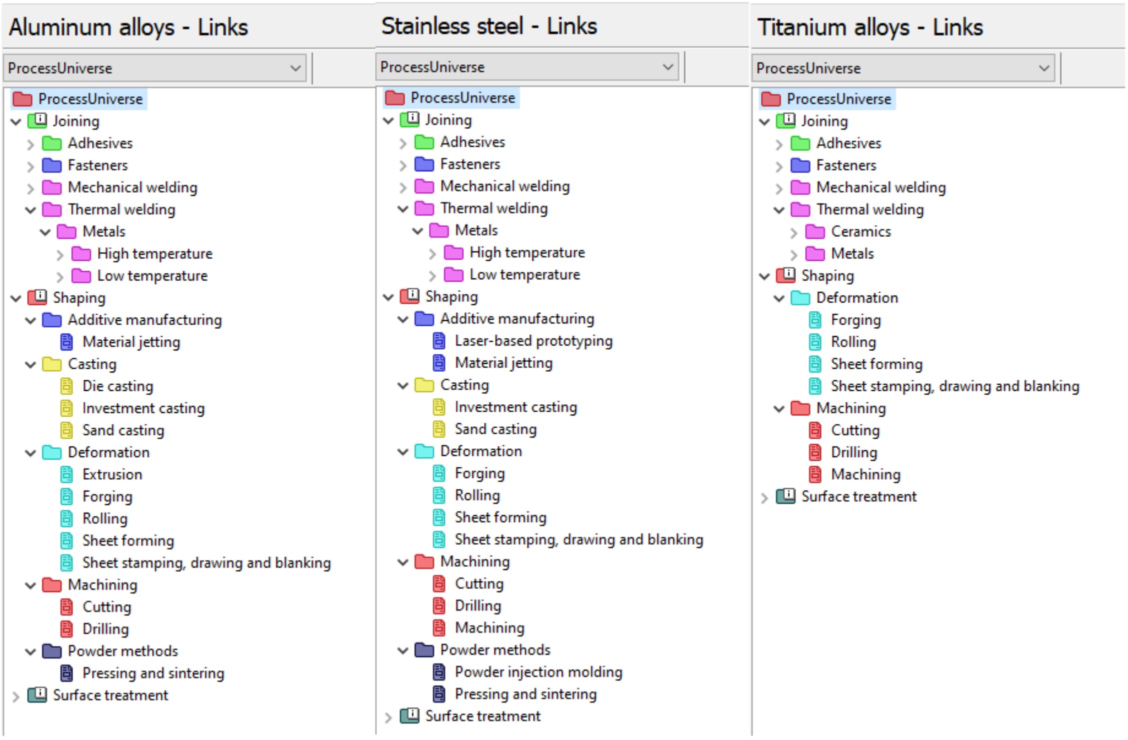
Stainless steel 18-8 (source)

$$\sigma_y/\rho = 205 * 10^6/7930 = 25851$$

P

Titanium manufacturability

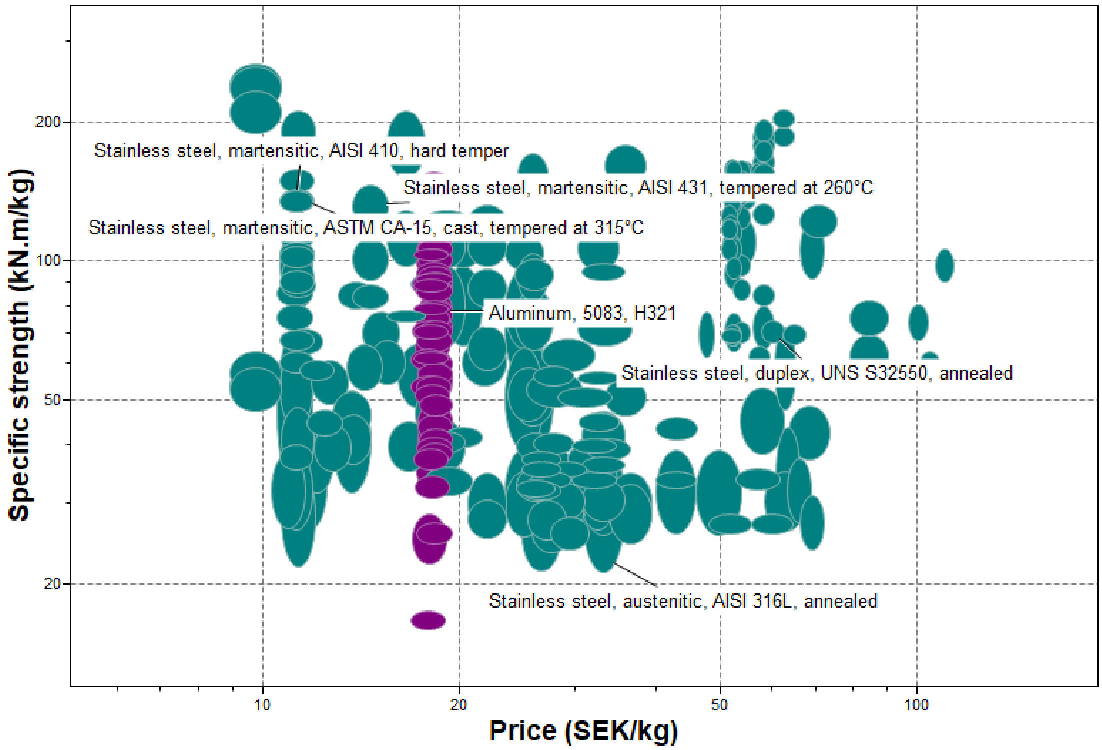
Available manufacturing methods suited for titanium is less than that of stainless steel and aluminum. (GRANTA Selector, Version 21.2.0; 2021)



Q

GRANTA Selector stainless steel chart

(GRANTA Selector Version 21.2.0; 2021)



R

Stainless steel 316 material properties

(GRANTA Selector, Version 21.2.0; 2021)

Stainless steels	**AISI 316**	**AISI 316L**	**AISI 316H**	**AISI 316Ti**
Price [kr / kg]	30.8 – 35.6	30.8 – 35.6	30.8 – 35.6	30.8 – 35.6
Density [kg / m³]	7870 – 8070	7870 – 8070	7980 – 8000	7910 – 8010
Young's modulus at 23°C [GPa]	195	195	193 – 200	195 – 205
Yield limit at 23°C [MPa]	259	226	205 – 260	220 – 270
Specific strength [kNm / kg]	25.7 – 38.9	21.3 – 38.9	37.3 – 47.1	27.6 – 33.9
Elongation [% strain]	30 – 50	30 – 50	29 – 40	30 – 50
Fatigue strength At 10^7 cycles [MPa]	228 – 252	256 – 299	200 – 220	267 – 317
Toughness [kJ / m²]	15.7 – 28	14.5 – 25.8	52.1 – 74.8	72.6 – 248
Thermal expansion Coefficient [µstrain / °C]	15 – 18	15 – 18	16	16 – 18
Galvanic potential [V]	(-0.18) - (-0.1)	(-0.18) - (-0.1)	(-0.16) - (-0.08)	(-0.18) - (-0.1)
Magnetic	No	No	No	No
Salt water durability	Excellent	Excellent	Excellent	Excellent

S

Weight estimation in different materials

HDM - Weight estimation

Early weight estimate - Numbers will shift by more than ±10%.
Theoretical weight calculated solely by multiplying with a specific strength factor conversion factor. Not controlled for implementability.
Steel -> Aluminum factor: 0.293; Steel -> Carbon fiber factor -> 0.054.
All weights in [kg]. [-] entails that component should not or cannot be manufactured in that material.

Material → Component ↓	**AISI 316L Stainless steel**	**5083-H321 Aluminum theoretical**	**Carbon fiber prepreg weave theoretical**	**Comment**
Brace – Upper	10.2	2.9886	0.5508	Needs rework if change to carbon fiber, becomes impossibly thin at 0.55 [kg].
Brace – Lower	12.8	3.7504	0.6912	Current design under-dimensioned.
Shaft 2x	3	0.879	-	Exact weight based on OEM specifications.
Shaft block 4x	4.4	1.2892	-	Current design under-dimensioned.
Wagon	14.4	4.2192	0.7776	Current design both under- and over-dimensioned.
Bearing unit 4x	4.2	1.2306	-	Exact weight based on OEM specifications.
Hub	-	-	16.905	Rough estimation.

Not material bound		
Trim	4.8	Ewellix CAHB-22-x4E.
Winch	12.5	Harken FlatWinder FW250EA24H.
Winch pulley system	2	Harken 4:1 32 mm Traveler kit 3190.
Pulley block 2x	6.4	Harken 200mm aluminum teardrop block C9225.

Total steel	**126.805**
Total aluminum	**65.4004**
Total carbon fiber + steel	**85.0246**
Total carbon fiber + aluminum	**56.4618**

T

Requirement specification evaluated

System-level requirement specification

No.	Requirement	D/W 1-5	Unit	Value marginal	Value optimal	Evaluation	Reference	Evaluation notes
Green: Optimal probably fulfilled | Lime: Marginal probably fulfilled | Yellow: Borderline | Red: Probably unfulfilled Grey: Unevaluated due to no / minor work done in area.								
R	Global requirements							
R1	Mass	D	kg	<100	<50	OEM specifications + CAD measurements	SSRS	Weight estimation shows combination of metal + CFRP brings weight clearly <100 [kg]. However, initial OEM contact shows required drivetrain might have significant weight.
R2	Initial purchasing cost	W5	SEK	<100 000		OEM invoice	SSRS	Previous OEM contact indicates all purchased components are <100 000 [SEK]. However, manufacturing costs (welding, etc) are unknown.
R3	Yearly maintenance cost	D	SEK	<10 000		OEM statements + cost estimations	SSRS	
R4	Chunk width	D	m	<0.9		CAD measurements	SSPA	Design fits inside geometry.
R5	Chunk length	D	m	<0.6		CAD measurements	SSPA	Design fits inside geometry.
R6	Chunk height	D	m	<1.5		CAD measurements	SSPA	Design fits inside geometry.
R7	Time between minor maintenance	D	year	<1		OEM statements	SSRS	
R8	Time between major maintenance	D	year	<12.5		OEM statements + lifespan calculations	SSRS	
R9	Minor maintenance by laypeople possible	W5	Y/N	Y		Subjective evaluation	SSRS	Wear-and-tear components expected to be linear bearings and rope. Probably replaceable by laypeople.
R10	Major renovation by professionals possible	D	Y/N	Y		Subjective evaluation	SSRS	No components are permanently joined. High maintainability.
R11	Water-proof	D	IP	IP54	IP67	OEM specifications + Subjective evaluation	SSRS	Highly likely to be implementable.
R12	Marine environment	D	Y/N	Y		OEM statements	SSRS	Galvanic corrosion between metal and CFRP, and any sacrifical anodes, not sufficiently investigated.
R13	Voltage	D	V	1 ↔ 800	12 ; 24 ; 48	OEM specifications	Micropower	Motors in specified voltage and loads common.
R14	Operating temperature	D	°C	-10 ↔ 70		OEM specifications	SSRS	Exact components not yet decided, cannot evaluate.
W	Winch							
W1	Brake downwards line (automated)	D	N	>32 000	>48 000	OEM specifications	Mantaray	Loads handable by commercially available components.
W2	Pull upwards line (automated)	D	m/s	>0.06	>0.09	OEM specifications	SSRS	Loads and speed might bring wattage requirement over Micropower specifications (out of scope of thesis project).
W3	Pull downwards line (automated)	D	N	>32 000	>48 000	OEM specifications	Mantaray	Loads handable by commercially available components.
W4	Pull downwards line (automated)	D	m/s	>0.06	>0.09	OEM specifications	SSRS	Loads and speed might bring wattage requirement over Micropower specifications (out of scope of thesis project).
W5	Manual winch force input	D	N	<150	<50	Lever + frictional losses calculations	Student	Long enough lever and gearbox makes requirement trivial to fulfill.
W6	Pull upwards line (manual)	D	N	>4 000	>6 000	OEM specifications	Student	Loads handable by commercially available components.
W7	Pull upwards line (manual)	D	m/s	>0.01	>0.015	Assumptions	Student	Uncertain if gear ratio and handle-lever will allow for a low enough manual exertion at high enough speeds.
W8	Brake upwards line (manual)	D	N	>4 000	>6 000	OEM specifications	Student	Loads handable by commercially available components.
W9	Pull downwards line (manual)	D	m/s	>0.01	>0.015	Assumptions	Student	Uncertain if gear ratio and handle-lever will allow for a low enough manual exertion at high enough speeds.
W10	Lock foil (retracted)	D	N	>4 000	>6 000	OEM specifications	Student	Trivial to implement even though exact solution not yet decided.
W11	Lock foil (deployed)	D	N	>52 000	>78 000	OEM specifications	SSPA	Enough space exists to make it highly reasonable to assume requirement fulfillable.
W12	Lock foil in user-chosen depth	W1	Y/N	Y		Assumptions	SSRS	Presumably not fulfilled as winch-locking will presumably not be implemented.
W13	Tension line	D	Y/N	Y		Design	Student	Commercially available rope tensioners tried and tested technology.

W14	Emergency foil retraction	W4	m/s	>0.6	>3.6	Calculations	Student	Retraction speed depends on unlocking speed of locking solution.
W15	Positional awareness	D	Y/N	Y		Sigma Embedded evaluates	Sigma Embedded Engineering	Trivial to implement mechatronically.
W16	Measured volumed near crewmates	W4	dB	<95	60 ↔ 40	OEM specifications	SSRS	
W17	Rope operating strength	D	N	>52 000	>78 000	OEM specifications	Student	Synthetic rope capable of handling significantly higher loads.
W18	Drum ↔ Rope diameter ratio	D	-	>20:1		CAD measurements	Dynamica ropes	Designed to be fulfilled.
W19	Pulley ↔ Rope diameter ratio	D	-	>10:1		CAD measurements	Dynamica ropes	Uncertain if pulleys fit inside compartment with required size.
W20	Groove ↔ Rope diameter ratio	D	-	>1.1:1		CAD measurements	Dynamica ropes	Designed to be fulfilled.
W21	Number of output ropes	D	-	2		Design	Student	Designed to be fulfilled.
W22	Number of output directions	D	-	2		Design	Student	Designed to be fulfilled.
W23	Structural strength	D	N	>52 000	>78 000	FEA simulations	Student	Simulations show design promising.
T	**Trim**							
T1	Allow trim-unit movement	D	Y/N	Y		Design	SSRS	Designed to be fulfilled.
T2	Provide trim movement	D	N	>8 760	>13 140	Calculations + OEM specifications	Student	Commercially available linear actuators capable of fulfilling requirements are plentiful.
T3	Provide trim movement	D	°/s	<0.5	<1.5	OEM specifications	Mantaray	Commercially available linear actuators capable of fulfilling requirements are plentiful.
T4	Lock trim	D	N	>8 760	>13 140	Calculations + OEM specifications	Student	Commercially available linear actuators capable of fulfilling requirements are plentiful.
Wa	**Wagon**							
Wa1	Withstand radial force	R	N	>4 000	>6 000	Calculations + OEM specifications	Student	Should radial force exceed specifications, a larger model of linear bearing can be chosen.
Wa2	Hold trim unit	R	N	>8 760	>13 140	Calculations + OEM specifications	Student	Simulations show design promising.
Wa3	Hold upwards line	R	N	>4 000	>6 000	FEA simulations	Student	Trivial magnitudes to fulfill.
Wa4	Hold downwards line	R	N	>52 000	>78 000	FEA simulations	Student	Interface relatively easy to strengthen by re-design to fulfill requirement.
Wa5	Hold hub	R	N	>52 000	>78 000	FEA simulations	Student	Uncertain if hub's flanges can avoid too high stress concentrations by merely increasing thickness / width.
H	**Hub**							
H1	Hold foil	R	Pa	Yield limit * 2 FoS	Yield limit * 3 FoS	FEA simulations	Student	Yield limit of the hub's material.
H2	Allow foil axle rotation	R	Y/N	Y		Design	Mantaray	Designed to be fulfilled.
H3	Hold trim unit	R	N	>8 760	>13 140	Calculations + OEM specifications	Student	
H4	Hold wagon	R	N	>52 000	>78 000	FEA simulations	Student	
H5	Allow hub rotation	R	N	>26 000	>39 000	Calculations + OEM specifications	Student	
H6	Hydrofoil change possible	W3	Y/N	Y		Assessment	SSRS	
F	**Frame**							
F1	Counter drag-torque	R	Nm	>3 600	>5 400	FEA simulations	Student	Simulations show design promising.
F2	Counter twisting-torque	R	Nm	>2 000	>3 000	FEA simulations	Student	Simulations show design promising.
F3	Transfer axial upwards forces	R	N	>52 000	>78 000	FEA simulations	Student	Uncertain if Bracing – Lower macro-geometry is capable of handling loads without stress concentrations exceeding yield limit of material.
F4	Counter leaning	R	N	>6 000	>9 000	FEA simulations	Student	Simulations show design promising.
F5	Foil travel angle	R	°	12		CAD measurements	SSPA	Designed to be fulfilled.

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