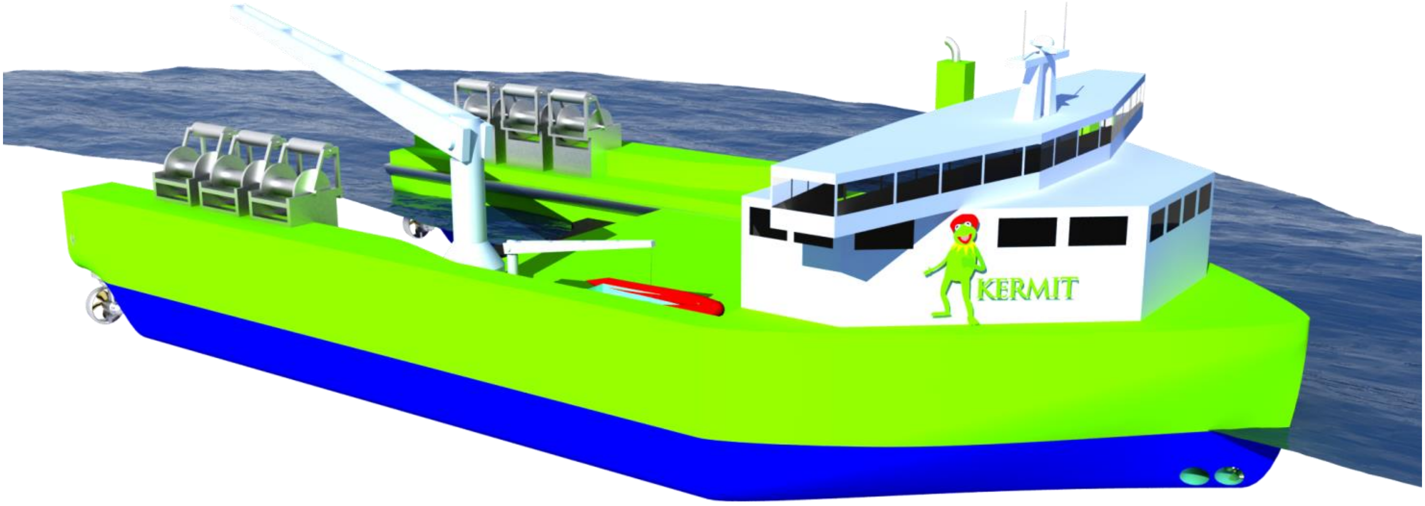

Kermit
Wind farm installation vessel
Designed for Blekinge Offshore

Marine Design Project 2014

Course within the International Master program of Naval Architecture and Ocean Engineering (15 HEC)
Department of Marine Technology
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2014

KERMIT

Special Purpose Ship designed for Blekinge Offshore

Marine Design Project 2014

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Abstract

In order to meet growing energy demands, the world can no longer rely solely on conventional energy sources of the industrialization era. The renewable energy industry is growing fast to compensate for the energy deficit, while keeping the negative impact of this growth on the environment to a minimum.

The 2014 Marine Design Project client Blekinge Offshore AB has proposed a 700 windmill farm in the Swedish waters of the Bay of Hanö. As part of this national-scaled project, the design and production of a vessel capable of transferring and installing the gravity foundations as well as the windmills themselves needs to be designed.

This report presents a special purpose vessel which uniquely combines principles of heavy cargo lifting by ballast water management, winches and the ability to ground The Vessel during loading of the cargo. The 7983 DWT vessel will shuttle between the coastal assembly site and the wind farm site at a speed of 10 knots and facilitate the lifting, transporting and mounting of the foundations as well as the already assembled windmill towers. The Vessel itself is designed in such a way to have the smallest possible impact on the environment, as The Baltic Sea is among the most tightly controlled regions in the world.

The vessel is designed with special human factor considerations in order to cater for the safety and comfort of the crew and the installation technicians, by means of sea keeping and accommodation considerations. Despite the vessels size and the highly advanced operations it will be performing, it only requires a crew size of 8 people.

Also equipped with state of the art Dynamic Positioning technology, the vessel insures accuracy of operations while maintaining the shortest possible time schedule, thus minimizing operational costs.

Preface and Acknowledgement

The Marine Design Project, MMA 150, is a mandatory 15 HEC course within the master program Naval Architecture and Ocean Engineering at Chalmers University of Technology. The course is organized by the department of Shipping and Marine Technology at Chalmers.

The objective is to develop a conceptual design of a vessel.

The project members would like thank our supervisors Senior Lecturer Per Hogström, Prof. Em Anders Ulfvarson and Prof. of the practice Bengt Ramne. We would also like to thank the client, Blekinge Offshore and their representatives Anders Nilsson, Hans-Olof Svensson and Johan Edvardsson, Naval Architect, from Offshore Väst specialist representing the client.

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Executive summary

The following section is a brief outline of the work accomplished in the project. The mission profile, mains particulars for the vessel and a general overview of the vessel are presented in this section.

Mission Profile

The vessel is designed to install an offshore wind farm outside the coast of Karlshamn, Blekinge in a more cost efficient way compared to existing solutions. The vessel shall be able to transport and install windmill foundations as well as fully assembled windmill towers with generators and blades. It must also incorporate an ability to be flexible with regard to cargo size, due to varying dimensions of both foundations and windmills.

Blekinge Offshore plan to install 700 complete windmill units during 8 years.

Main particulars for vessel

Type	Special Purpose Vessel	
Flag	Swedish	
Class	DNV GL	
Ice class	Swedish – Finnish 1C	
Dimensions	Length over all, LOA	80 m
	Rule length, Lpp	77.5 m
	Draught	4 m
	Moulded depth, D	10 m
	Beam, B	38 m
	Frame spacing	100 mm
	Lightship weight	2499 mt
	Deadweight	7983 mt
Gensets	2xWärtsilä 9L20DF + 1xWärtsilä 9L34DF	
Propulsors	4xWärtsilä – WST-16 + 2xWärtsilä - CT125H	
Accommodation	8 Single cabins	

General overview

The vessel is arranged with seven decks as can be seen in Figure 1. Top to bottom the decks are: Monkey Island, Bridge deck, Accommodation deck, Upper deck, Crane deck, Main deck and E.R. deck. The vessel has a large length to breadth ratio in order to provide sufficient stability when carrying cargo with a high vertical point of gravity, like fully assembled wind turbines in a vertical position. The propulsion system is diesel electric with one main engine and two smaller auxiliary engines. The vessel is fitted with a total of four azimuth thrusters placed in pairs in each of the pontoons, complimented with two bow thrusters to improve the maneuverability

On the main deck there’s space to store the cargo. To allow for controlled movement of the cargo a skidding system is placed on the main deck. To lift fixtures and load and unload cargo a crane is fitted on the port side of the crane deck. The lowering of the foundations is done by winches placed on the sides of the cutout, three on each side. The winches are supported by a structure stretching from the craned deck to the upper deck distributing the loads acting on the hull.

In the superstructure at upper deck level the fire station, emergency generator, lockers laundry, HVAC and gymnasium is placed. On the accommodation deck level the crew cabins, offices, first aid room, galley, mess and conference room is placed.

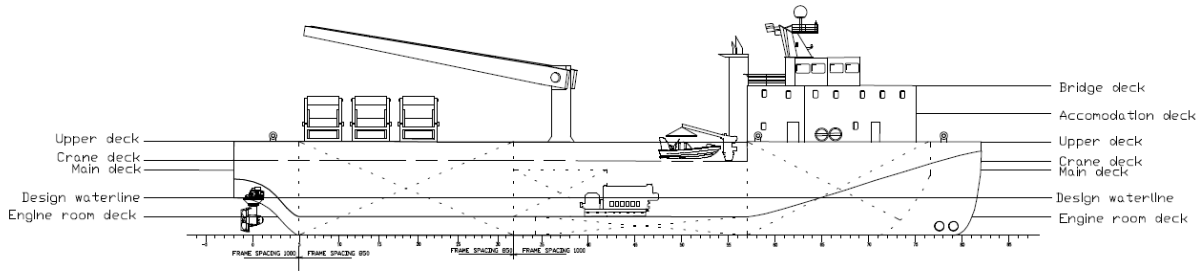

Figure 1. Profile view

The bridge is made wide with control stations in both forward, aft, port and starboard direction. This will provide possibilities to overview and control the vessel in all phases of the operation.

To provide the possibility to ballast the vessel in accordance to the different operational modes the majority of the hull structure consists of ballast tanks. There are two tanks in each of the pontoons. Two are placed on the sides of the aft part of the engine room and five placed in the bow area.

The engine room is located on the engine room deck, below the cargo on the main deck. The fuel tanks are place on the sides of the engine room, with a methanol tank on the starboard side and a MDO tank on the port side. The ballast pumps are located in the port side of the engine room.

List of Abbreviations

AHU – Air Handling Unit

AP – Aft Perpendicular

B – Beam

B_m – Moulded Beam

C_B – Block Coefficient

CO_2 – Carbon Dioxide

D – Depth

D_m – Moulded Depth

DNV – Det Norske Veritas

DP – Dynamic Positioning

DW– Dead Weight

ECA – Emission Control Areas

Fn – Froude Number

FP – Forward Perpendicular

GM – Metacentric Height

HVAC – Heat Ventilation and Air Conditioning

IACS – International Association of Classification Societies

IMO – International Maritime Organization

LCF – Longitudinal Center Floatation

LCG – Longitudinal Center of Gravity

LNG – Liquefied Natural Gas

L_{OA} – Length over all

L_{PP} – Length between perpendiculars

MARPOL – Marine Pollution Act

MGO – Marine Gas Oil

NO_X – Nitrous Oxides

SCR – Selective Catalytic Reduction

SECA – Sulphur Emission Control Area

SOLAS – Safety Of Life At Sea

SO_X – Sulphur Oxides

SPV – Special Purpose Vessel

T – Draught

TCG – Transverse Center of Gravity

VCG – Vertical Center of Gravity

Table of Contents

1 Introduction

The world energy consumption steadily increases every year, which in turn leads to growing implications for the environment. In order to cease this trend, more and more investments supporting renewable energy sources are made. Off the coast of Sweden a large offshore wind farm is planning to be built in the bay of Hanö. Most of the existing ships for installing offshore wind power are however designed for harsher conditions, such as the conditions in the North Sea. Thus, there is a need for offshore wind power installation vessels designed and cost optimized for more benign waters.

1.1 Background

In order to decrease the use of fossil fuels consumed, the European Union decided that the share of EU energy consumption produced from renewable resources should be 20% by 2020 (European Commission, 2014). Sweden already today produces a lot of energy from renewable resources, mostly hydropower from the northern parts of the country, and therefor sets a national goal of having 49% of the energy produced from renewable resources (Swedish Government, 2014). To achieve this goal, significant investments in areas such as offshore wind power.

Offshore wind power is more reliable in terms of energy production compared to the land based kind. However, it is significantly more expensive to install and maintenance offshore wind power. Today many offshore wind farms exist in the North Sea, where Great Britain, Germany and the Netherlands all have installed offshore wind farms. The conditions in the North Sea is however much harsher compared to the benign Baltic Sea, thus are vessels constructed for installing wind mills in the North Sea sturdier and more expensive.

Blekinge Offshore is planning an offshore wind farm in the bay of Hanö and have requested the assistance of students at the master programme Naval Architecture and Ocean Engineering at Chalmers University of Technology to make a conceptual design of a vessel more suitable for installing wind mills in the Baltic Sea.

1.2 Objective

The objective of the project is to develop a conceptual vessel design, including relevant technical documents, outlined in a specification defining an offshore installation vessel for Blekinge Offshore.

The main focus of the project is to design an installation vessel capable of installing 80 complete units per year for at least 8 years, for as low a cost as possible.

1.3 Methodology

The work within the team has been divided among four groups of different engineering disciples in order to run the project efficiently. The four subgroups are General Arrangement, Structural Design, Hydromechanics and Machineries.

The General Arrangement group is responsible for logistics of operations, management, layout and general arrangement including layout of the living quarters.

The Hydromechanics group make the hull lines, hydrostatic and stability calculations for all relevant loading conditions, as well as resistance calculations and sea-keeping properties.

The Structural Design group are responsible for the structural arrangement of the vessel and should ensure that the ship complies with issued rules and regulations.

The Machineries group responsibilities are the power balance, propulsion and engine setup, auxiliary systems and layout of the engine room.

All of the mentioned groups contributes with data from their discipline in order to assess a light ship weight estimation. The work has been carried out according to the Design Spiral, see Figure 1.1 below. During the course of the project two evolutionary loops have been executed.

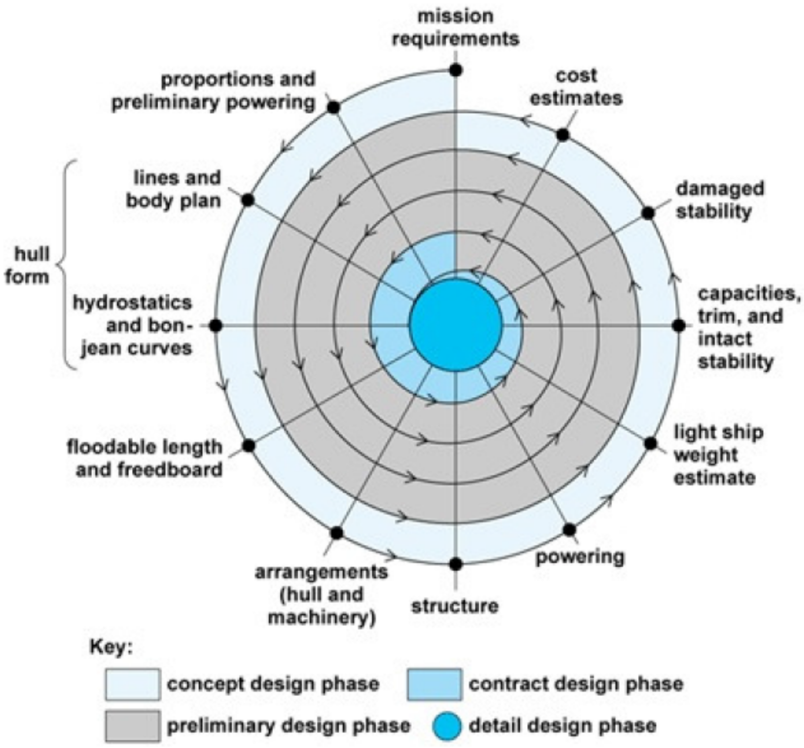

Figure 1.1. Design Spiral

1.4 Limitations and assumptions

Due to a limited time schedule only two laps in the Design Spiral was made possible. The time shortage also restricts the amount of optimization to the proposed design. This consequently leaves room for further adjustments and future development of the design.

Another limitation is the lack of engineering experience of the project team. The scarce number of reference vessels complicated the establishment of the initial concept, hence some assumptions had to be made in order to start the design loop.

2 Design Basis

The following chapter describes the outline of the project and contains information regarding the ships mission, the stakeholders of the project, specifications and requirements of the project and the class rules used.

2.1 The mission

A windmill farm is going to be built outside the region Blekinge, close to the island Hanö. To do this, the vessel is designed to be able to transport gravity foundations in varied sizes for different water depths. It should also be designed to transport and install windmills on top of the gravity foundations, see Fig. 2.1 below. The vessel shall be as cost efficient as possible, making it the obvious choice for this type of operations in relatively calm waters in the Baltic Sea. Solutions for tougher weather conditions already exist on the market, but they are expensive and more advanced than necessary for use in the Baltic Sea.

The design of the vessel should be relatively small and easy to manoeuvre in an offshore wind farm. The operational time is very important factor in the design, since many windmills are going to be installed in a relatively short period of time. The vessel is therefore designed with focus on this task. Maintenance of the wind farm is to be carried out by another vessel, why this is not handled in this project. Sustainability is very important for a wind farm, since it is considered “green energy”. The environmental aspects are taken in account when designing the ship, specifically when designing the propulsion system. The goal is to install 700 windmills with complementing foundations during a timeframe of 8 years.

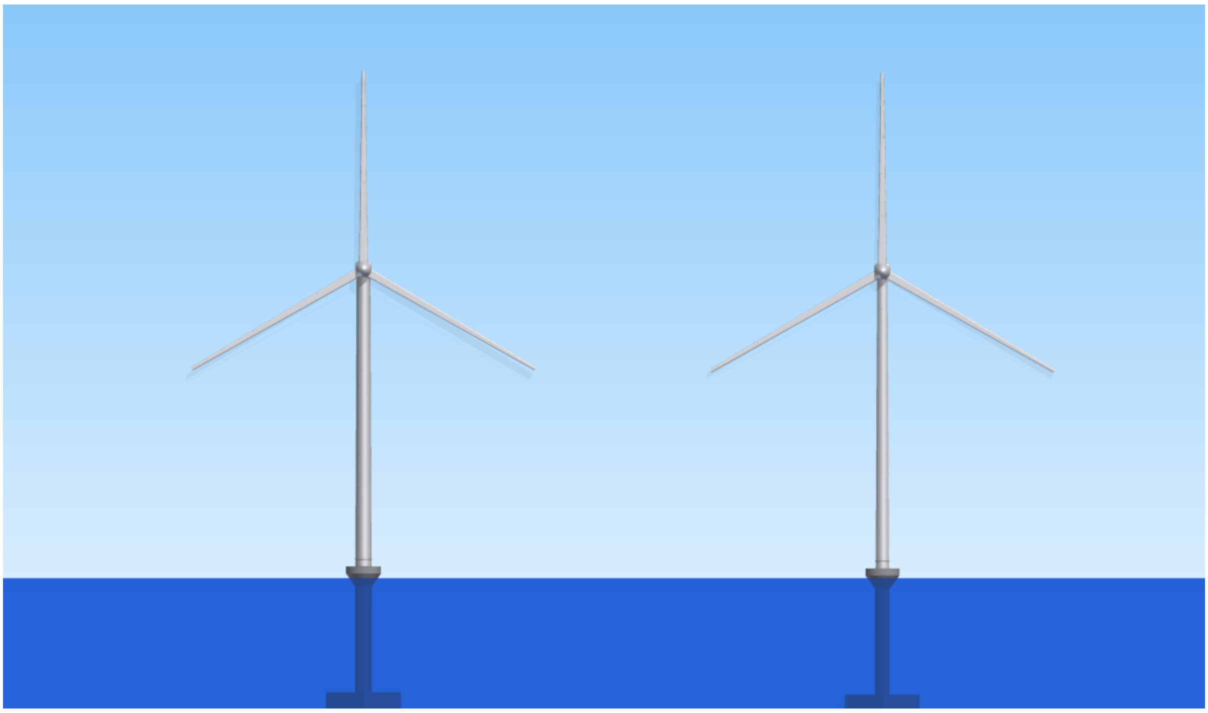

Figure 2.1. Windmill mounted on gravity foundation

2.2 Stakeholders

The following table shows identified stakeholders and their influences on the requirements on the design of the vessel:

Table 2.1 Stakeholder map

Stakeholder	Influences on requirements
Blekinge Offshore	Costs, performance, dimensions, installation time, Health, safety and Environment (HSE) aspects
IMO	Rules and regulations regarding emissions and safety at sea (MARPOL & SOLAS)
Classification society DNV GL	Rules of the design and construction
Flag state authorities, Transport Agency of Sweden	National rules and regulations
Port state control	Local rules and regulations in ports
Insurance companies	Class certificate to be updated and approved
Shipyard	Ship design possible to construct, low complexity
Crew/union	General safety and on board working environment
The public/environment	Environmental aspects
Stilleryd Port	Water depth limitation
Windmill manufacturer	Impact force, accelerations
Foundation manufacturer	Pushing force

2.3 Specification of requirements

The port Stilleryd (56°9'25.4"N 14°49'46.1"E), in which the ship will have its main operational time, has a limitation of a maximum draft of 10m. The windmills are quite sensitive to impact forces and a smooth installation process is therefore necessary. The foundations can withstand a maximum force of 3MN (COWI, 2014).

The wind energy sector wants to be as environmental friendly as possible. The energy it delivers as well as the equipment and processes related to it have to be environmentally sustainable. This makes the demands and requirements for an installation vessel like this high. Since the Baltic Sea is an environmentally controlled area (ECA) the ship must fulfil all regulations regarding NOx and SOx emissions etc. stated in MARPOL (IMO, 2005).

People are going to have this vessel as their job site for long periods of time. The working environment and conditions for the crew must therefore be the best possible. This means for example low noise, high safety (SOLAS regulations) and pleasant accommodation facilities.

2.4 Class

The vessel is designed to be classed by DNV GL as a S*pecial Purpose Vessel*. DNV GL was selected early in the project since they are well reputed in the offshore area and attractive to the client. Additionally are the class rules easy to apply and accessible online. In the rules, Part 7 chapter 5, offshore service vessels, tugs and special ships and many different kinds of ships are found, though the special purpose vessel is the only vessel type applicable for the project with the specific pre-defined requirements (DNV, 2010).

Regarding Ice Class, the vessel is classed as 1C according to the Finnish-Swedish Ice Class Rules (Sjöfartsverket, 2012). Ice maps were studied during the selection to see what the ice conditions are and have been in the area. It is assumed that the vessel will not operate if there is ice thicker than 0.15 meters which makes Ice Class II sufficient, but the vessel could be in transit to a new site during periods with thicker ice and therefore 1C is selected.

3 Logistics and operation

The wind power plant installation vessel has four main tasks, to load and transport foundations and windmills to site, to install foundations and to position windmills on the foundations. The purpose of this vessel is to install the wind power plant to a lower cost than already existing vessels can do. In order to keep costs low, cranes is not to be used to lift foundations or windmills. Instead the whole vessel will be lifted and lowered through ballasting and deballasting of water. For lowering of the foundation down to the seabed, winches are installed; since the ballasting system does not submerge the vessel enough for this operation see fig. 3.1 below.

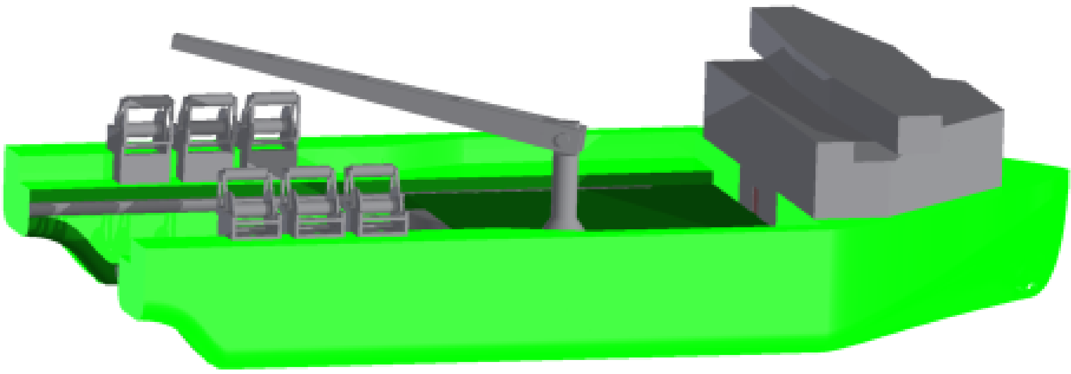

Figure 3.1. The Vessel with winches, crane and superstructure.

For transport on board the vessel and for attaching the winches to the foundation, fixtures have been designed, see section 4.4.1. In port the fixtures will be attached to the foundation or windmill. On board the ship there will be rails on which the fixtures will slide. When it is time to lower the foundation onto the seabed the winches will be attached to the fixture. When the foundation is at place the fixture is detached from the foundation and winched back up to the vessel.

The port is designed to fit the vessel, which will moor with starboard side to the quay and with the stern inwards as Fig. 3.2 indicates. There will be an extra wharf built which the stern of the vessel will moor against. The wharf can also be seen in Fig. 3.2. This means that when the ship arrives in port it will turn around and reverse against the wharf. When the vessel is in the right position it is grounded by ballasting and then the loading procedure starts. The seabed will be arranged in such a manner so that the vessel will not be damaged by the grounding procedure.

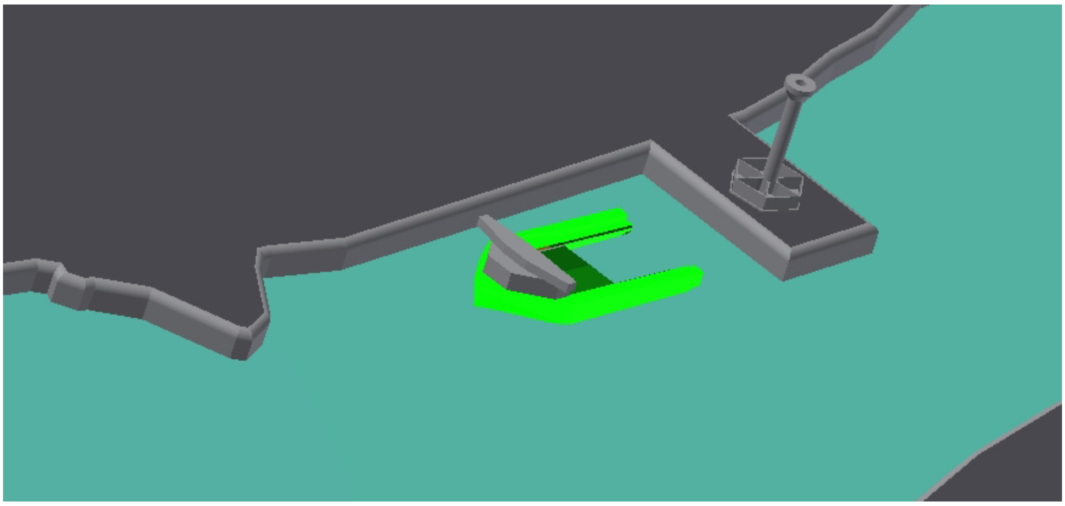

Figure 3.2. The port of Karlshamn with the specially designed wharf.

When installing the windmill the installation crew will be transported to site by small, high speed vessels. The separate transport of the installation crew is due to the low speed of the installation vessel and the fact that the ship only needs to hold the windmill at its position for approximately one hour after the installation crew has started the mounting procedure. At this stage it is assumed that the amount of fastened nuts is enough to keep the windmill in position. When the installation vessel leaves site the mounting of the windmill will continue. The installation crew will be transported back to shore by the same type of vessel that brought them out to the site.

The location of the wind farm site is a relatively protected area of the Baltic Sea, see map in Fig. 3.3 below. The area is also spared from ice most winters. The estimated number of days with suitable weather conditions is estimated to be 6 day/week during April-September and 1day/week October-March. This means that there will be 182 working days/year. Assumed that half of those days the vessels carries two units and the rest it is carrying one unit, the result is 273 units or 136 ready to use windmills per year. If the installation has this pace and the goal is 700 complete windmills the installation will take just over 5 years.

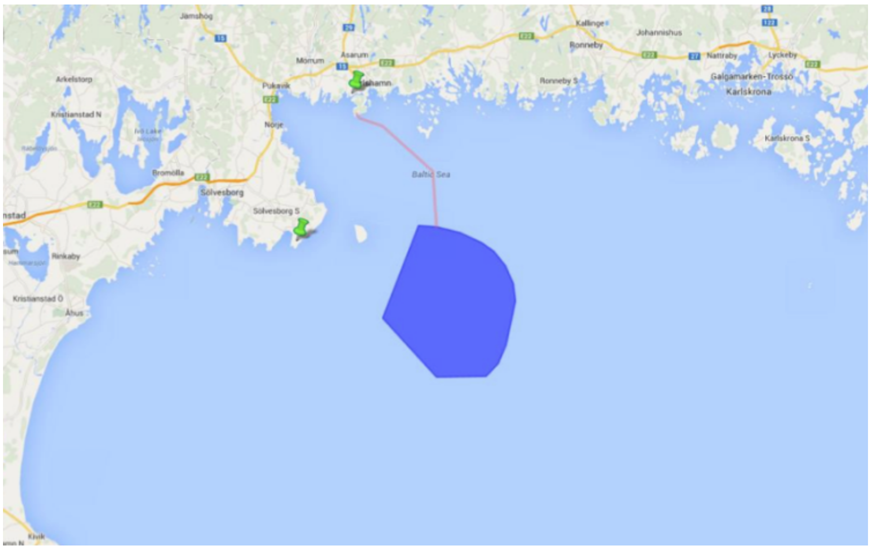

Figure 3.3. Map over the site where the foundations and windmills will be installed.

3.1 Operational description

There are four different operations in the operational scope. The vessel can carry one foundation or one windmill separately; but is also designed to also carry two of each. When it comes to carrying two foundations the procedure is limited to combinations of foundations that weigh maximum 3600 metric tons together. The time to complete the operations has been estimated as shown in Table 3.1 below. A complete time schedule of the different operations is presented in Section 3.1.5.

Table 3.1 Summary of the time required for the different operational alternatives with the average and the longest distance.

Operation	Time (shortest distance)	Time (average distance)	Time (longest distance)
One Foundation	11,8 hrs	13,2 hrs	14,1 hrs
One windmill	11,3 hrs	12,7 hrs	14,1 hrs
Two foundations	15,2 hrs	16,6 hrs	18,8 hrs
Two windmills	16,4 hrs	17,9 hrs	19,3 hrs

3.1.1 Loading in port

The windmills and foundations will be attached to fixtures and placed in position for loading when the vessel arrives to port. There will be several fixtures available, so that the loading can start immediately when the vessel arrives in port instead of having to wait for offloading of the fixture and installing it to the next unit. To enable safe loading of the heavy units the vessel will be grounded in the special designed port before the loading starts. This is to make the rails on the quay and the rails on the vessel to be on the same height and to minimise the stability effects that the heavy units do to the vessel. A principal rendering of this can be seen in Fig. 3.4 below. When the vessel is ready, the fixture together with a unit will slide onto the rail in the cut-out. When the fixture has been placed on the rail a skidding system (see Section 4.4.2) will pull the fixture to the mid part of the vessel. If two units are transported at the same time the second one will be stored above the cut-out during transit. The fixtures positions will be locked in place during transit.

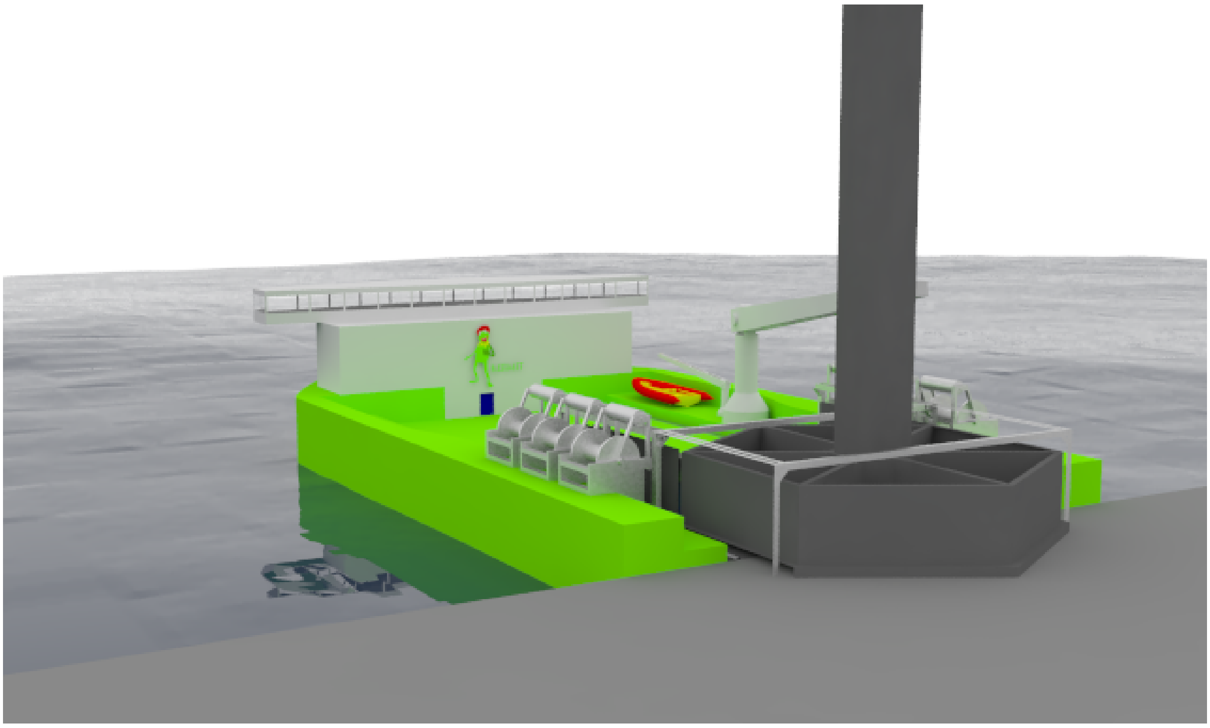

Figure 3.4. Loading of foundation in port.

For the procedure with two foundations, two of the medium sized (1700 metric tons) is the maximum to be carried at the same time. The operation of carrying one windmill and one foundation has not been evaluated in this report, since the windmills will be delivered in batches of 15 pieces at the time. The best operation is therefor considered to get these out as quickly as possible, carrying two items at a time. After the vessel has been loaded it will be re-floated from its grounded position by de-ballasting of the water ballast.

3.1.2 Transit and positioning

During loaded transit the ship will sail with a speed of 10 knots and during the ballast transit it will also have a speed of 10 knots. Due to the time it takes to travel to site it can be favourably to carry two units.

When a foundation is to be placed at site the position is maintained by the DP-system. For installation of the windmill an arced support is placed in the cut-out. The support, further described in Section 4.5.3, fits around the top of the foundation and enables the ship to press against the foundation with up to 3MN, see Fig. 3.5.

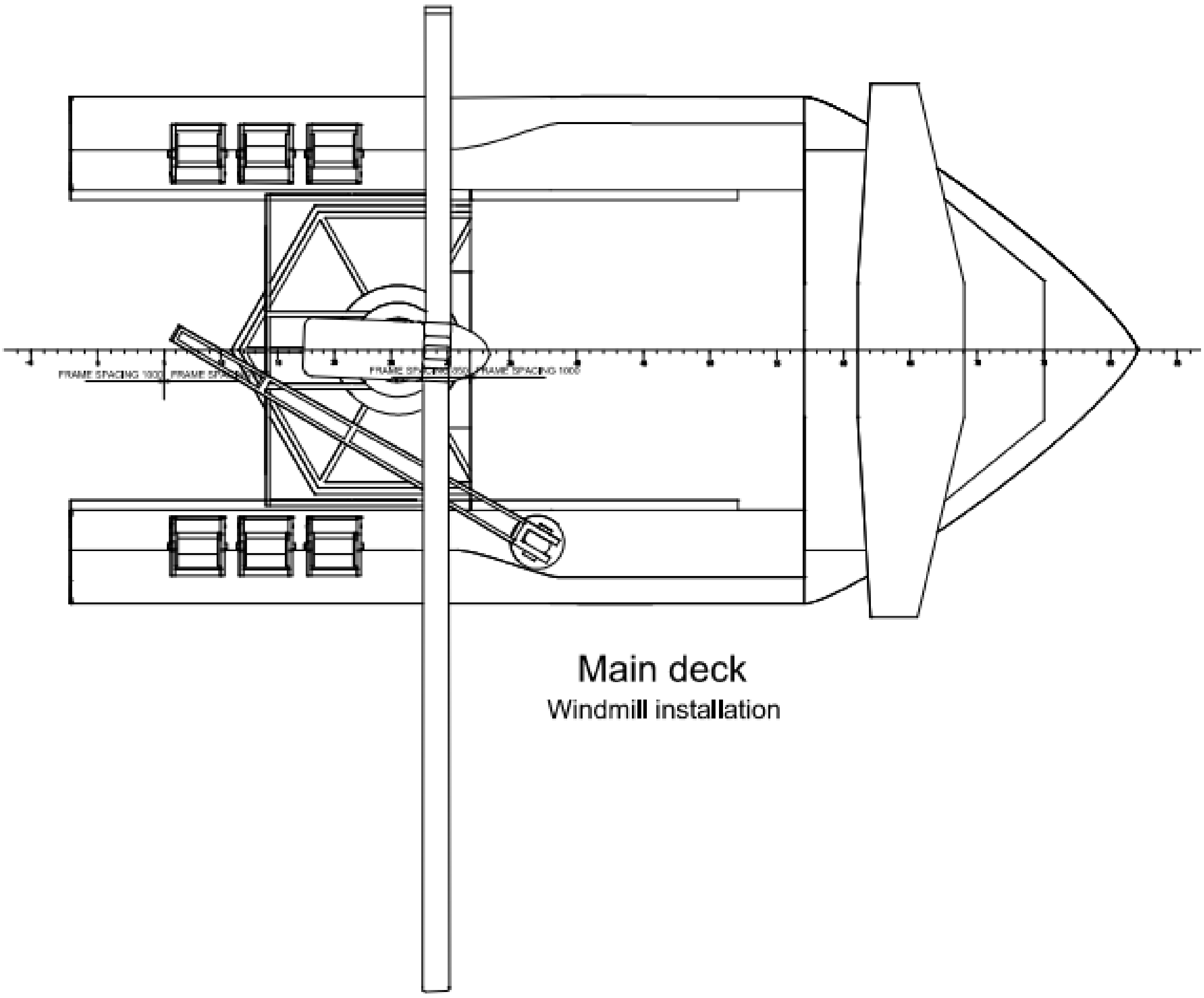

Figure 3.5. Principal drawing of the installation of a windmill on a foundation.

3.1.3 Off-loading and installing

The unit, which can be either a foundation or an assembled windmill, will slide to the area over the cut-out before off-loading or installation, if not the unit has been stored there during transit. If the unit is a foundation, the winches will be attached to the fixture and the rails in the cut-out will fold inside the hull and the winches will start to lower the foundation. When the foundation reaches the seabed, (see Fig. 3.6 below), the attachment between the fixture and the foundation will be released via a remote link from the manoeuvring station on deck. When the fixture is free from the foundation it will be winched up to the main deck and the rails will fold out so that the fixture can be placed on them.

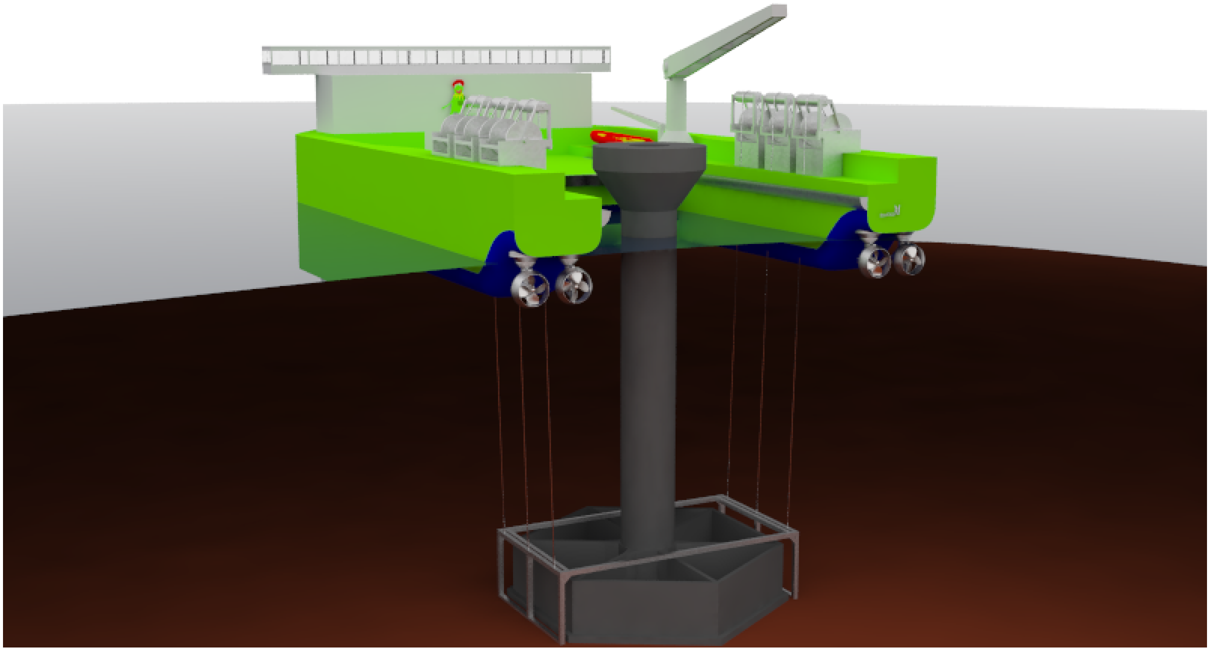

Figure 3.6. Installation of foundation on the seabed.

If the unit is a windmill the ship will reverse into position and push against the foundation as can be seen in Fig. 3.7. The windmill will be held over the cut-out with the help of the special designed fixture. The ship will then be lowered using ballast water to place the windmill on the foundation. The windmill has to be held on place for one hour after the installation crew has started the mounting procedure. The fixture will then be loosened from the windmill and the ship will rise to the transit draft.

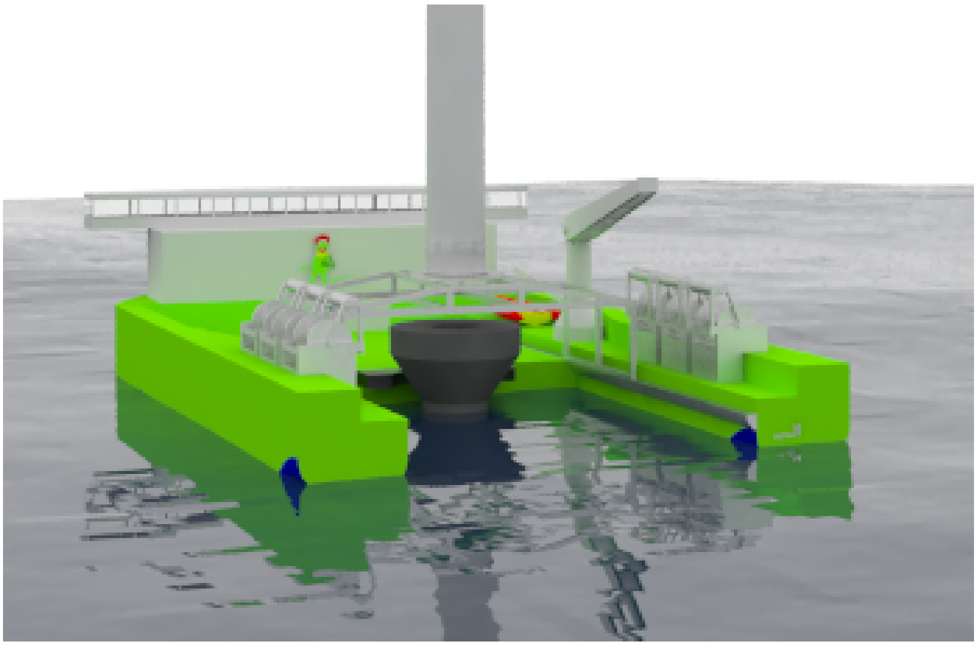

Figure 3.7. Installation of a windmill on top of foundation.

If the ship is carrying two units the empty fixture above the cut-out will be removed with the crane, mounted on starboard side, when it is time to move the second unit to the space above the cut-out.

3.2 Time schedule

Time schedules for the operations have been created to estimate the amount of time each operation will take. In this chapter the longest distances to the site from port have been considered and the transit speed of 10 knots.

Table 2.2 below shows how much time different part of the operation takes for the operation of carrying one foundation. The part that takes the longest time is the total transit time that is 5 hours and 20 minutes, 40% of the operation time. The part that takes the second longest time is loading the foundation to the vessel, 2 hours and 45 minutes, which is 20% of the operational time.

Table 3.2. Time schedule for carrying one foundation, hours and minutes

Lower the vessel in port:	00:26
Load foundation on vessel:	02:45
De-ballast the vessel:	00:34
Transit in port:	00:06
Transit to the site:	02:30
Position vessel, at site:	00:20
Move foundation to the cut-out:	01:30
Lower the foundation to the seabed:	01:20
Fixation of the foundation:	00:40
Loosen and recover the fixture:	00:40
Transit to port:	02:30
Transit in port:	00:15
Moor:	00:32
Total time:	14:08

The time schedule of the operation has also been visualised with a graph, see Fig 3.8. The x-axis shows elapsed time since the operation has started and the y-axis shows the distance from port. The largest distance from port to site is 25 nautical miles and the time schedule is calculated with this information.

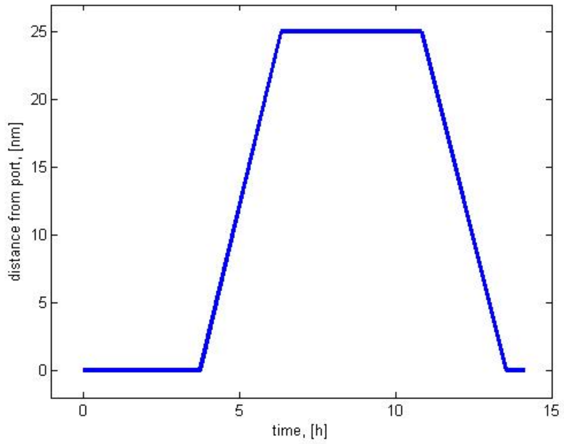

Figure 3.8. The graph shows the operation of carrying one foundation

The time for the transit in port when entering is longer compered to exiting. The reason is that the vessel has to reverse into position.

In Table 3.3 the time for the operation of carrying one windmill is estimated, and it has been visualised in Fig. 3.9. The time for this operation is the same as the time for carrying one foundation. The reason for this is mainly that the most time consuming parts are the same for both operations.

Table 3.3. Time schedule for carrying one windmill, hours and minutes

Lower the vessel in port:	00:26
Load windmill on vessel:	02:45
De-ballast the vessel:	00:33
Transit in port:	00:06
Transit to the site:	02:30
Position vessel, at site:	00:40
Move windmill to the cut-out:	01:30
Place windmill on foundation:	00:30
Fixation of the windmill:	01:00
Loosen fixture:	00:40
De-ballast before transit	00:10
Transit to port:	02:30
Transit in port:	00:15
Moor:	00:32
Total time:	14:07

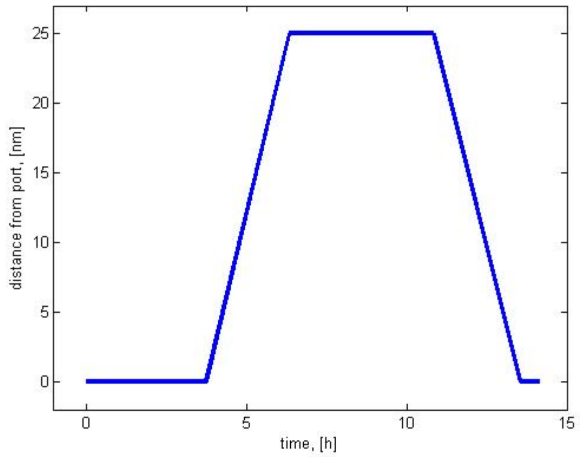

Figure 3.9. The graph shows the operation of carrying one windmill

As can be seen in the figures and tables, a lot of time can be saved by carrying two foundations, see Table 3.4 and Fig. 3.10. The total transit time for carrying two foundations is 5 hours and 30 minutes, compared to two times 5 hours and 21 minutes. The loading time of the second foundation will also be lower, 1 hour and 30 minutes instead of 2 hours and 45 minutes.

Table 3.4. Time schedule for carrying two foundation, hours and minutes

Lower the vessel in port:	00:26
Load foundation on vessel:	02:45
Load second foundation on vessel:	01:30
De-ballast the vessel:	00:34
Transit in port:	00:06
Transit to the site:	02:30
Position vessel, at site:	00:20
Lower the foundation to the seabed:	01:20
Fixation of the foundation:	00:40
Loosen and recover the fixture:	00:40
Transit to second position at site:	00:10
Position vessel, at site:	00:20
Move the second foundation to the cut-out and the first fixture away from the cut-out:	01:30
Lower the foundation to the seabed:	01:20
Fixation of the foundation:	00:40
Loosen and recover the fixture:	00:40
Transit to port:	02:30
Transit in port:	00:15
Moor:	00:32
Total time needed:	18:48

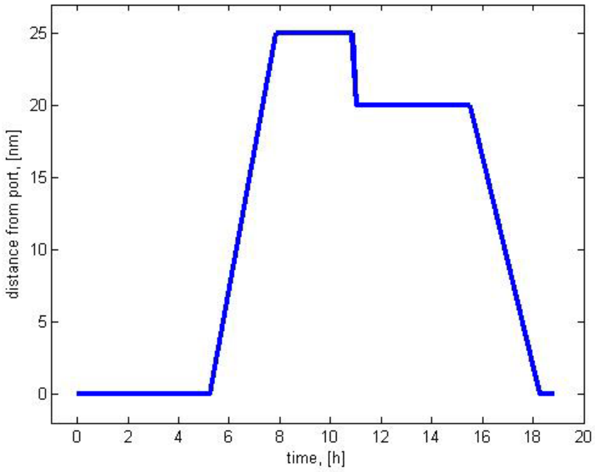

Figure 3.10. The graph shows the operation of carrying two foundations

There is also time to save by carrying two windmills in the same operation, see Table 3.5 and Fig 3.11. When it comes to transit and loading of the vessel the conditions are the same as the operation of carrying two foundations. The part that differs and makes the operation of two windmills longer is the procedure of installing the windmills.

Table 3.5. Time schedule for carrying two windmills, hours and minutes

Lower the vessel in port:	00:26
Load windmill on vessel:	02:45
Load second windmill on vessel:	01:30
De-ballast the vessel:	00:34
Transit in port:	00:06
Transit to the site:	02:30
Position vessel, at site:	00:40
Place windmill on foundation:	00:30
Fixation of the windmill:	01:00
Loosen the fixture:	00:40
De-ballast before transit	00:10
Transit to second position at site:	00:10
Position vessel, at site:	00:40
Move the second windmill to the cut-out and the first fixture away from the cut-out:	01:30
Place second windmill on foundation:	01:00
Fixation of the windmill:	01:00
Loosen the fixture:	00:40
De-ballast before transit	00:10
Transit to port:	02:30
Transport in port:	00:15
Moor:	00:32
Total time needed:	19:18

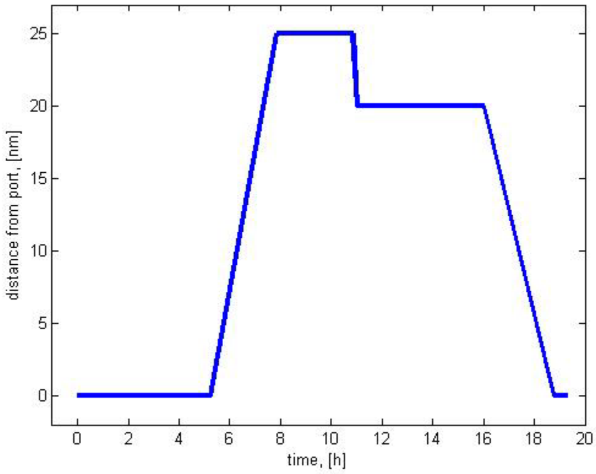

Figure 3.11. The graph shows the operation of carrying two windmills

3.3 Risk analysis

A general risk analyses have been conducted for the operations of installing foundations and windmills. In this analysis operational failures and the economic risk connected to these operational risks are analysed. The risks for the installation crew and other personnel working on board are not considered in the risk analysis since the installation procedure is still not known in detail. The details will be delivered by the windmill manufacturer in a later stage of the design. There are only analysis concerning one unit at the time in the tables, but operations' installing two foundations or windmills includes the same operations meaning that the risks are covered in the other analyses. But, carrying two units result in larger capital risk especially when carrying two windmills. However, the risk is regarded as small compared to the benefits of carrying two units.

When considering the operation with the foundations, the largest risks are when the foundations are to be winched down. If a winch breaks while lowering a foundation this can cause damage to the ship as well as to the foundation. The extent of the damage depends on where in the procedure it happens and if the winch is jammed or the wire breaks etc. But this risk should be minimized by maintenance and inspections of the winches with short regular intervals.

For the windmill the largest identified risk is waves that affect the positioning of vessel at site and the safe and smooth lowering of the windmill. Furthermore a wind gust can as well affect the installing procedure. However, these risks should be eliminated by close weather monitoring.

In general the risks in port and during transit are considered low compared to installation of the units at site. The port is considered protected compared to the site and if something goes wrong the vessel is close to shore. The transit is a conventional procedure and should not include any large risks. The reason there are no risks for the step of removing the fixtures when the vessel is back in port is because if there are issues with weather the operation can wait until the conditions are better.

4 General Arrangement

The purpose of the vessel is to transport and install offshore wind power plants, this includes both foundations and windmills. This purpose has formed the basis for the concept generation and the design of the vessel. The main challenges regarding the general arrangement are how to move, fixate and install the foundations and the windmills. Therefore two different fixtures are designed for the windmill and foundation as well as a skidding system to move the units on board the ship.

The superstructure layout with accommodation area, galley, mess and bridge are designed for the vessel according to TSFS 2013:68 (Transportstyrelsen, 2013). Lifesaving appliances, lifeboats and fire safety systems are all designed according to SOLAS, Safety of Life at Sea (IMO, 2004).

The following sections describe all the systems mentioned above in detail.

4.1 Deck specifications

The vessel has six decks in different levels, which are limited by the many, large ballast tanks. The ballast tanks make it possible to ground the vessel for loading and unloading of the cargo and their size and position is therefore very important.

The first deck is situated below the main deck, it contains the engine room and ballast tanks. The main deck is located seven meters above the keel. Above the main deck on 8 m above the keel there is a deck referred to as the crane deck since it facilitates the crane and the winches used to lowering the cargo. Ten meters above the keel the upper deck is located, on which the superstructure is placed. The first deck in the superstructure contains changing rooms, gym etc. The second deck of the superstructure is the accommodations deck. The third and last deck is the bridge deck.

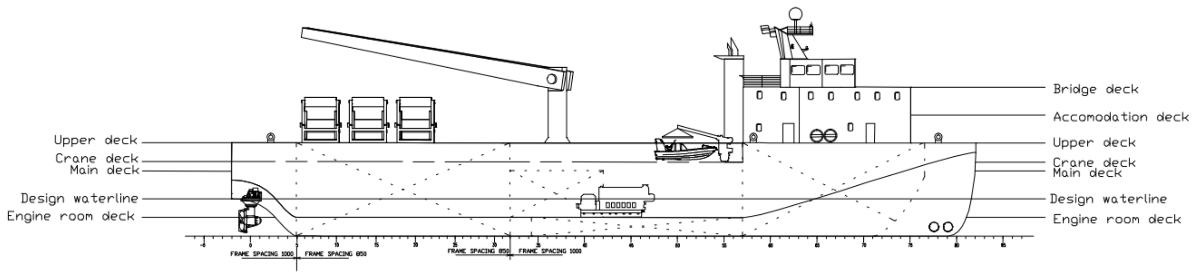

Figure 4.1. Side view of the vessel.

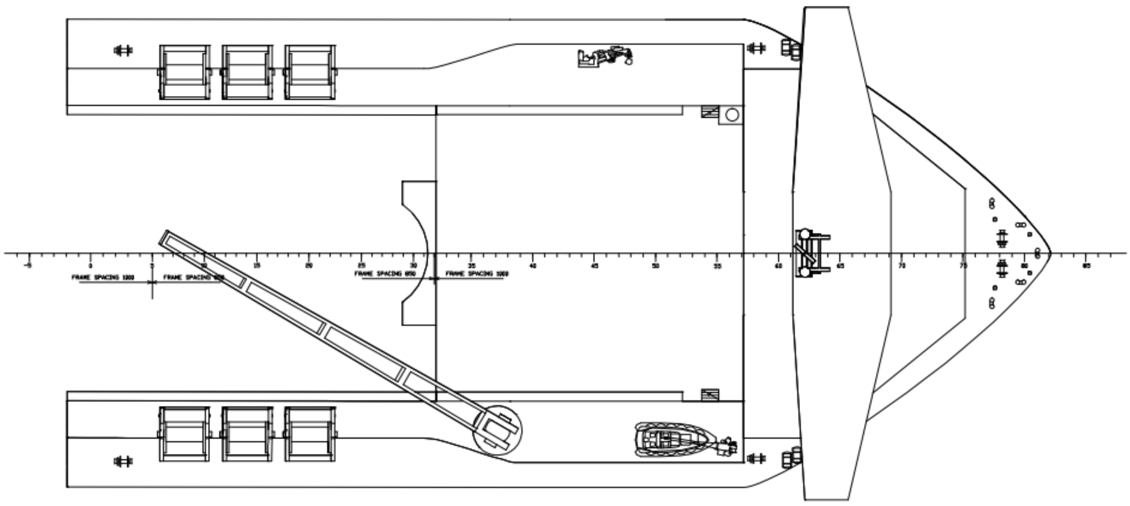

Figure 4.2. Top view of the vessel.

4.1.1 Engine room deck (deck 1)

On the first deck the engine room and ballast tanks are located. The first priority for this deck was to make room for the ballast tanks in order to fulfil the demands of lowering the vessel in port and at site. The space of the engine room was decided when the amount of ballast water needed was determined. Since the vessel is propelled by electrical pods, there is no propeller shaft to consider and the location of the engine room is more flexible compared to conventional propulsion system. Though, there is a constraint that the engine room has to access the elevator shaft and the staircase that goes all the way up to the accommodation deck. In the engine room, 13 meters behind the superstructure, two emergency exits are placed, one on starboard side and one on port side. These are in the form of two emergency ladders going up to the crane deck. The engine room complies with safety appliances and is described further in Section 4.8.2.

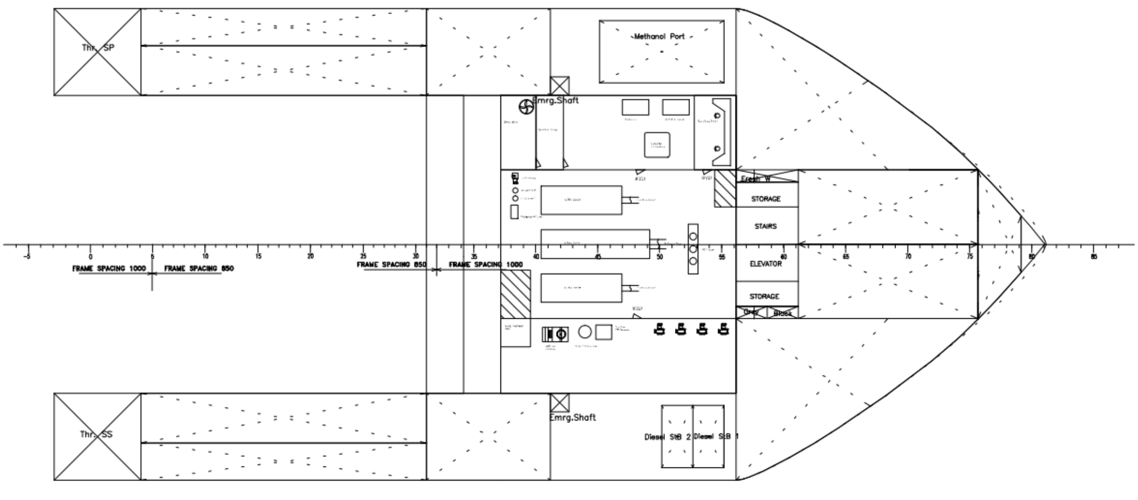

Figure 4.3. 1st Deck, Engine room deck. 2000mm ABL

4.1.2 Main deck (deck 2)

The main deck, which also applies as the working deck and freeboard deck, is 7 meters above keel and is where the windmills and the foundations are placed during transit. The aft part has a catamaran shape with a 30 meters long and 24 meters wide cut-out. In the cut-out there are foldable rails for the fixtures to slide on. In the middle of the vessel there is a large open deck, equipped with rails, where the cargo is placed during transit. If two units are carried at the same time, one of them is placed on the working deck and the other on the foldable rails in the cut-out. The foldable rails are described more thoroughly in Section 7.7.4.

4.1.3 Crane deck (deck 3)

The crane deck is located 8 meters above the keel. On this deck close to the midship a crane is placed, see section 4.4.4. On the same deck level but further aft, in the cut-out, the 6 large winches used for lowering the foundations are placed, three on each side of the cut-out, see Fig. 4.2 above.

4.1.4 Upper deck (deck 4)

On the upper deck, 10 meters above keel, most of the ship equipment is placed. At the bow the anchors with associated anchor winches are located. They are not to be used in the daily operations but are required according to the rules in DNV Pt.3 Ch.3 Sec.3 (DNV, 2011a). On both sides of the vessel mooring stations are located, one in the aft and one on the prepared area next to the superstructure. Next to the superstructure is also where the life rafts are found, see Section 4.6.1.

4.2 Superstructure

The superstructure is located on the upper deck at the bow, on top of the large ballast tanks. It consists of three decks, two larger and one smaller on top where the navigation bridge is located. The superstructure does not cover the entire deck area making it possible to walk around it. This is both because that there is plenty of space for the superstructure and the space for mooring stations and lifesaving equipment is located outside the superstructure. The shaft containing elevator and stairs starts in the engine room on deck 1 and goes up to the accommodation deck, deck 5, and then there is separate stairs up to the bridge deck.

4.2.1 Upper deck (deck 4) indoor

The first deck in the superstructure can be seen in Fig. 4.4 below. It contains changing rooms with lockers for the ship crew and the installation crew. There is also a gym for the crew and four toilets. The HVAC room with all installations is found on this deck as well as a laundry room. Only reached from outside, on port side is the garbage room and on starboard side are the fire station and emergency generators located. In the front of the superstructure there are some empty rooms for which the usage can be decided according to future needs.

Figure 4.4 also shows that there are four corridors with exits in the ends for easy access to the upper deck. The front doors lead to the front mooring station and the side doors leads to the smaller mooring stations on the sides and the life rafts.

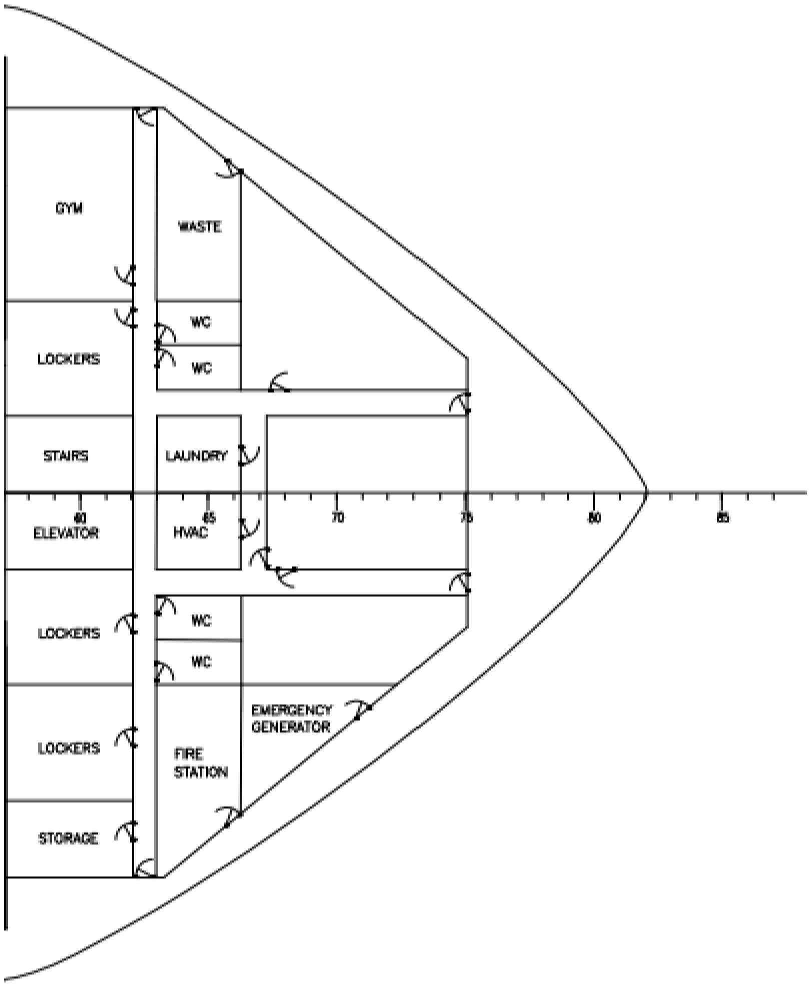

Figure 4.4. 4th deck, Upper deck. 1000mm ABL

4.2.2 Accommodation deck (deck 5)

The accommodation deck is shown in Fig. 4.5. It contains cabins, a dayroom, a mess and a galley with a separate room for food storage and a separate room for dishes. This deck also contains a conference room, first aid room, a room with cleaning equipment and stairs to the bridge deck.

For the daily operations a crew size of four is sufficient, but for longer journeys, for example to the yard, a larger crew is needed and therefore are there eight cabins on board, six smaller ones for the crew and two larger ones for the captain and the first engineer. The cabins for the crew have a size of 11.25 square meters and contain bathrooms. The captain and the chief engineer have cabins with the size of 15 square meters and an office connected to the cabin.

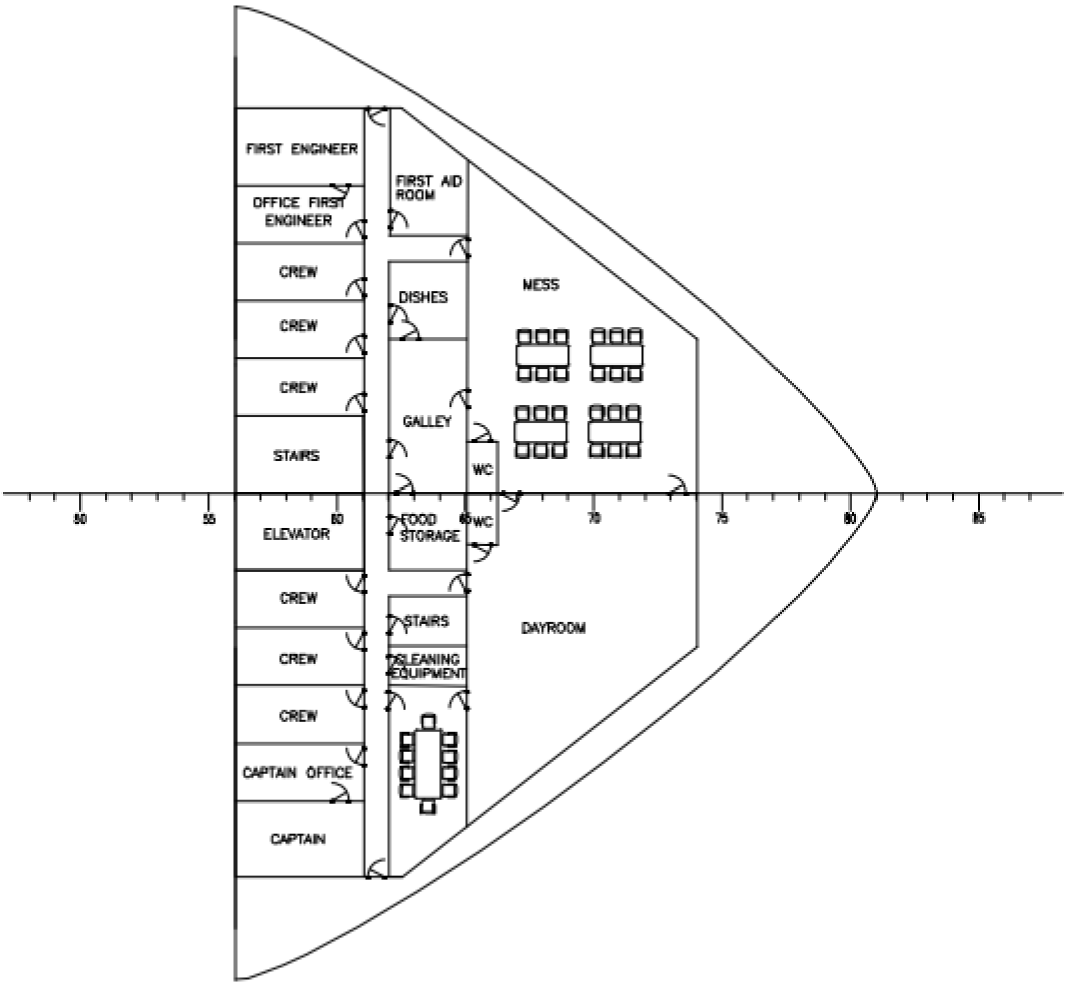

Figure 4.5. 5th deck, Accommodation deck. 13000 ABL

The food storage room is located opposite to the elevator and next to the galley for an easy transportation of goods. Next to the galley on the other side is the room for dishes. In front of these rooms are the mess and the dayroom located. These have a lot of windows to create a nice and light environment. Both the mess and the dayroom are sufficiently large to fit both the crewmembers and the installation crew. In connection to the mess and dayroom there are toilets.

A first aid room to be used in case of emergency is located on the port side of the superstructure and a conference room is on the starboard side. These are also placed to get a lot of daylight in to the room. There are emergency exit doors on both sides of the superstructure with ladders making it possible to access the upper deck and the life rafts.

4.2.3 Bridge deck (deck 6)

The bridge is located on the top deck of the superstructure to provide good view for navigation, manoeuvring and working operations on deck. To ensure good vision during mooring the bridge is equipped with bridge wings. To make the risks as low as possible when fixing the position at site during the windmill installation operation the bridge is design so that a separate panel can be used when heading astern. The transit control system is placed in the front of the bridge and on each wing the vessel can be manoeuvred during mooring and cargo operations. The DP system control panel is also located at the bridge. The navigation bridge is shown in Fig. 4.6 below.

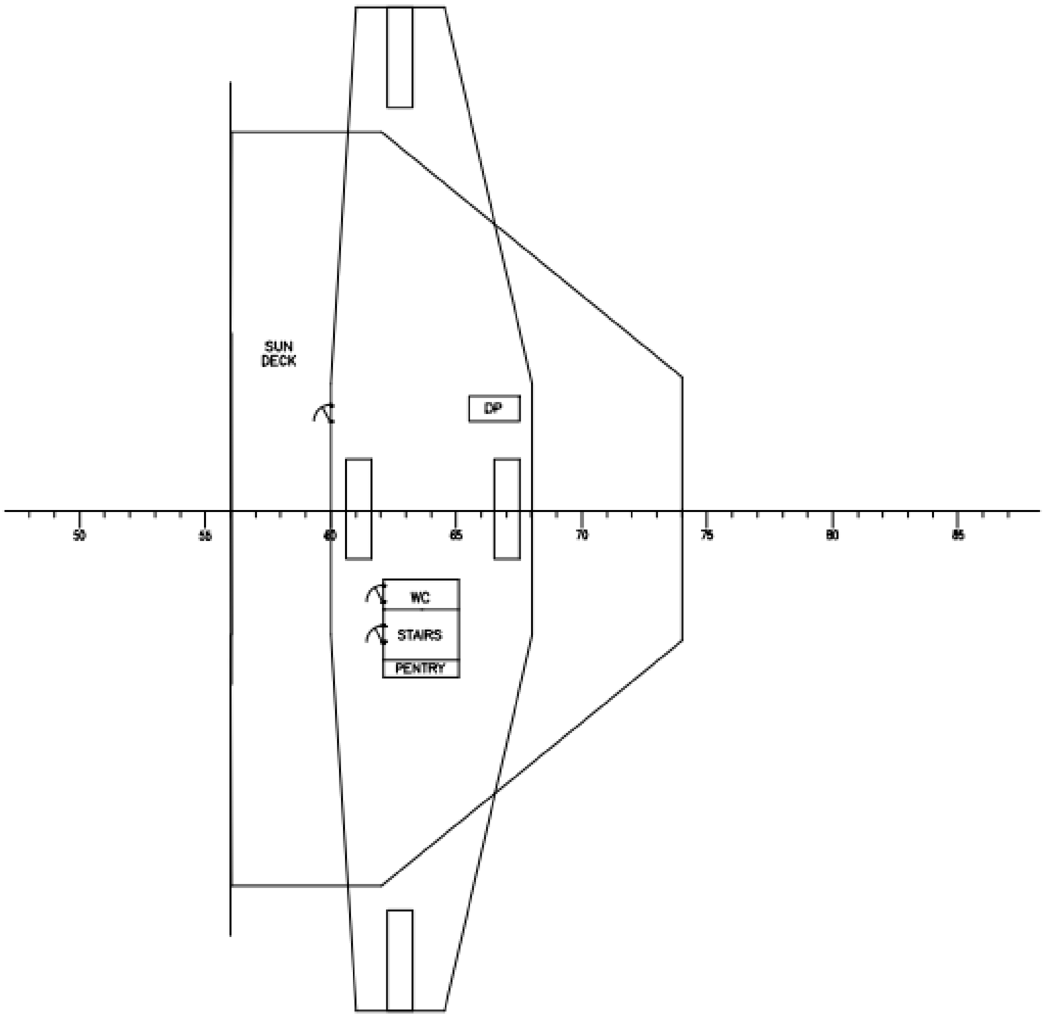

Figure 4.6. 6th deck, Navigation bridge. 16000 mm ABL

All windows on the bridge are equipped with windscreen wipers, de-icing and washing systems. The windows are inclined in 15 degrees to improve the vision effects further. At the bridge a kitchenette and a toilet is found. Behind the bridge there is a sundeck where the personnel can have a nice and relaxing break.

4.3 Navigation and Communication

The navigation and communication equipment on the special purpose vessel complies with SOLAS chapter V- Safety and Navigation Regulation 19 and chapter IV Radio communications (IMO, 2004). The DP-system installed on the vessel is connected to the navigation system. The navigation system is equipped with the following items:

- Magnetic compass independent of power supply
- Pelorus or compass bearing independent of power supply
- ECDIS, Electronic Chart Display System
- Backup charts
- GPS
- Radar S- and X-band
- Autopilot
- Gyro compass
- Echo sounder
- Speed log
- Wind speed and direction sensor

For the internal communication on board, radios are used in combination with cameras. The cameras are placed at the working deck in the aft, next to the mooring stations on each side of the vessel and at the mooring station in the bow

4.4 Deck equipment

Different deck equipment is necessary to perform the task of the vessel. As presented in the operational description, foldable rails are used in the cut-out to move the windmills and the foundations on board. On the rails, described in Section 7.7.4, fixtures are used as a tool for keeping the windmills or foundations in place. On board the ship the fixtures are moved by a skidding system, which can transport the heavy load. The fixtures have different designs for the foundations and the windmills. To lower the foundations, winches in the cut-out will be used. When collecting the fixture back to the vessel, a crane is used to put it out of the way for the second lowering operation if two items are going to be installed. A support in the cut-out is also used to make it easier to stay still while installing a windmill.

4.4.1 Fixtures

Two types of fixtures are used when installing the foundations and the windmills. The fixtures are made of steel and are sliding on the rails. The same type of fixture works for all the different foundation sizes. The fixture is connected to the reinforcement in the bottom of the foundations. This makes it possible to connect the fixture at a low point of the foundation. The fixture has an opening, large enough for all sizes of foundations, so it can pass the top of the foundations and be lifted back to the vessel. Figure 4.7 shows a principal sketch of the fixture which is further described in Section 7.7.3.2.

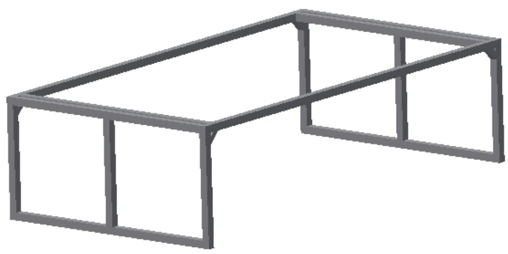

Figure 4.7. Principle sketch of the fixture to be used during operations with foundations.

The fixture for the windmills looks similar to the one for the foundations, see Fig. 4.8. On top of it, there is a circular plate which is connected to the upper flange on the windmill. When the windmill has been placed on the foundation the circular plate is disconnected and opened. The fixture will also be opened before the vessel can sail. Between the fixture and the circular plate, a number of hydraulic jack-ups are placed; see Fig. 4.9. The purpose of the jack-ups is to obtain damping effects making it possible to install the windmills even when the vessel is exposed to some heave.

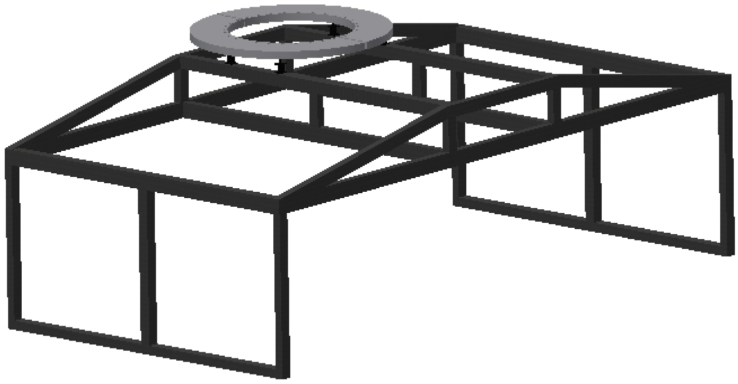

Figure 4.8. Principle sketch of the fixture to be used during operations with windmills.

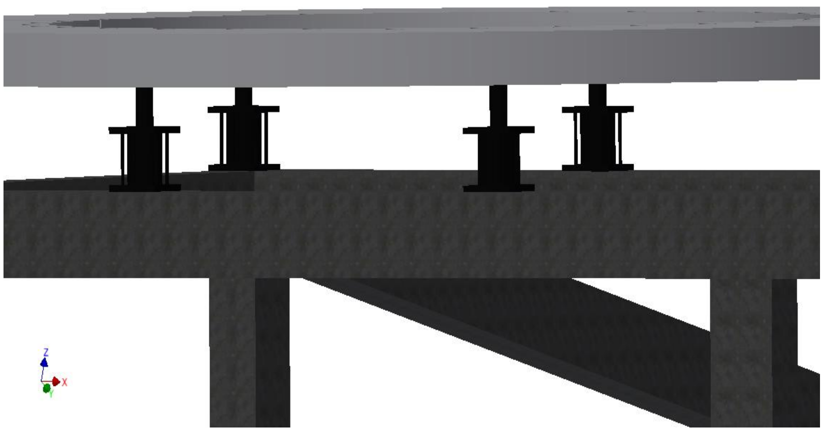

Figure 4.9. Principle sketch of the jack-ups on the windmill fixture.

4.4.2 Skidding system

The ship is equipped with a hydraulic skidding system for transportation of foundations and windmills on deck. The system uses sliding materials and low friction to provide a durable and highly controllable method for moving heavy loads. Two skidtracks with skidshoes are mounted on the deck. The fixture with a foundation or windmill will then be placed on these skidshoes, which can be moved along the rail. A hydraulic push-pull unit is also placed on each rail. This unit moves the skidshoe in both directions by pushing or pulling. Then it follows the skidshoe, connects to the track, and pushes again. This system is able to transport a load of 3000 metric tons, 1500 metric tons on each track, and has a speed of 20 m/hour. In the cut-out the skidtracks continues in two foldable rails (see Section 6.10.1), which are connected at the side of the catamaran part. When the fixture is standing on these tracks and connected to the winches, the tracks can be folded into the hull to let the fixture be lowered. The exact design of the system is not decided, but the rails and the push-pull unit could look something like the left picture in Fig. 4.10 and the skidshoe similar to the right picture in Fig. 4.10. This skidshoe is connected to the hydraulic arm of the push-pull unit.

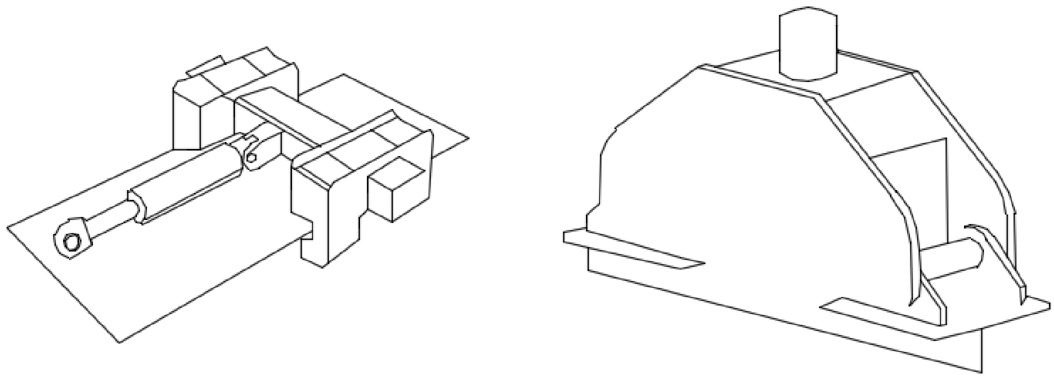

Figure 4.10. The left figure shows the principals of the push/pull unit and the right figure shows the principals of the skidshoe.

4.4.3 Winches

The vessel has six hydraulically or electrically driven winches installed onboard, three on each side of the cut-out. The fixture with a foundation will be connected to the winches via cables when it is standing on the foldable rails. When the rails are folded into the hull, the winches will lower the foundation to the seabed. The winches are then going to lift the fixture back to the vessel. This requires a high braking capacity, but a bit smaller lifting capacity. The winches are designed to make sure that the loads are the same in each wire. The weight is 24 000 kg for each winch. For more detailed information regarding the winches see Section 6.10.4.

4.4.4 Crane

A heavy lift crane is placed on starboard side of the ship, 6 meters in front of the cut-out on crane deck. The crane is used for handling of the fixtures for the foundations and the windmills, but it can also be used for other deck operations as well. This could for example be changing a broken winch or handling equipment for the engine. The crane could also be used for handling of provision. The crane weighs about 160 metric tons, has a diameter of 4 meters and a slewing range of 360 degrees. It can lift a maximum of 250 metric tons and has an outreach of 30 meter. This is enough since the largest fixture weighs about 200 metric tons and the cut-out is 30 meter long. However, the superstructure is closer than 30 meter to the crane so the operator has to take that into account when he operates the crane.

4.5 Mooring and Positioning Systems

The possibility to position the vessel in different operations is very important. In port it is essential that the rails in the cut-out align properly to the rail on the quay so the loaded fixtures can be transported on board. To do this the ship will be grounded but is has also the possibility to use mooring equipment. Anchoring equipment is also available if necessary. When installing foundations and windmills the positioning is also highly important. A DP system is therefore used to keep the vessel at the right place for all installations. A support in the cut-out will also be useful when installing the windmills.

4.5.1 Dynamic Positioning System

The vessel is equipped with a Dynamic Positioning (DP) system. This will make sure that the vessel can stay at the correct position when the foundations and windmills are installed. The system has to respond very fast to rapid changes in sea condition. Placing a windmill on top of a foundation is an operation that is highly sensitive to motions. Large ship motions could also cause problems when a heavy foundation is hanging in the winches. The DP system consists of four different sub systems, which work together to perform satisfactory; sensor system, control system, thruster system and power system. Since the same thruster system is responsible for the transit of the vessel and it may be conflicting with the DP requirements, a reasonable compromise had to be done. For more detailed information regarding the DP system see Section 6.6.

3.5.2 Mooring and Anchoring

It is possible for the vessel to fix its position by using the seabed, or to secure to a permanent structure like bollards on a quay or mooring buoys. The system is designed to hold the vessel in good holding ground and in moderate sea condition. There are four mooring stations and one anchoring station. There is one mooring stations behind the winches on the upper deck of both sides and one on both sides of the superstructure. The anchoring station is situated at the bow. This station can also be used for mooring by adding mooring lines to the station.

In the case of normal operation, these mooring stations will not be needed since the ship will be grounded in port. In other cases when it is not possible or suitable to ground the ship, the mooring will be used instead. The anchor won’t be used in normal conditions but is a class requirement.

4.5.3 Support

When installing a windmill on top of a foundation, some support is needed to guide the ship to the correct position but also to make it possible to keep still on the same position during the installation. To do this the ship is equipped with a support in the cut-out in the shape of an arc. This support makes it possible for the ship to head against the foundation, with a maximum push force of 3 MN, and in this way makes it easier to place the windmill on top. The supporting arc has a core of steel and is covered with a rubber material, making the grip between the arc and the foundation useful. The support has the height of 1m and the top of it is in the same level as the main deck. This makes it possible for the skidding system to pass with a foundation or windmill over the support, but at the same time it is in the right height to be able to grab the foundation even when the ship is lowered when installing a windmill. The support is shown in Fig. 4.11.

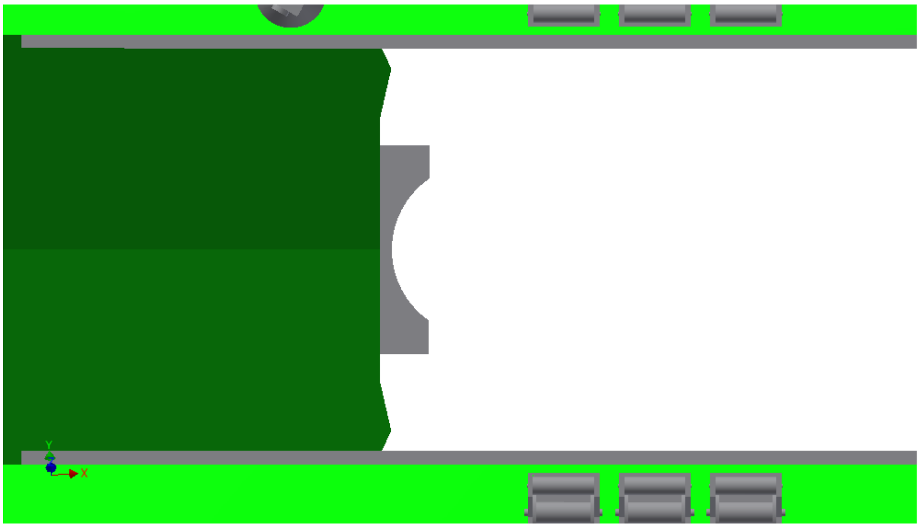

Figure 4.11. A principal figure showing the support against the foundation from above.

4.6 Tanks for hotel facilities

The vessel is equipped with three different water tanks; freshwater, blackwater and greywater tanks. All three tanks are placed on the engine room deck, 2 meter above baseline. The freshwater tank is 18,8 m^3 and placed on port side of the stairs/elevator to the superstructure. This should be enough for a total crew of 18 persons, eight for the ship and ten for installation operations, for 2,5 days. If the crew consists of less people, the water will of course last even longer. Freshwater could also be used for some deck operations but the tank will have enough water for that as well.

The blackwater and greywater tanks are place on starboard side of the stairs/elevator. The blackwater tank is 9,4 m^3 and the greywater tank is 9.5 m^3. Blackwater and greywater is used freshwater from galley, laundry, showers, toilets etc. Usually there are regulations regarding discharging of these tanks, which is the reason that they together can hold all the water from a fully loaded freshwater tank.

4.7 Funnel

The funnel from the engine room is placed in the front left corner of the main deck. It has the bottom shape of a rectangular with the breadth 1,5 meter and length 2 meters. To ensure a pleasant working environment for the crew and to minimize the amount of exhaust gas on deck, the funnel is made as high as the bridge.

4.8 Safety

The safety equipment and safety in general on this vessel is designed according to SOLAS, Safety of Life at Sea (IMO, 2004).

4.8.1 Lifesaving appliances

The lifesaving appliances are governed by SOLAS chapter III (IMO, 2004) and Life Saving Appliance code, (van Dokkum, 2008). According to the regulations the vessel needs to have lifejackets, lifebuoys, survival suits, life rafts and a man-over-board boat (MOB).

Since the length of the vessel is less than 85 m no lifeboats are needed to meet the regulations. The vessel will have one life raft placed at the upper deck on each side of the superstructure. Each life raft is able to carry all persons on board at the same time. There will be at least one lifejacket for each person who will be on board the vessel during the same time. The lifejackets will be stored by the life rafts. On starboard side of the crane deck (deck 3) there will be a MOB. A separate crane will carry out the launching of the MOB, it will also be able to pick-up the MOB back to the crane deck.

4.8.2 Fire safety

The ship will carry all necessary safety equipment according the latest SOLAS requirements and international FSS code. Lifesaving appliances will be stored where it is necessary and easy to reach for all of the crew on board in an emergency. This means both on the main deck but also in the accommodation and on the bridge.

The way fire is treated is different for different areas of the ship. Accommodation will typically require water or soda-acid systems, and breathing apparatus as the area will most certainly fill up with smoke in case of fire. The galley in particular, requires additional provisions such as foam, dry powder or CO_2 extinguishers as the materials such as cooking oils and such are Class B (Taylor, 1992).

Machinery space fires mostly involve Class B materials as well, so using water will lead to expanding the fire. Here, foam extinguishers are used. If the fire is electrical, then only dry powder or CO2 extinguishers are permitted. If the situation of extinguishing the fire becomes hopeless, then an evacuation of the entire area, followed by isolation of fuel lines, electrical lines and ventilation ducts, and then a gas flooding of the sealed out space is performed. Meanwhile cooling from the outside must continue using water hoses (Taylor, 1992).

The fire fighting systems are able to detect both fire and smoke, and will be easy handled and controllable from the bridge. Push buttons for the systems are placed on all decks and in the corridors, which allows everyone on board to start the systems manually if an emergency occurs. The systems work together with the ventilation system, this makes it possible to prevent fresh air from reaching the fire and in that way stop the fire propagation.

Every deck has fire hoses and in the accommodation area is it important that every cabin can be reached. The ship is also equipped with hand held fire extinguishers at every 250m2. They weigh maximum 23kg and contain 9 litres. There are at least two exits from every hallway, which makes it possible to escape in the case of fire. Those exits are marked with lights that will work even during a power failure.

A fire safety classification that complies with SOLAS is defined by DNV-GL (DNV, 2011a) as additional class notations of F-A, F-M and F-C depending on whether they are equipped according to accommodation, machinery and cargo spaces respectively. The vessel will have to fulfil both F-A and F-M notations. DNV-GL may or may not be authorized to issue safety certificate, as it is within the authority of SOLAS. Nevertheless, DNV-GL’s own requirements need to be fulfilled. (DNV, 2011a) SOLAS Safety Certificates are issued upon approval of a number of documents including but not limited to: Fire Control Plan, Structural fire protection plan, Ventilation system drawing, Escape route plan, Fire main system drawing, etc. These documents must be included in future developments.

4.9 Human factors

The human factors perspective has influenced the general arrangements in this project.When designing the superstructure natural daylight was high priority in the cabins, and especially in the mess and the day room since this influences how people feel, their mood and energy. The cabins, corridors and other rooms are made larger than the rules require as well as the room height is increased to create a more inviting environment onboard. Furthermore the superstructure is in two different decks, where the first contains areas connected to work while the second deck is for accommodation, breaks and free time. This arrangement is to establish a work free and relaxed environment on the accommodation deck.

The accessibility varies between different areas of the vessel. In the superstructure there are wide corridors and the different rooms and cabins are placed in connection to each other to make it easy to reach them. From the superstructure it is easy to reach the upper deck from where the winches can be reached as well as mooring stations and the life rafts. The working deck is easiest accessed from the elevator or indoor staircase, but could also be reached by smaller outdoor ladders between the different decks. The engine room is also accessed by the elevator or the main staircase, and has two emergency exits 13 meters behind the superstructure, one on each side.

The anchoring station, the mooring stations close to midship and the mooring station in the bow are placed close to doors in to the superstructure. This makes it possible for the person operating the mooring station to wait indoors until the station is to be used. This does not apply for the stations in the aft, but they could be improved with rain covers. All the mooring stations are designed so that the snap back zone can be avoided.

The safety equipment installed onboard complies with SOLAS, but to improve the design further regarding safety and operation issues experts are contacted for consultation. For the bridge design Johan Magnusson, bridge officer, assisted answering questions and had input on the design. The same did Linda de Vries, human factors researcher and former bridge officer, regarding the mooring stations. Both of these exports are employed at the department of Shipping and Marine Technology at Chalmers University of Technology.

The equipment used during the installation operation of the windmill is not analysed in this project and is left for future investigations by experts within the field. Though, it has been considered to the extent that there are storage rooms available for equipment. Also the elevators are installed to ensure the possibility to transport equipment and material between the different decks, for example from main deck to engine room deck where the workshop is located.

An area where closer investigations are necessary is on the two decks next to the cut-out part of the vessel. The winches are large and might cause problems passing by to the mooring stations in the sterns. One solution, if this after closer investigations turns out being a problem, is to make the deck area larger but keep the hull shape of the submerged part. To make this area safe, it is important to make railing that can be opened to enter the vessel easily. If these analysis show that the accessibility in the aft is limited, then an accommodation ladder will be added on starboard side of the vessel to enter close to midship.

4.10 Economy of the project

An estimation of how much it will cost to build and run the vessel is evaluated in this section. Since this report is written in an early stage of the development of the project, this estimation can be considered as non-conservative. There are several parameters that have not been considered and some are still unknown. The sole purpose of the estimation is to give an indication of the different costs.

The building cost of a vessel can be divided into several items. The ones that have been chosen for this case can be seen in Table 4.1.

Table 4.1. Building costs of the vessel expressed in million SEK

Steel cost of the ship:	64,1
Machinery:	68,6
Bridge:	10,0
Fixtures:	24,0
Accommodation facilities:	10,0
Total investment cost:	176,7

The cost of building a ship without equipment or machinery can be estimated using the steel weight of the structure; this is also applied for the fixtures. The weight of the structure was multiplied with 35 000 SEK/metric tons, the price is based on a price form a Polish shipyard. The weight of the fixtures was multiplied with 60 000 SEK/metric tons. It is assumed that 8 fixtures of each type will be built. The cost of the machinery can be estimated in a similar manner, the effect of the system was multiplied with 9 300 SEK/kW. The costs of the bridge and the accommodating facilities are estimated after discussions with experienced naval architects.

Some cost items that should have been included in the new building cost have been excluded from the analysis since they are difficult to estimate in this step of the process. The items are:

- The 6 winches
- The crane
- The mooring and anchoring stations
- The skidding system

To make this vessel comparable to other vessels the yearly cost has been estimated, see Table 4.2.

Table 4.2. The yearly costs of the vessel expressed in million SEK

Investment cost:	39,6
Maintenance, certificates etc:	5,5
Insurance:	1,8
Administration:	1,9
Crew of the vessel:	21,9
Fuel cost:	6,7
Total yearly cost	77,4

The investment cost has been calculated for the case of taking a loan to cover for the new building cost. In the calculation the interest has been 6% and the number of years that the debt should be paid in is 5 year, since this is the time it will take to finish the wind farm, see chapter 3.

The costs of maintenance, certificates, insurance and administration are estimated by compering cost for other Swedish ships. The crew cost is estimated to be 1250 SEK/hour/person and there are 6 crewmembers that run the vessel. The cost for the installation crew has been disregarded. The fuel cost has been estimated for running the vessel on methanol with the fuel consumption for 182 working days per year, 91 days carrying two units and 91 days carrying one unit.

4.11 Light Ship estimation

To be able to estimate the light ship weight of the vessel, different weights from the superstructure, machinery equipment and steel weight distribution have been added together. The light ship weight is the weight of only the vessel, without cargo, fuel, fluids or passengers/crew on board. Figure 4.12 below illustrates the different weights together with their position on the vessel. Summing the weights from machinery, structural and superstructure gives the ship a total light ship weight of 2499 metric tons.

4.11.1 Superstructure

The weight of the superstructure is calculated using standardized measurements for living quarters, galley etc. The weight used for regular areas is 125 kg/m^3 and gives a total of 378 metric tons. The longitudinal length this weight is distributed over can be seen in fig 4.2, Topview.

4.11.2 Machinery equipment

The Azimuthing thrusters, engines and bow thrusters, together with the deck crane and the six winches are the heaviest pieces on board the vessel. Their position can also be seen in fig. 4.2. Topview together with fig. 4.1. Sideview. The total weights of the engine room have also been estimated with reference to similar ships. The total contribution to the light weight is 667 metric tons.

4.11.3 Structure

The weight of the ship structure is 1452 metric tons. More about the total steel weight can be found in section 7.6 *Structural weight distribution.* As can be seen in fig. 4.12 below, the peak in weight is 37 meters from the stern, where the crane, engines and steel structure interact.

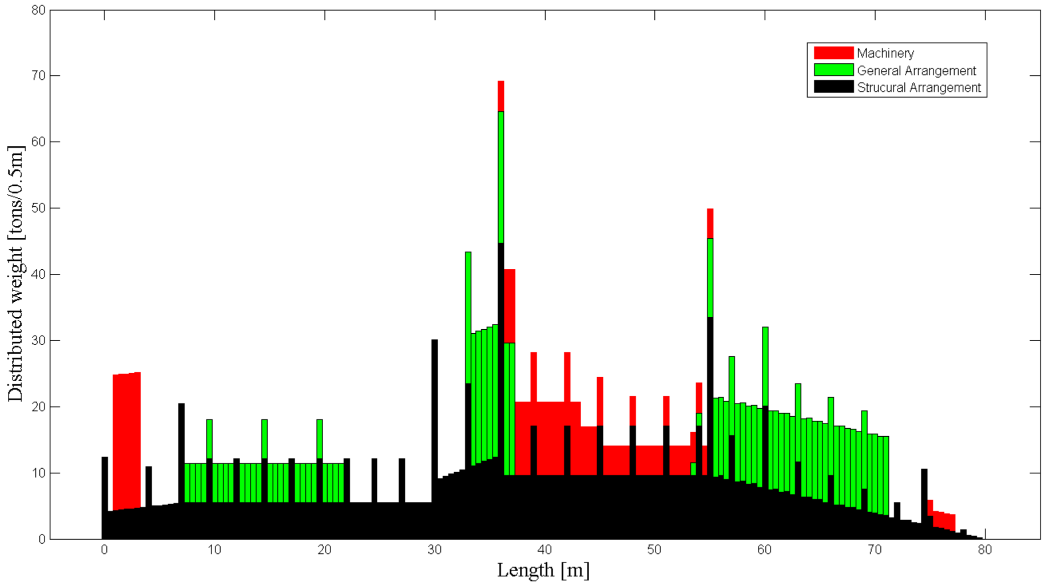

Figure 4.12. Light ship estimation of Kermit

4.12 Future work

In the future, before a vessel like this can be build, much more detailed work has to be done. The following list mentions some of the necessary development that has to be made before production.

Detailed time schedule: The time schedule in this project is in many ways based on assumptions. The operational time for the installation procedure has to be further analysed but also the possible time, weather window, in which installation operations could be carried out, has to be more analysed.

Accurate cost calculations: It is challenging to estimate the cost for a ship like this. Since not all the necessary development in finished with this project, more extensive work has to be done before production. The cost for this development should also be taken into account when calculating the total cost to build this ship. It is also difficult to predict the time it will take to build the ship, and the time at the yard will have a large influence on the cost.

Installation operations: The installation procedure described in this project is still in a development stage. For example, the possibility to carry more than one windmill or fixture and the handling of the fixtures in these combinations could be developed further.

Fixtures: Exactly how the fixture with the damping and ring for the windmills are going to look like is not completely designed. The fixtures do also need some attachment, in which the winches could connect.

Superstructure: The size and shape of the superstructure could be changed if the client would like to. Also the inside of the superstructure could have another layout if the amount of crew changes or some rooms would like to be removed or added.

Port: This vessel has some requirements on the port. The seabed must be in the way that the ship could be able to ground. Also the interface with the rails on port and on the vessel must be designed. This is something that needs to be done in future work.

5 Hydromechanics

In this chapter the shape of the hull and its characteristics are explored. The hull is designed to comply with relevant DNV rules and to be optimized for the task. The hull´s hydromechanics is designed to minimize cost while being able to operate in several modes. The coordinate system used in this chapter is illustrated in Fig 5.0. The zero point is located at the intersection of the baseline and the extreme aft of the vessel.

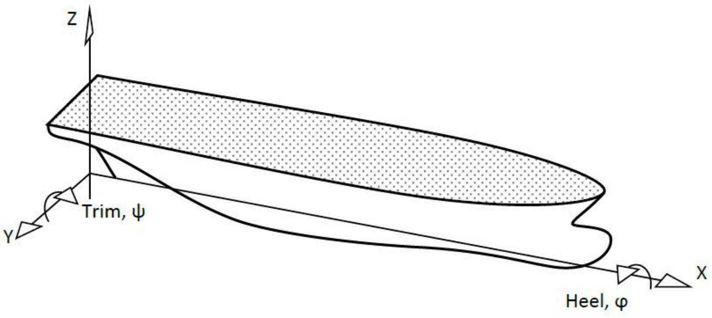

Figure 5.0. Coordinate system

5.1 Main design particulars

A critical part of the concept is that the foundation is to be lowered with wires attached to the base of the foundation. For this to be possible the cargo has to move through the ship. The opening in the ship; here on called the cut-out, has to be at least as wide and long as the base of the largest foundation. Since the cargo is to be carried on deck, there is a need for a cargo deck large enough to fit the largest foundation.

Keeping cost as low as possible was a main restriction on this project. As the cost for the ship is directly linked to ship size the ship is made as small as practically possible. With the established set of class rules, a vessel with LPP bigger than 80 m introduces a set of rules for damaged stability that was deemed unfavourable, see Section 5.7 .The length is therefore set to 80 m. The cut-out needs to be 24 m wide in order for the large foundation and its fixture can pass through it. To have space on deck where the cargo is lowered, the beam was set to 38 m. The total beam results in a beam for the catamaran hulls of 7 m each; see Fig 5.1 below.

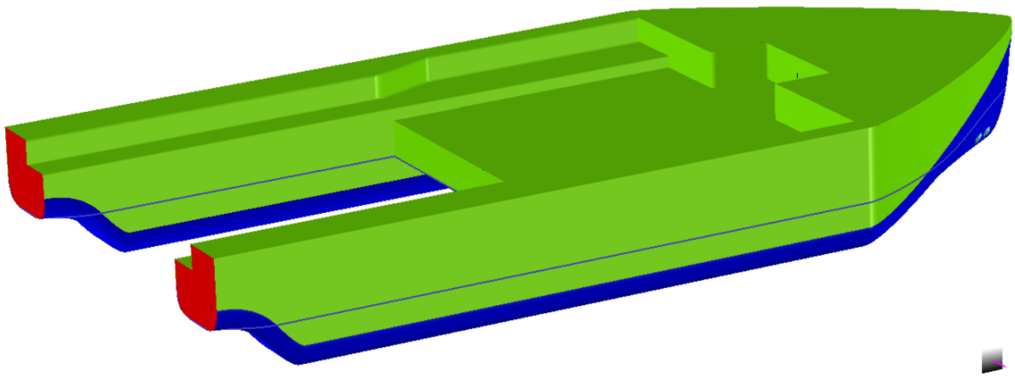

Figure 5.1. Semi-Cat

The ship in front of the moon pool was not made into two separate hulls for a number of reasons. It would be structurally complicated, it would increase the wetted area of the hull, resulting in an added resistance, -and it would leave little space for machinery. The length of the cut-out was therefore extended from its minimum length to 30 m. This was done to give space for propulsors in sterns without depriving too much of the needed buoyancy. The issue of keeping the ship on even keel will be further explained in the following chapters.

In order for the ship to be able to be grounded easily, the keel line is kept at the same level throughout the ship. The design draft is set to 4 m although the draft varies a lot depending on the load case. The beam of 38 m is set because it gives a reasonable compromise between centre of buoyancy, COB, and the beam. If the beam would be less, the draft in the stern would have to be greater. This would relocate the COB further to the front as the displacement in the mono-hull front increases faster with the draft than the catamaran stern does. Centre of buoyancy close to the front is unfavourable because the cargo is kept over the cut-out, near the stern, when offloading.

Several more concepts than the semi-catamaran were initially tested, such as catamaran and SWATH (Small Water-plane Area Twin Hull), see Fig 5.2 and Fig 5.3 respectively. The SWATH had a too big draft and poor longitudinal stability added to the disadvantages with the use of a catamaran mentioned earlier.

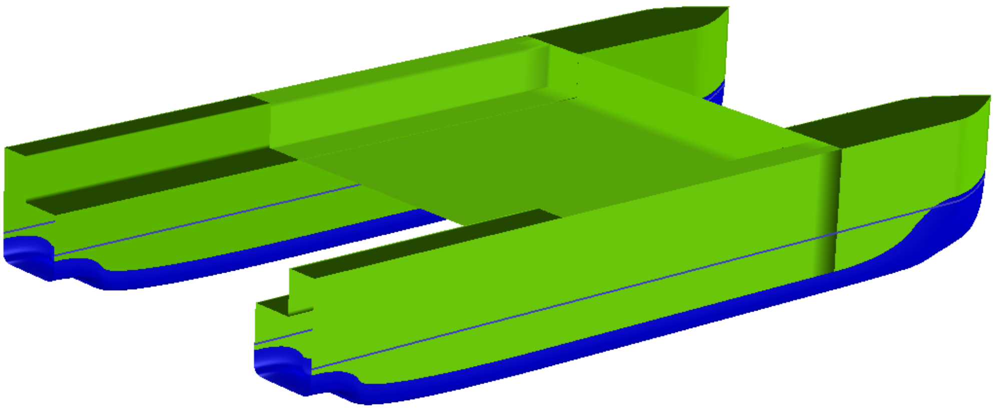

Figure 5.2. Catamaran

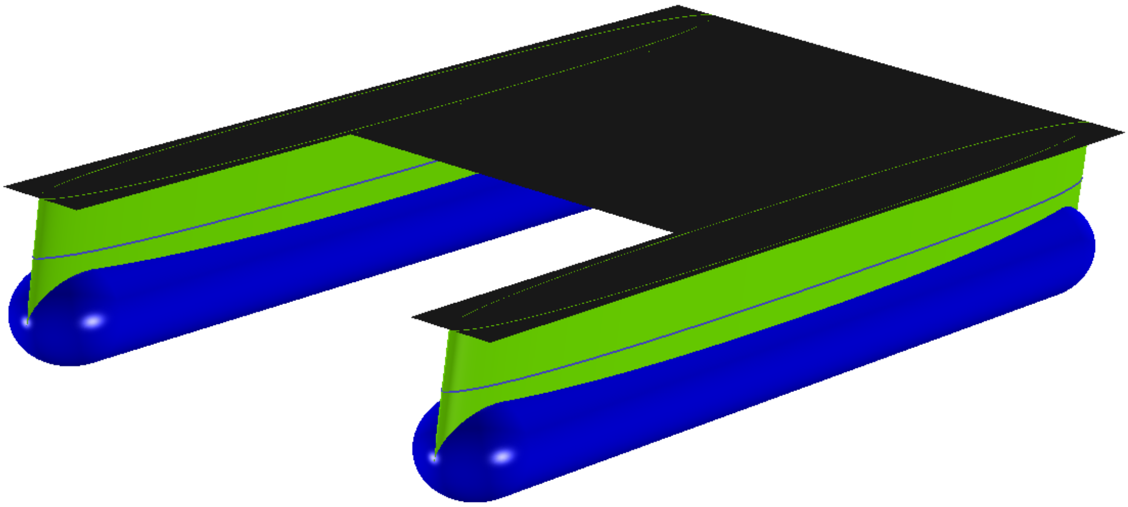

Figure 5.3. SWATH

Further description of the chosen design will come in the following chapter Hull lines.
In Table 5.1 the main dimensions of the ship is listed.

Table 5.1. Vessel main particulars

Length over all (LOA)	80 m
Length between perpendiculars (LPP)	77.5 m
Waterline length (LWL)	79.5 m
Cut-out length	30 m
Beam	38 m
Cut-out beam	24 m
Design draft	4 m
Design Displacement	6592.9 m^3
Depth to upper deck	10 m
Depth to main deck	7 m
Deadweight	7983 ton

5.2 Hull lines

A main goal of this project is to keep the cost of the vessel low. The price of double bended steel is several times more expensive than single bended steel, since manufacturing of double bended is more demanding. In Fig 5.4 the double bended surfaces are displayed while the hull is upside down; all the green surfaces are only single bended.

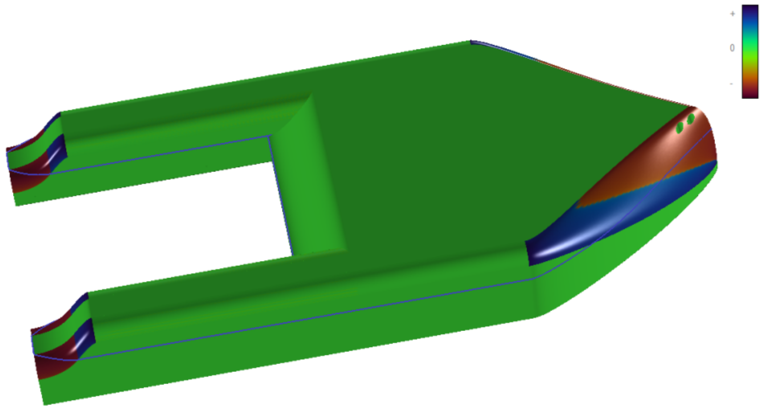

Figure 5.4. Gaussian curvature where double bended surfaces are blue or red and single bended are coloured green

This issue is brought up because it has a large impact of the design of the hull. The full lines drawings are in Appendix B Lines Drawing. The hull is made with a constant outside wall shape alongside 60% of the ship's length, from the stern slope to the shoulders. In Fig 5.5 and Fig. 5.6 the curvature in longitudinal and transverse direction of the hull surfaces is shown, surfaces that are green in both figures are flat. The hull is designed in such a way that most surface area is flat, this is to keep the manufacturing process simple and cheap. As a result of this the only double bended surfaces are the bilges in the slopes in the sterns and the bilges in the bow.

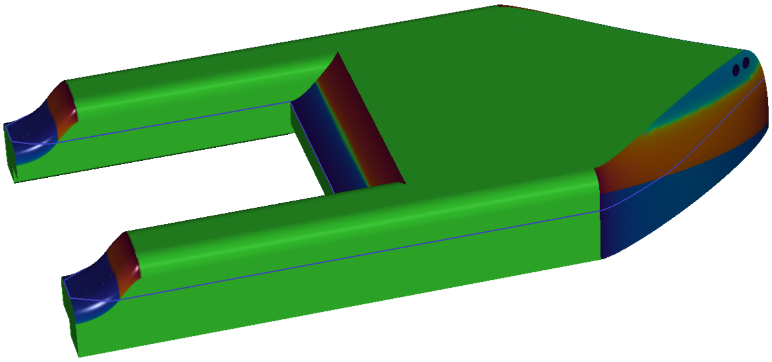

Figure 5.5. Longitudinal curvature where longitudinally bended surfaces are blue or red other are green.

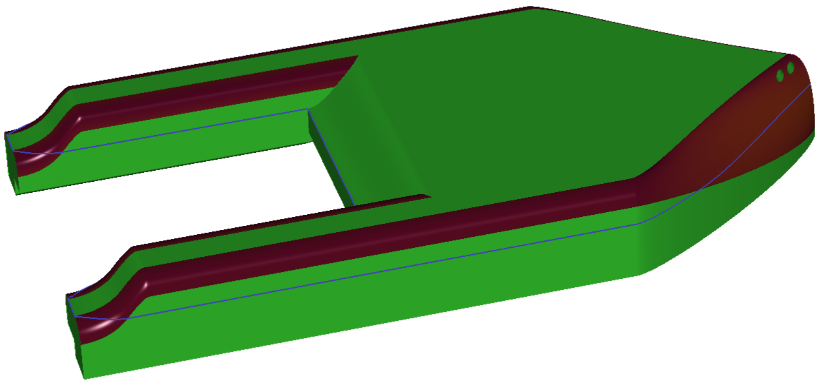

Figure 5.6. Transverse curvature where transversely bended surfaces are blue or red other are green.

The bilge radius is 2m both on the inside the cut-out and outside to reduce the wetted area. The sterns are sloped up to the water line in order to house the four azimuth thrusters and to eliminate wet transoms. The slopes have a mean angle of 35.5°, elevating 4 meters on 7 meters longitudinal distance. The mono-hull end has a similar shape with a mean angle of 39.8° and is there to form a good inflow into the cut-out by eliminating the wet transom. There are sharp inside corners in the very front of the cut-out instead of rounded to further simplify the building process.

The bow is made slender to counteract the large beam of the ship. A more slender bow generally decreases the height of the bow wave system which will lower the wave resistance. The bow is also made narrow to reduce buoyancy and to relocate the COB further astern. The shoulders are located 25 m from the bow giving it a wedge-like shape with a forward corner of 75°. The bow has a slight V-shape, both in and outside of the water. This is to earn more room for ballast tanks in the bow while reducing the possible effects of slamming connected to a step.

5.3 Human factors

The main concerns regarding human factors from a hydromechanics point of view involves movements of the vessel and the response of the crew to them. The short rolling time of the vessel will make the people working on board more prone to sea sickness.

Another thing to consider is the movements of the vessel during the installation process, especially when the wind turbine will be installed due to the small tolerances. The movements are investigated further in the seakeeping section.

5.4 Tank plan

When designing tank arrangement, the main things under consideration are to have enough counterweight for the offloading procedure and to ground the vessel. Dimensions of fuel-, lube oil-, fresh water-, grey water- and black water tanks were decided in coordination with the consumptions that GA group and Machinery group estimated, presented in section 4.5.4. Maxsurf Stability was used to calculate the stability properties of the vessel. The tank sizes and tank COG’s are presented in Table 5.2, the vessel totals are also shown.

Table 5.2. Tank Plan with sizing, capacity and centres of gravity.

Tank name	Load(mt)	Vol(m^3)	LCG(m)	TCG(m)	VCG(m)	Perm.
BW Bow Port P	1272,5	1241,5	61,6	-10,6	5,3	0.95
BW Bow Port S	825,8	805,6	67,2	2,9	5,0	0.95
BW Bow StB P	825,8	805,6	67,2	-2,9	5,0	0.95
BW Bow StB S	1272,5	1241,5	61,6	10,6	5,3	0.95
BW Bow mid	170,5	166,4	76,0	0	7,2	0.95
BW Stern Port P	655,3	639,3	18,5	-17,4	5,1	0.95
BW Stern Port S	704,6	687,4	18,5	-14,0	4,0	0.95
BW Stern StB P	704,6	687,4	18,5	14,0	4,0	0.95
BW Stern StB S	655,3	639,3	18,5	17,4	5,1	0.95
BW DB P	766,5	747,8	43,7	-10,1	1,0	0.95
BW DB S	766,5	747,8	43,7	10,1	1,0	0.95
BW Mid P	340,8	332,5	35	-15,5	4,5	0.95
BW Mid S	340,8	332,5	35	15,5	4,5	0.95
Methanol Port	150,5	190	49	-15,5	4,0	0.95
Diesel StB 1	39,9	47,5	50,3	15,5	4,0	0.95
Diesel StB 2	39,9	47,5	52,7	15,5	4,0	0.95
Fresh W	18,8	18,8	57,5	-5,5	4,0	0.95
Grey	9,7	9,5	58,8	5,5	4,0	0.95
Black	9,7	9,4	56,3	5,5	4,0	0.95
Lube	0,88	0,95	40,5	0	2,5	0.95
Slop	34,694	38	45	0	1.0	0.95
Total	**9645,1**	**9436,4**	**46,4**	**1**	**4,0**	

5.5 Hydrostatic calculations

The hydrostatic properties of the ship can be seen in Table 5.3 below. These calculations are valid for the design draft of 4 m and were carried out using the software Maxsurf Modeler.

Table 5.3. Showing Hydrostatic Calculations for design draft of 4 m

Dimensions:		
L.O.A	80	m
Lpp	77,5	m
Beam max	38	m
Design Draft	4	m
Displacement volume	6735,5	m^3
FW displacement	6735,5	ton
SW Displacement	6904.0	ton
Centroids:		
LCB	43,7	m
TCB	0	m
LCF	42.5	m
VCB	2,1	m
Coefficients:		
Block (Cb)	0,563	
Prismatic (Cp)	0,557	
Midships (Cm)	0,99	
Water plane (Cwp)	0,626	
Areas:		
Waterplane area	1891.1	m^2
Wetted Surface	2610,2	m^2
Ratio:		
L/B Ratio	2,03	
B/D Ratio	9,5	
Immersion:		
Load Increment / Draft	3150,6	ton/m
Metacenter:		
BM transverse	37,25	m
BM longitudinal	95.37	m

5.6 Intact Stability

Stability criteria for intact conditions for the vessel are examined for seven different loading conditions. The calculations are performed according DNV criteria for special purpose vessels which include basic limitations according to IMO 2008 Intact Stability Code , IMO Res. MSC.267 (85)(MSC 2008) covering IACS UR L2(IACS 2013). All calculations are internally performed by the software Maxsurf Stability which has integrated IMO, DNV and IACS rules.

5.6.1 Intact Stability Criteria

According to IMO MSC (267) 85 general criteria, these specific requirements are the ones to be complied with this vessel

Area under GZ-curve shall not be less than:

- 0.055 metre-radians from 0-30°
- 0.090 metre-radians from 0-40°
- 0.030 metre-radians from 30-40°
- Max rightening lever at angle not less than 15°.
- Initial GM not less than 0.15m at Equilibrium.

5.6.2 Loading Conditions

The seven different loading conditions which have been evaluated for this vessel are the following,

- **LC1 Lightship**: The vessel is without consumables and loads except the machinery and piping fluids used at all operating levels.
- **LC2 Ballast Leg**: The vessel has its fuel and its waste water tanks filled plus some ballast percentage in the ballast tanks in order for the propellers to be fully submerged. Thus the vessel stays as lightweight as possible without any loads
- **LC3 Grounding**: The vessel's tanks are filled as much as possible in order to achieve maximum draft capability, without carrying any loads but ballast water.
- **LC4 Sailing with 1 of the largest foundations**: The cargo is place in the deck area almost midship. Fuel tanks are full and waste water tanks. Some ballast tanks are filled in certain percentage in order to achieve 0 heel and a draft conservative enough to submerge the propellers.
- **LC5 Sailing with 2 large turbines**: The cargoes are placed one on deck and the consequent over the cut-out. Same condition as LC4 but with different capacity on ballast tanks in order to achieve the same conditions as LC4.
- **LC6 Offloading of 1 of the biggest foundations**: Cargo is placed over the cut-out, only fuel and waste water tanks are filled and all bow ballast tanks but one, were filled approximately at 80% to achieve equilibrium and stable conditions concerning heel. Trim and positioning issues will be analysed thorough in Section 6.6 with the use of dynamic positioning.
- **LC7 Sailing with 2 of medium foundations**: Cargoes are placed one standing on deck and one hanging over the cut-out area stabilized with fixtures. This case yielded the maximum draft after the grounding case. At this scenario, all bow located ballast tanks were filled to a percentage of 60-70% in order to achieve zero heel and this included a safety margin of 20 cm available distance to reach draft of grounding case.

5.6.3 Equilibrium data for different loading conditions

This result will depict the values of the equilibrium condition and the measured KG at different loading cases. The following table, Table 5.4, will indicate and clarify the attained values of GM and KG for the different cases examined.

Table 5.4. Loading Cases and Maximum KG, GM, Heel values.

Load Cases		Draft (m)	Trim (deg)	GM (m)	KG (m)	Heel (deg)
LC1	Lightship	1.68	0.72	82.2	6.7	0.7 stbd
LC2	Ballast Leg	4,11	-0,04	30,1	8,41	0
LC3	Grounding	6,13	0,05	20	7,2	0
LC4	Sailing w/ 1 largest foundation	4.20	-0.01	23.11	14.6	0
LC5	Sailing w/ 2 largest turbines	4.29	0.04	15.32	21.8	0
LC6	Offloading of largest foundation	5.53	-0.08	19.25	10.5	0
LC7	Sailing w/ 2 medium foundations (1900tn)	5.59	0.08	17.72	11.78	0

The condition that will be evaluated is sailing with two medium foundations, which is the most severe one in terms of weight lifting and ship stability. This case can be described as one foundation of 1900 tons, included the fixture weight, on the deck and one over the cut-out. The thought behind this can be justified by the estimation that if the vessel passes all the required criteria for this case then it will be more than eligible to sail on the other load cases. In table 5.5 the displacements and COG's for all loading condition can be seen. The calculation results for the criterion from section 5.6.1 are presented in Table 5.6. Calculations are performed by Maxsurf Stability. In Fig 5.5 the GZ curve is shown. Hydrostatic calculations are presented in Table 5.7 for offloading conditions which is the worst possible loading case of the vessel.

Table 5.5. Displacement and centres of gravity.

Item	Displacement (mt)	LCG(m)	TCG(m)	VCG(m)
Lightship	2633.28	42.386	1.072	6.695
Displacement	9643.709	43.638	0	11.741

Table 5.6. IMO MSC (267) 85 stability criteria calculations results.

Limit	Min/Max	Actual	Pass
Area from 0-30°	>0.0550mrad	75.54	Yes
Area from 0-40°	>0.0900mrad	85.4	Yes
Area from 30-40°	>0.0300mrad	9.8	Yes
Rightening Arm at 30° or Max RA	>0.0200m	1.81	Yes
Angle from 0° to Max RA	>15.00 degree	15	Yes
GM at Equilibrium	>0.150m	17.72	Yes

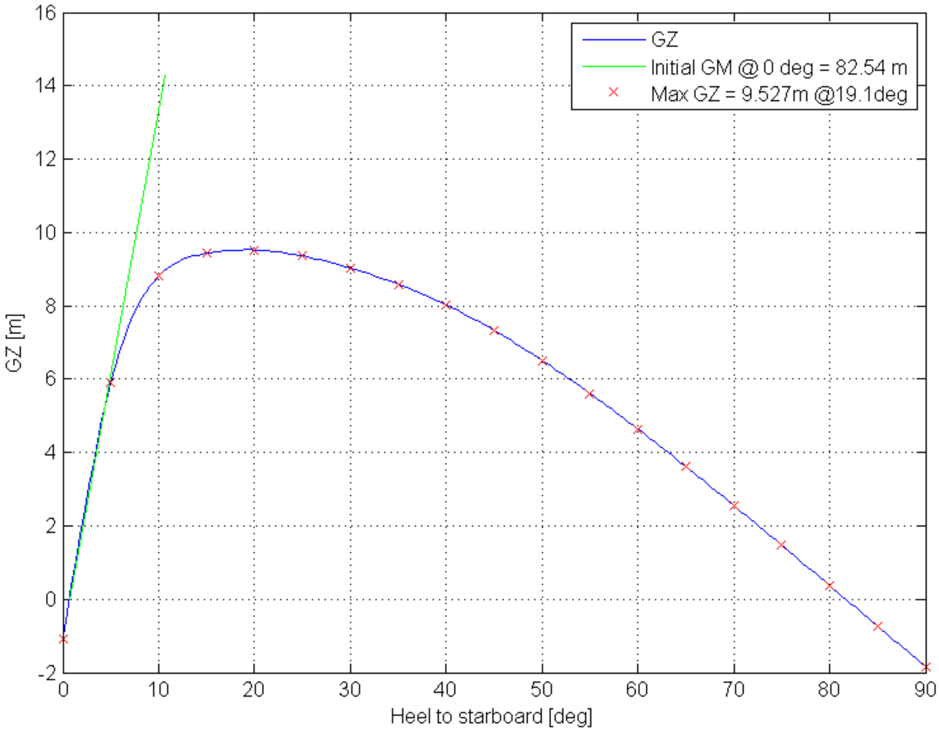

Figure 5.5. GZ curve for lightship condition with Max value at 9.527m at 19.1 deg. including initial GM= 82.54 m at 0.0deg.

Table 5.7. Floating Status for the loading conditions.

Conditions	Lightship	Grounding	Ballast Leg	Sail. 1 large foundation	Sail. 2 medium foundation	Offload. 1 Large foundation	Sail. 2 large Turbines
Draft midship (m)	1.68	6.13	4.12	4.20	5.59	5.53	4.29
Displacement (mt)	2633.00	10646.00	6949.00	7115.00	9644.00	9543.00	7262.00
Heel (deg)	0.70	0.00	0.00	0.00	0.00	0.00	0.00
Draft at FP (m)	1.32	6.13	4.09	4.21	5.55	5.57	4.27
Draft at AP (m)	2.04	6.13	4.15	4.20	5.63	5.48	4.31
Draft at LCF (m)	1.65	6.13	4.11	4.20	5.58	5.53	4.29
Trim (+ by stern) (m)	0.72	0.00	0.06	-0.01	0.08	-0.08	0.04
WL Length (m)	74.20	79.84	79.55	79.58	79.78	79.78	79.58
Beam max extents on WL (m)	37.99	38.00	38.00	38.00	38.00	38.00	38.00
Wetted Area(m^2)	1882.73	3053.62	2555.64	2576.09	2922.46	2902.75	2598.70
Waterpl. Area (m^2)	1677.35	1802.62	1771.18	1774.03	1797.97	1797.92	1776.66
Prismatic coeff. (Cp)	0.54	0.56	0.55	0.55	0.56	0.56	0.55
Block coeff. (Cb)	0.41	0.56	0.54	0.55	0.55	0.55	0.54
Max Sect. area coeff. (Cm)	0.87	0.99	0.99	0.99	0.99	0.99	0.99
Waterpl. area coeff. (Cwp)	0.60	0.59	0.59	0.59	0.59	0.59	0.59
LCB from zero pt. m	42.33	43.66	43.82	43.89	43.63	43.78	43.81
LCF from zero pt. (m)	43.40	43.36	43.20	43.20	43.28	43.28	43.17
KB (m)	0.87	3.16	2.12	2.17	2.88	2.85	2.21
KG fluid (m)	6.70	7.10	8.39	14.64	11.74	10.45	21.84
BMt (m)	88.08	24.10	36.38	35.59	26.58	26.86	34.94
BML (m)	185.11	63.63	92.14	90.47	69.74	70.47	89.17
GMt corrected (m)	82.26	20.16	30.12	23.11	17.72	19.26	15.31
GML (m)	179.29	59.69	85.87	78.00	60.88	62.87	69.55
KMt (m)	88.94	27.26	38.50	37.76	29.46	29.71	37.15
KML (m)	185.96	66.79	94.26	92.64	72.62	73.32	91.38
Immersion (TPc) (tonne/cm)	17.19	18.48	18.16	18.18	18.43	18.43	18.21
MTc (ton.m)	60.50	81.43	76.48	71.12	75.24	76.89	64.72
RM at 1 deg (ton.m)	3780.33	3745.85	3652.67	2870.10	2982.23	3208.34	1940.87
Max deck inclination (deg)	0.91	0.03	0.06	0.01	0.06	0.07	0.03
Trim angle (deg)	0.53	0.00	0.04	-0.01	0.06	-0.06	0.03

5.7 Damage Stability

According to the IMO SPS Code (IMO 2008), the particular vessel is unique and does not abide by in any damage stability conditions, since it is not over 80 m. However, a number of damage stability cases are evaluated, as seen in Table 5.8. The used loadcases and compartments damaged are seen in Table 5.4 and 5.2 respectively.

Table 5.8. Damage Stability Cases

Loadcase	Grounding	Sailing 1 large foundation	Sailing 1 large foundation
Compartment damaged	BW DB P- and S	Collision Bulkhead	BW Stern Port P and BW Stern StB S
Trim	-0.005	-0.233	-1.875
Heel	0.4	0	0
Draft	Damaged 6.501	4.224	4.681

The results indicate that the most extreme damage case is when the stern tanks are damaged. Despite the damage, the vessel remains floating with a trim of 1.875m towards the stern.

5.8 Seakeeping

The task this vessel is designed for sets very high requirements on its movements. In order for the wind turbine to be installed, two flanges have to get fixed within a tolerance of millimetres. The operational limit was a significant wave height of 0.9 m (H_S). Maxsurf Motions was used to perform the seakeeping calculations. This software uses strip theory and panel method to define the hull shape. Unfortunately, the software is not able to handle the catamaran part of the vessel, which resulted in just strip theory being available for this ship.

The approach taken, was that it would be more conservative, concerning heave motion to disregard the moon pool and use a mono-hull for the analysis. Also that would result in a more conservative result than if strip theory would understand the real hull shape. Since heave motion is the biggest concern this analysis was the only one performed.

Figure 5.6 shows heave in meters in relation to speed and heading at the limiting operational sea state. The reference point in graphs 5.6 to 5.8 is at the bottom of the wind turbine, since that is the most critical position during the installation.

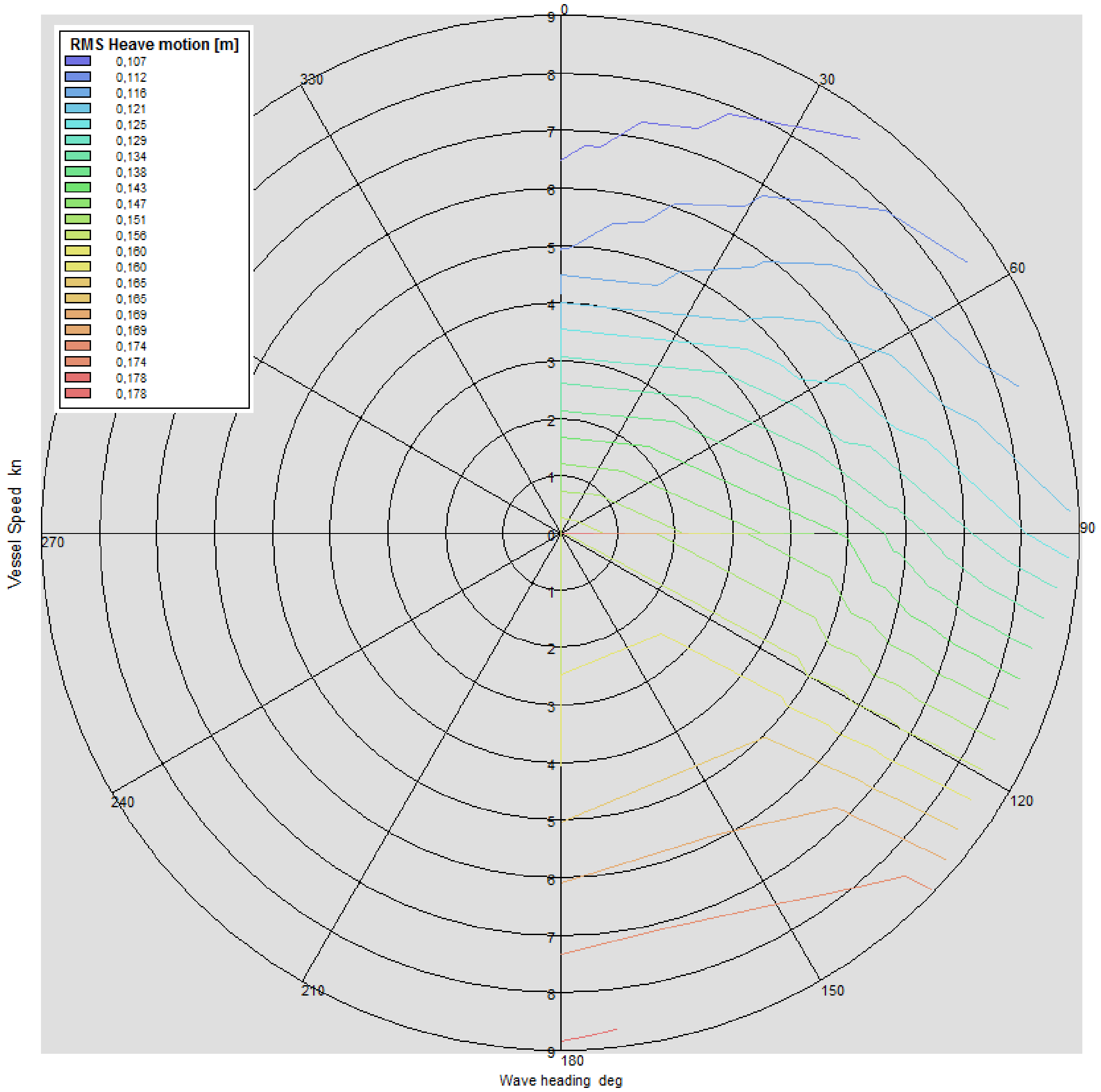

Figure 5.6. Heave at the limiting operational sea state

As can be seen from the graph above the heave will be about 13 cm at the limiting sea state if the vessel is positioned head seas and zero speed. It is known that water plane area contributes a lot to heave motions and with our conservative approach the water plane area is 50% larger than the real water plane area.

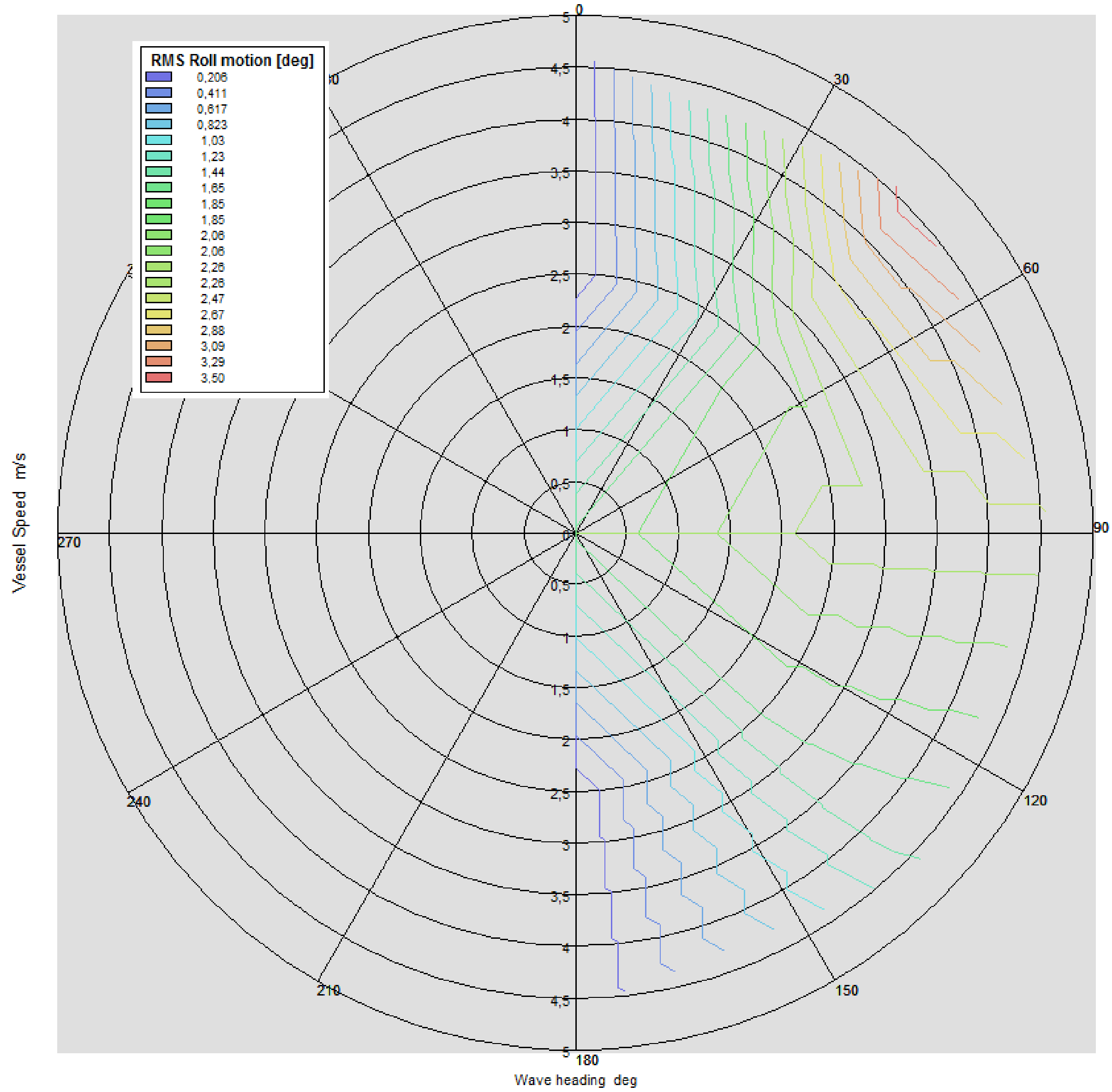

Figure 5.7. Roll motions at the limiting operational sea state

Figure 5.7 above shows the worst case in rolling of the vessel which is if the vessel is sailing in the operational limit with significant wave height of 0.9m. These results are preliminary since strip theory does not recognize the hull correctly.

Figure 5.8 shows the worst case in pitch motions of the vessel if the vessel is sailing in the operational limit with significant waive height of 0.9m. Since the analyses are done on a barge looking vessel other analyses than heave are very difficult to trust. But since GM for the barge looking vessel and the real vessel is similar it can be used to have something about motions of the vessel. Required sea state is very small as well so motions should not be a big concern during other operations than offloading the wind turbine and foundation.

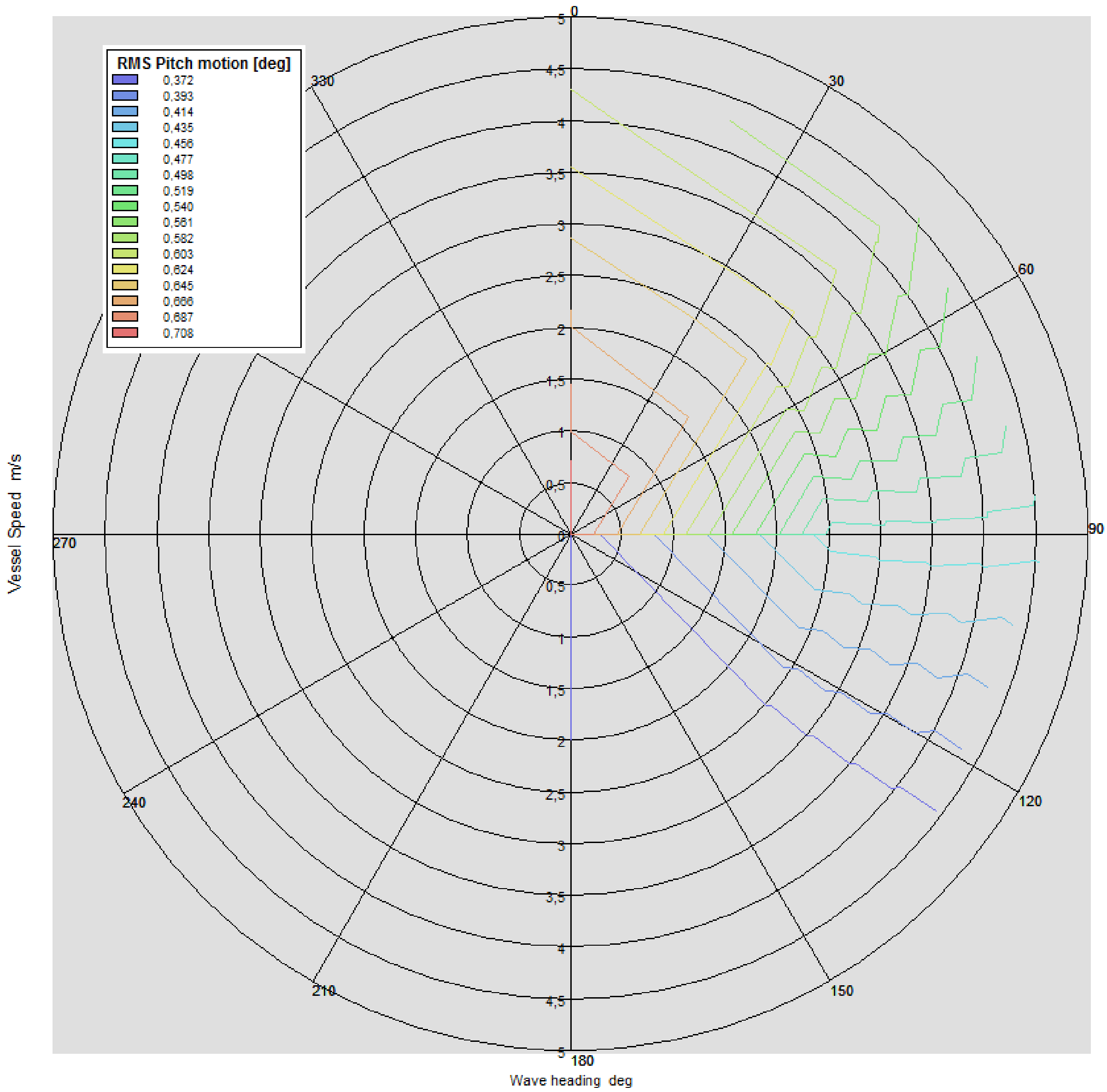

Figure 5.8. Pitch motions at the limiting operational sea state

5.9 Approximate Resistance and Propulsion Power Requirements

The approximate resistance estimation gives an indication to how much power that is needed for the propulsion of the vessel. This required resistance, called total resistance (C_T), is commonly calculated using the procedure of the 1978 ITTC Performance Prediction Method (Larsson & Raven, 2010).

The ITTC-78 method decomposes the total resistance into two contributing parts, wave and viscous resistance. The viscous contribution to the total resistance can be approximated using empirical formulas while the wave making resistance require either data from model tests or CFD simulations. Unfortunately there is no CFD software available, capable of performing accurate calculations within a reasonable time limit, due to the odd shape of the vessel. In addition to this no model testing has been executed, meaning that the wave resistance cannot be measured.

In order to make a qualified estimation of what the total resistance of the vessel would be, the results from the viscous contribution was fitted to typical data from three different ships, a tanker, a containership and a fishing vessel.

5.9.1 Viscous Resistance and Power Approximation

The formula for the viscous resistance according to ITTC-78 is expressed as

$$C_v = (1 + k) \cdot C_f + \Delta C_f$$

Where C_f, is the friction coefficient of the ship, k is the form factor and ΔC_f is the roughness allowance. The power required to overcome the viscous resistance reads:

$$P_v = \frac{C_v \cdot \rho \cdot S \cdot V^3}{2}$$

Where ρ, is the density of the water, S is the wetted surface area and V is the speed of the vessel.

5.9.2 Total Resistance Approximation

For the approximation to be valid, typical data from three different ships are used to estimate the total resistance, taken from Ship Resistance and Flow by Larsson and Raven (2010).

Table 5.8. Typical resistance data from three different ships

Ship type	Froude number	Viscous resistance [%]
Tanker	0.15	7.5
Containership	0.24	17.5
Fishing vessel	0.34	62.5

By matching Froude numbers to the ones from the table above, a total resistance is achieved simply assuming that the calculated viscous resistance have the same percentage distribution as the one in the table. Since there is only three Froude numbers being matched, a polynome of degree four is fitted in order to obtain a graph of the effective power versus ship speed and Froude number, see Fig 5.9.

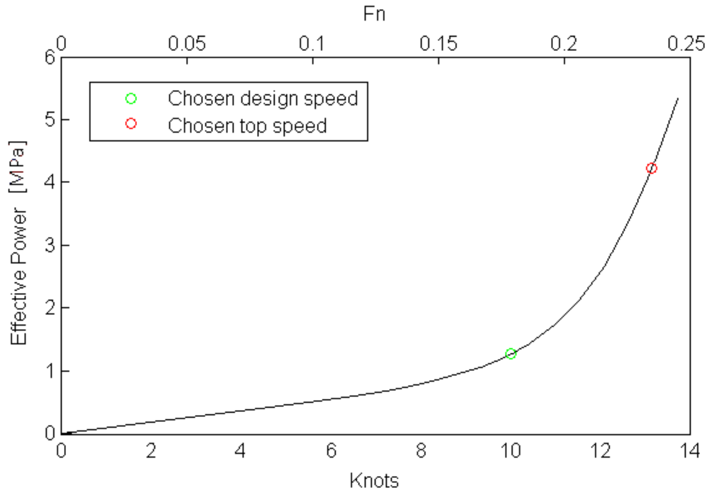

Figure 5.9. Power estimation

The figure is showing the range of relevant speeds for this vessel. Based on this a design speed of 10 knots was chosen, while 13 knots is the top speed.

5.10 Propeller

A propeller for the Azimuth Thrusters was designed and evaluated in the software OpenProp which is a plugin tool for Matlab. Based on inputs such as propeller geometry, ship speed, draft, rpm and required thrust, the program is able to calculate the efficiency of the propeller. The propeller was evaluated for two speeds, 10 and 13 knots, and the results are tabled below in Table 5.9.

Table 5.9. Propeller data

Speed in knots	Rpm	Required thrust (per propeller)	Efficiency
10	150	48.8 kN	67.9 %
13	200	124.6 kN	58.15 %

The efficiency is satisfactory and there is no cavitation on the blades but it should be noted that the inflow to the propellers is basically unknown, meaning that the results are approximate. A schematic picture of the propulsion setup can be seen in figure 5.10 below.

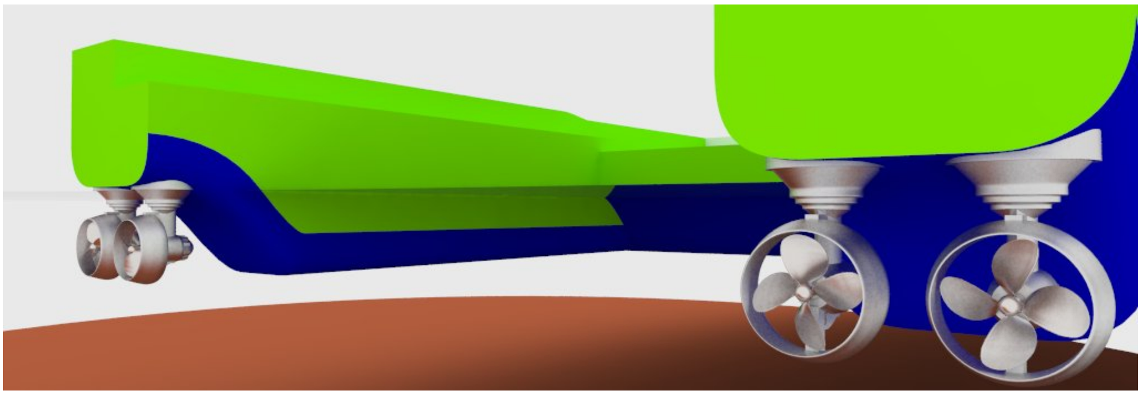

Figure 5.10. Visualization of the propellers

5.11 Future work

Due to the unusual shape of the vessel, a proper CFD calculation was not performed. Instead of that the evaluation was based on estimations from existing data from other vessels. This report therefore lacks a valid resistance calculation. A calculation from a software and not an empirical estimate would be more reliable, or even better a towing tank test. Accordingly, due to geometry input misalignments, the basic seakeeping calculations were performed based only on heave and roll motions. A wave basin test for the seakeeping would be necessary to have accredited results.

More extensive studies concerning the accelerations in nacelle during transit should also be performed, since the seakeeping analyses are not accurate enough. Wave basin test would be preferable to get a very good estimation on all effects on the cargo.

6 Machinery and propulsion system

The purpose of the machinery section is to evaluate a reliable machinery system that fulfils requirements and regulations to as low cost as possible. A conceptual design for propulsion and power supply has been evaluated and decided. The total power demand for the power consumers on-board is calculated and compared to ice class requirement. Machinery system design has been evaluated and chosen with respect to functionality, economy and environmental impact.

The design limitations and criteria, as well as a concept evaluation are discussed in Section 6.1. Special considerations with regards to Ica Class of the vessel are explained in Section 6.2. The electrical power balance is presented in Section 6.3. The discussion on choice of fuel and its impact on the design parameters and the environment are included in Section 6.4. The selection procedure of the main engines is presented in Section 6.5. The main characteristics of dynamic positioning are investigated in Section 6.6 followed by the designed concept including choice and placement of thrusters in Section 6.7. Section 6.8 covers the important ballast system which will be performing the heavy lifting. Auxiliary machines are discussed in Section 0. Machinery on deck including the installation winches and the foldable railing are described in Section 6.10. The details of power consumers on board are studied in detail in Sections 6.11 and 6.12. A preliminary failure analysis is introduced in Section 6.13 and finally the desired task to be performed during future design cycles are briefly touched upon in Section 6.14.

6.1 Design parameters

The main challenges of this design are related to efficiency and power delivery. The propulsion units and propeller diameter are limited by the height of the aft stern slope. Manoeuvring is critical meaning optimization of propulsion units are of highest interest. The Vessel will be grounded which requires a relatively high capacity ballast system. High power consumption in some operation modes and relatively low power consumption in others requires a machinery system operating with high efficiency over a wide range of power demands.

6.1.1 Stakeholder requirements that have influenced the design

The design of the machinery systems and propulsion has been influenced by several requirements from different stakeholders. The main stakeholders and their requirements are presented in Table 6.1 below.

Table 6.1. Stakeholder and their requirements

Ship owner	**Regulations**	**Infrastructure (society)**
Low price	DNV ice class 1C	Depth restrictions in harbour
DP (Dynamic positioning)	IMO Tier III	Supply of fuel

6.1.2 The process of concept generation

The main conceptual ideas that where discussed and investigated during the generating concept procedure where the following

- Number of engines
- Mechanical or diesel electric machinery configuration
- Combination of mechanical and diesel electric machinery configuration
- Pods or shaft driven propulsion
- Electrical or mechanical pods
- Voith Schneider propulsion
- Water jet propulsion
- Tunnel thrusters
- Azimuth thrusters
- Type of fuel

The Voith Schneider and water jet propulsion where eliminated in an early design stage, this due to the depth restrictions when grounding The Vessel and the lack of high speed operations which the water jet is beneficial for.

To be able to comply with the DP (Dynamic positioning) requirement regarding the positioning of The Vessel when installing the foundations and the windmills, a dynamic positioning system needed to be installed. The propulsion system must be able to produce a substantial amount of thrust at the stern and the bow.

To be able to merge the best bow and stern configuration with the best machinery system, the machinery system and propulsion configurations where divided to one aft and one forward part respectively, which were investigated separately, those concepts where then investigated separately. The concepts can be seen in Tables 6.2 and 6.3 below.

Table 6.2. Aft propulsion configuration and machinery system

Machinery system and Aft propulsion configuration	Concept names
Diesel electric machinery with Azimuth thrusters	Concept 1
Diesel electric machinery with mechanical thrusters	Concept 2
Diesel engine machinery with shafted propellers and controllable pitch /rudder	Concept 3
Diesel electric machinery with electrical motors driving a shafted propeller with controllable pitch / rudder	Concept 4

Table 6.3. Forward propulsion configuration

Forward propulsion configuration	Concept name
Electric Pods	Concept 5
Tunnel thrusters	Concept 6

6.1.3 Machinery concept evaluation

To evaluate the different concepts two evaluation matrix where used. The concepts where rated, weighted and ranked separately towards its specific selection criteria where the highest outcome value in the matrix is the best suitable one (D.Eppinger, 2012).

The table below illustrates the rating of the relative performance of a specific selection criterion.

Table 6.4. Ratings of the relative performance

Relative Performance	Rating
Inferior	1
Poor	2
Adequate	3
Good	4
Superior	5

The Selection Criteria for this concept evaluation is the following:

- **Reliability:** It is always important to be able to rely on the machinery system, if a failure should occur. The reason why the reliability criteria do not get a higher importance weight factor are due to that The Vessel will be operating 3nm SE of the maintenance station at Hanö.

- **Running costs**: There are several costs that is needed to be covered, the fuel cost, cost of maintenance, crew and the capital cost of the investment. If those costs can be as low as possible, some of the revenue from the project can be re invested in The Vessel or in other equipment and the enterprise will be commercially attractive.

- **Investment costs**: The investment cost is an important issue; if it is too high the ship owner won't be able to invest in The Vessel.

- **Manoeuvrability**: The manoeuvrability is one of the most important criteria for The Vessel. It has to cope with the dynamic positioning system when installing the foundations and the windmills as well as it has to be easily manoeuvrable in port.

- **Maintenance:** A machinery and propulsion system will need maintenance; it is beneficial to keep the need of maintenance as low as possible since it will be an extra cost for the ship owner. The human factors should always be kept in mind when design the engine room to make the maintenance work of the machinery as ergonomic and safe as possible for the crew.

- **Complexity:** The complexity of the system should be kept as low as possible, this to minimize the possibility of a failure.

The evaluation matrices, Table 6.5 and 6.6, where the concepts where rated, weighted and ranked are presented below.

Table 6.5. Evaluation matrix

Evaluation matrix of the aft concepts									
Concepts		Concept 1		Concept 2		Concept 3		Concept 4	
Selection Criteria	Weight	Rating	W.S[1]	Rating	W.S	Rating	W.S	Rating	W.S
Reliability	5%	5	0,25	1	0,05	4	0,2	2	0,1
Running costs	5%	2	0,1	1	0,05	2	0,1	2	0,1
Investment costs	20%	1	0,2	3	0,6	3	0,6	2	0,4
Manoeuvrability (DP)	50%	5	2,5	5	2,5	1	0,5	1	0,5
Maintenance	5%	5	0,25	2	0,1	3	0,15	2	0,1
Complexity	15%	4	0,6	2	0,3	3	0,15	1	0,15
SUM	100%	**3,9**		3,6		1,7		1,35	

[1] Weighted Score

Table 6.6. Evaluation matrix

Evaluation matrix of the forward concepts						
Concepts		Concept 5		Concept 6		
Selection Criteria	Weight	Rating	W.S	Rating	W.S	
Reliability	5%	5	0,25	5	0,25	
Running costs	5%	2	0,1	3	0,15	
Investment costs	20%	1	0,2	4	0,8	
Manoeuvrability (DP)	50%	5	2,5	4	2	
Maintenance	5%	5	0,25	5	0,25	
Complexity	15%	4	0,6	5	0,75	
Total	100%	3,9		**4,2**		

Based on the evaluation of the selection matrix, discussing within the group and with external experts one conceptual configuration were chosen. The chosen machinery system best fitted for The Vessel and its operational profile is a Diesel electric machinery system with Azimuth thrusters in the stern and tunnel thrusters in the bow. By applying this combination of the best suitable concepts, The Vessel will get highest possibilities manoeuvrability and a reliable system with a low complexity to the lowest possible operational lifetime costs.

6.2 Ice Class

The Vessel shall meet the Swedish Finish ice class 1C. The requirements are given by DNV report (DNV, Ships for Navigation in Ice, 2013). The requirements cover sensitive machinery systems to maintain propulsion and steering when operating in ice or ice brass described by ice class 1C. The class regulates important units in the propulsion and steering system. All components must comply with the ice class 1C requirements.

6.2.1 Ice class engine power requirement

For ice class 1C the minimum installed power cannot be lower than 1000 kW. To fulfil requirements for ice class 1A Super, 1A, 1B or 1C, the ship needs to deliver the power, P, given by equation 6.4.1 on the propeller shafts. The input parameters of K_e and R_{CH} depends on ice class level and are calculated using different properties, constants and calculation methods.

$$P = K_e \frac{(\frac{R_{CH}}{1000})^{3/2}}{D_p} [kW] \quad \text{(eq. 6.4.1)}$$

K_e is given by propulsion system properties. D_P is the propeller diameter. R_{CH} is the ship resistance in ice brass channel. The resistance R_{CH} is based on the ship properties, angles for breaking ice, areas and lengths that are of interest when operating in stated condition, given by (DNV, Ships for Navigation in Ice, 2013). The equations for calculating R_{CH} are semi-empirical meaning there is a validity range given for ship properties as well as relations between ship properties. If the properties of The Vessel are invalid there are two ways to continue. Either DNV accepts a widened range for ship properties, or ship resistance can be model tested in an ice brass channel. For calculating R_{CH} the constant C_a needs to be calculated given by the ship properties:

$$C_a = (Length * Draught / Breadth^2)^{3/2} \quad \text{(eq. 6.4.2)}$$

The range for the value of C_a given by eq. 6.4.2 is from 5 to 20. The value for this vessel is 0.0283. The properties for eq. 6.4.2 are not valid for this specific vessel according the given range. This means that the resistance for the ship in ice condition might be incorrect. The Vessel is considered as a concept and therefore the equations for ice resistance will be accepted at this stage. A more accurate value will have to be estimated together with DNV for a final vessel. The calculation resulted in a demand on the propeller shaft delivering 5650 KW to fulfil the requirements for Swedish-Finnish ice class 1C.

6.2.2 Ballast requirements

In order to attain ice classification for the ballast system the following arrangements must be made (DNV, 2013b):

1. According to DNV ice class rules, ship side ballast discharge valves and ballast tanks located partly or fully above the designed lightest load line must have a heating arrangement to prevent freezing. If the tank is situated partly above the designed lightest load line, an air-bubbling arrangement or a vertical heating coil, capable of maintaining an open hole in the discharge valve can be used.
2. To ensure supply of water to the sea chest when navigating in ice, the sea inlet shall be placed near the centre line of the ship as far aft as possible. The inlet grids shall be specially strengthened.

6.3 Electrical power balance

In the electrical power balance, the power consumption of the main electrical power consuming components was identified. The Electrical power requirement has been investigated for the following operational modes.

- Port Loading
- Ballast/Deballast
- Transit to site 7 knot / Transit to port
- Positioning
- Inst. Windmill
- Inst. Foundation
- Ice Condition
- Break
- Emergency

The consuming power at the different modes has been developed through calculations regarding that specific component, by engineering assumptions comparing data with similar type of vessels and by discussing with experts within the area.

To insecure that no power blackout will occur during any of the operational modes the peak values for each mode has been the value that has been taken under consideration. For the complete electrical power investigation see Appendix In the table below the peak values for each operational mode is showed.

Table 6.7. Peak loads [kW]

Port Loading	Ballast / Deballast	Transit to site 7 knot / Transit to port	DP	Inst. Windmill	Inst. Foundation	Break	Ice Condition	Emergency
482	5897	3059	4672	4653	5320	743	6996	292

6.4 Selection of fuel

The main purpose with the wind farm project is to install renewable energy for a long-term environmentally sustainable production. The ship operation area, the Baltic Sea, is considered sensitive for environmental impact therefore it is adapted to the ECA (Emission Control Area), by IMO. The ship needs to fulfil these regulations.

6.4.1 Environmental regulations

NOx emissions are regulated in MARPOL Annex VI (IMO, 2013a). The regulation is valid for all diesel engines with installed power of more than 130 kW. This vessel will have more than 130kW installed power. Today there are three levels of NOx emission Tier I, Tier II and Tier III. New ships built after 1 January 2016 shall meet Tier III when operating in the ECAs, which will be the requirement for this vessel.

SOx emissions are regulated by MARPOL Annex VI (IMO, 2013b). Burning fuel with sulphur content creates SOx, The regulation applies to the level of sulphur content in the fuel. Valid since January 2012 the limit for operating within ECA is 1.00% sulphur in the fuel, by weight. The allowed sulphur content will decrease to 0.10% by January 2020. Alternatively an exhaust gas cleaning system can be installed to reduce the SOx emissions to the equivalent level. In the design of The Vessel low-sulphur fuel will be considered. All requirements will be fulfilled by designing the machinery system for fuel within the approved range.

6.4.2 Fuel types

When selecting fuel one must realize carrying fuel is in essence a way of storing energy that is needed to enable the engines to produce power. The overall operation goals are to minimize cost, impact on environment as well as on human health. In addition one must evaluate if supply will be sufficient in order to secure availability in various situations and over time. The storage on board needs to be safe and preferably kept on a low complexity level to meet all requisites. Different fuels have varying properties and the goal is to find the optimum solution for this vessel.

The prerequisites given are amongst others:

- Sulphur restriction of fuel
- NOx emission regulations
- Fuel distribution infrastructure in southern Sweden

In a first evaluation traditional marine fuels were investigated. The prerequisites limited the selection of fuels and the tree most interesting fuels was to be compared. The remaining fuels see Table 6.8, comply with the sulphur restrictions of maximum 0,10% sulphur content by weight within the ECA (Emission Controlled Area). The fuel consumption represents 18% of the running costs for the vessel, see Section 4.10. The fuel prices in Table 6.8 are calculated from fuel properties and fuel prices in Gothenburg.

Table 6.8. Fuel properties

Fuel	Price SEK/KWh	Shore infrastructure complexity	On-board fuel system complexity	Availability
MGO	0.50	Low	Low	High
Methanol (fossil)	0.44	Low	Medium	Medium
LNG	0.46	High	High	Low

Price of MGO (Marine Gas Oil) has fluctuated during a period of time and is considered low. The availability is high; MGO is the most common fuel around the area of operation. MGO is a liquid fuel with no need for pre-treatment. The fuel system is of low complexity compared to methanol and LNG.

MGO is a fossil fuel. There are renewable alternatives available but because of bacteria growth in the fuel tanks, renewable diesel is not always a sufficient alternative. MGO can comply with required NOx regulations without an after treatment system. However a SCR (Selective Catalytic Reduction) system is to be preferred in order to optimise the engine for high efficiency and not for low NOx emissions.

The advantages using methanol is fuel price and low environment impact. Methanol is a clean fuel complying with MARPOL NOx and SOx regulations, (Stenhede, 2013). It is also the most cost effective fuel considered price per energy unit. Methanol is widely used fuel for industries, making the availability and distribution sufficient for ship fuel application, especially in the region of operation. It can be produced from wood as a renewable fuel. This means the ship owner has the possibility to run the ship on a clean fuel with no CO_2 contribution. However most of the produced methanol on the market today is refined from fossil fuel.

Methanol is a liquid fuel meaning storage on shore and on board is of similar complexity level as MGO. Methanol energy density is 20 MJ/kg while energy density for MGO is 42 MJ/kg, this means the ship will consume 2.1 times as much fuel by weight in methanol mode compared to MGO mode. A direct affect is higher fuel consumption by weight compared to MGO. The engine technology is known, see reference vessel Stena Germanica.

The operational cost for LNG (Liquefied Natural Gas) is in the lower range, comparable with methanol and MGO. The shore infrastructure is insufficient in the southern parts of Sweden. It is possible to have it delivered, but it will affect the overall cost.

The combustion properties are of high quality. The fuel is clean, complying with all requirements without pre- or after treatment systems. LNG is a gas that is delivered and stored as a liquid in a pressure close to atmospheric. This storage requires a fuel temperature of approximately – 160 C that makes the fuel system highly complex compared to MGO or methanol. The fuel is common on the global market and the technology for engine and sub systems are established.

The IGF code is a regulation given by the IMO for gas and low flashpoint liquid fuels. The code regulates the technical requirements for security on-board with philosophy to minimize risk to the ship, crew and the environment. The first draft of the updated IGF code, including alcohols and LPG fuels, was approved by IMO and SOLAS on the MSC 94 meeting in November 2014. Methanol and LNG is considered as low flashpoint fuels meaning it will be treated under the IMO regulation IGF code. Areas where the regulation will affect the fuel systems are fire protection in engine room, safe packaging of fuel tanks and fuel piping.

Given these prerequisites LNG is dismissed due to the lack of infrastructure in southern Sweden and the complexity needed of the fuel system, on shore and on board.

This vessel is designed for dual fuel system allowing MGO (Marine Gas Oil) and Methanol operation. The main fuel is methanol. The technology for methanol combustion requires a low content of MGO acting as a pilot fuel; this is why a second fuel system is needed. The advantages of using two fuel systems are the availability for fuel and possibility of operate on the most cost efficient alternative. The additional fuel system increases weight and investment cost.

6.5 Selection of Main Engine

The engine combination is configured and optimized for low fuel consumption and to match power- and emission requirements.

The operation power demand and the installed power requirement is given from the engine load diagram, see Fig 6.1. This figure illustrates the different load cases where the power demand range spans from port loading of 800 kW to 7080 kW. The peak demand of 7080kW represents the required installed power given by ice class regulation plus the additional active power consuming systems in transit operation.

The ice condition should be considered as a criterion for minimum installed power. The other operation modes illustrated in Fig 6.1 represents power demand during normal operation. The requirement for minimum installed power is reached when all engines are operating simultaneously. Power demand during normal operation varies from 800kW to 5700 kW, see Fig 6.1.

The engine load illustration is based on peak values for the systems in operation. This means that the power required in each operation mode is dimensioned and illustrated with all systems active. In reality however this will not be the normal case, meaning the power demand from the engines will be lower than indicate.

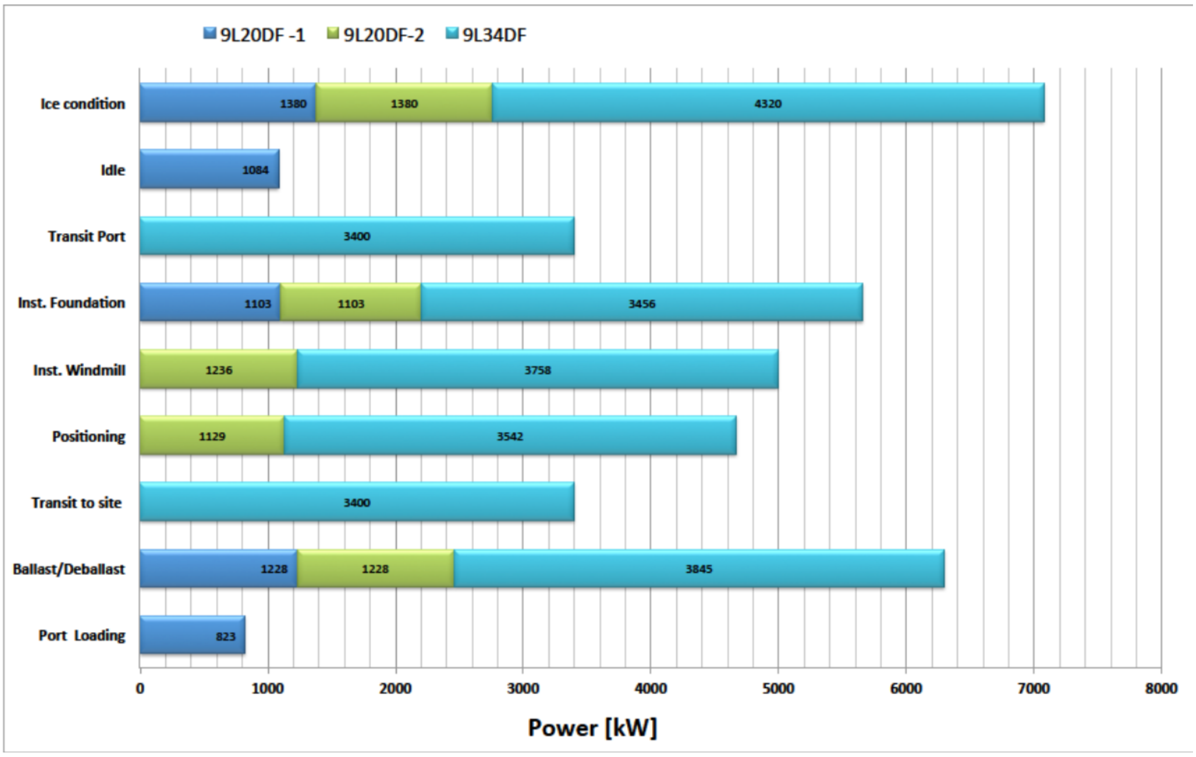

Figure 6.1. Power requirements

To dimension the engines it is important to analyse the most efficient operation load. The engines are most fuel efficient when operating at higher loads; it is common to operate an engine at 70-90% of maximum load. See Fig 6.2 and Fig 6.3 for fuel consumption compared with engine load (Wärtsilä, 2014).

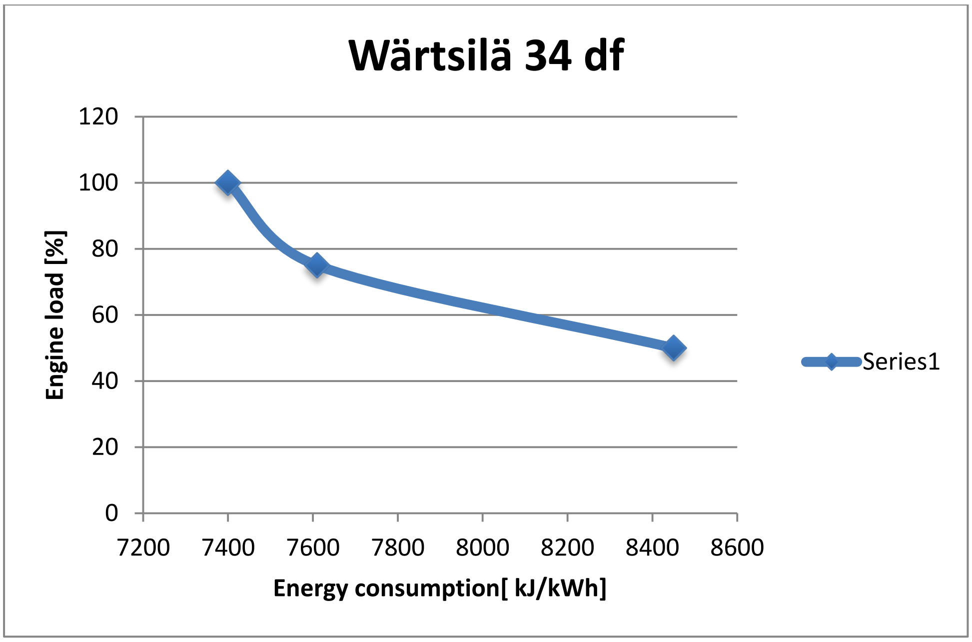

Figure 6.2. 34 DF Energy Consumption

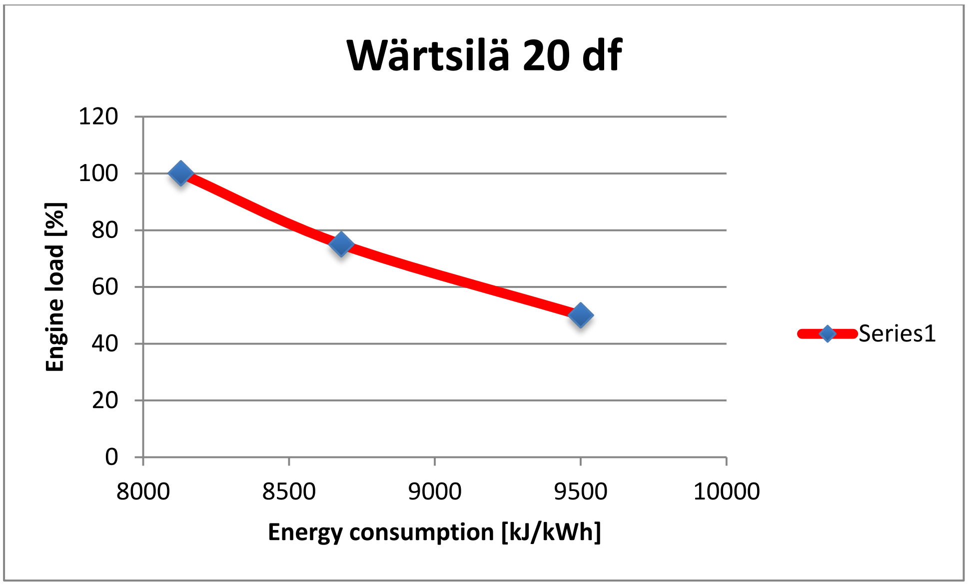

Figure 6.3. 20 DF Energy Consumption

The engine layout is three power units, two smaller and one larger generator sets, see Table 6.9. This configuration satisfies a fuel-efficient combination of active engines for all power demand cases. During transit the only activated engine is Wärtsilä 34 DF, during port idling the only activated engine is one Wärtsilä 20 DF.

Table 6.9. Engine selection

Manufacturer/model	Numbers installed	Max. engine power [kW]	Max. generator power [kW]
Wärtsilä/ 9L20DF	2	1440	1380
Wärtsilä/ 9L34DF	1	4500	4320

There are a few manufacturers on the market offering dual fuel technology for marine diesel generator sets within the given power range; the two leading companies are Caterpillar and Wärtsilä. Wärtsilä participate in the first methanol conversion project, Stena Germanica, the research is considered valuable. The engine supplier for this vessel is Wärtsilä.

Wärtsilä 34DF and Wärtsilä 20DF are medium speed four stroke engines. DF stands for "dual fuel" and means they are optimized for alternative fuels such as LNG combined with MGO or HFO. Methanol is not included in the product description. The engines are used as reference engines to obtain data for fuel consumption and power output.

The fuels MGO-diesel and methanol have different properties of interest for combustion, see Table 6.10 (Heywood, 1988). The octane rating for the fuels can be discussed since there are different ways of measure but when comparing MGO and methanol there is a clear difference. A low octane number describes how easily ignited the fuel is. The MGO has a lower octane rating than methanol and it therefore more easily ignited.

Table 6.10. Combustion properties comparison between MGO and Methanol

Fuel	MGO diesel	Methanol
Lower Heating Value MJ/kWh	42	20
(Air/Fuel) Stoichiometric	14,6	6,5
Auto ignition temperature C	210	460
Flash point temperature C	74	11
Octane	15-20	~105
Flame speed m/s	0,38	0,52

For a methanol engine the MGO is used as a pilot fuel, a more easily ignited fuel to start the chemical reaction. The cylinder head therefore has one additional injector, one for methanol and one for the pilot fuel oil. See figure 6.5.6.

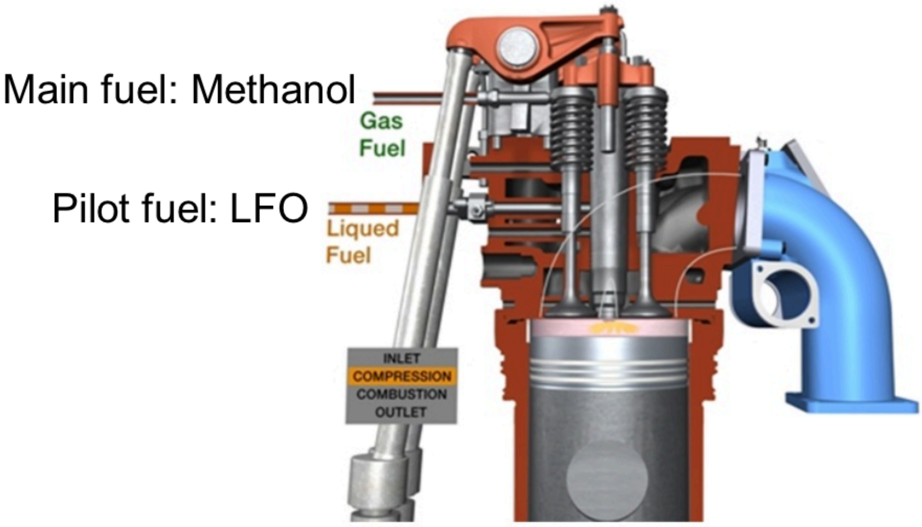

Figure 6.4. Fuel injection system (Used with permission from Wärtsilä Finland Oy)

The engine Wärtsilä 34 DF is based on a Wärtsilä 32. The number 34 respectively 32 refers to the bore of the cylinder in cm. It is given that the stroke is kept constant at 400 mm for both engines, meaning the cylinder volume is increased for 34 DF. The power output is kept constant and the mean effective pressure is lowered 22 bar for the 34 DF respectively 24.9 bar for the low power version of 32 DF, (Wärtsilä, 2013) and (Wärtsilä, 2014b).

Burning time and burning temperature are key factors for NOx formation. Lower mean effective pressure gives lower peak combustion temperatures. The flame speed is higher for methanol, Table 6.10, and indicating shorter burning time. Shorter burning time and lower peak temperature is beneficial for low NOx-emissions.

The energy density and the air fuel ratio for stoichiometric combustion for the two fuels varies, see Table 6.10. It can be seen that the airflow is almost constant for the two fuels. The energy density is lower for methanol, and therefore the mass flow of fuel needs to increase by 2.1 times for the methanol compared to MGO. No modification is needed for the air intake system however the methanol fuel system needs to be configured for higher capacity than the MGO system.

A power management system is installed on the vessel. The system automatically compares available power with the power consumers. If the system detects a request for more power, it increases power generated by either increase the load on the engines or start additional engines. In a situation where the power generated isn’t enough for satisfying the power consumers, the system will prioritize the power delivery to important functions and limit the power to other. This means a blackout will never occur for important systems.

6.6 Dynamic Positioning

The objective of a DP system on a vessel is to maintain her heading and position by use of available thrusters. There are no heave control requirements on the DP system as it is generally unconventional to navigate the vessel vertically except in some research applications. This is often achieved by a combination of equipment such as computer controls, environmental sensors and the propulsors. IMO DP Classification rules according to publication 645 states that a DP1 notation has no redundancy. This means that loss of position may occur upon a single failure. Motivated by the close distance of operation sites to the coastal facilities, the DP1 is deemed satisfactory, but some redundancy capacity is available on-board which allows the vessel to continue DP operation, even after certain failures have occurred (See Failure Mode and Effect Analysis, Ch. 6.6.3.

6.6.1 DP subsystems

The typical dynamic positioning system will be comprised of four subsystems: Sensor system, Control system, Thruster system and Power system (Virk, et al., 2000). The particulars of these subsystems will be explained in more detail below.

Sensor system

The sensor system is responsible for mapping of the area as well as measuring environmental forces such as wind, wave and currents. The design condition is often a storm. However, in the operational definitions of the vessel a moderate situation of Sea State 3 $(H_S = 0.9m/s)$ and maximum wind velocities of 12 m/s are required by the client. Current velocities are a function of, among other factors, the wind velocity. This makes it possible to estimate overall forces by only considering the wind speed during design stage (See HSSC). Additional swell and wave information is however crucial when operating the DP system, which is fed from the sensor system.

Selection of the sensor system is dependent on the level of accuracy desired as well as the working environment of The Vessel including but not limited to water depth, weather severity, etc. For the present project, a combination of three different sensory systems is proposed in consultation with the Kongsberg Maritime department. This includes a light taut wire crane, see Fig 6.5, suitable for the medium to shallow water depth operating conditions of the vessel, a DGPS system complete with carefully located triangulation antennas, and finally a hydro acoustic system. These three systems will complement each other and not act in redundancy. The referencing system is an on-board gyrocompass.

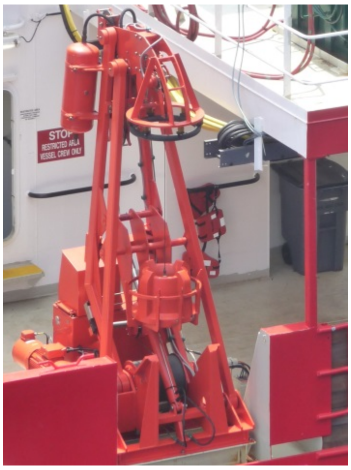

Figure 6.5. Light Taut Wire Crane (CC-BY-SA License by BoH)

Controller system

The controller system is often purchased as a whole from manufacturers. The controller system suppliers often provide their own assessment of the power, thruster and sensor systems in order to propose an optimized solution. The DP control console is positioned inside the bridge for better visibility, but may as well be placed anywhere in the ship even if there is no visibility. Employment of a DP officer according to the standard DNVGL-ST-0023 (Competence of dynamic positioning operators) is crucial in delivering the most out of the capabilities of the system.

Since the controller system is responsible for mapping the area and keeping track of forces on the ship it is crucial that the power feed to the recording devices will resume after a total power shutdown on the ship. This is to ensure that the mapping continues, until the main power comes back on. The Uninterrupted Power Supply should be sized to last for at least 30 minutes.

Thruster system

When selecting an adequate thruster system, the basic requirement is that the vessel should be able to maintain heading as well as transverse station-keeping during design weather conditions. Since the same thruster system is responsible for transit of the vessel which may be conflicting with the DP requirements, a reasonable compromise has to be made. The system is thus comprised of four aft azimuth thrusters, plus two bow tunnel thrusters capable of yawing the ship around any vertical axis along the centre line as well as performing manoeuvres such as crab and reverse.

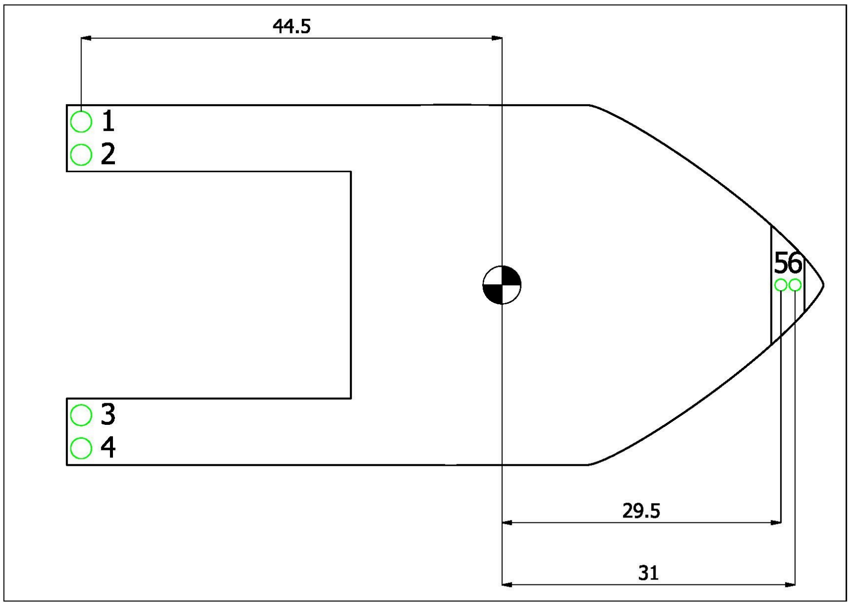

Figure 6.6. Transverse Thruster Configuration

Power system

A DP system is expected to efficiently respond to relatively rapid changes in the weather conditions. This makes the sizing of the power plant an optimization problem. A diesel-electric power plant is beneficial in that it allows for each thruster to take just enough power as is required at different times by making use of a well-designed switch board. Power prediction for a DP system is often complex and challenging. The different influencers on the station keeping capability of a vessel are often wide-ranged and overlapping. From wind effects to wave and current, and the numerous combinations of these variables it is possible that the worst design conditions are actually never met in operation. Then there is the possibility that the DP system proves to be inadequate under a condition that was deemed to be insignificant compared to design conditions. This was the case with the Discoverer Seven Seas (Steddum & Herrmann, 1997). It is therefore preferable to invest in extensive simulations and model experiments on the sea keeping and station keeping behaviour of the ship, which is at this stage prohibitively costly and time consuming.

6.6.2 DP Performance Evaluation

Various estimation methods are being implemented in the industry to make safe assumptions in order to calculate the power necessary for keeping of heading and position of vessels. Some are classification societies requirements, and others common practice. With the exception of the ern numbers, the methods presented herein are all valid for preliminary calculations. However, due to the computational or experimental limitations of these methods they may not be applied until a later stage of design where test facility models and elaborate computer simulations are developed.

ern

In accordance with class notation requirements of DNV-GL Pt.6, Ch.7, a DP capability criterion known as Environmental Regularity Numbers or ern is evaluated (DNV, 2013b). The ern is based on different performance indicators representing the least and most severe failure modes. Balance of forces and balance of moments under these failure modes shall be maintained for evaluating the ern. This method requires full scale sea trials and is not an estimation method, but rather a class notation requirement that should be performed upon completion of vessel construction.

HSSC

The Howard Shatto Sanity Check (HSSC) number is a dimensionless ratio for DP system performance. The general idea is that 80% of the available thrust on the ship should be equal or greater than the forces exerted on the vessel in a 61 knot beam wind. An HSSC value of greater than 1 is considered satisfactory (Steddum & Herrmann, 1997). In order to calculate the HSSC value for the vessel, a hydrodynamic computational model is required, which is yet to be developed for this unconventional hull form.

Transverse Speed Criteria

A more absolute way of deciding whether a ship is capable of handling the forces at sea, is assuming a wind driven beam current of 1.0 – 1.5 knots and using the reasoning that as long as the vessel is capable of doing 2 or more knots of transverse speed, then it should be able to handle the sea currents it encounters. This measure is named Transverse Speed Criteria (von Ubisch, 2004). As effective as this criterion is, the fact that it requires full scale measurements in real sea conditions makes it an impractical approach at this stage of design. The general concept of Transverse Speed Criteria however, is used in the novel approach of TTTI, which is described below.

TTTI

The cheapest, and coincidentally simplest, available estimation method is finally introduced here as the Thrust to Transverse Area Indicator or TTTI (Herdzik, 2013).

$$TTTI\left(\frac{kN}{m^2}\right) = \frac{T(kN)}{L(m).D(m)}$$

Where:

T= Maximum available transverse thrust
L= Vessel length at water level
D= Vessel draft in worst DP operating mode (Maximum operational draft)

The criterion is 2 kN per unit transverse surface area [m^2] which will as a rule of thumb fulfil the minimum 2 knots transverse speed.

$2\frac{kN}{m^2} = \frac{T}{80 \times 6.2} \Rightarrow T = 992kN$ Required transverse thrust

This value is used as the required minimum transverse thrust when selecting thrusters.

DP power to displacement ratio

Another indicator of the station keeping performance of a vessel is the ratio between DP power and tons of operating displacement. It is then beneficial to compare the ratio for similar existing ships where the larger ratio represents better power availability during handling of situations such as hurricanes, sudden squall and eddy currents (Virk, et al., 2000).

Table 6.11. - A comparison of power to weight ratio with similar vessels

Vessel Name	Vessel Type	DP/DWT [kW/Tonnes]
The Vessel	Special Purpose Vessel	4.25 MW/7983 Tonnes=0.53
OSA Goliath (DP III)	Offshore Construction Vessel	10 MW/35600 Tonnes=0.28
MV Lone (DP II)	Heavy Cargo Ship	5.6 MW/12500 Tonnes=0.448
Seven Borealis (DP III)	Crane Vessel	5.7 MW/47000 Tonnes=0.12

It would appear that the vessel has similar DP power compared to similar sized vessels and considerably more compared to larger vessels. Thus, the power is deemed to be sufficient

6.6.3 Failure Mode and Effect Analysis (FMEA)

A detailed analysis of the failure modes and their impact on the performance of the DP system is required in accordance with DNV-RP-D102 and in more detail with DNV's Rules for Classification of Ships Part 6, Section 7.

A calm water analysis of various thruster failure modes is presented here as an overview, not to be confused with the DNV-GL required FMEA documents. For thruster notation refer to Fig 6.9.

Table 6.12. Thruster Failure Mode Analysis

Failure Mode	Maintain Surge Control	Maintain Sway Control	Maintain Yaw Control
Single failure of 1,2,3,4,5 or 6	YES	YES	YES
Double failure of any combination of 1,2,3 and 4	Possible Heading Control Reduction*	YES	YES
Failure of 5+6	YES	NO	YES
Double Failure of (1,2,3,4)+(5,6)	YES	YES	YES
Triple failure of any combination of 1,2,3 and 4	Possible Heading Control Reduction	YES	YES
Failure of 1+2+3+4	NO	NO	Possible Yaw Control Reduction**

* If both failed azimuths are on the same pontoon.
** If speed is above tunnel thrusters effective speed of 2-3 knots.

6.6.4 Closing Remarks and Design Considerations

The loading conditions upon the final phase of deploying the heavy cargo are likely to cause unpredicted instability in the DP control system. This is one motivation why this stage is best handled manually by experienced DP officers. Additionally, the more number of thrusters available the more control the DPO will have over the situation, which may motivate during future development the addition of 1 or more transverse thrusters, possibly near the bow.

6.7 Propulsion

The propulsion system design is driven primarily by Ice Class requirements, succeeded by hydrodynamic thrust requirements while considering DP performance throughout the design.

All elements of the drivetrain need to satisfy the Swedish-Finnish Ice Class rules from the power plant to the propeller blades and everything in-between (Transportstyrelsens, 2009) The power that is required by the Ice Class notation requirements is larger than the power required from hydrodynamic calculations. This means that for normal operating mode (non-ice breaking) there should be some measure to reduce propulsion power output without sacrificing fuel efficiency. To that aim either a variable rpm electric motor or Controllable Pitch Propellers or a combination of both is needed. The fast response of CPP will somewhat alleviate the capital costs. Operational costs for CPP are greater than that of fixed pitch propellers due to more machinery. Variable rpm is in general not significantly more complex as it only relies on a variable frequency electric motor, which is readily available.

The slowest steaming, if required, will be made possible by using only two of the aft thrusters.

6.7.1 Azimuth Thrusters

Four aft azimuth thrusters are responsible to deliver the necessary forward thrust and steering while in transit mode. Furthermore these aft thrusters will contribute to the position keeping of the vessel while operating on site using the DP control system. IMCA states that *"Stern thrusters may be sized for transit speed and may operate at a fraction of their rating on DP"* (IMCA, 2010). Naturally the azimuth thrusters will be operating at a lower point than their rated power during DP operations. The performance of the aft thrusters under DP mode may be enhanced if a power management system is implemented in the drivetrain.

In order to power these thrusters an electric motor and gearbox combination (L-drive) will provide the required torque to the propeller shaft. A combined electric to mechanical efficiency of 94% is assumed when calculating required electrical input.

According to manufacturer specifications (See Appendix C), the indicated powers are short time allowed maximum input powers. The selected thrusters allow for a range of 1300-1600 kW input power, placing the desired 1500 kW well within the operating envelope while avoiding the high rpm values.

In many azimuth thruster arrangements, the design suffers from so-called forbidden zones. This is defined as the operating directions in which two thrusters will face each other and flush each other's suction side, reducing the performance drastically. In the present case the unconventionally wide breadth of the vessel, combined with the presence of the moon-pool along the centre is considered to resolve this issue to a great extent. However, careful hydrodynamic investigation of the phenomena is strongly advised for future stages of development. Each pair of azimuths on the pontoons will operate in contra rotating directions in order to reduce the paddling effect, while rotating the inflow for their respective counterparts. This will effectively improve the performance of the azimuth operating downstream from the other, similar to the principle of counter-rotating propellers.

There exists a different kind of forbidden zone in this case and that is the directions in which the azimuth thrusters will create jets of stream towards the cargo while it is being lowered into the water. In order to take this effect into account, the required transverse thrust for the position keeping of the vessel during DP operations was calculated for 2 out of 4 available azimuths. This was done in order to eliminate the effect of the thrusters that are operating upstream from the hanging cargo. Failure to consider aforementioned effects will result in possible heeling or more importantly oscillation of the hanging cargo into a hazardous resonance.

6.7.2 Tunnel Thrusters

The responsibility of the four tunnel thrusters positioned near the bow is to provide transverse thrust both for DP operations and berthing. Furthermore these have been sized and located in order to provide yawing moments around the centre of floatation of the vessel for heading control.

Equipped with the azimuth thrusters, the vessel already has the capacity to produce 1080 kN transverse thrust (two aft-thrusters, see the discussion on forbidden zones in Section 6.7.1). Two tunnel thrusters with the capacity to produce a combined 1192 kN are fitted at the bow in order to provide the DP control system with the required yawing moments for heading control against side seas.

Tunnel Thruster Dimensioning

In order to size the tunnel thrusters, equilibrium of moments calculation is performed.

$$\sum M = 0 \rightarrow T_1 \bullet d_1 + T_2 \bullet d_2 = T_5 \bullet d_5 + T_6 \bullet d_6$$

Where:

T_1= Azimuth no. 1 thrust [kN]
d_1= Longitudinal arm to CoF
T_2= Azimuth no. 2 thrust [kN]
d_2= Longitudinal arm to CoF
T_5= Thruster no. 5 thrust [kN]
d_5= Longitudinal arm to CoF
T_6= Thruster no. 6 thrust [kN]
d_6= Longitudinal arm to CoF

The DP requirement for available transverse thrust is 992kN (see TTTI, Sec. 6.8).

$$(T_1 + T_2) + (T_5 + T_6) = 1080 + 1192 = 2272 \geq 992kN$$

This amount of thrust when translated into power, for the required transverse speed of 2 knots, is 1022kW.

$$P(kW) = T(kN) \bullet V_A(m/s)$$

A recommended combination of one aft azimuth at 32% rated power and one bow thruster at 82% rated power will sufficiently provide this transverse thrust. The remainder of thruster capacity offers a double redundancy for the bridge to use as needed while manoeuvring.

Maximum available yawing moment, around the centre of floatation is calculated as:

$$(T_5 + T_6) \bullet \frac{d_5 + d_6}{2} + (T_1) \bullet d_1 = (596 + 596) \bullet 30.25 + (540) \bullet 44.5 = 60088kNm$$

Table 6.13. Thruster Summary

No. of units	Model Name	Type	Max input Power (kW)	Propeller Speed (rpm)	Propeller Diameter (mm)	Bollard Pull/Thrust (kN)
4	Wärtsilä – WST-16	Azimuth-CPP	1600	277	2200	540
2	Wärtsilä - CT125H	Tunnel-CPP	621	524	1250	596

6.8 Ballast and Sea Water System

The vessel is to be grounded at port for loading of both the windmill and the foundation. This can only be achieved with ballasting. On arrival at the loading quay, the ship positions itself over the blocks on the sea floor. Large volumes of ballast water is taken into the various ballast tanks located forward and aft to increase the draft till it sits on the blocks.

For the purpose of operation, all the tanks are grouped according to their location on the vessel. There are three groups of tanks that need to be filled together during ballast operations, see Section 5.4. When loading is complete and the vessel is ready to sail, the ballast water is discharged rapidly till a desirable draft is attained. The volume of ballast water required is given in Tables 6.14 and 6.15:

Table 6.14. Group ballast tank capacities (Tank plan in Appendix)

Dedicated Group	**Number of Tanks**	**Total Volume (m^3)**	**Total Mass (tons) @ρ=1.025tons/m^3**
Bow	4	4247	4354
Amidships	2 ballast & 2 DB	2649	2717
Stern	4	2115	2168

Table 6.15. Ballast Water Volume at different loading conditions

LOADING CONDITION	TANKS	BALLAST REQUIRED(m3)
Grounding	Bow	2865
	Amidships	2058
	Stern	2470
Sailing	Bow	2422
	Amidships	0
	Stern	0
Offloading	Bow	3510
	Amidships	418
	Stern	0

6.8.1 Pump Capacity and Power Requirement

The ballast pump capacity is governed by the volume of water in tons that has to be discharged in a given time as required by the operation of the vessel. System layout can be seen in the Appendix C.

The rules DNV Ballast Water Management rules require each pump connected to the largest tank or a group of dedicated tanks to be able to discharge 95% of the ballast water in the tank within 3 hours.

The dedicated group of tanks are connected to two pumps each. This is to ensure continuous operation without interruption in case of failure. Each installed pumps and their capacities are given in the table below:

Table 6.16. Ballast Pumps capacity and power required

Dedicated Group	Pumping Capacity	95% Discharge Time	Estimated Power (Installed)
Bow	2x 3000-3300 m^3/hr.	approx. 1.3 Hours	314kW
Amidships	2x 2000-2400 m^3/hr.	approx. 1 Hour	232kW
Stern	2x 2000-2400 m^3/hr.	approx. 1.3 Hours	232kW

6.8.2 Ballast Water Treatment

Ballast water is treated to prevent the invasive aquatic species from destroying marine ecosystems. The vessel will operate its ballast system in the same region. The International Convention for the Control and Management of Ship's Ballast Water and sediments does not apply to this vessel.

6.9 Auxiliary Machinery

The auxiliary machinery are responsible for supporting the main engines as well as provide subsystems that are required for managing emergency situations.

6.9.1 Fire pumps

Two fire pumps in accordance with DNV-OS-D301 are responsible for delivering fire fighting water to the fire mains and sprinkler system. It is permitted that one of these pumps is shared with other services, and as such the bilge pump in the engine room is designated as the shared fire pump. The other pump, which needs to be dedicated to fire fighting and available at all times on an independent power supply will be located with the emergency generator. The two pumps will be of similar configuration.

6.9.2 Start Air system

Air at high pressure is required for starting the main engines. The starting air system has two compressors and two air reservoirs. The air reservoirs must have sufficient capacity to allow a sufficient number of consecutive engine starts without the need for the compressors to replenish it.

6.9.3 Emergency Generator

The main generators will provide electricity for normal operation of the vessel. In the event of failure of the power supply set an emergency supply is needed for the continuous operation of essential services required for the safety of the ship, passengers and crew. The emergency generator is rated to provide power for the emergency switchboard that supplies power to the emergency bilge pump, watertight doors and the fire fighting equipment. Emergency lighting for the accommodation, navigation lights, communications systems and alarm systems must also be supplied. The required power output for the emergency generator must be approximately 300kW as given in the Appendix C.

6.10 Deck Equipment

The equipment installed on the upper, main and crane deck are the ones that is used to carry out operations. These are discussed in this section.

6.10.1 Foldable rails

In order for the foundation or the wind tower and their fixtures to be positioned for offloading rails that extend throughout the grooved stern part of the hull must be fitted. When foundation and fixture are in place and secured for offloading these rails will inhibit the lowering procedure, unless the rails retract in some manner. This problem can be solved by a concept that is based on folding the rails in to the ship's hull by rotating the complete rail structure around a shaft. Retractable rails will also benefit the loading procedure, since they then can be easily folded away during the docking.

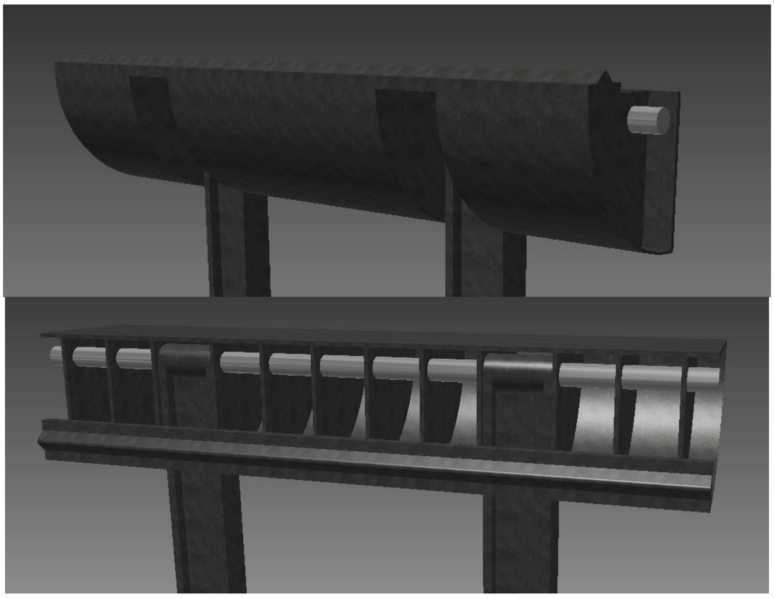

Figure 6.8. Concept visualization

The mechanism consists of an array of quarter circle shaped plates welded to a long stripe of steel plating on which the rails are laid. A shaft runs orthogonally through these plates, whereas any relative motion between the elements is prohibited. The purpose of the shaft is to serve as an axis for the rotary motion of the assembly, thus realizing a lowering of the rails combined with their retraction to the pontoons. The plates should be arranged in a way that vertical elements, such as web frames on the inner side structures, do not constrain the rotation of the part. The geometry of the structure is influenced by the rotary motion, the size of the fixtures and the need for adequate spacing for the installation operations. The dimensioning is attained from the numerous analyses with varying thicknesses, spacing and stiffening. A FE model is presented in Section 7.7.4. For details on dimensions see Appendix D.

The function of the structure sets certain requirements. The main one is the retraction of the rails by entirely going of the way during lowering the foundations and the wind towers. An important necessity is the small deformation of the rails in order to allow the sliding of the fixture without difficulties. A small corner plate is inserted at the free end.

6.10.2 Anchor Windlass and Mooring Gear

The equipment's and installation for anchoring are supposed to meet requirements for fixing the position of The Vessel in shallow water by using the sea-bed. The mooring equipment is for securing the ship to a permanent structure. The structure could be bollards on quay side or mooring buoys at sea. It can also be used for ship-to-ship mooring during transfer of fuel, provisions or crew.

The anchoring must be designed to hold The Vessel in good holding ground and in moderate sea condition. The size and power of an anchor is given by an equipment number (EN). The class rule for an equipment number is given by (DNV, 2013a):

$$EN = \Delta^{2/3} + 2BH + 0.1\,A$$

Δ = moulded displacement in tonnes to the summer load waterline

B = moulded breadth in metres

A = the area in square metres, in profile view of the hull superstructures and houses above summer load line and also greater than $B/4$ in breadth

H = effective height in metres from summer load line to the top of uppermost deckhouse. It is measured as:

$$H = a + \sum h_i$$

a = distance in metres from midship summer load line to upper deck at ship side.

h_i = height in metres on the centre line of each tier of house having a breadth greater than $B/4$.

Table 6.17. Minimum weights dimension of anchors, chains and ropes

Calculated Equipment number 1534												
Equ. No.	Equ. letter	Stockless bower anchors		Stud-link chain cables				Towline (guidance)		Mooring line (guidance)		
		No.	Mass per anchor	Total length		Dia. and steel grade		Steel or fibre ropes		Steel or fibre ropes		
			kg	*m*	NV K1 *mm*	NV K2 *mm*	NV K3 *mm*	Min. length *m*	Min. breaking strength *kN*	No.	length of each	Min. Break-ing length
1483	C	2	4590	550	68	60	52	220	888	5	190	324

The mooring operations at the stern part will be carried out by two capstans with a pull of 10 tons installed on the aft of each of the catamarans. The use of the capstan is to limit the space needed for installing mooring equipment.

The layout of the mooring system for the fore deck is given in Appendix A.

6.10.3 Chain Locker

The anchor chains are stacked in the chain locker at the bow of the vessel. There is space provided in the superstructure for this purpose. The chain locker has to be high and narrow to prevent the stacked chain from falling over in bad weather.

The space for the chain locker is dimensioned using the information from the chain cable as given from the Table 6.17 above.

The volume of space needed to stack the chain locker from the calculation should not be less than 2 cubic metres. This is dimensioned using the biggest diameter for the stud-link and the minimum total length of the chain cable. The dimensions of the chain locker compartment given as Length x breadth x Height (LxBxH) is 0.85mx0.85mx1.7m.

6.10.4 Installation Winches

Six winches are fixed on the catamaran part of the hull for installation of the foundation and the windmill. The winches are hydraulically or electrically driven to give a constant render value. The render force is the force at which the winch begins to turn in the opposite when it is set to heave and the driving force is applied. All winches must operate at a constant tension load (self-tensioning) to eliminate the need for line tending. This will ensure that the foundation is lowered upright and in a stable condition to the sea floor. They must be designed to a pre-set tension value so that all winches operate at the same tension value. In the case when a winch experiences a greater external force greater than the pre-set value then the rope will be released effectively from the drum (render) on the other hand the winch will heave to the pre-set load value when the line tenders.

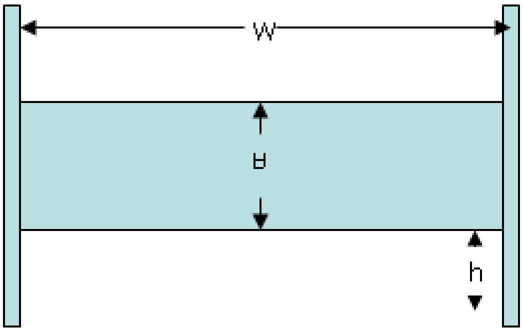

Figure 6.9. Dimensions and Power Rating

The winch drum capacity is calculated using the formula from UNOLS Handbook of Oceanographic Winch, wire and Cable Technology.

$$L = \left(\frac{0.2618\, Wh\,(B+h)}{d^2}\right) * 0.304801[m]$$

Where L is the length of the line in metres and d is the diameter of the line also in metres. The others are given in figure 6.5 above with units in feet. Given the above formula, the winch is to be dimensioned for the drum to be capable of storing a line of at least 60m.

To be on the safe side, each winch must be dimensioned to be able to pull 600 tons at first line even though there will be a reduction of the load due to buoyancy from the water.

The estimated power for each winch is 335kW.

Table 6.18. Winch Specifications

Winch Type	Winch Capacity	Line Diameter	Nominal Speed	Approximate weight	Dimensions LxBxH	Rated Power
Electric Towing winch	600 tons at first layer[2]	109mm	25m/min	24 tons	4000 x4420x3377	335kW

6.10.5 Redundancy

The installation winch is very critical equipment for the operation of the vessel. The function of the vessel is mainly dependent on the performance of the installation winches. The winches must be able to work together to perform this task. This means that there is no tolerance for failure or malfunction of the equipment. For this reason the winches must be preferably of hydraulic type. A hydraulic winch has the characteristic of a simple design. It has few components consisting mainly of a hydraulic motor, a valve and a couple of hoses or pipes. Faults can be repaired quickly, usually by changing a hose or replacing a valve. A short redundancy lag of the winch is provided by designing the system to allow for quick fix in case of failure of any of the components. This can be done by providing easy access to components and simplifying the repair procedures.

6.10.6 Bilge and Oily water separator

The main purpose of the bilge system to drain the mixture of oil and water that accumulates in the bilge space – mainly under the engines. Maritime regulations require bilge water to be treated to an acceptable quality. The vessel will be fitted an oily water separator that meets regulations.

6.11 Utility system

To make the life comfortable for the crew on board the vessel, some utility systems are added. These systems are presented in the following Sections.

6.12 HVAC (Heat, Ventilation and Air Condition)

HVAC is the systems regarding the Heat, Ventilation and Air Condition on the vessel. The HVAC systems are designed according to the ISO7547 and ISO8861, some of the most important design criteria's for those systems are presented in the following Sections.

Air Condition

The air condition system is important to maintain a comfortable indoor climate for the crew. In order to create at comfortable climate, limits are set by ISO7547. The air condition system

[2] *The winch holding capacity is always designed to be higher than the pulling capacity therefore there is an excess of power for lowering however the pulling capacity is of interest due to the constant tension requirement.*

can be defined as a form of air treatment whereby temperature and humidity are controlled. According to ISO standard 7547 there are some limits that the system has to comply with, which are specified in the table below. (ISO7547, 2002)

Table 6.19. Summer Design criteria for air condition system (ISO7547, 2002)

Summer condition	Temperature	Humidity
Outdoor air	+35 C°	70%
Indoor air	Max +27 C°	50%

Table 6.20. Winter design criteria for air condition system (ISO7547, 2002)

Winter condition	Temperature	Humidity
Outdoor air	-20 C°	-
Indoor air	Min+22 C°	-

Ventilation

The ventilation system is important to maintain the necessary airflow that is required, and can be defined as the provision of air to an enclosed space to meet the needs of occupants and the requirements of the equipment installed on The Vessel. In order to fulfil the ISO standard 7547 and 8861 all accommodation spaces and engine rooms must be equipped with sufficient ventilation.

For spaces such as the mess and other common day-rooms the ventilation should be designed in a way that the supplying airflow is the same as the exhaust airflow. The airflow in sanitary rooms such as W.C and showers should be designed for minimum 10 air changes per hour or 0,02 m^3/s. (ISO7547, 2002) (ISO8861, 1998)This is done by an AHU (Air handling unit), the AHU is the unit that controllers the air supplied to The Vessels accommodation spaces. The AHU regulate the temperature, humidity and sanitary of the air to meet the requirements from the ISO standards. (Aeron, 2014)

The ventilation system in the engine room has to provide the enclosed area with cooling air in order to protect the machinery and electronics from overheating. It is also of big importance that the ventilation system provides the engine with the right amount of air needed for a sufficient combustion, which is a driver of the ventilation capacity. The engine room ventilation system should also prohibit the accumulation of flammable and toxic gases, as well as contribute to a comfortable and safe working environment for the crew.

The air flow needed for the combustion is based on the service standard power of the main generators, following calculations has been done according to the procedure and assumptions of ISO8861.

Table 6.21. Maximum continuous power

Item		[kW]
Wärtislä 9L20DF -1	$\boldsymbol{P_{dg1}}$	1310
Wärtislä 9L20DF-2	$\boldsymbol{P_{dg:2}}$	1310
Wärtislä 9L34DF	$\boldsymbol{P_{dg:3}}$	4500

The density of air at +35C at 101.3kpa (ISO8861, 1998)
$\rho_{Air} = 1.13$

The required amount of air needed for combustion for 4-stroke diesel engine (ISO8861, 1998)
$m_{ad} = 0.002$ Kg/(kW*s)

The airflow for combustion for diesel generator engine where calculated according to the following equation (ISO8861, 1998)

$$Q_{dg} = \frac{(P_{dg} \times m_{ad})}{\rho_{Air}}$$

Table 6.22. The air flow for combustion for the different diesel generator sets (ISO8861, 1998)

item	Air flow	[m^3/s]
Wärtislä 9L20DF -1	$\boldsymbol{Q_{dg:1}}$	2.3
Wärtislä 9L20DF-2	$\boldsymbol{Q_{dg:2}}$	2.3
Wärtislä 9L34DF	$\boldsymbol{Q_{dg:3}}$	8.0

The total air flow needed combustion (ISO8861, 1998)
$Q_{Total} = 1.5 \times (Q_{dg:1} + Q_{dg:2} + Q_{dg:3}) \approx$ 19.0m^3/s

To provide the engine room with sufficient combustion air an air flow of 19.0m^3/s is needed. Suitable configuration for the air supply are 5 × (CLZ12-J) marine Axial ventilation Fans at 4.4m^3/s, 37.5kW is the power needed to be generated with this configuration. (HI-SEA)The air flow needed for the evacuation of heat emissions from the machinery and equipment within the engine room need also be determined according to ISO 8861, this calculations are to be done in future work since more correct data of the equipment are needed.

The fans ventilation and exhaust fans should be dimensioned and be operated in a way that a slightly positive pressure within the engine room is achieved. (ISO8861, 1998)

6.12.1 Heat system

Generation of heat and a reduction of fuel consumption can be achieved by using a Waste Heat Recovery system (WHRS). This system has many positive aspects, both financial and environmental.

The WHRS reuses the exhaust heat by letting it pas through an exhaust gas economizer, the economizer are designed as a heat exchanger which produces steam. The steam from the economizer will be used for heating the vessel.

6.12.2 Sewage Treatment System

Just like the bilge system, the sewage water produced from the galley (grey water) and sanitary spaces (black water) must also be treated before it is discharged overboard.

A biological treatment system is fitted to allow the discharge of sewage in any waters to ensure continuous operation of the vessel. This treatment system has automated operation and comes with minimum maintenance. The system uses an aeration system to naturally produce a clean and safe sludge suitable for discharge.

6.12.3 Freshwater and hotel system

The hotel system will supply the crew with fresh water for showering, drinking, cooking and for sanitary spaces. The required amount of fresh water needed are a function of the number of crew and are dependent on which sanitary system the vessel are equipped with. The estimated need of fresh water can be determined according to Marine Diesel Power plant Practices, the amount of fresh water need where estimated to 4.5 MTPD (Metric Ton Per Day), (Rowen, 1990) this for a crew size at 8 and 10 people working at site. The required amount of fresh water where reduced by using a vacuum sanitary system.

Since the vessel will operate close to land, fresh water will be bunkered in port and not mainly be generated on board. If the vessel must stay out for a longer period fresh water could be generated by evaporation of sea water, this is done by installing a Wärtsilä Serck Como Fresh Water Generator. The system utilizes the heat from the exhaust gas or the low temperature return cooling water.

The sea water evaporates at low temperature due to a created vacuum in the chamber, the distillate water then goes through a salinometer to ensure that it is absolutely free of salt. In the next stage, the distillate is effectively sterilized using UV light to eliminate all microbiological organisms. This technology is simple and has automated operation with low maintenance requirements (Wärtsilä, 2014a).

6.13 Failure analysis

By making a failure analysis, potential emergency situation can be identified, and the right action could be taken. If one of those situations would occur the action taken can be crucial to maintain the safety of the crew and the vessel. In this chapter a failure analysis are made over the main systems regarding the machinery and propulsion.

This failure analysis helps identify the consequence and the action that follows after one single failure. In the analysis of the engine room equipment it was found that the switchboard is one weak function. If the switchboard is out of function the consequence will be a total blackout of all equipment on board. This could be solved by installing one additional switchboard, located isolated from each other in event of fire. This would require power connection to the switchboards with enough supply to maintain functions in case of one switchboard fails. The engine and the thruster layout can withstand one single failure.

Table 6.23. Failure analysis

	Failure	Consequence	Action
Port Engine	No start	Less total power output	Reduce load on systems
Starboard Engine	No start	Less total power output	Reduce load on systems
Large main Engine	No start	Less total power output	Reduce load on systems
Switchboard	No power	Power Blackout	Start emergency engine, call for help
Port Thruster 1	No Start	Reduced thrust	Finish operation
Port Thruster 2	No Start	Reduced thrust	Finish operation
Starboard Thruster 1	No Start	Reduced thrust	Finish operation
Starboard Thruster 2	No Start	Reduced thrust	Finish operation
Tunnel thruster 1	No Start	Reduced thrust	Finish operation
Tunnel thruster 2	No Start	Reduced thrust	Finish operation

6.14 Future work

The calculations constituting the vessel's eligibility to bear the Ice Class 1C notations with regards to power requirements is at the present based on empirical formulas. In order for the classification society to approve the Ice Class notation, the design will need to fulfil more stringent criteria which are a result of the odd hull shape.

The ballast piping system as it is now only comprises of pumps and schematic pipelines with sizing on main branches. The exact routes of the pipelines as well as all branch sizes are required for manufacturing of the ballast system. A hydrodynamic simulation of the vessel in design conditions can greatly enhance the accuracy of the design of DP systems; in case the azimuth thrusters have a negative effect on each other's performances, it is plausible to replace 2 of them with retractable thrusters. A complete heat balance will also be needed, this to ensure that as much as possible of the heat generated from the machinery will be reused. Further work will also be needed regarding the HVAC. Discussions on special arrangements for safety of ships using low flash point fuels such as methanol are underway which may require their implementation in the future development phases (IMO, 2014) .

7 Structure

The unusual geometry of the vessel is attributed to the concept of combining a mono- and a multihull in the design. From structural point of view the monohull part of the ship provides the needed strength and stiffening in the connection between the catamaran hulls and carries loads from fundaments and wind towers during transit. The multihull cross-section allows for an easy access to operation relevant locations, while carrying the cargo during installation. The design complies with the DNV rules for classification of ships, part 3, chapter 2 (DNV, 2012) and the Finnish-Swedish ice class rules (Sjöfartsverket, 2012). The ship is classified as a special purpose vessel with the ice class 1C.

7.1 Structural arrangement

The vessel's uncommon shape clearly sets the need for a division between two significantly different ship cross-sections. The main concern during this differentiation is to provide sufficient continuity in geometry and material while allowing for the full development of the concept.

7.1.1 Ship cross-sections

On one hand, there is the prismatic monohull part, which is 19 meters long. It has a 2 meters high double bottom in order to increase the reliability of the vessel, given the regular grounding. Within the double bottom, vertically stiffened floors are fitted under every web frame and transverse bulkhead and girders under every longitudinal bulkhead. There are also additional girders in-between. Furthermore, the safety of crew, cargo and vessel is ensured by the side structure together with the outermost watertight longitudinal bulkheads. Additional bulkheads are fitted in the space between the inner bottom and the main deck. They are of a structural significance as they provide a solid support for the rails on the deck and decrease the large span in between. In this ship cross-section the frame spacing is 1 m with a web frame occurring at every third frame. They limit the deformations of the deck in order to keep a sufficient level of serviceability during installation work on deck. They are arranged with smooth curves and brackets to avoid stress concentrations and contribute to a proper transfer of all forces. Drawings of the two prismatic sections can be seen in Appendix D. Figure 7.1. provides an overview of how the different sections are defined along the hull girder.

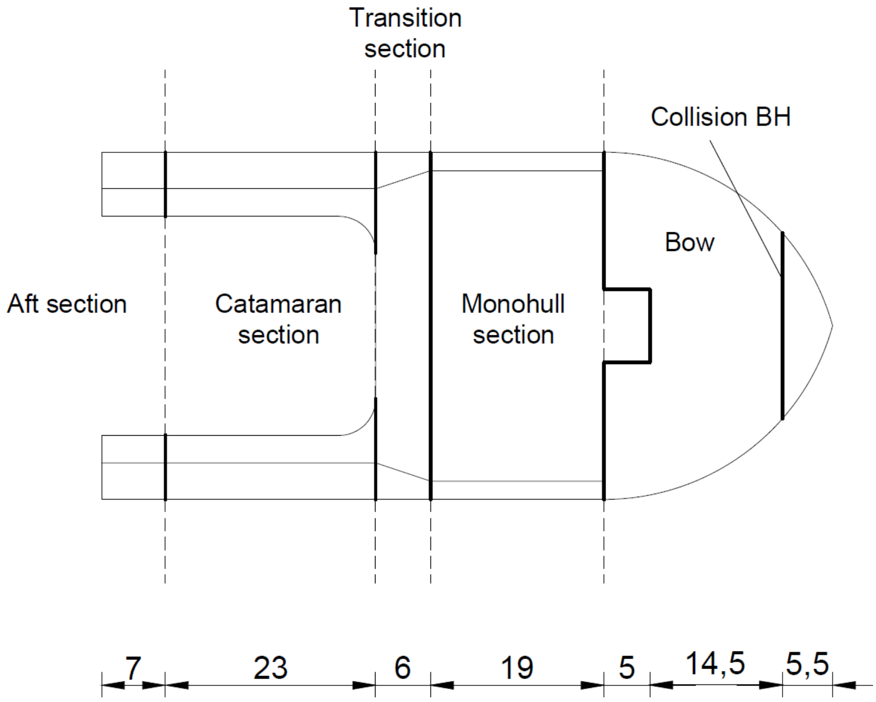

Figure 7.1. Longitudinal division of hull girder sections

On the other hand, the catamaran section of the ship consists of two parallel hulls. They are mirrored at the vertical plane lying in the centreline. The large open space in-between allows for an easy access for the placing of gravity foundations and installation of windmills. The challenges in this part arise from the relative slenderness of the hulls compared to the length of the section and the loads acting on the structure. The asymmetry only manifests itself above water level: The upper decks are located on the outside, while the crane decks are oriented on the inside. Since the slenderness of this catamaran ship cross-section is an important characteristic, extensive consideration is made in order to reach the required strength and stiffness.

The catamaran ship cross-section is 23 meters long and it is arranged as elongation of the mono-hull part: The sides, the outer bilges, the bottoms, the crane decks and the longitudinal bulkheads have precisely matching longitudinal elements over the full length of the vessel, thus satisfying the requirements for continuity. Geometrically speaking, a heavily expressed difference is the width of the upper deck: 4 meters here, as opposed to the 2 meters in the mono-hull ship cross-section.

The frames within the catamaran part have a spacing of 0.85 meters and the web frames here occur at every third frame, as well. No double bottom is introduced here in order to free up enough space for the arrangement of the ballast tanks. The longitudinal watertight bulkhead completely separates this section in two: outer side envelopes the space below the upper deck and the inner side – the one below the crane deck. Furthermore, the crane deck is continuous and watertight throughout the full width of the pontoons, thus horizontally dividing this compartment. This allows for a reduction in hydrostatic pressures in the outer ballast tanks. The side structures and the longitudinal bulkheads are additionally supported by cross-ties, which are arranged as I-beams and integrated in the web frames. These are important elements as they provide a solid support for the inner sides, which are subjected to great loads creating multiaxial normal and shear stresses.

The transition between the two ship cross-sections spreads over 6 meters. As previously mentioned, the main longitudinal elements remain constant over the full length of the vessel. The upper deck is reduced by 2 meters to form the main deck in the other section. The outer bottom of the mono-hull ship cross-section is smoothly trimmed as it fuses to the inner bottom until they both meet the closing transverse bulkhead at the end of the transition. The inner side of the connections with the pontoons is curved so, that the stress concentration in the area can be reduced.

The subdivision of the vessel is achieved with consideration of the significant structural characteristics described so far. The DNV rules (DNV, 2012) state a clear requirement on the minimum number of transverse watertight bulkheads. For ship with length between 65 and 85 meters that is 4 bulkheads. In the design of the installation vessel, an additional transverse bulkhead is arranged, thus giving a total of 5. The following is a discussion on the subdivision of watertight compartments.

According to the rules (DNV, 2012), the collision bulkhead should be located within a specific interval in longitudinal direction. For the arrangement of it, the presence of bow thrusters must also be taken into consideration. For this reason, it is located at 5.5 meters from the bow tip.

The next watertight bulkhead is arranged at the beginning of the mono-hull ship cross-section (25 meters from the bow). It is a reasonable location marking the change of the global cross-section and it has a structural significance in the support of the main deck and the superstructure. About a quarter of the length amid the bulkhead has a 5 meter offset in the direction towards the bow. This is to allow for access to elevator and staircase to the machinery room.

There is another transverse watertight bulkhead at the other end of this prismatic cross-section. The two bulkheads define the mono-hull ship cross-section as a watertight subdivision on its own. The engine room is located within this subdivision; hence the DNV requirements (DNV, 2012) for a watertight enclosure of the engine space are met.

Another bulkhead is located at the beginning of the catamaran ship cross-section. It is aligned with the closure of the transition as previously described. This defines the transition section as a watertight compartment.

The fifth transverse bulkhead marks the end of the pontoon ship cross-sections and isolates the rear thrusters from the ballast tanks. It is located 7 meters from the stern. This distinguishes two more watertight compartments. For an illustration of the longitudinal positions of the transverse bulkheads see Figure 7.1.

7.1.2 Ice belt

The choice of Ice class 1C (Sjöfartsverket, 2012) imposes some additional requirements on the hull structure that need to be considered. The local strengthening of the side structure, within a so called ice belt, is one of these requirements. The ice belt is a part of the outer side structure that extends a certain vertical distance above the waterline at maximum draught (LWL) and above the waterline at minimum draught (BWL). For this vessel these draughts corresponds to sailing with maximum payload (T=5.6 m) and in ballast condition (T=4.0 m) respectively. Worth noting is that only draughts during transit conditions are considered for the ice reinforcements. The grounding condition in port is assumed as non-applicable for the ice class requirements. The ice belt is also longitudinally divided into three parts with different structural requirements (Sjöfartsverket, 2012); forward, midship and aft region. The required longitudinal outlines of the ice belt for the vessel with ice class 1C are presented in Figure 7.2. As a consequence from the relatively short stern the aft region becomes very small and is thus treated as a midship region, which has more stringent requirements. The scantling of the ice belt can be found in Appendix D.

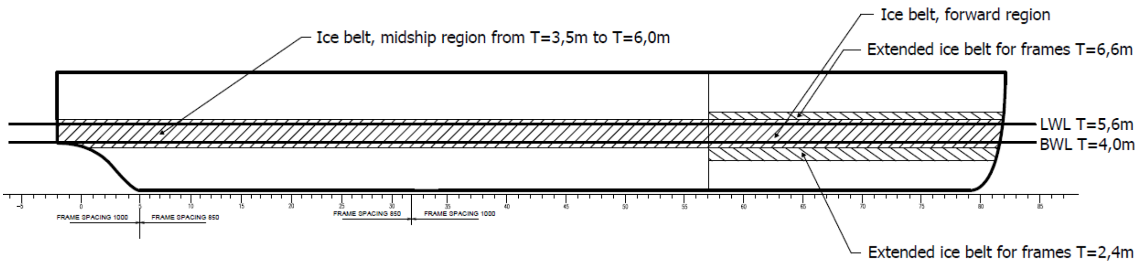

Figure 7.2. Longitudinal outlines of the different ice belt regions

7.2 Loading conditions

The vessel will be exposed to a large variety of different loading conditions. However when it comes to structural integrity five critical conditions can be distinguished. See Table 7.1.

Table 7.1. List of loading conditions

Condition number	Operation	Cargo
1	Offloading	Largest foundation and fixture
2	Transit	Largest foundation and fixture
3	Transit/Offloading	Two medium foundations with fixtures
4	Ballast leg	Fixture
5	Grounding	Ballast

The first one corresponds to the offloading procedure of the largest expected single payload; a 2500 ton foundation together with a 200 ton fixture. This loading condition is characterised as hogging and will induce a very large global bending moment in the hull girder as a result of the unevenly distributed net load. Figure 7.3 below illustrates the longitudinal distribution of the hull girder loads during the described loading condition in still water. The position of the payload will also inflict a large torque on each of the two sterns hulls.

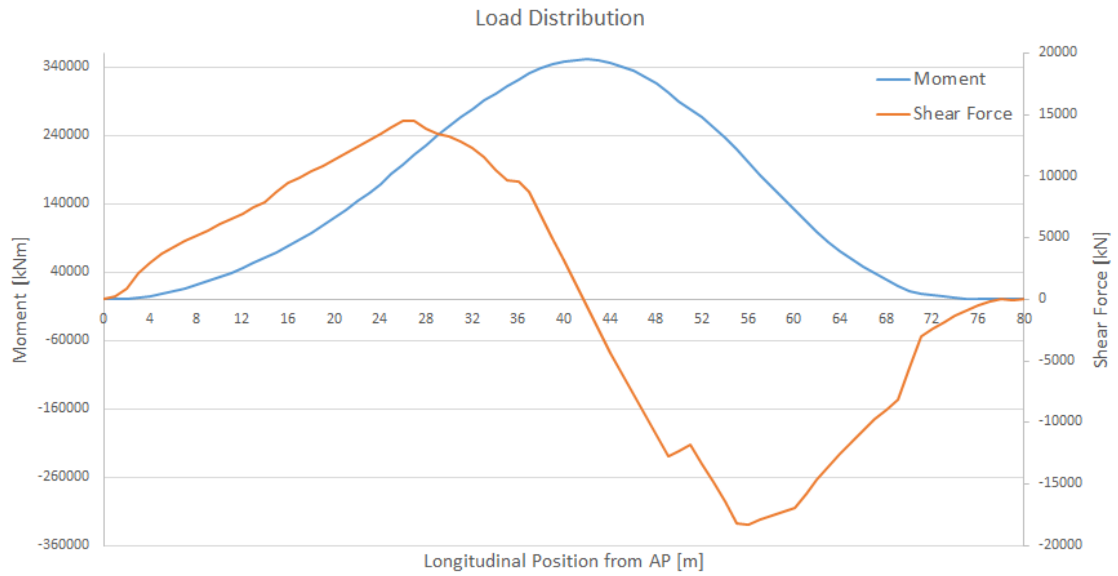

Figure 7.3. Stillwater bending moment and shear force for largest single payload in offloading position.

Figure 7.4 shows the load distribution for the condition when a single unit of the largest foundations is placed on the main deck. It can be seen that the bending moment is of sagging characteristics with a lot lower extreme values. Thus this is a more favourable condition with regards to longitudinal strength. It does on the other hand inflict the largest expected loads on the structural members supporting the rails on the main deck.

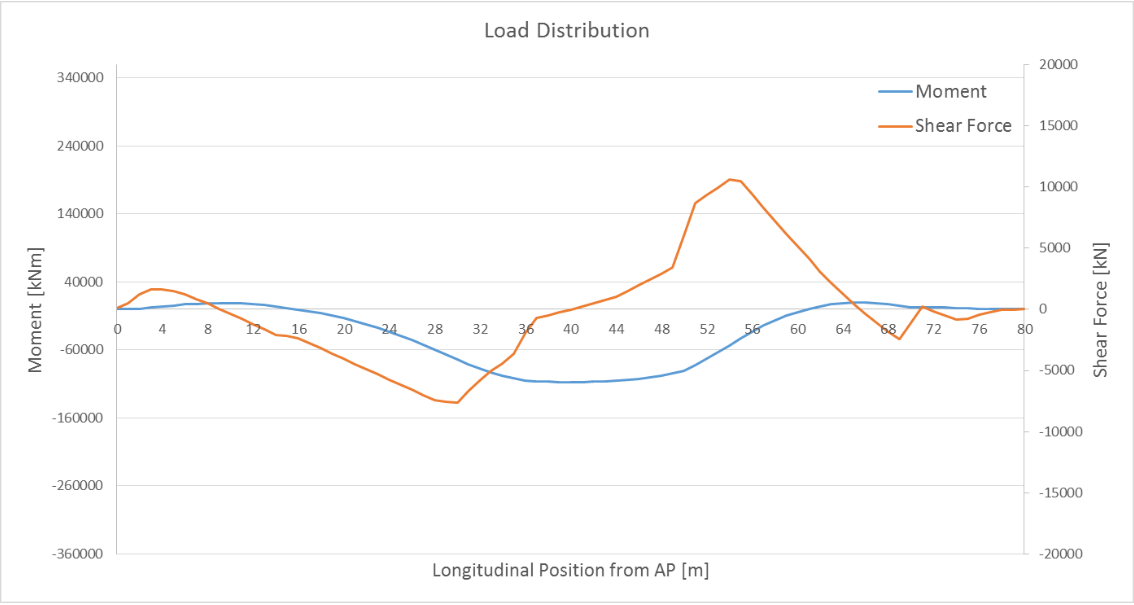

Figure 7.4. Stillwater bending moment and shear force for largest single payload in transit position.

The third loading condition corresponds to the scenario when carrying the largest total payload, i.e. two 2000 ton foundations (one on main deck and one over the cut-out) together with their fixtures, which can be seen in Figure 7.5.

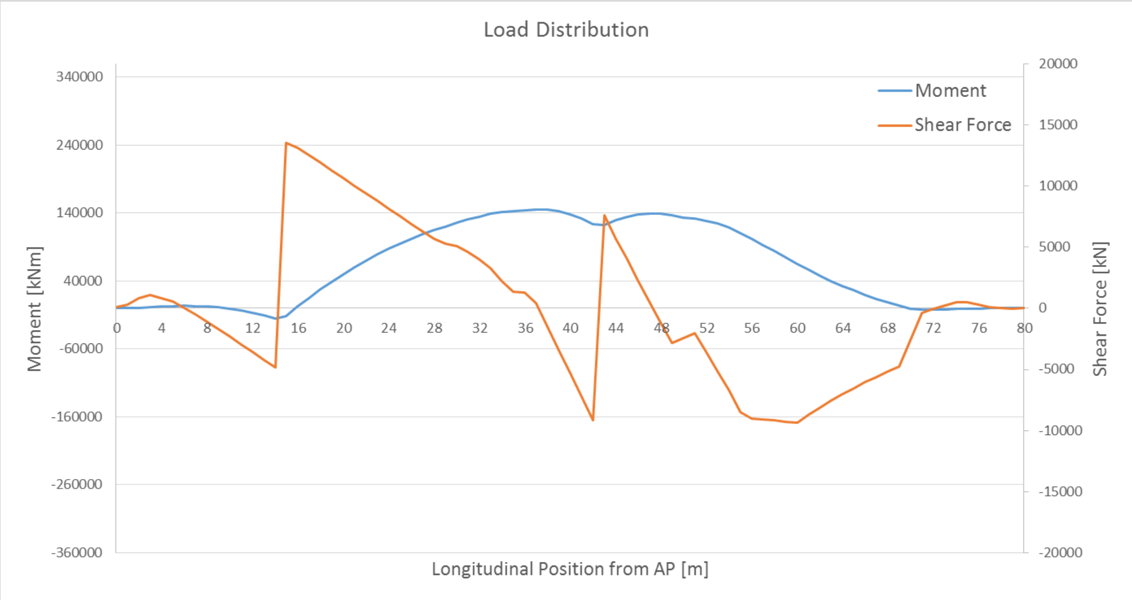

Figure 7.5. Stillwater bending moment and shear force at largest total payload

Also the ballast condition is investigated. However Figure 7.6. shows that this condition will be of minor interest for the structural design, since the longitudinal loads are relatively small.

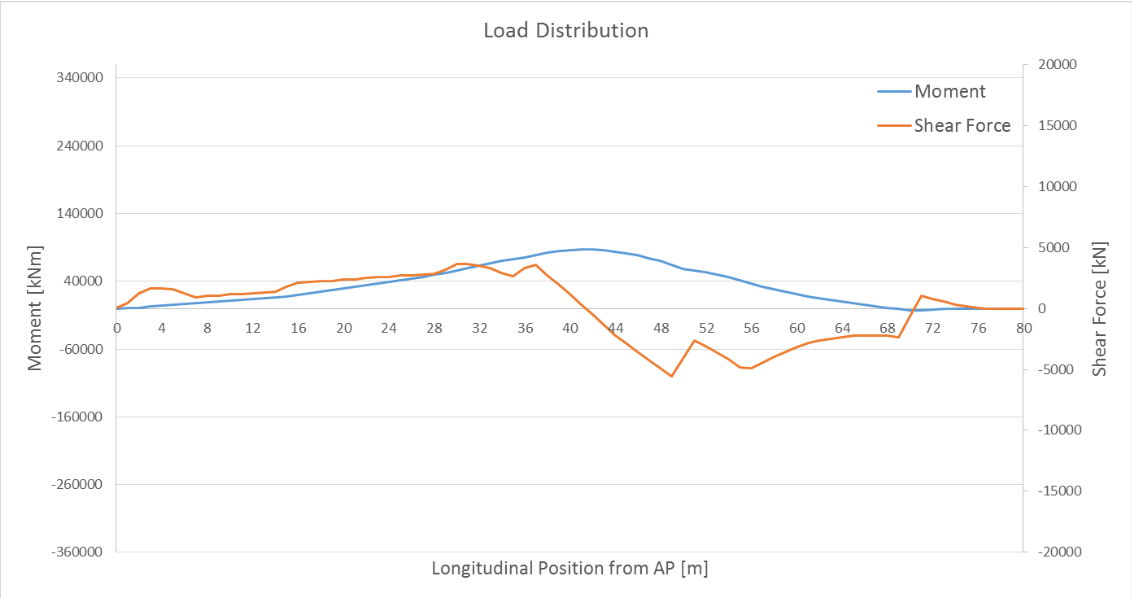

Figure 7.6. Stillwater bending moment and shear force at ballast condition

The final critical loading condition is taken as the grounding procedure. In this condition the vessel reaches its maximum expected draft. This is only of interest when assessing the transverse structural integrity of the hull and thus not evaluated with regards to longitudinal strength. The design draught for this condition is taken as 6.2 m.

To sum up the loading conditions and their significance for the hull girder strength, Table 7.2. provides information about the shear forces and bending moments in the two cross-sections of the ship.

Table 7.2. Maximum moments and shear forces for the defined loading conditions

Monohull section		
Loading condition	**Maximum bending moment [kNm]**	**Maximum shear force [kN]**
1	350 642.7	-18 147.4
2	-107 087.1	10 615.42
3	145 611.0	-9 132.6
4	86 965.9	-5 548.3
Catamaran section		
Loading condition	**Maximum bending moment [kNm]**	**Maximum shear force [kN]**
1	252 766.8	14 474.7
2	-74 219.6	-7 620.3
3	125 892.4	13 502.5
4	56 111.5	3 633.4

7.3 Longitudinal strength

The class requirements regarding the global longitudinal strength of the hull girder is governed by the main class structural rules for ships with length less than 100 m (DNV, 2012). These rules state that the section modulus, 0.4 L amidships, of the hull girder is not to be less than:

Required section modulus:

$$Z_R = \frac{M_w + M_S}{175} * 10^3 \quad (cm^3)$$

Here the wave bending moment is taken as the larger of:

$$M_{W0} = 0.11 C_W L^2 B (C_B + 0.7) = 222385 \quad (kNm)\ in\ sagging$$
$$M_{W0} = 0.19 C_W L^2 B C_B = 179178 \quad (kNm)\ in\ hogging$$

Where:

$$C_W = 5.7 + 0.022L, minimum\ 7.0$$

The stillwater bending moment (M_S) is taken as the larger value between the maximum calculated value, within 0.4 L amidships, and the rule design stillwater bending moment (M_{S0}), which is calculated accordingly (DNV, 2012):

$$M_{S0} = 0.0052L^3B(C_B + 0.7) = 132737\ (kNm)$$

From Table 7.2. it can be seen that the largest calculated stillwater bending moment is $M_{S,max} = 350\,642\ (kNm)$. Thus this will be the dimensioning moment used for evaluating the least required longitudinal strength.

Finally the required section modulus that must be achieved for both prismatic sections and throughout the transition zone is calculated to:

$$Z_R = 3.028\ m^3$$

The achieved section moduli for the top and bottom of the prismatic cross-sections are presented in Table 7.3.. It can be seen that the catamaran section is on the margin of fulfilling the requirements on the longitudinal strength. This borderline value can however be justified by the fact that the highest maximum bending moment is located further towards the bow in the monohull section, which is well above the requirements. Due to the non-prismatic shape of the transition zone the section modulus will depend on the longitudinal position and is therefore harder to evaluate analytically. It can however safely be assumed that the required section modulus will be maintained throughout this part of the hull girder.

Table 7.3. Achieved section moduli for prismatic hull sections

Location\Section	**Catamaran section modulus [m^3]**	**Mono-hull section [m^3]**
Top	3.514	9.622
Bottom	3.577	13.75

7.4 Materials

The vessel is mainly designed to be constructed using normal strength structural steel with a yield limit of 235 MPa in accordance with the DNV main class rules (DNV, 2012). A certain degree of stiffness in the structure is required to limit the deflections in the hull girder. Such stiffness is to a great extent influenced by the cross-sectional properties of the structural elements. The material grade plays a minor role in this aspect. Higher strength steel would therefore be excessive. Elements such as longitudinals, transverse frames and plates limited by buckling could be investigated further though.

In some parts of the structure there are other steel classes. The main deck has a higher strength steel under the rails to have sufficient yield limit. There are also bulkheads in the monohull part with higher strength steel.

In order to distinguish between the material grade requirements for different hull parts, various material classes are applied as defined in DNV Hull Structural Design, Section 2, Table B1 (DNV, 2012). The primary structural members including bottom plating, crane deck plating, continuous longitudinal members, uppermost strake in longitudinal bulkhead, should follow the Class III within 0.4L amidships or Grade A/AH outside 0.4L amidships.

7.5 Scantlings

The DNV rules and regulations (DNV, 2012), as well as the Finnish-Swedish ice class rules (Sjöfartsverket, 2012) give essential information on the requirements for the geometry and/or the cross-sectional properties of the different elements. All these values are considered as the first instance for dimensioning of the scantlings during the design of the vessel. The implementation of the strength calculations and safety margins serve as the second instance giving more meaningful insight into the structure specifics. Sizes and geometry of the various elements are referred to in the following subchapters.

7.5.1 Plate thicknesses

The plating on different places in the structure has varying thickness values, depending on the rules (DNV, 2012), calculations and assumptions.

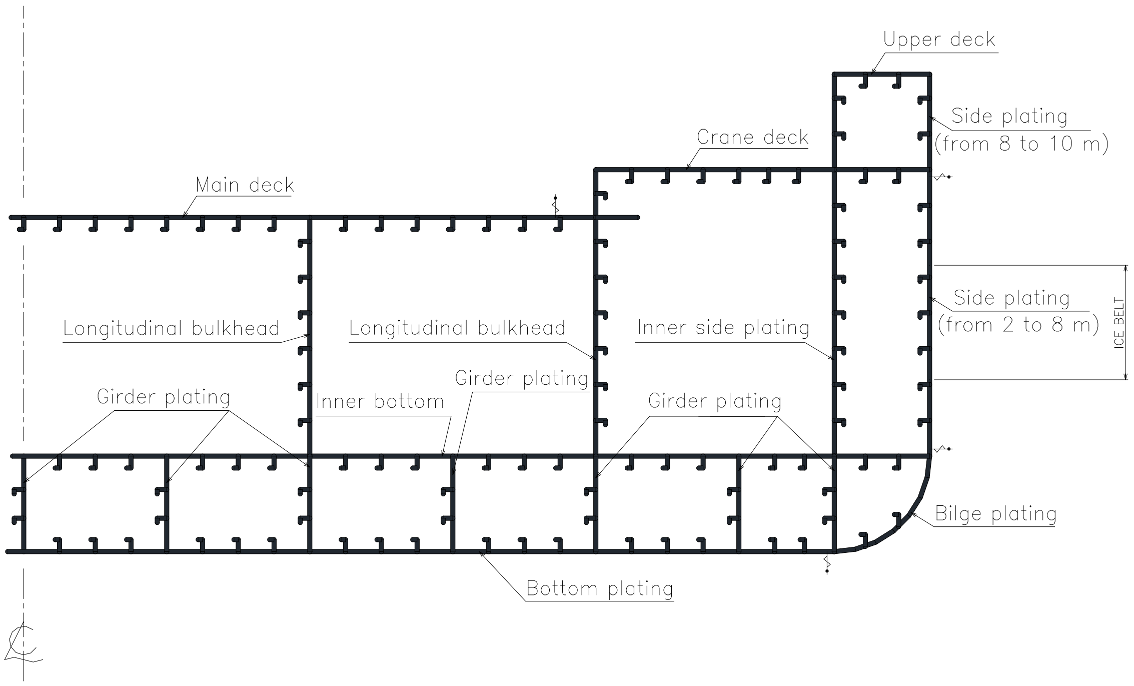

Figure 7.7. Platings in the monohull cross-section

Table 7.4. Scantling: Plate thicknesses in monohull part

Monohull part	
Location	**Thickness [mm]**
Bottom plating	10
Inner bottom	8
Bilge plating	10
Side plating from 2 to 8 m from bottom	8.5
Side plating from 8 to 10 m	12.5
Ice belt from 3.5 to 6 m	28
Inner side plate (on-board)	8.5
Main deck	8
Crane deck	9.5
Upper deck	11.5

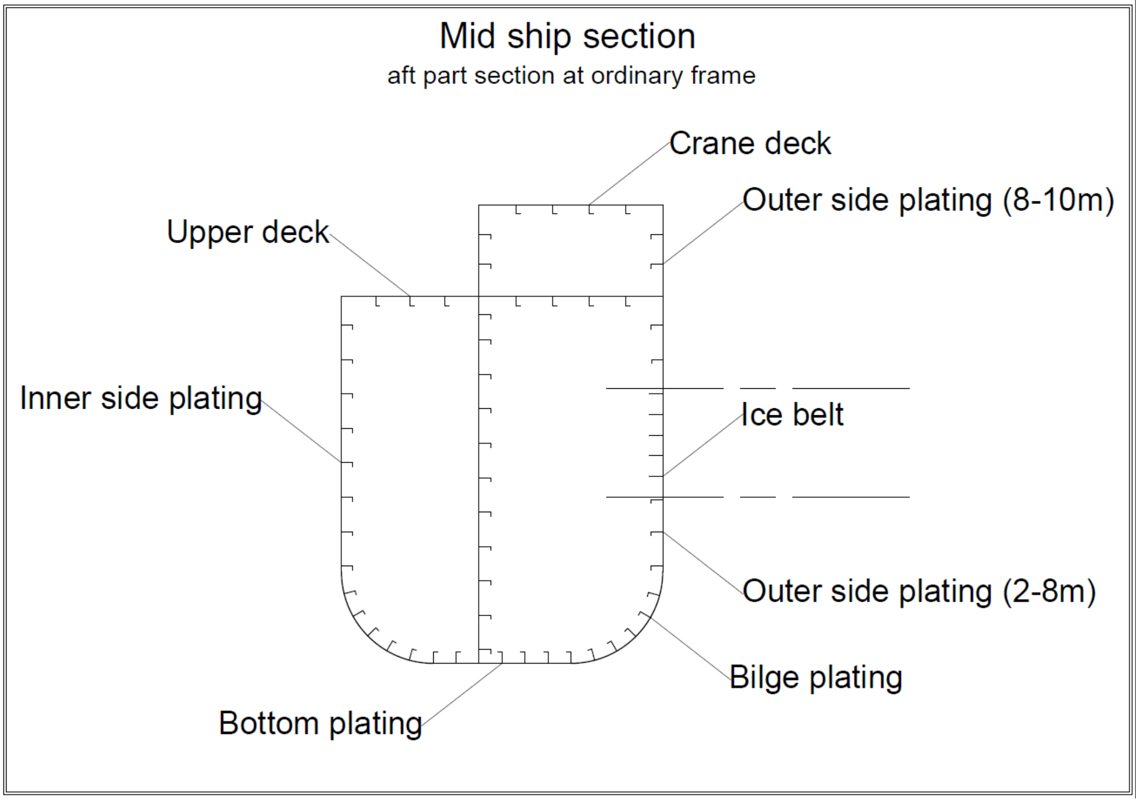

Figure 7.8. Platings in the catamaran cross-section

Table 7.5. Plating thickness of catamaran part

Catamaran part	
Location	**Thickness [mm]**
Bottom plating	13
Bilge plating	13
Outer side plating from 2 to 8 m from bottom	11.5
Outer side plating from 8 to 10 m	14
Ice belt	26.5
Inner side plating	11.5
Crane deck	9.5
Upper deck	12.5

7.5.2 Girders and floors

The girders are designed to be continuous in length. In the double bottom there are 13 girders, which are 6.5 mm thick and 2 m high. The floors in the double bottom have the same thickness and height.

7.5.3 Stiffeners

The stiffeners are a significant contributor to the overall strength of the ship. The DNV rules and regulations (DNV, 2012) are clear about the minimum requirements on the section moduli of these elements depending on their location. Furthermore, standard sized profiles are chosen to accommodate the rules and the loading conditions on both local and global levels. The main stiffener type is the L-profile, with exception of the flat bar profile stiffening of the floors in the double bottom and the web plates in the side structure of the monohull part.

The watertight transverse bulkheads are vertically stiffened and the stiffeners are supported by horizontal T-profiles with dimensions 500x14+200x20 mm. On the bulkheads in the pontoons two supports are employed with even distances in-between. In the monohull part only one such support per bulkhead is used, because of the smaller vertical dimension. A summation of all stiffening elements is presented in Table 7.6.

Table 7.6. Scantling: Stiffeners

Monohull part	
Elements	**Dimensions [mm]**
Longitudinal bottom and side stiffeners	L 250x90x9/14
Other longitudinal stiffeners, stiffeners of the transverse bulkheads and the girders	L 200x90x9/14
Stiffening girders of the transverse bulkheads	T 500x14+200x20
Stiffening of the floors and web plates, flat bar profiles	300x15
Catamaran part	
Elements	**Dimensions [mm]**
Longitudinal bottom, side and longitudinal bulkhead	L 250x90x9/14
Longitudinal deck stiffeners	L 200x90x9/14
Stiffening girders of the transverse bulkheads	T 500x14+200x20

7.5.4 Frame design monohull

The frames in this section are designed to withstand the applied loads while maintaining the large spacing between the longitudinal bulkheads. Web frames occur at every third frame with spacing of 3 meters in total. The scantling of the web frame is visible in Table 7.7 with reference to the girders as defined in Figure 7.9. More details on the dimensions can be found in Appendix D.

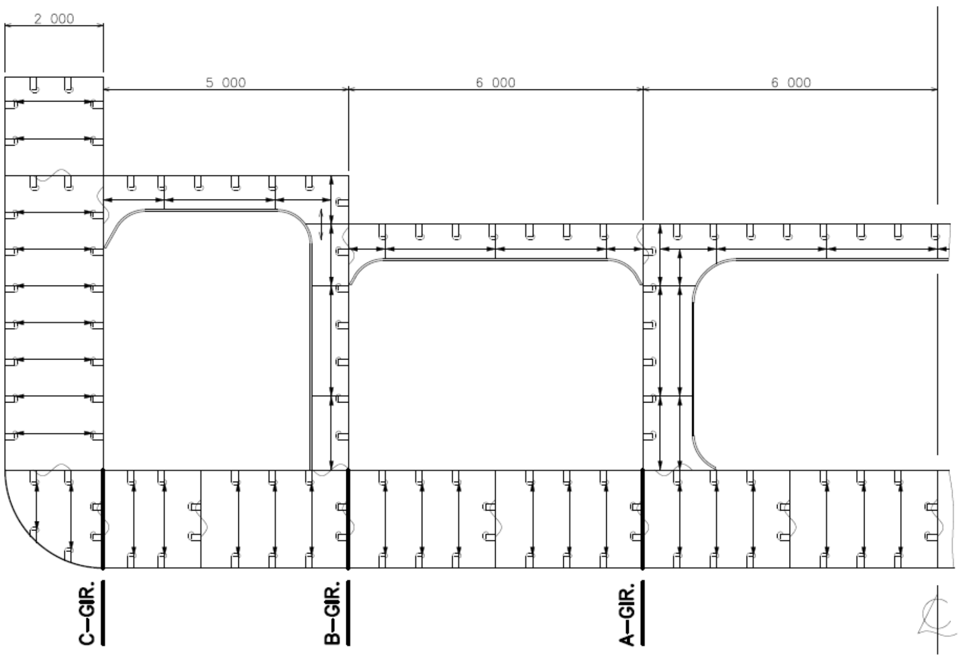

Figure 7.9. Web frame and girder reference

Table 7.7. Scantling: Transverse frames in the monohull section

Between girders	Girders CL-A	Girders A-B	Girders B-C
Horizontal web height [mm]	700	700	700
Vertical web height [mm]	1000	–	750
Web thickness [mm]	12	16	12
Flange width [mm]	200	200	200
Flange thickness [mm]	14	14	14

7.5.5 Frame design stern

The transverse frames in the prismatic catamaran section are dimensioned in accordance to the main class rules (DNV, 2012). The design is chosen as T-Profiles running along all plate fields including the longitudinal bulkheads. In order to reduce the scantlings of the frames, crossties are introduced at a distance of 4000 mm above the baseline. This reduces the span of the frames and consequently allows for less stringent requirements. A reduction in frame weight of approximately 25% is achieved with the design with crossties compared to a frame design without them. The crossties are incorporated in the frame structure with flanges running from side and bulkhead frames to each side of the crosstie web. Thus the crosstie design will be represented by a symmetric I-beam. They are conservatively dimensioned against Euler II buckling. Table 7.8 summarizes the frame and crosstie scantlings together with the dimensioning properties.

Table 7.8. Scantling: Transverse frames in the prismatic catamaran section

Structural dimensions and properties	Frame side/deck		Crosstie	
	Required	Chosen	Critical	Actual
Required section modulus [cm^3]	2109	2228	-	-
Required web thickness [mm]	13,7	14	-	15
Chosen web height [mm]	-	400	-	500
Chosen flange thickness [mm]	-	16	-	16
Chosen flange width [mm]	-	200	-	200
Euler II buckling stress [MPa]	-	-	194	114

7.5.6 Bulkheads

The bulkheads are arranged as plates with constant thickness for the purpose of dividing the ship in watertight compartments. See Table 7.9.

Table 7.9. Scantling: Bulkhead thicknesses

Monohull part	
Orientation	**Thickness [mm]**
Transverse	9
Longitudinal	13
Catamaran part	
Orientation	**Thickness [mm]**
Transverse	9.5
Longitudinal	13

7.5.7 Reinforced scantlings within the ice belt

The major ice reinforcements for the present vessel correspond to an increase in the shell plate thickness, which can be seen in Table 7.10. This reinforced belt is required to extend from 6 m down to 3.5 m above the baseline (Sjöfartsverket, 2012). In addition to the plate thickness the stiffening elements are subjected to ice reinforcements. Firstly the spacing is taken as the maximum allowed value of 450 mm. Secondly the web thickness is taken as 15 mm due to the requirement on this being at least half the plate field thickness (Sjöfartsverket, 2012). Thus 300x15 flat bar profiles are used throughout the whole reinforced plate field.

Table 7.10. Plate thickness within ice belt

Region	**Ice belt plate thickness [mm]**
(Aft)	(20.5)
Midship (prismatic catamaran)	26.5
Midship	28.0
Forward	29.5

Intermediate transverse frames are introduced at every frame position in the forward ice region in order to reduce the required plate thickness here. These frames are required to be extended at least from 6.6 m down to 2.4 m from the baseline (Sjöfartsverket, 2012). At each of the terminating ends a supporting stringer is fitted. Both the intermediate frame and the stringer are chosen as T-profiles with scantlings as stated in Table 7.11.

Table 7.11. Scantlings for intermediate frames and their supporting ice stringers

Structural dimensions and properties	Intermediate frame	Ice stringer
Required section modulus [cm^3]	1056	1808
Required web thickness [mm]	14.75	14.75
Chosen web thickness [mm]	15	15
Chosen web height [mm]	400	500
Chosen flange thickness [mm]	15	15
Chosen flange width [mm]	120	160
Actual section modulus [cm^3]	1093	1878

7.5.8 Buckling Control

The critical buckling stresses of plating and longitudinals have been calculated in accordance with DNV class rules (DNV, 2012). It is found that the buckling stress is the dimensioning requirement for the scantlings on deck and bottom plating. This is the case for both of the prismatic sections of the vessel, but the bottom and bilge plate in the mono-hull are an exception, since these are dimensioned in accordance with the class rules regarding bottom structures. On the other hand for the longitudinals the critical buckling stresses are well above actual levels for all plate fields. These elements are therefore not dimensioned with regards to buckling.

In the calculation process, two loading conditions corresponding to hogging and sagging have been taken into consideration. The two conditions are used to calculate the longitudinal bending stress for the bottom and different decks respectively. The hogging condition corresponds to loading case #1 and the sagging condition to loading case #2, as described in chapter 7.2. Deck scantlings are dimensioned towards sagging and bottom towards hogging. Furthermore the stillwater bending moments, used for calculating the actual bending stress, are taken from Table 7.2. Here the maximum value within each of the prismatic hull sections are used as dimensioning value for that corresponding section respectively. The critical and actual buckling stresses for the decks and bottom plates in both hull girder sections are presented in Table 7.12.

Table 7.12. Critical and actual buckling stresses in bottom and deck plating's

Plate Field	Critical Buckling Stress [MPa]	Actual Buckling Stress [MPa]
Catamaran		
Bottom/Bilges	129.7	120.7
Upper deck	89.5	84.4
Crane deck	54.7	50.9
Mono-hull		
Bottom/Bilges	78.0	24.0
Upper deck	61.0	55.1
Crane deck	41.7	36.7
Main deck	29.5	27.6

7.6 Structural weight distribution

The uncommon shape of the vessel strongly characterizes the weight distribution. A lot of material is employed in the monohull part of the ship due to its relatively large width. In comparison, the slender pontoons demand much less steel in total, as their dimensions are significantly smaller. This, combined with the large span in-between, sets the trends in the weight distribution, making the sections easily recognizable in Figure 7.10. It is also easy to recognize the web-frames, because of their sizable geometry attributed to the large loads acting on the vessel. The largest peaks are the result of the watertight bulkheads.

The information of the bare hull weight is essential. It is the basis for the total weight distribution, which is superimposed with the buoyancy distribution in order to calculate the global moments.

The weight distribution is delivered in tons per half meter, because the smaller the intervals are, the more precise the approximation of the local weights is. As a result, the global moments are evaluated more accurately.

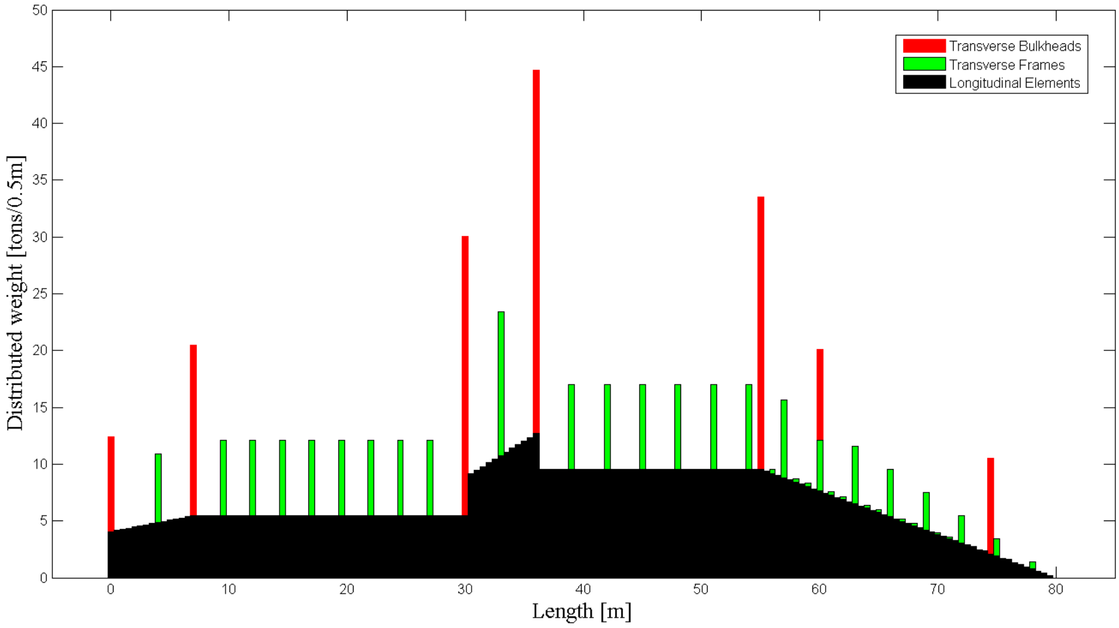

Figure 7.10. Steel weight distribution of bare hull

7.7 FE-Analysis

The structural scantlings on the vessel are based on minimum requirements from class and relatively simple analytical calculations. In order to evaluate the structural integrity of the vessel on a more comprehensive level and validate its capacity to withstand the expected loads a number of FE-Analyses is conducted. These are presented in the following chapters.

7.7.1 Model1 – Stern and transition connection.

The transition zone between the two prismatic hull sections yields a discontinuity in the longitudinal strength of the ship structure. Such a discontinuity will inevitably lead to certain degrees of stress concentrations at the interfaces between transition zone and the prismatic sections. This together with the fact that the expected loading conditions is of a very complex nature poses difficulties for analytical strength calculations. Therefore an FE-Analysis is done to more accurately evaluate the design with respect to the expected loading conditions. An FE-Analysis can also pinpoint special regions that may be subject to redesign.

The evaluated FE-model can be seen in Figure 7.11. It is a half beam global model, which spans from the aft part of the prismatic catamaran section to the aft part of the prismatic mono-hull section. This gives a model capable of describing the complex loads acting on the structure. The investigated loading conditions are based on the most severe loading condition with regards to longitudinal strength during stillwater conditions. This corresponds to the cargo offloading procedure of the largest foundation, which will generate the highest possible hull girder bending moment for this vessel. The condition also takes the largest torsional loads on the catamaran pontoons into account.

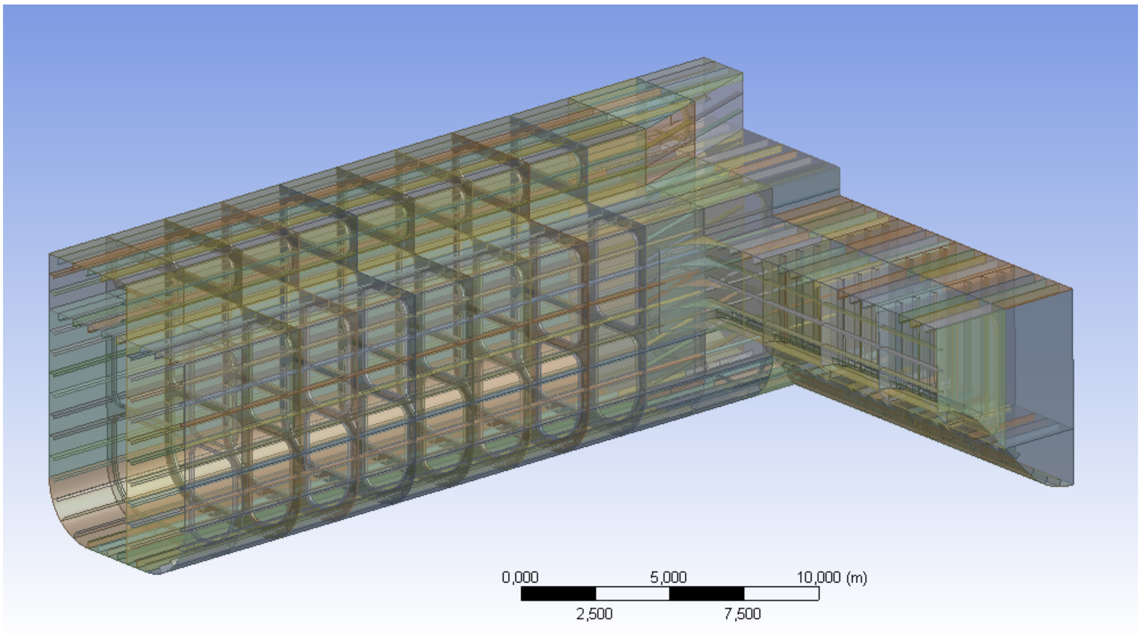

Figure 7.11. Geometry representation of the evaluated FE-model

It should be noted that the evaluated loading condition is based on the scenario when the payload is standing on the cargo rails. This is assumed to yield approximately the same global loading as when the payload is attached to the winches. There will however be local differences in the load situation, since the structural parts acting as the major load bearing members will differ between the cases. Therefore an additional local analysis of one winch together with its supporting hull structure is advised. Due to lack of time and information regarding winch attachments etc. this is excluded from the scope of this project.

The model's boundary conditions are conservatively taken; allowing free rotation around cross sectional NA in forward and aft end of the model, thus disregarding the structural bending stiffness at these boundaries. In addition the aft boundary is allowed to translate in transverse and longitudinal directions and rotate freely around all axes. This gives the model the possibility to capture global displacements caused by the weight of the cargo and asymmetric cross-section of the catamaran pontoon. A symmetry boundary is taken at centreline in order for the model to capture the full beam behaviour of the evaluated hull section.

Shell elements with assigned thicknesses and material data are used to represent the geometry. In order to achieve manageable computation times the maximum mesh element size is set to 200 mm. This gives a mesh with an unsatisfying density in the interface between the transition zone and pontoon. Therefore a local refinement is introduced in this region, so that the maximum element size is 50 mm.

7.7.1.1 Results

The following presents the results from the FE-analysis made on the catamaran/transition zone interface together with suggestions for potential redesign.

From Figure 7.12. and Figure 7.13. it can be clearly seen that the overall structural integrity is maintained for the major parts of the structure. Figure 7.12. illustrates the equivalent stresses present in the outer shell structures. Red colour in the contour plot indicates elements with higher stress levels than the yield limit. As can be seen the levels are well within recommended levels from Chinese Class Society (CCS, 2003). In direct vicinity of the connection between the pontoons inner side plate and the mono-hull transom plate there is however a significant increase in stresses. The maximum value of this local effect is indicated by a probe in Figure 7.12. Even though the acceptable level of 220 MPa, for the shell structure and current material, is maintained, this region should be considered as subject to redesign. Firstly the uncertainties in the model, both with regards to loading and boundary conditions, require some margin to the acceptable levels. Secondly the dynamic loads from a wave environment are disregarded in this static FE-analysis, which also adds to the uncertainties.

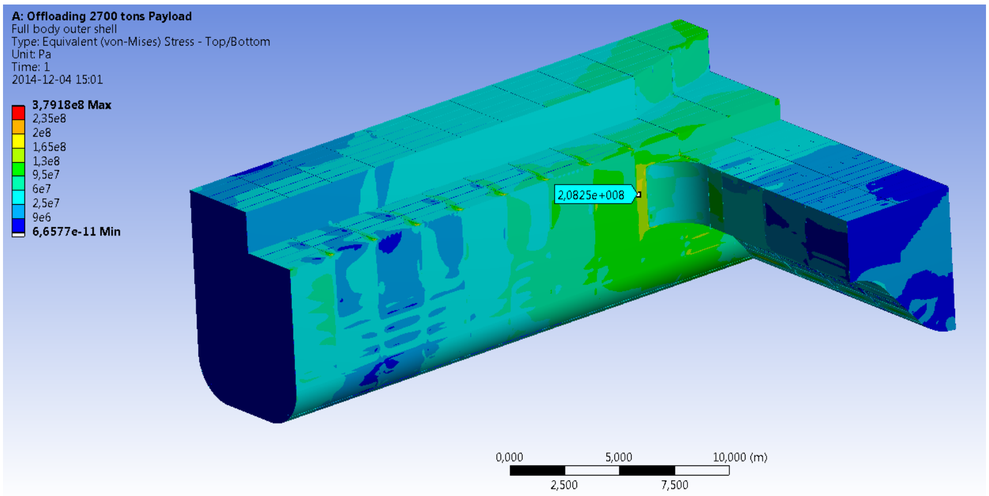

Figure 7.12. von-Mises Equivalent stress for the outer shell on the global model (inner side and deck).

Figure 7.13. illustrates the equivalent stress levels in the internal members such as web frames, bulkheads and stiffening elements. Also here the stress is well below the acceptable limits on a global scale. Some additional local hotspots with stress levels exceeding the yield limit can however be found for the internal members. These are discussed further on in the text.

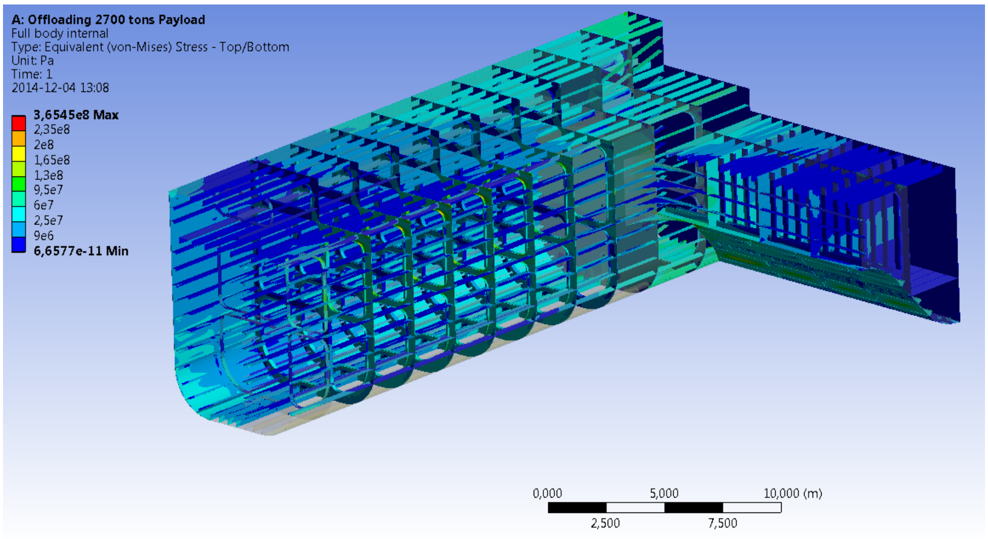

Figure 7.13. von-Mises Equivalent stress for the internal members on the global model.

As expected some of the local stress concentrations occur at the direct interface between the catamaran pontoon and transition zone. Especially exposed regions are the corners between the water tight bulkhead and the mono-hull transom, which can be seen in Figure 7.14. The stress at this location can easily be reduced by introducing a bracket or radius. Such elements would however be limited in size because of the immediate proximity of the cargo rails.

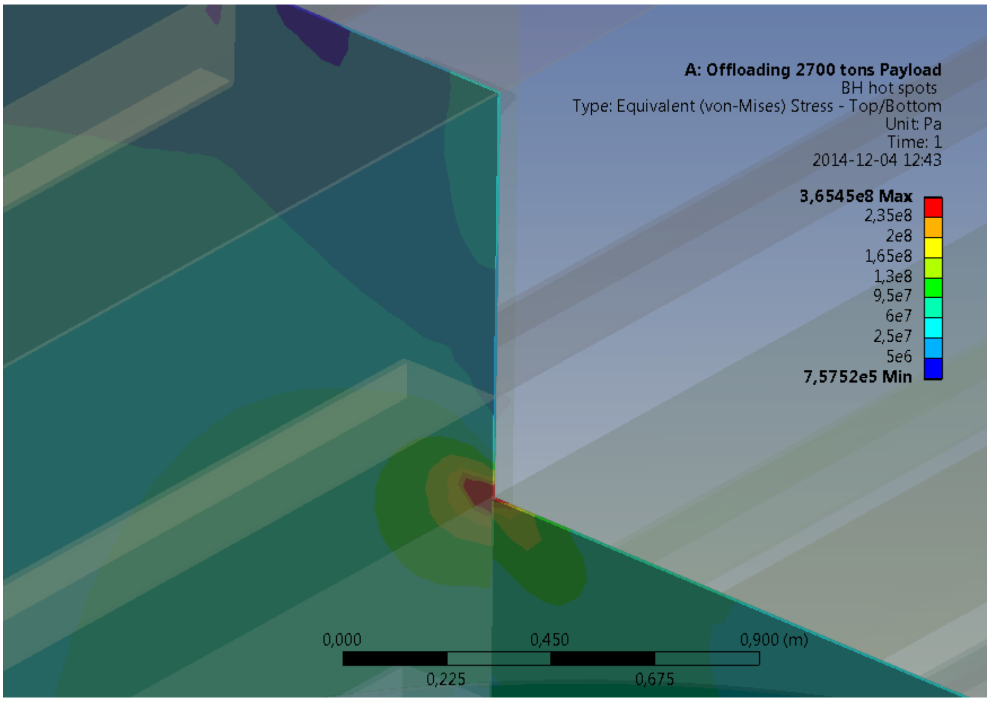

Figure 7.14. Stress concentration in bulkhead at catamaran/transition zone interface (upper corner)

Figure 7.15 shows that the frames in vicinity of the cargo load also suffer from stresses at yield level and above. It can be seen that the highest stresses only occur locally in the flanged radiuses. Increasing the flange radius is most likely sufficient to decrease the stress to acceptable levels.

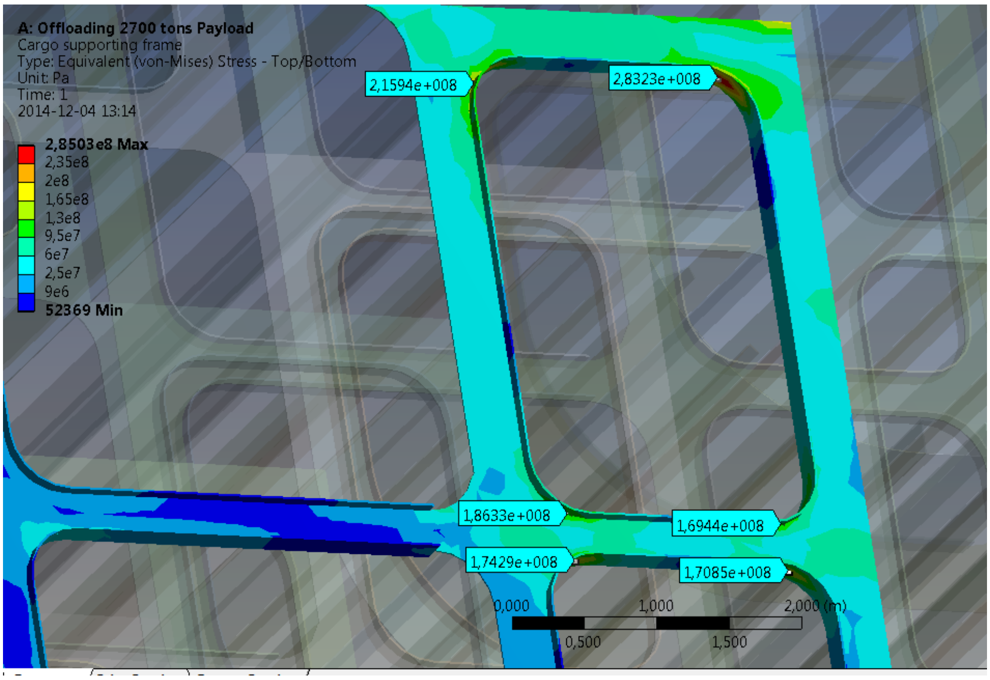

Figure 7.15. Single frame at position below the cargo.

7.7.2 Direct strength analysis in the Monohull region

The direct strength analysis is carried out to evaluate the stress level of the plates in the region of monohull (30 meters to 55 meters from the stern). The analysis is based on the finite element method, which evaluates the longitudinal strength members, the primary support members and the transverse bulkheads.

7.7.2.1 Coordinate system

The coordinate system used within this analysis is shown in Figure 7.16.. The X-axis aligns with the length of the ship and is considered positive in the forward. The Y-axis represents the width direction of the ship and the positive direction is pointing to the port side. The upwards vertical direction defines the positive z-axis.

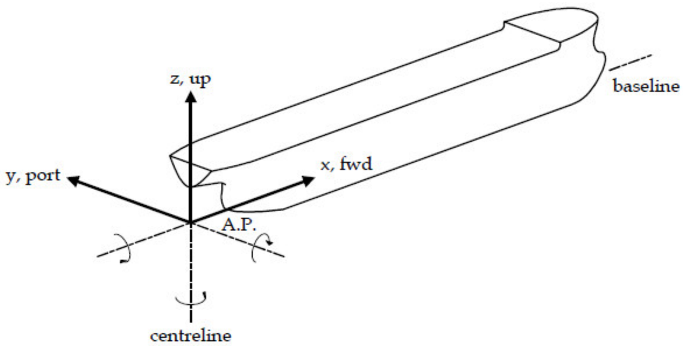

Figure 7.16. Coordinate system definition (IACS, 2012)

7.7.2.2 The range of the finite element model

The analysis model consists of target (monohull) and boundary (bow and catamarans) parts. The target model contains the monohull region (36 to 55 meters from the end), the transition zone (30 to 36 meters from the end). The rest of the model contains and partial bow (55 to 61 meters form the end) and catamaran stern (24 to 30 meters from the end). The strength evaluation is only carried out in the target part.

The model covers the whole depth of the ship. During the analysis of global longitudinal strength, only starboard side is modelled. The boundary parts of the model have the same length in order to keep the middle of the target model equal to the middle of the whole model.

The finite element model includes all of the transverse and longitudinal structural members. The SHELL181 (type of shell element in Ansys) is used to model the outer shells, inner shells, longitudinal bulkheads, transverse bulkheads, floors, girders, transverse frames and stringers. The longitudinal, vertical stiffeners and face plate of web frames are defined by BEAM188 (type of beam element in Ansys). The gross thickness of plates and stiffeners are inputted as elements properties.

The mesh is controlled by the arrangement of the stiffeners, which represents the real shape of the plate panels. In this analysis, the default mesh size is set to 750 mm, which is the space between the longitudinal stiffeners. As recommended in the IACS Common Structural Rule for Bulk Carriers (IACS, 2012), that mesh size is adequate for a coarse mesh analysis. At least three elements are arranged on the bottom floor and girder along the vertical direction. The pipe openings and holes for weight optimisation are not included in this model. In principal, quadrilateral shell elements are used. In the complex region, the high order triangle shell elements are adopted. The aspect ratio of quadrilateral shell elements is smaller than 3. The whole model is shown in Figure 7.17.

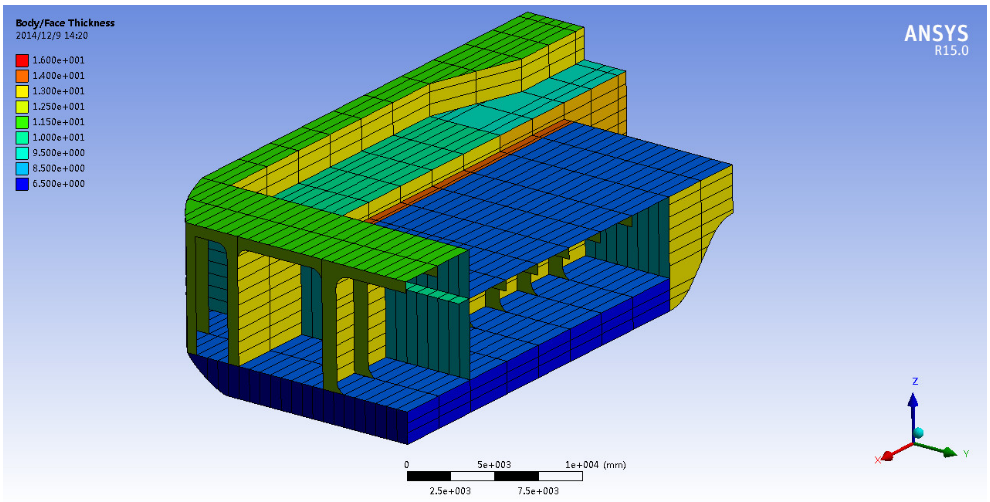

Figure 7.17. The FEM model in front side view

7.7.2.3 Boundary conditions

The global longitudinal strength is evaluated based on the beam theory, thus two independent points are defined as the boundary in the forward and in the backward cutting plane of the model. In the software ANSYS, the remote point feature implements this application. At both end sections of the model, all longitudinal members are rigidly connected to the independent points by MPC (multiple points constrain) and all degrees of freedom except y-axis translation and x-axis rotation are fixed. All nodes in the centre line plane are constrained by translation in y-direction and rotation in x-direction. The front independent point is fixed in all degrees of freedom except the rotation around the y-axis; the same boundary conditions are introduced in the aft with addition of a free translation in x-direction. All of the boundary conditions are summed up in Table 7.13 and visualized in Figure 7.18. (CCS, 2003)

Table 7.13. Boundary conditions of the model

	Translation			**Rotation**		
	x	**y**	**z**	**x**	**y**	**z**
Forward point	Link	Free	Link	Free	Link	link
Backward point	Link	Free	Link	Free	Link	Link
Forward end	Constrain	Constrain	Constrain	Constrain	Bending moment	Constrain
Backward end	Free	Constrain	Constrain	Constrain	Bending moment	Constrain
Centreline	Free	Symmetry	Free	Symmetry	Free	Free

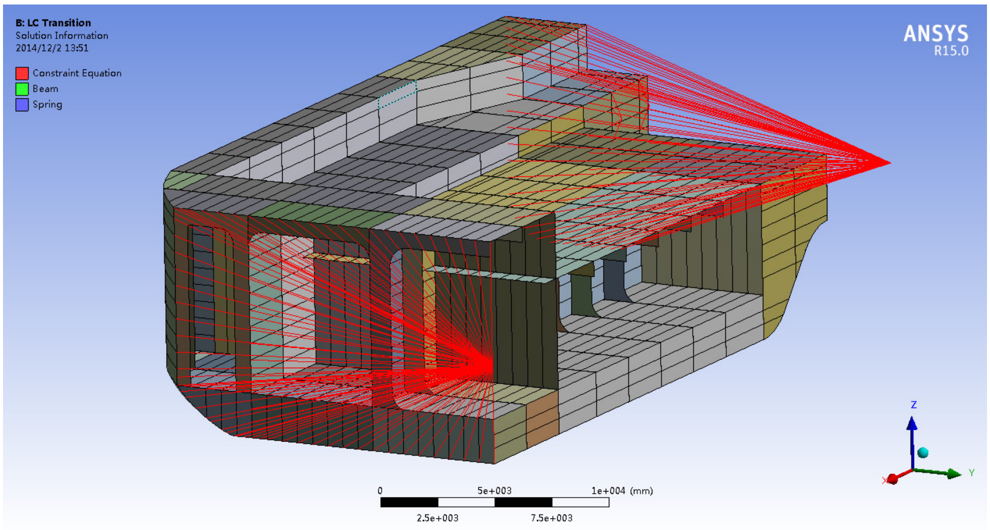

Figure 7.18. MPC boundary condition setting at the ends of the model and mesh visualization (red lines represent connections between nodes and the remote points)

The whole model contains 14298 nodes and 15176 elements. The aspect ratio distribution of the model implies a good control over the quality of the mesh. The most of quadrilateral elements (6287 elements in total) have the aspect ratio between 1 and 1.25. The maximum aspect ratio of the element is controlled below three.

7.7.2.4 Loading conditions

In this analysis, two loading conditions are used to evaluate the structure's strength – foundation transition and foundation offloading. The weights of vessel and foundation, the buoyancy, the pressure of ballast water, the water pressure on the outer shell are considered.

The weight of the vessel is defined as Standard Earth Gravity in ANSYS. The load from foundation acts in two ways. The first contribution is the weight of the foundation, which is distributed along the rail in the monohull region. The other one is the inertial force due to ship motion. From conservative point of view, the transverse and the longitudinal accelerations are set to half the earth acceleration.

The hydrostatic pressure implements the buoyancy following the different draughts during each loading condition. The water pressure on the outer structure is a combination of hydrostatic and dynamic pressure. A method is given to estimate the value of it (CCS, 2003). The pressure for different draughts and locations is given in Table 7.14.

Table 7.14. The hydro pressure on the out- shell in full loaded loading condition

Position	**Pressure (kN/m^2)**
Base line	$P_b = 10d + 1.5C_w$
Water line	$P_w = 3C_w$
Free board	$P_s = 3P_0$
Upper deck	$P_s = 2.4P_0$

The water plane coefficient is equal to

$$C_W = 0.0412\mathrm{L} + 4 \ (\mathrm{L} \leq 90)$$

The value of P_0 equals to $C_w – 0.67*(D - d)$, where D is the depth of ship and d is the draught of the considered loading condition.

Table 7.15. The outer shell pressure in different loading conditions

Pressure	**Transition**	**Unloading**
(kN/m2)	**d = 4m**	**d = 5.6m**
P_b	50.94	66.94
P_w	21.89	21.89
P_s	9.83	13.04
P_{s_deck}	7.86	10.44

The applied end bending moments are determined as the combination of still water bending moment (M_s) and wave bending moment (M_w). In the hydrodynamics chapter, the M_s under sagging and hogging can be found. The M_w is determined by the rule requirement (DNV, 2012).

$$M_{W\_sagging} = 0.11C_W L^2 B(C_B + 0.7) \ (kNm)$$

$$M_{W\_hogging} = 0.19 C_W L^2 B C_B (kNm)$$

The parameters in above equations can be found in the DNV rules (DNV, 2012). The summary of the end section bending moment can be found in Table 7.16**Error! Reference source not found.**.

Table 7.16. Bending moment at the model's ends

Load case	**draught**	**Condition**	**M_s_front**	**M_s_back**	**M_w_front**	**M_w_back**
	(m)		**(kNm)**	**(kNm)**	**(kNm)**	**(kNm)**
Transition	4	sagging	4572	-29745	-222385	-222385
Offloading	5.6	hogging	112577	107539	179178	179178

The load implements in the two loading conditions are shown in Figure 7.19 and Figure 7.20.

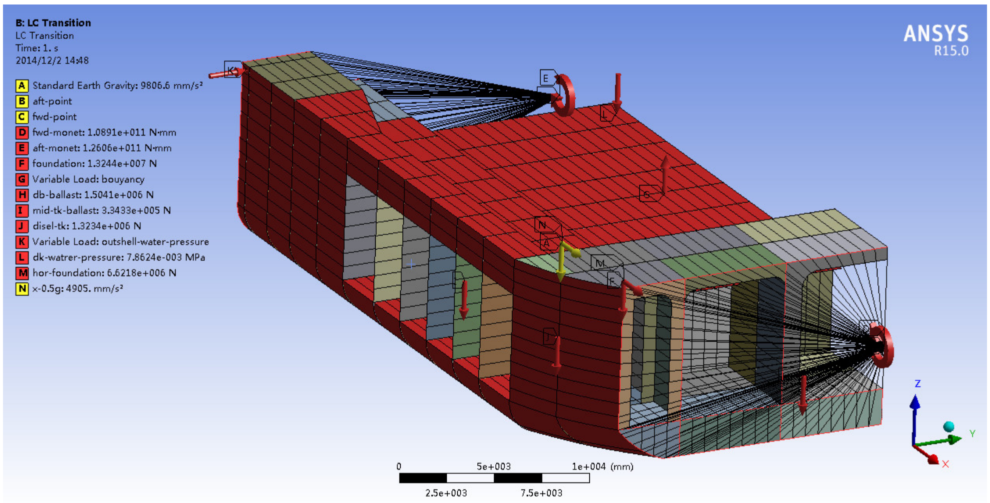

Figure 7.19. The load implement under one foundation transition condition

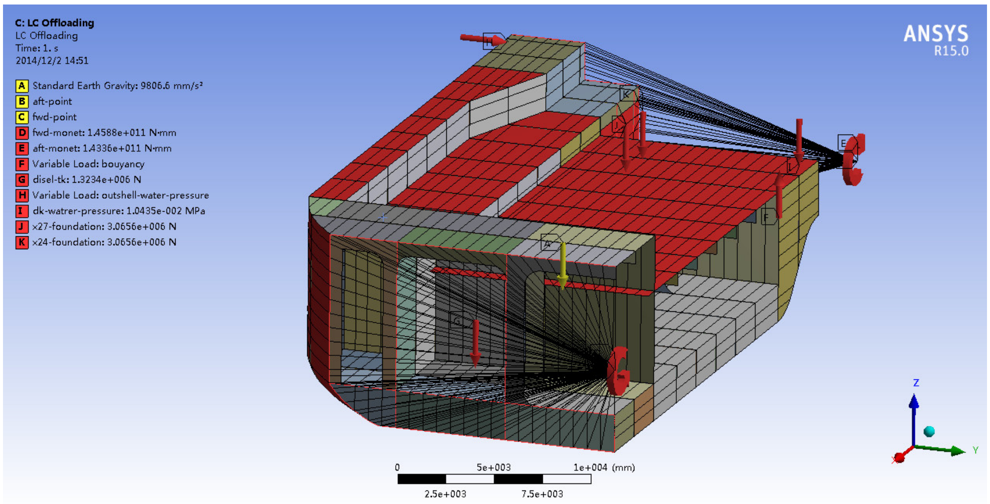

Figure 7.20. The load implement one foundation offloading condition

7.7.2.5 Result

From the post process in ANSYS, the von Mises stress of the plates can be computed. The structures in the target model mentioned before are evaluated and compared with the criteria. For the shell elements, the middle surface stress is selected, while the axis stress is investigated in beam element. The criteria can be found in Table 7.17 (CCS, 2003).

Table 7.17. Allowable stress [N/mm2] level for different structural members

Type	Deck	Shell plate	Longitudinal bulkhead	Girder	Floor	Transverse bulkhead	Web frame
NVA	220	220	220	235	175	175	195
NVAH32	282	282	282	301	224	224	250
NVAH36	305	305	305	326	243	243	270

The maximum stress is 235 MPa on the transverse frame. On the deck under the rails, the stresses reach 233 MPa. The third biggest of those stresses occur on the transverse bulkhead with the value of 186 MPa. The unity check (UC) is carried out to evaluate the capability of the strength. By dividing the von Mises stress by the criterion values, the UC value is shown in Table 7.18.

Table 7.18. Maximum three von Mises stresses with unity check

Stress (MPa)	Unity check	Location	Loading Condition
235	0.87	Transvers frame	Transition
233	0.83	Deck plate	Transition
186	0.83	Transverse BHD	Transition

The von Mises contour figures for the three locations are shown in Figure 7.21, Figure 7.22 and Figure 7.23 respectively. The transverse structural members near the rail withstand the load from the foundation; the stresses of these components stay on a high level. High tensile steel will be used in these locations to reduce the weight of the vessel.

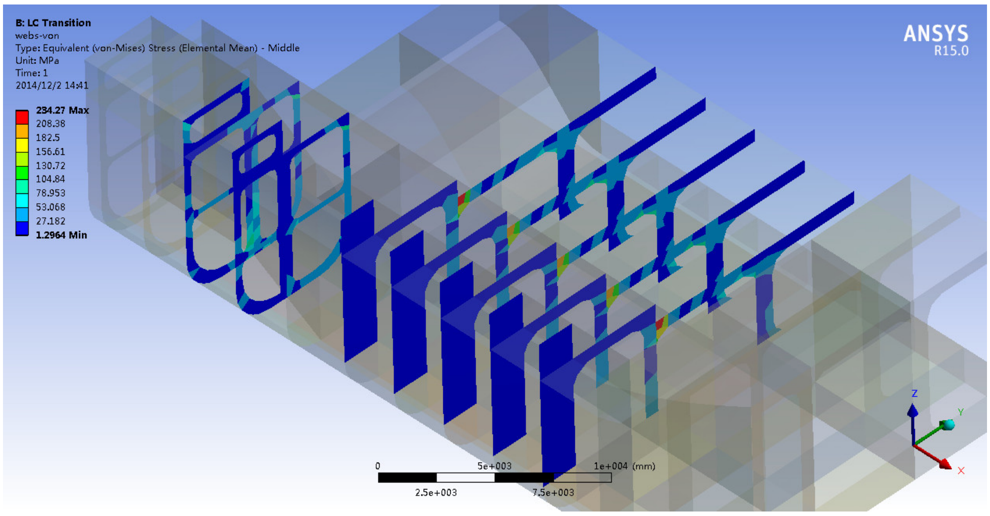

Figure 7.21. Transverse frame plates von Mises stress contour

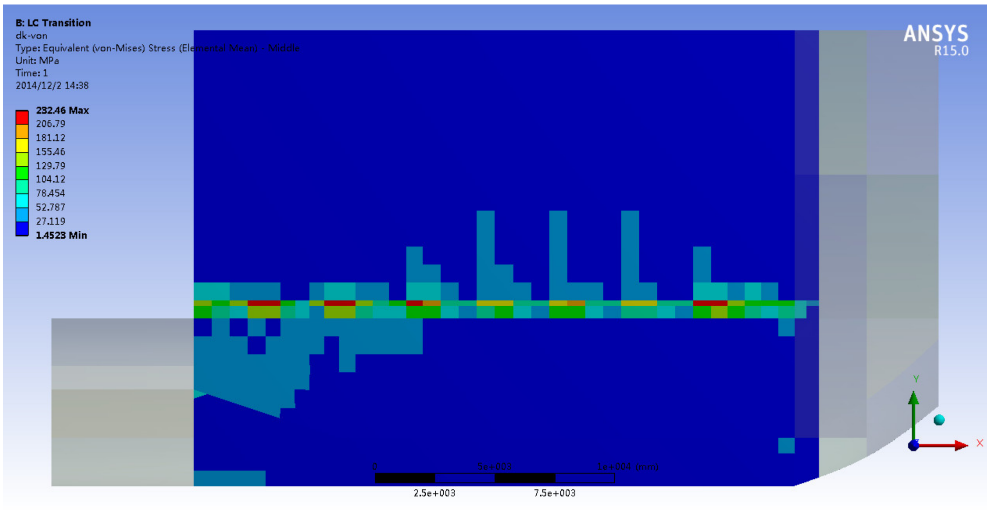

Figure 7.22. Deck plates von Mises stress contour in transition load condition

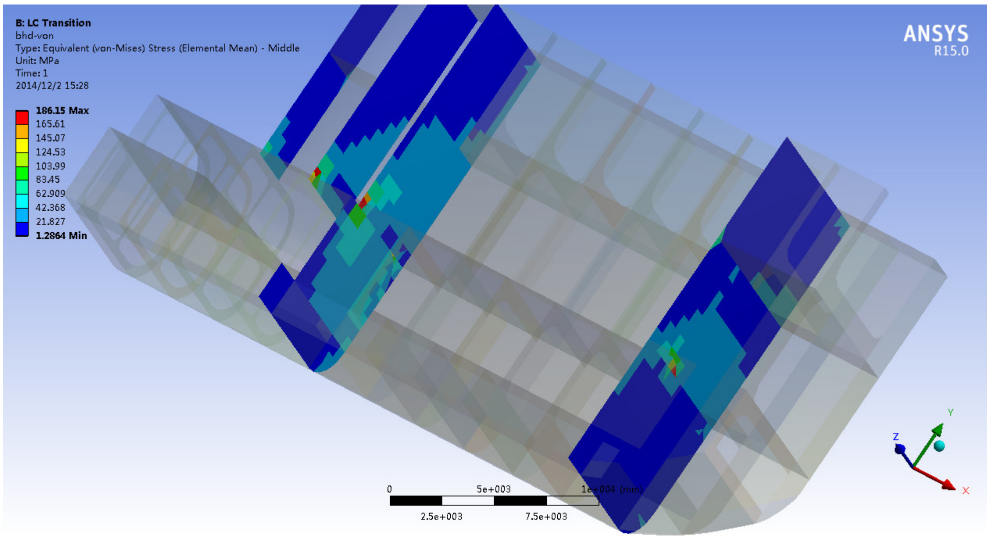

Figure 7.23. Transverse bulkhead plates von Mises stress contour

The results demonstrate that the structures of the vessel satisfy the requirement. The exploitation of the structure stays within a reasonable range considering both the safety and economical requirements. The minimum UC value is 0.46 and the biggest one equals to 0.87. The worst condition happens at the web frames when the vessel is in the transition loading condition. The weight of the foundation and the longitudinal acceleration create this situation. By using high tensile strength steel, the thickness of the plate can be maintained in an acceptable range.

The horizontal acceleration due to ship motion cannot be solved in the same model. An anti-symmetry boundary is set to a new model. Only transverse acceleration, which equals to 0.5 times the gravity constant, and transverse inertial force of the foundation are defined in the model. The load condition can be found in Figure 7.24..

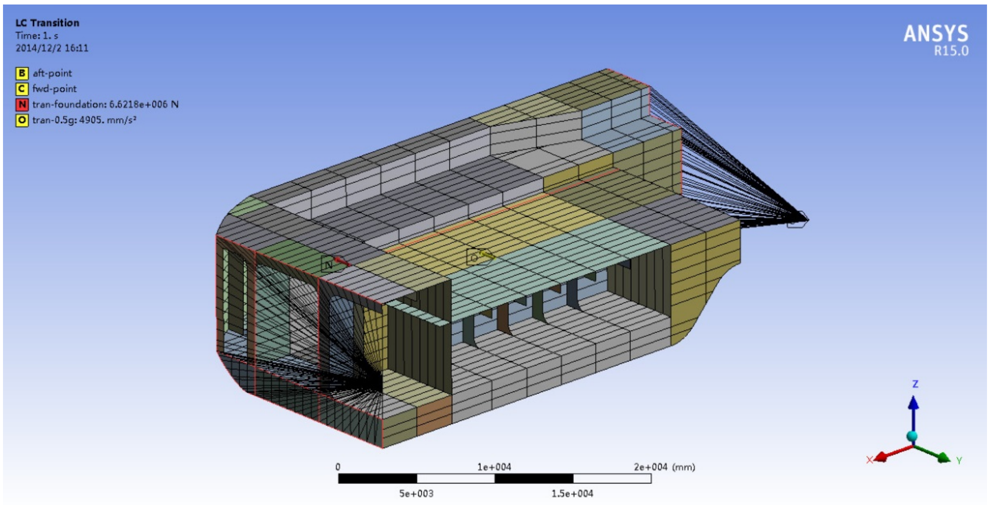

Figure 7.24. Transverse loading due to ship roll motion

The result shows that these transverse loads mainly influence the deck, transverse bulkheads and web frames. Taking these facts into account, the stress level of the vessel still satisfies the yielding requirement. The stress contribution of transverse loads is shown in Figure 7.25., Figure 7.26. and Figure 7.27..

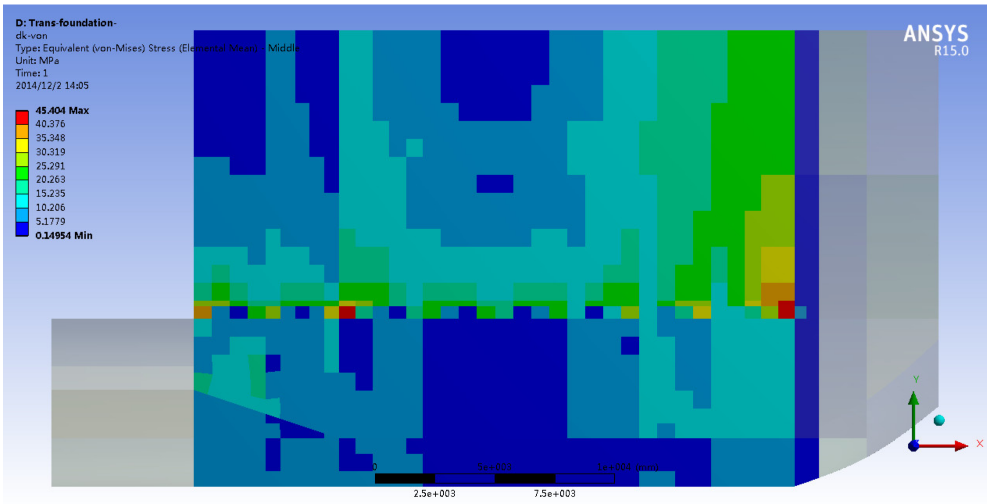

Figure 7.25. Deck von Mises stress contour under transverse loads

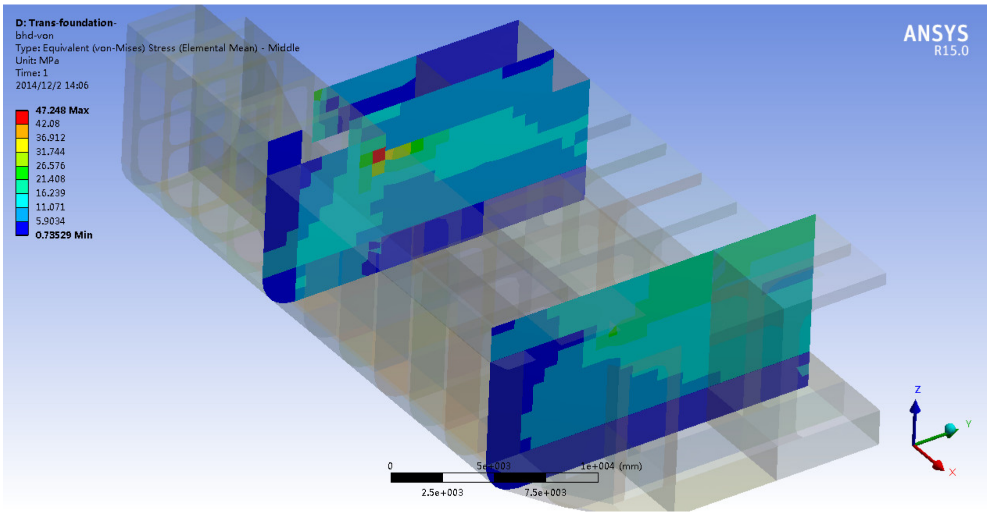

Figure 7.26. Transverse bulkhead von Mises stress contour under transverse loads

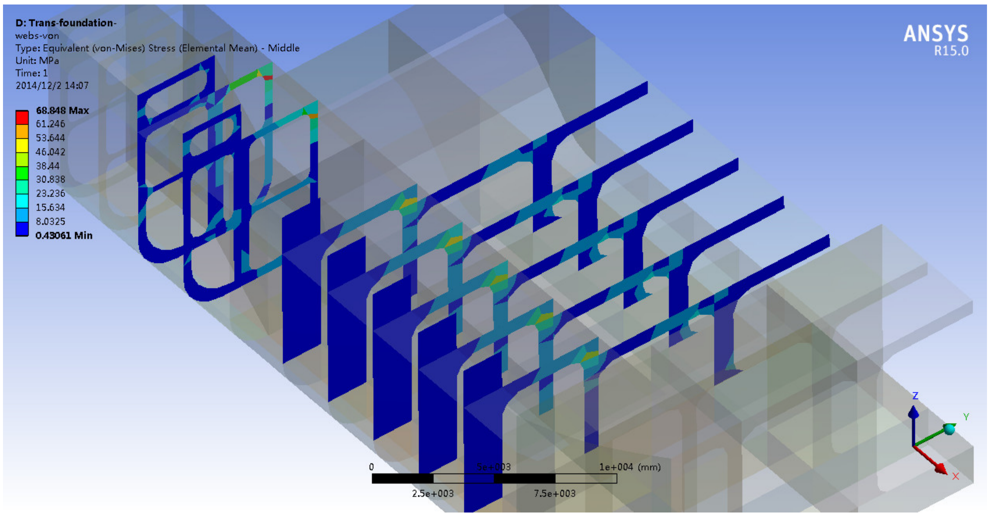

Figure 7.27. Web frame von Mises stress contour under transverse loads

7.7.3 Cargo Fixtures

To be able to give a weight estimation of the two different fixtures used in the offloading operation a quick analysis of two simple designs is done. To simplify the analyses solid beam elements are used, which does not give the optimal solution but is a conservative way to estimate the weight. The concepts should be further evaluated in order to arrange the steel beams in a more efficient way and thereby reducing weight.

7.7.3.1 Foundation fixture

To be able to withstand the weight of the foundation large bending moments should be avoided. An efficient way to lower the moments is to apply the loads close to the vertical sides of the structure. In order to fit the rails the width of the lower part of the fixture was required to be less than 500 mm wide. To simplify the analysis a 400x400 rectangular cross section is chosen for all the beams.

The structure is analysed in ANSYS Workbench with the bottom side of the fixtures vertical part set as fixed and loads with a magnitude of 6.2 MN were applied in four corners as can be seen in Figure 7.28.. The assumption of a fixed boundary condition can be discussed but since the interface of the fixtures and the rails isn't decided it can be sufficient to give the weight estimation based on these calculations.

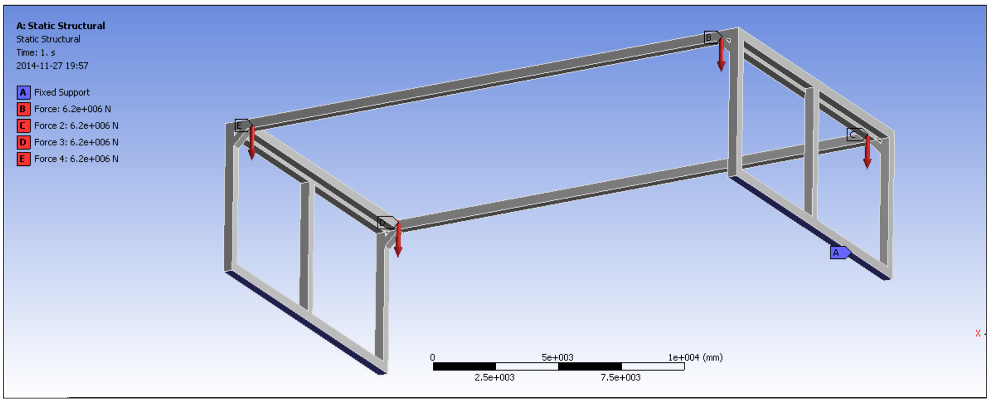

Figure 7.28. – Boundary conditions and loads on foundation fixture.

The von Mises equivalent stress is given as 266 MPa as can be seen in Figure 7.29.. The total deformation in vertical direction in the middle of the horizontal beam is calculated as 58 mm.

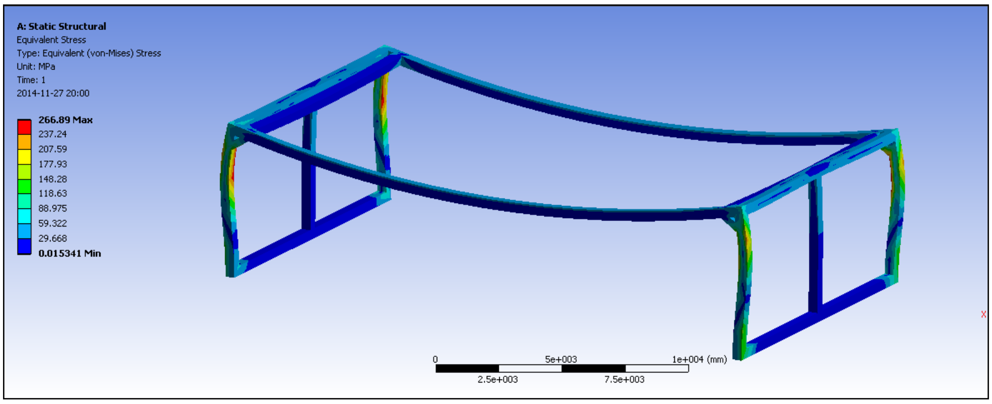

Figure 7.29. – Maximum von Mises stress of foundation fixture.

As the maximum stresses are lower than the yield strength of the normal construction steel the design could be seen as sufficient to validate the weight. The total weight of the structure was estimated to 200 tonnes. The deformation of the fixture is not expected to cause any problems, as this does not affect the lifting operation.

7.7.3.2 Windmill fixture

In order to ensure a safe operation for the offloading of the windmills the deformation of the fixture is of great importance. Since the windmill will be placed in the centre of the fixture there will be large bending moments acting on the fixture. The main focus of the design is to ensure a small deformation in the vertical direction in order to fit the windmill onto the foundation. Another issue to consider is to avoid to large differences in the deformation in the front and back of the fixture in order to avoid the windmill to tilt. This is also the reason to make the fixture reasonably long.

For the windmill fixture solid beam elements with different sized cross-sections are used. A load of 6.9 MN is applied to the fixture through a flange placed on top of the fixture.

The fixture is analysed with a fixed boundary condition in the bottom part of vertical structure of the fixture. The load was applied as a force of 6.9 MN acting downwards on the flange as can be seen in Figure 7.30..

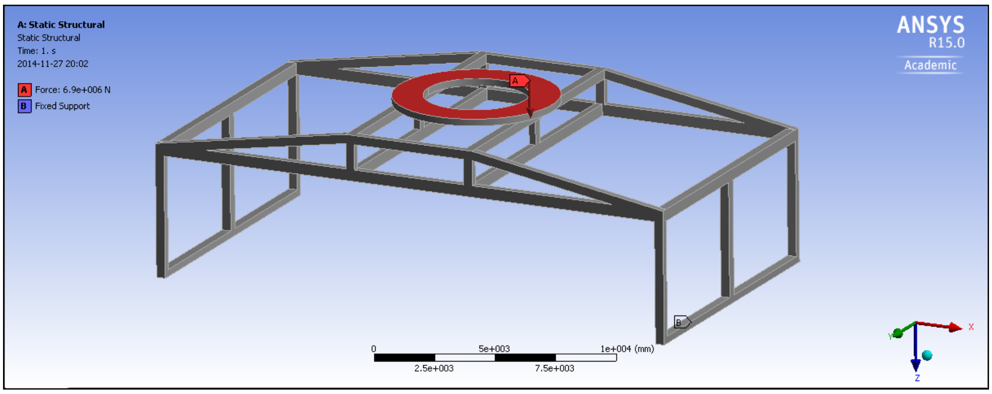

Figure 7.30. – Boundary conditions and loads on windmill fixture.

The results of the analysis give a von Mises equivalent stress of 166 MPa, which is not exceeding the yield stress of normal engineering steel. The allowed vertical deformation in the area of the windmill is limited due to the off-loading operation. The analysis gives a deflection of 58 mm as can be seen in Figure 7.31. which is within the requirement to perform the offloading operation. It would be reasonable to redesign the fixture to avoid stability problems since the centre of gravity of the windmill is placed at a quite large distance from the fixation point in the fixture.

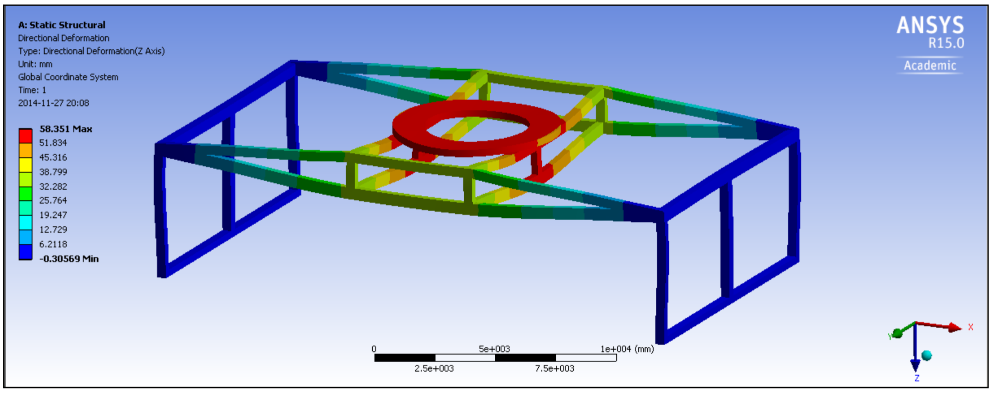

Figure 7.31. – Maximum deformation in z-direction for windmill fixture.

7.7.4 FE-Analysis on the foldable rails

The concept for the foldable rails is presented in Section 6.10.1. The model analysed here complies with the conditions of small deformations and sufficient supporting strength of the rails. Furthermore it provides an insight in the stress distribution during offloading. Because the rails are supported at multiple locations with the same span, it is sufficient to only model the first and second supporting elements, while isolating them by a plane going through the middle of a span. The boundary conditions include the support from the shaft and the corner plates, the load distribution from the fixture and the largest fundament, and the conditions on the edges from the cutting plane (no out-of-plane deformations; no rotation around the x-axis). The analysis is performed in ANSYS Workbench 15.0.

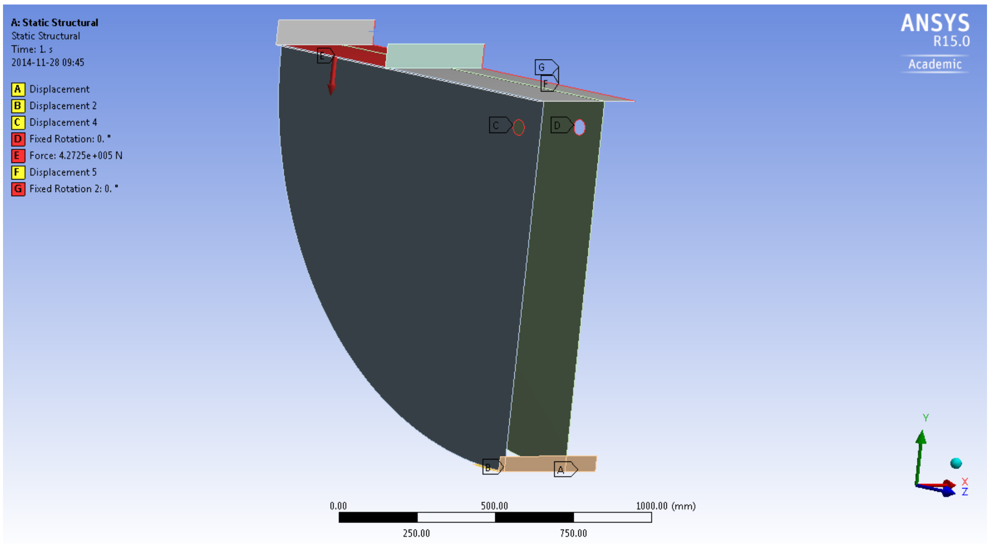

Figure 7.32. Boundary conditions on the retractable rail model

The results demonstrate the structural behaviour of the model. The rails provide a satisfying serviceability. The small deformations won't interfere with the sliding and handling of the fixture. A slight risk of buckling arises from the large size of the quarter circle plates, but it can be easily controlled in a later design loop when there is more clarity on the full operation, the needed equipment and additional factors. The stresses reach low values and suggest the possibility of even further reduction of the thickness of some elements. Higher values are present around the cut-out from the shaft support. The main dimensions from the current design can be found in Appendix D.

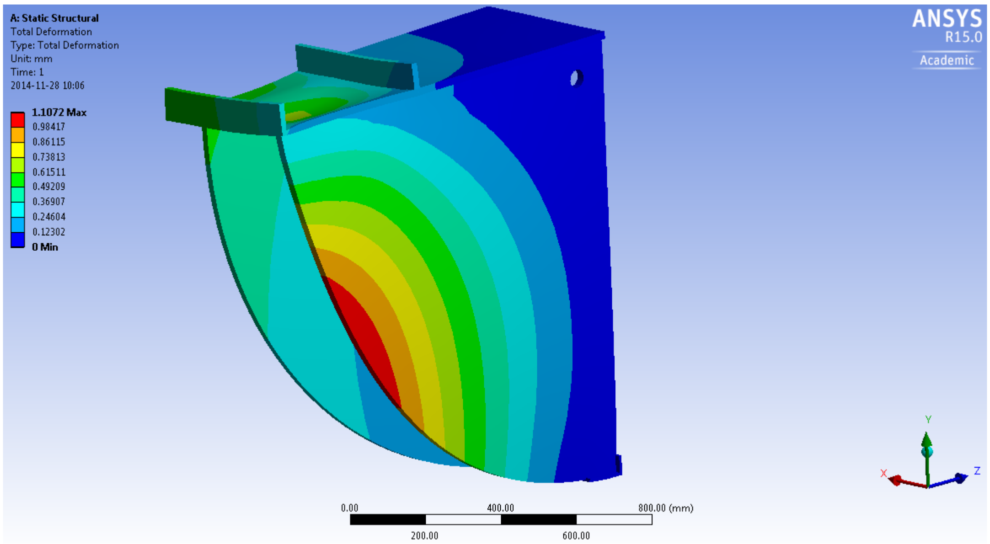

Figure 7.33. Total deformation of the retractable rails

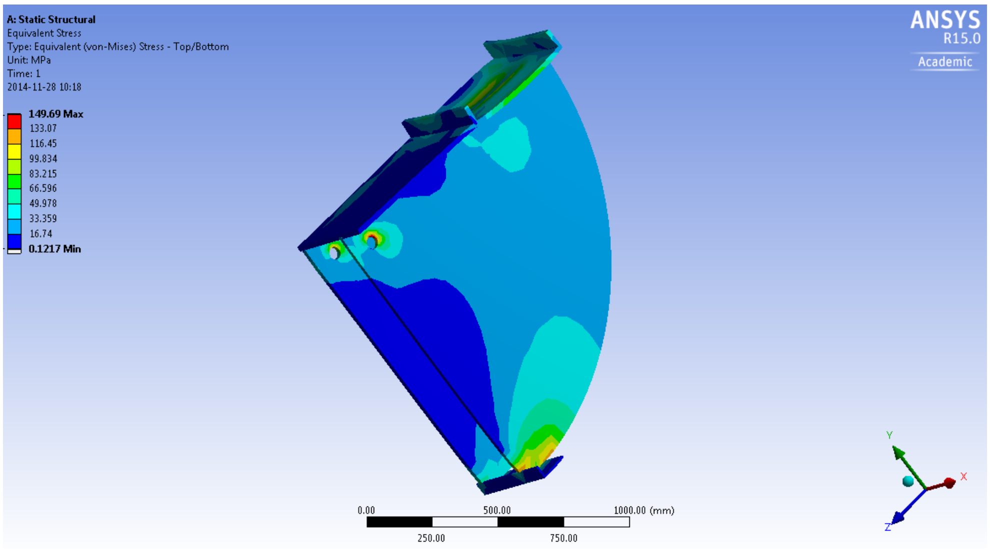

Figure 7.34 von Mises equivalent stress distribution of the retractable rails

7.8 Future work

The work described in the report has certain boundaries. On the one hand the limited time sets the need to prioritize the main tasks and leave out other aspects in need of development. On the other hand the missing pieces of information are replaced with assumptions. These limitations outline the whole of the presented work

The structural design of the bow part should be considered during the further development of the project. The scantling should be chosen and the stresses resulting from the mooring should be evaluated. Investigation of the structural response to slamming is also an aspect to be taken into account.

From the results of the FE-Analysis on the two global models it is clear that some redesign on local structural elements is required in order to withstand the expected loads. Some suggestions for redesign have been given. In the opposite, some locations are subjected to rather low stresses. This means that there is a poor utilisation of the material. Components and elements with low stresses should be optimized in order to reduce more weight. These will however need to be evaluated in the same manner as the initial design.

Further suggestion for future work is to incorporate the dynamic load from a wave environment in a global model of the complete aft to mono-hull part of the vessel. This would give essential information on the relative displacements between the catamaran pontoons. A possibility is that the misalignment between the cargo rails can reach unacceptable levels for the cargo offloading procedure to be executed. Therefore it is considered necessary to conduct such an evaluation.

The boundary conditions of the finite element model should be improved as well. A spring support boundary should be added to the transverse and vertical structure members at each end section. The end bending moment and shear force shall be corrected according to the global bending moment then the deformation of the vessel shall be more accurate. In the stress concentration regions, a submodeling shall be carried out to investigate the stress level at these areas; consequently, a fatigue analysis can be applied as well. Moreover, the buckling check shall be improved by following DNV buckling recommended practices (DNV, 2012) using the stress status from finite element analysis results.

Additionally there is a need to investigate local regions of the structure such as winch and crane foundations and members supporting the superstructure. This would preferably also be done by constructing local FE models of the structure in the vicinity of these locations.

The fixtures designed for the foundation and the windmill has to be reviewed. The current design is just to get an estimation of the weight. The fixture for the windmill in the final design will include hydraulic dampers as well which are not included in the current design.

The retractable rails are a feature of the vessel that is only remotely investigated in the current project. The lack of input information combined with the early stage of the ship design is the cause for a model based mainly on assumptions. At this stage, the arrangement of the supporting plates between transverse structural members of the hull and the consideration of an unfolding and a locking mechanisms present obvious open questions. Later on, when that is solved and more information about the rails and the operation procedures is obtained, the dimensions should be updated and a more thorough analysis should be performed. An important aspect of it would be the local investigation on the interaction between the catamaran hulls and the loaded unfolded rails.

At a later design loop, when the majority of structural problems are solved, considerations on the human factors should be implemented in the design. When the safety of the crew, the vessel and its cargo is ensured from structural point of view, local changes should be adopted. These will be influenced by the movement of the crew members during operation and break times, importance of survey locations, possible maintenance works, etc. However, even after sufficient information on that matter is provided, the priority of the design is the structural integrity of the ship and the compliance with the chosen rules and regulations.

Finally, any future work should deal with the problem of grounding the vessel during loading of the units. A design of a hull-shaped bed is needed for the ship to rest on and the respective structure response (i.e. from possible impact, static loads, etc.) should be considered.

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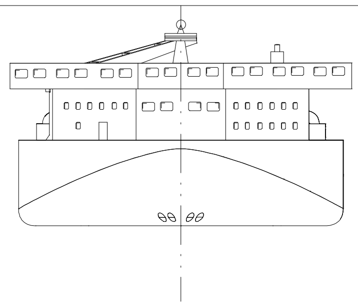
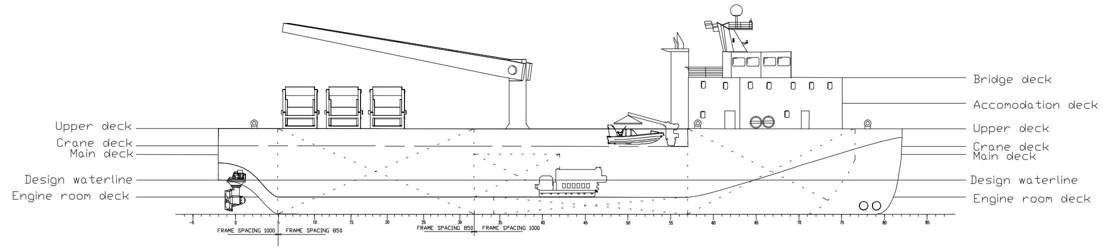
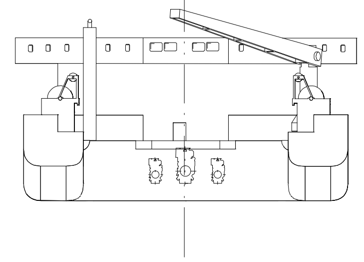
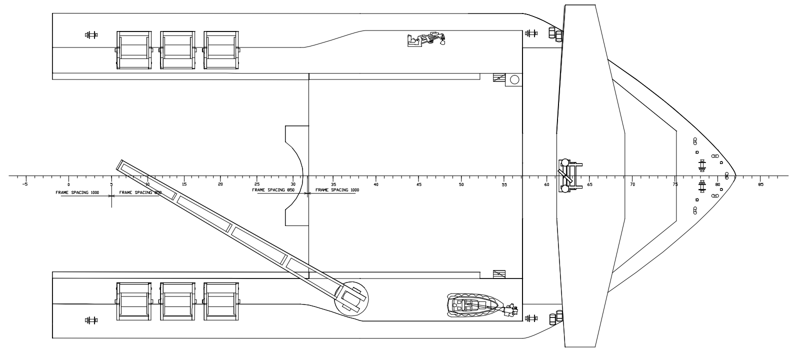
List of drawings

Appendix A









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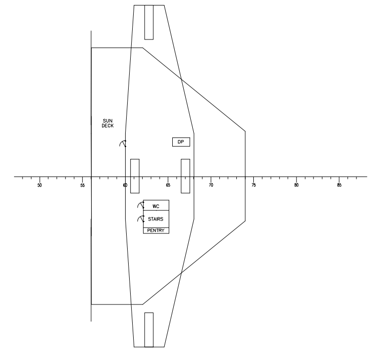

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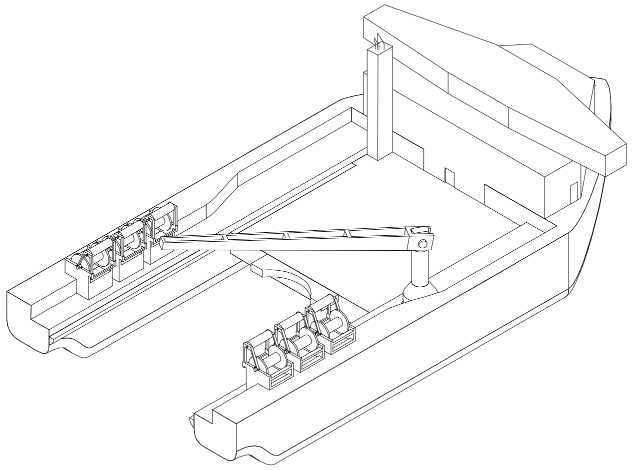

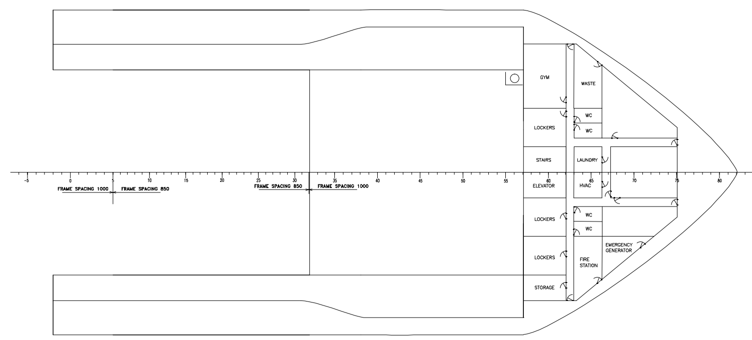

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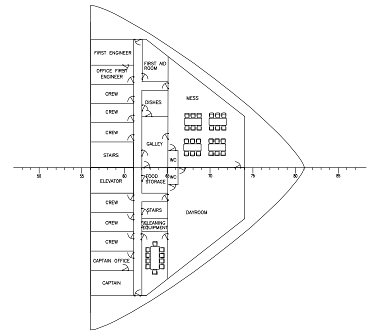

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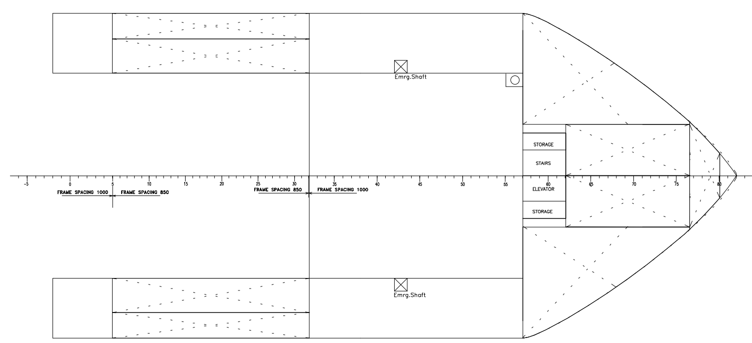

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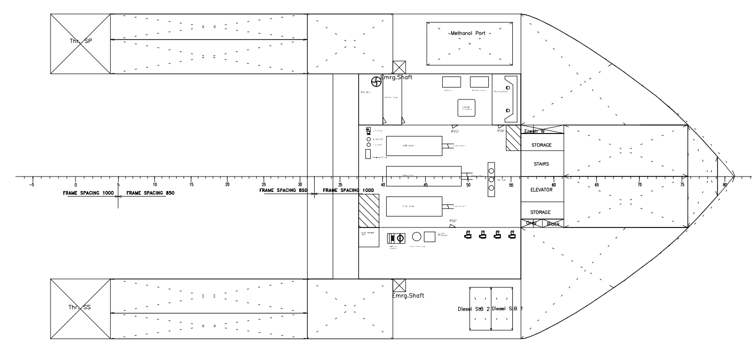

1st DECK, ENGINE ROOM DECK
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LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number MDP2014		
	OWNER Blekinge Offshore		
	Class DNV GL		
Wind Installation Vessel KERMIT General Arrangement	PREP. BY MF	DATE 02/12/2014	
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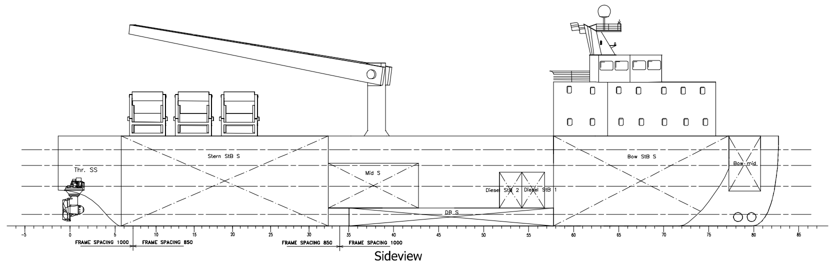

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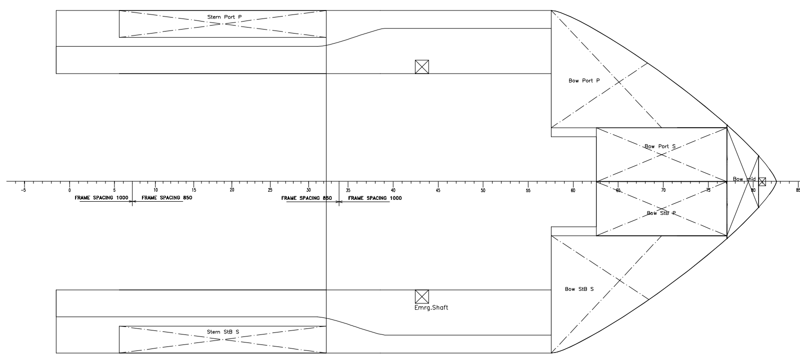

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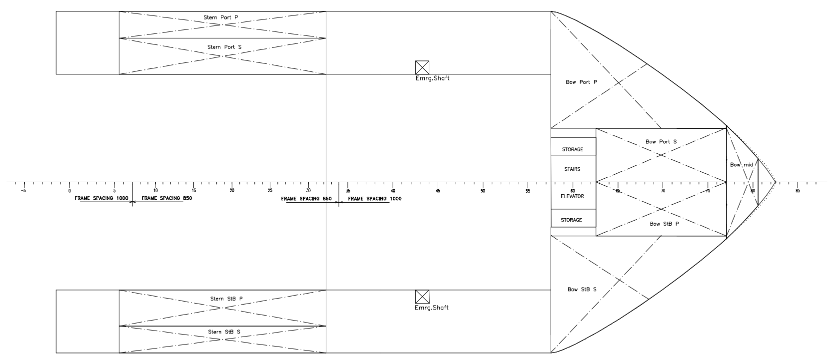

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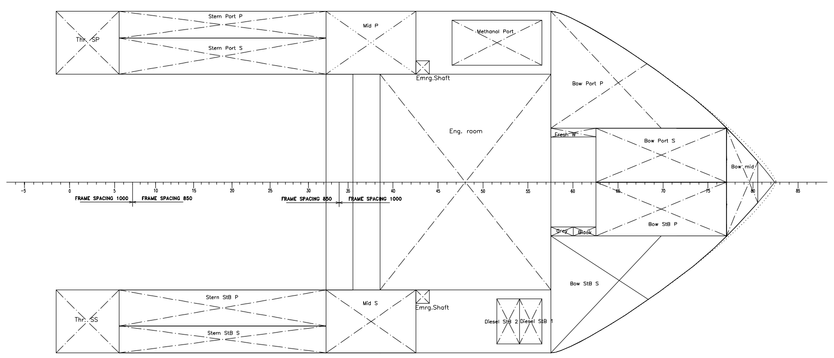

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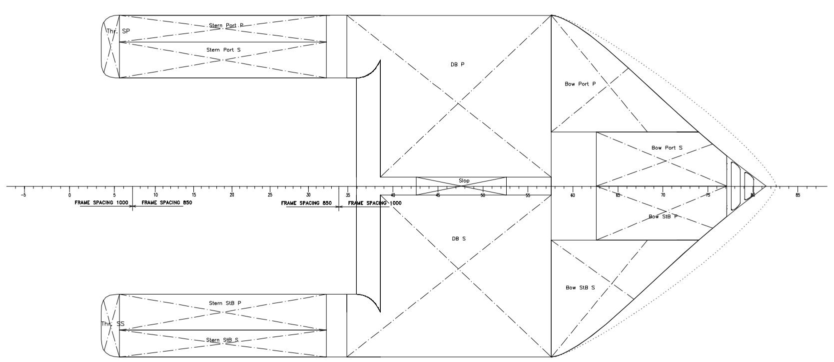

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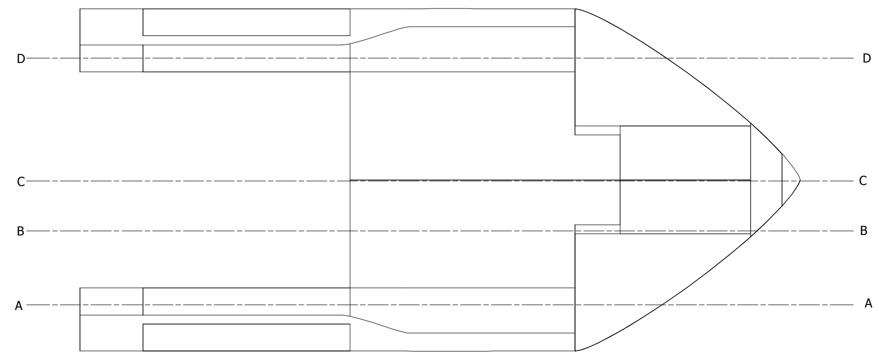

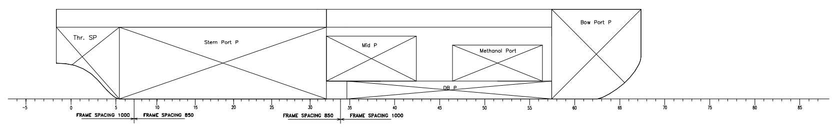

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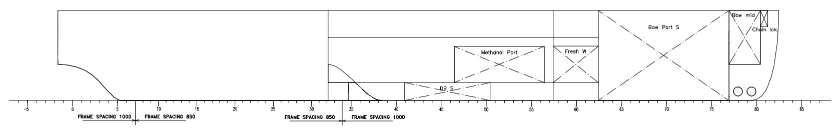

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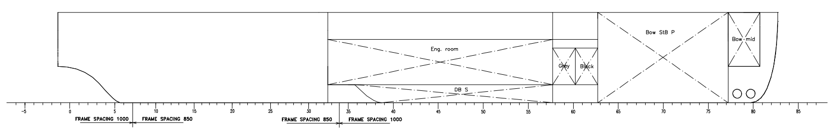

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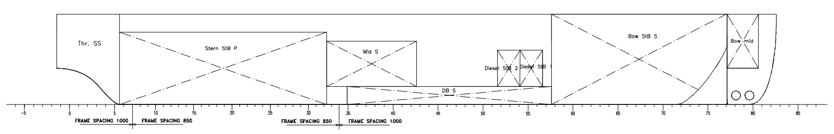

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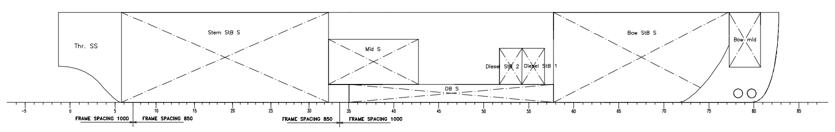

Sideview

Printed in A3, scale 1:800

Main dimensions Kermit

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m

Ballast water					
COMP. ID.	TITLE	VOLUME (m3)	VCG (m from B.L.)	LCG (m from A.P.)	TCG (m from C.L.)
Bow Port P	BALLAST	1241,447	5,323	61,628f	10,641p
Bow Port S	BALLAST	805,668	5,055	67,163f	2,961s
Bow StB P	BALLAST	805,668	5,055	67,163f	2,961p
Bow StB S	BALLAST	1241,447	5,323	61,628f	10,641s
Bow mid	BALLAST	166,372	7,204	76,001f	0
Stern Port P	BALLAST	639,333	5,115	18,498f	14,028p
Stern Port S	BALLAST	687,401	4,062	18,5f	14,028s
Stern StB P	BALLAST	687,401	4,062	18,5f	17,473s
Stern StB S	BALLAST	639,333	5,115	18,498	10,196p
DB P	BALLAST	747,83	1,028	43,717	10,196s
DB S	BALLAST	747,83	1,028	43,717	s15,5p
Mid P	BALLAST	332,5	4,5	35	15,5s
Mid S	BALLAST	332,5	4,5	35	15,5s
Methanol Port	METHANOL	190	4	49	15,5p
Diesel StB 1	DIESEL	47,5	4	50,25	15,5s
Diesel StB 2	DIESEL	47,5	4	52,75	47,5
Fresh W	FRESH WATER	18,838	4	57,521	5,5p
Grey	GRAY WATER	9,5	4	58,75	5,5s
Black	BLACK WATER	9,419	4	56,261	5,5s
Lube	LUBE OIL	0,95	2,5	40,5	0
Slop	SLOP TANK	38	1	45	0
TOTAL		9436,43	4,04	46,43	1

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number MDP2014		
	OWNER Blekinge Offshore		
	Class DNV GL		
Wind Installation Vessel KERMIT Tank Arrangement	PREP. BY MF	DATE 02/12/2014	
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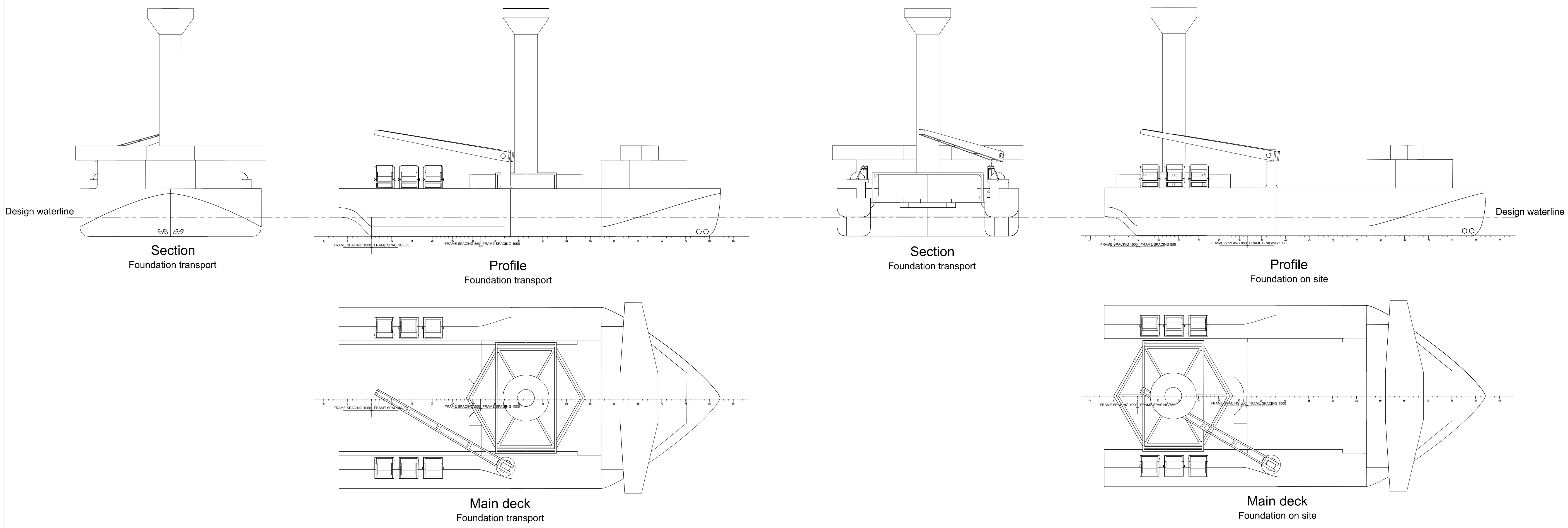
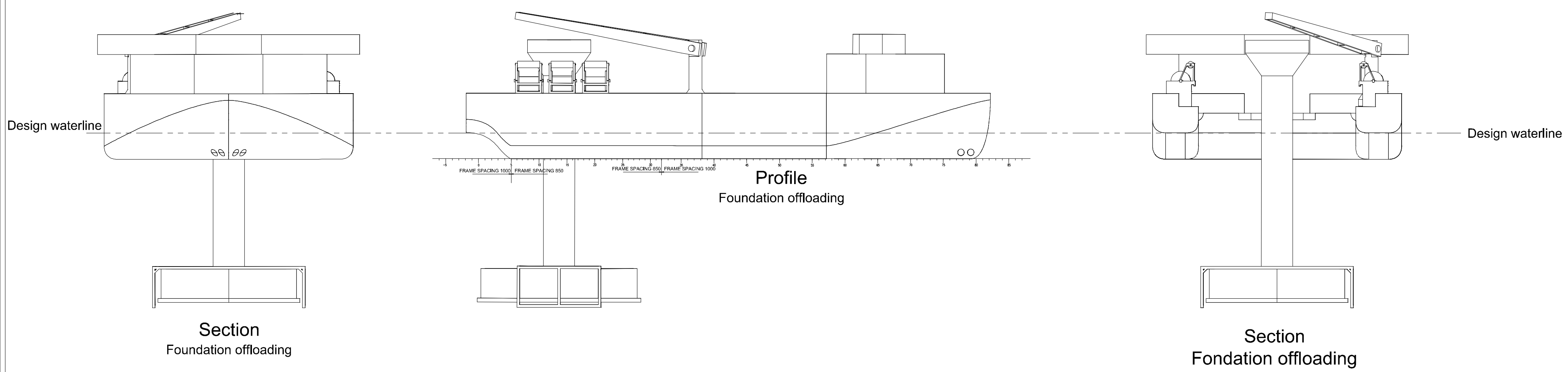
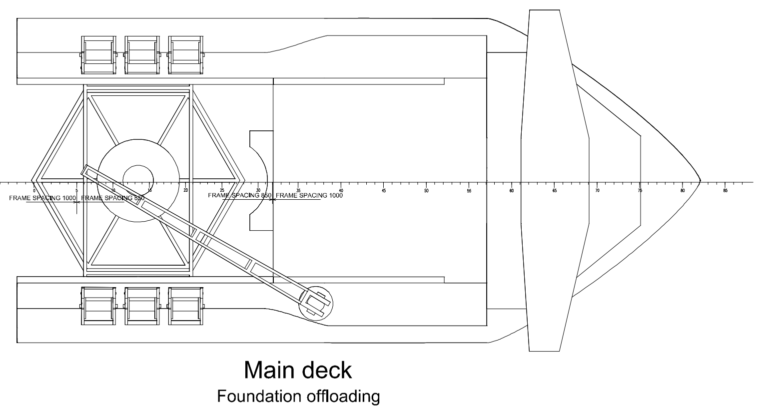






Printed in A3, scale 1:800

Main dimensions Kermit

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m

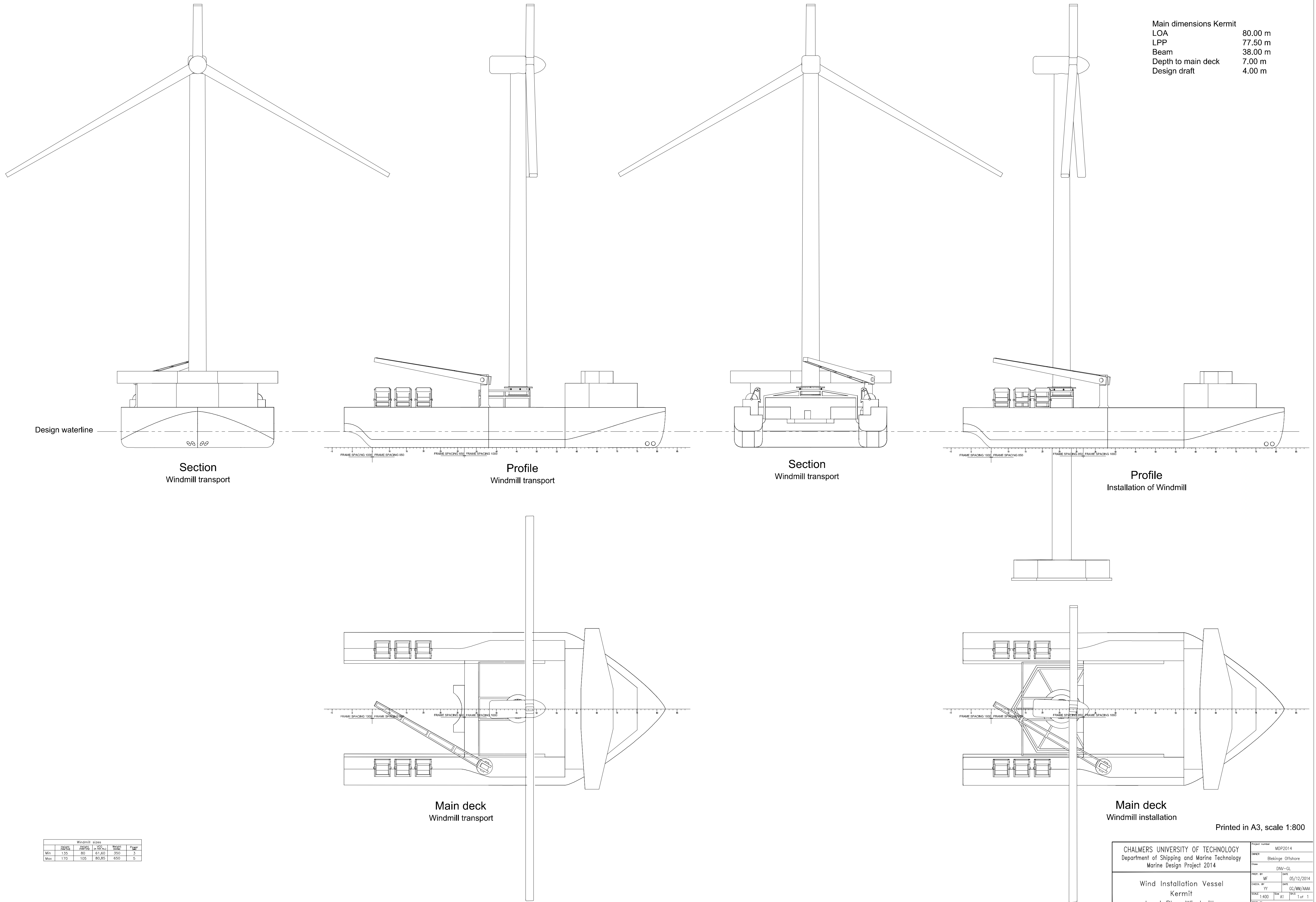
Foundation sizes				
	VOLUME	VOLUME (m3)	VCG (m from B.L.)	Weight (tonnes)
Min	15,000	541	6,740	1389
Max	40,000	987	14,665	2533

CHALMERS UNIVERSITY OF TECHNOLOGY
Department of Shipping and Marine Technology
Marine Design Project 2014

Wind Installation Vessel
Kermit
Load Plan Foundations

Project number	MDP2014
OWNER	Blekinge Offshore
Class	DNV-GL
PREP. BY: MF	DATE: 05/12/2014
CHECK. BY: YY	DATE: GG/MM/AAAA
SCALE: 1:400	Size: A1 / SH.S: 1 of 1
DRAW. N°	MDP2014_2-300-01

Main dimensions Kermit	
LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m



Printed in A3, scale 1:800

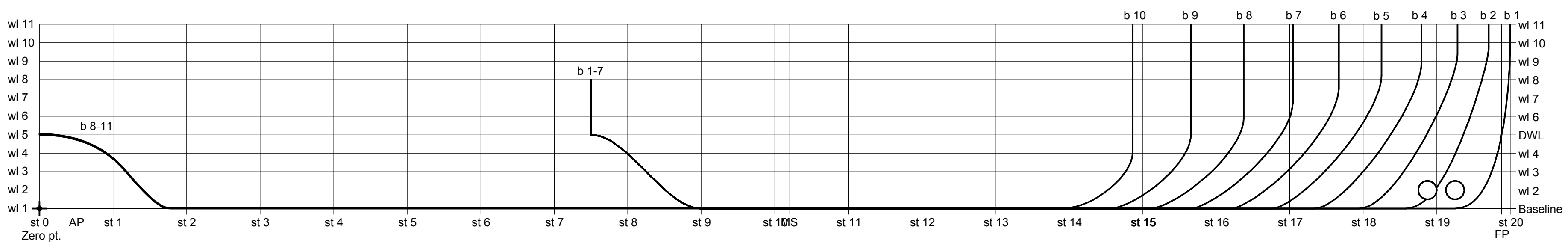
Windmill sizes					
	Height (Total [m])	Height (Tower [m])	VCG (m from B.L.)	Weight (tonnes)	Power (MW)
Min	135	80	61,60	350	3
Max	170	105	80,85	650	5

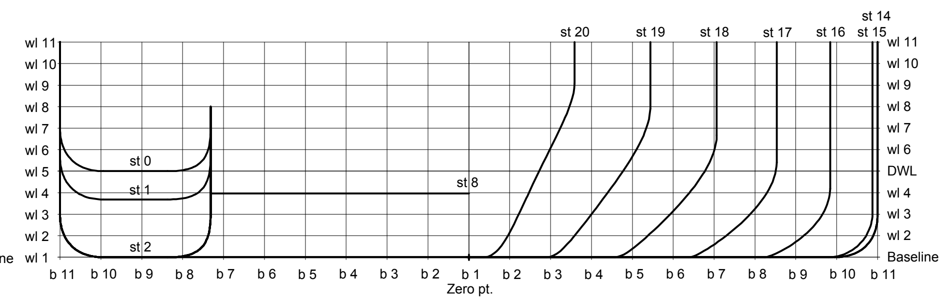
CHALMERS UNIVERSITY OF TECHNOLOGY
Department of Shipping and Marine Technology
Marine Design Project 2014

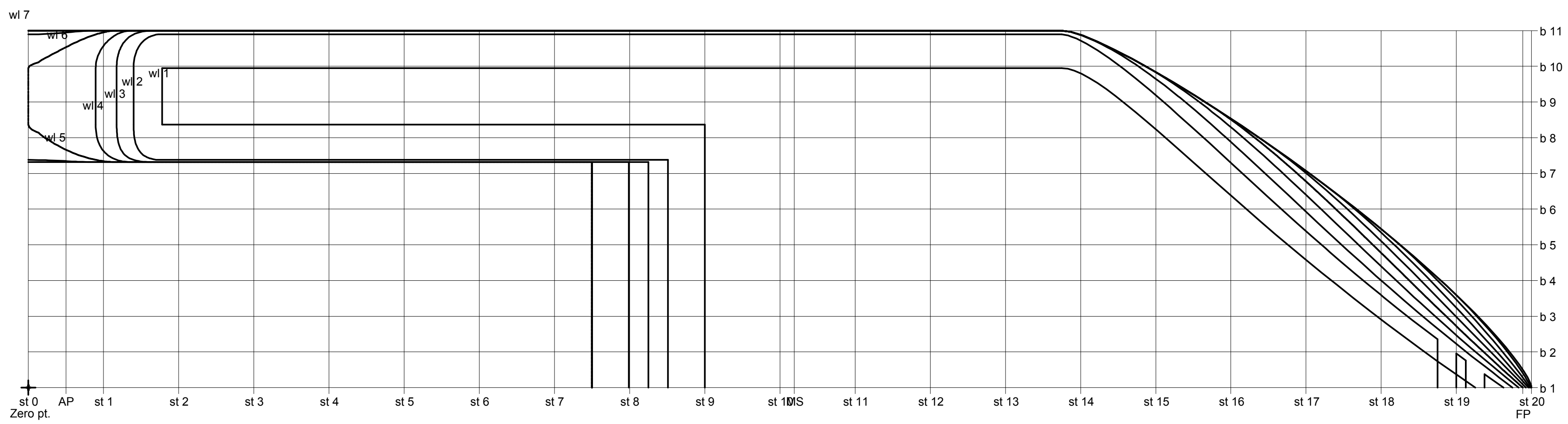
Wind Installation Vessel
Kermit
Load Plan Windmills

Project number	MDP2014				
OWNER	Blekinge Offshore				
Class	DNV-GL				
PREP. BY	MF	DATE	05/12/2014		
CHECK. BY	YY	DATE	GG/MM/AAAA		
SCALE	1:400	Size	A1	SH.S	1 of 1
DRAW. N°	MDP2014_2-300-02				

Appendix B



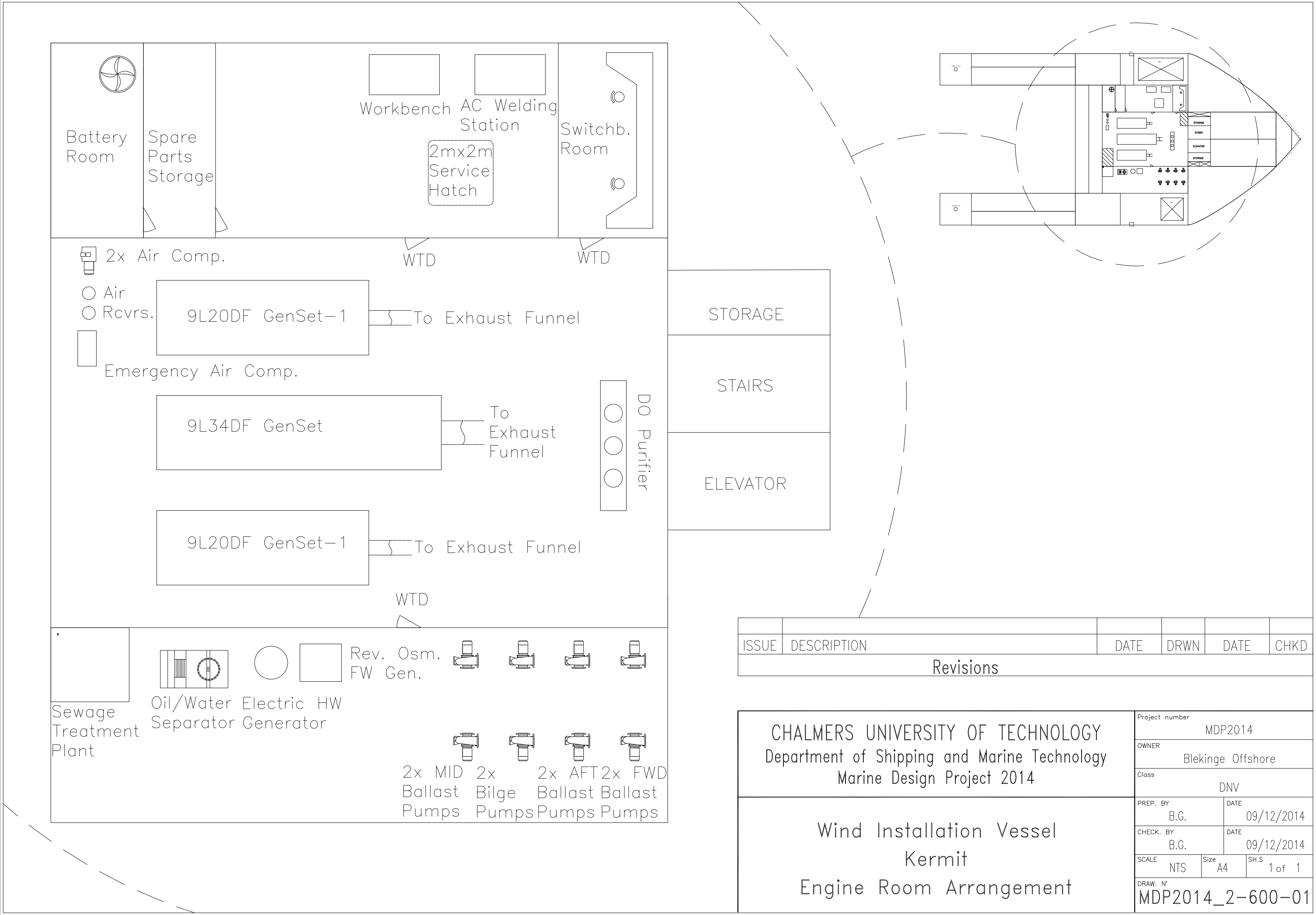




Main dimensions Kermit	
LOA	80.0 m
LPP	77.5 m
Beam	38.0 m
Depth to main deck	7.0 m
Design draft	4.0 m

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number: MDP2014 OWNER: Blekinge Offshore Class: DNV
Wind Installation Vessel KERMIT Hull Lines	PREP. BY: AP — DATE: 09/12/2014 CHECK. BY: KL — DATE: 09/12/2014 SCALE: 1:400 — Size: A3 — SH.S: 1 of 1 DRAW. N°: MDP2014_2-101-02

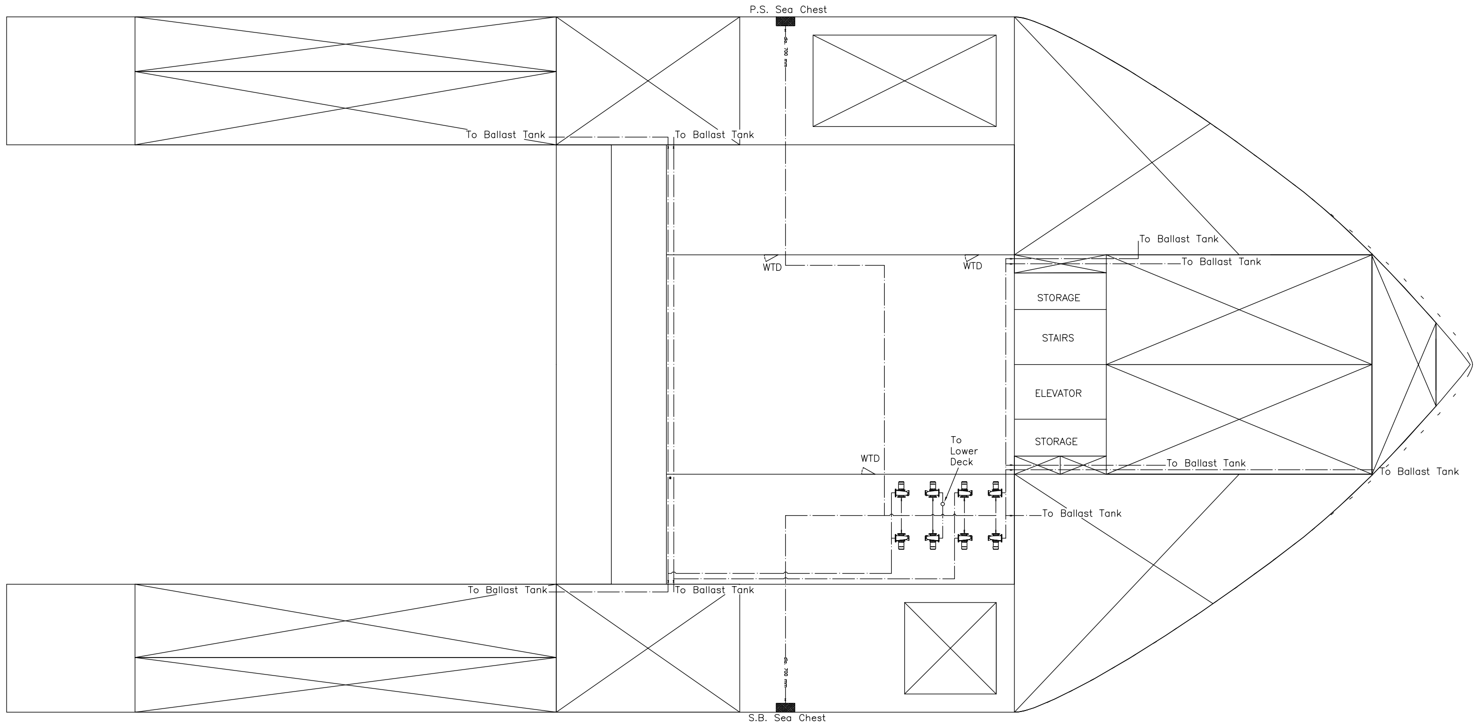
Appendix C



ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number: MDP2014		
	OWNER: Blekinge Offshore		
	Class: DNV		
Wind Installation Vessel Kermit Engine Room Arrangement	PREP. BY: B.G.	DATE: 09/12/2014	
	CHECK. BY: B.G.	DATE: 09/12/2014	
	SCALE: NTS	Size: A4	SH.S: 1 of 1
	DRAW. N°: MDP2014_2-600-01		



ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY
Department of Shipping and Marine Technology
Marine Design Project 2014

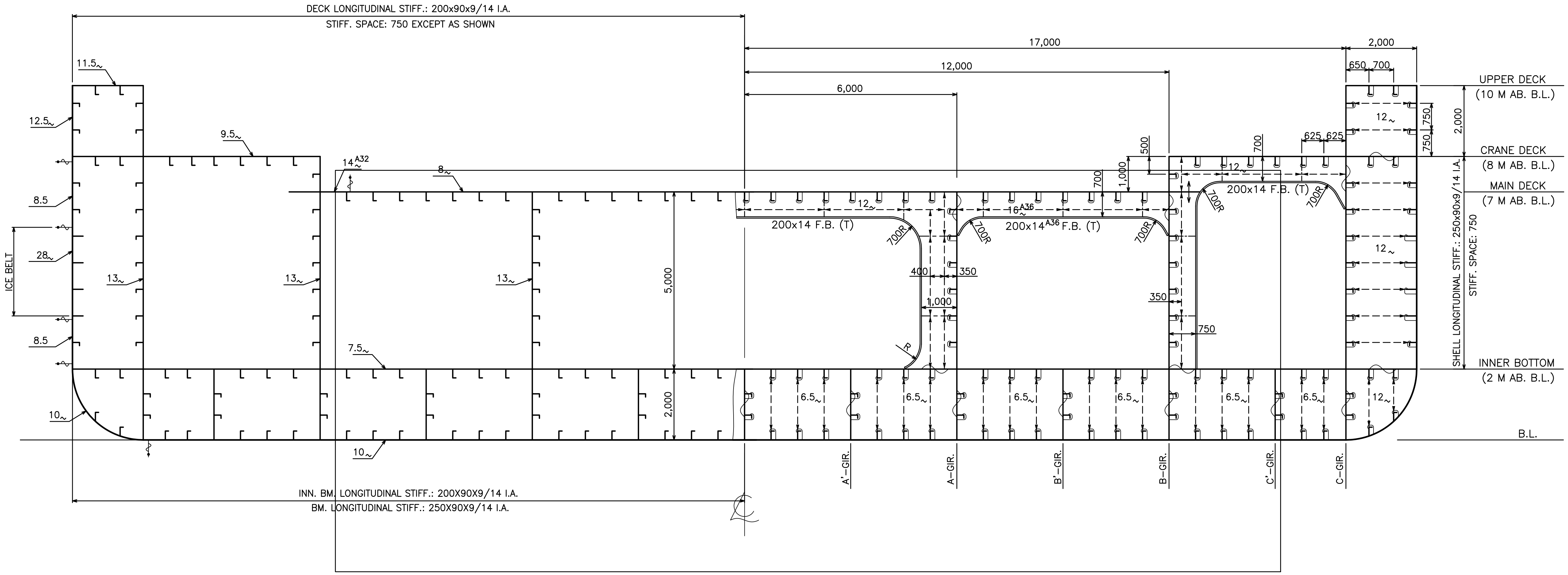
Wind Installation Vessel
Kermit
Ballast System Layout

Project number	MDP2014				
OWNER	Blekinge Offshore				
Class	DNV				
PREP. BY	B.G.		DATE	09/12/2014	
CHECK. BY	B.G.		DATE	05/12/2014	
SCALE	NTS	Size	A4	SH.S	1 of 1
DRAW. N°	MDP2014_2-600-02				

Appendix D

TYPICAL FRAME SECTION

WEB FRAME: STIFF.: 200x14 F.B.



NOTE:

1. BOTH SIDES ARE SYMMETRICAL UNLESS OTHERWISE SPECIFIED.

2. ALL TRANSVERSE SECTIONS ARE SHOWN IN LOOKING FORWARD.

3. STEEL QUALITIES

   A36.....MARKED TO BE GRADE "NV–36" STEEL.

   A32.....MARKED TO BE GRADE "NV–32" STEEL.

   UNSPECIFIED TO BE GRADE "NV–NS" STEEL.

Main dimensions Kermit installation vessel

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depthto main deck	7.00 m
Design draft	4.00 m
Scantlign draught	6.20 m

ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

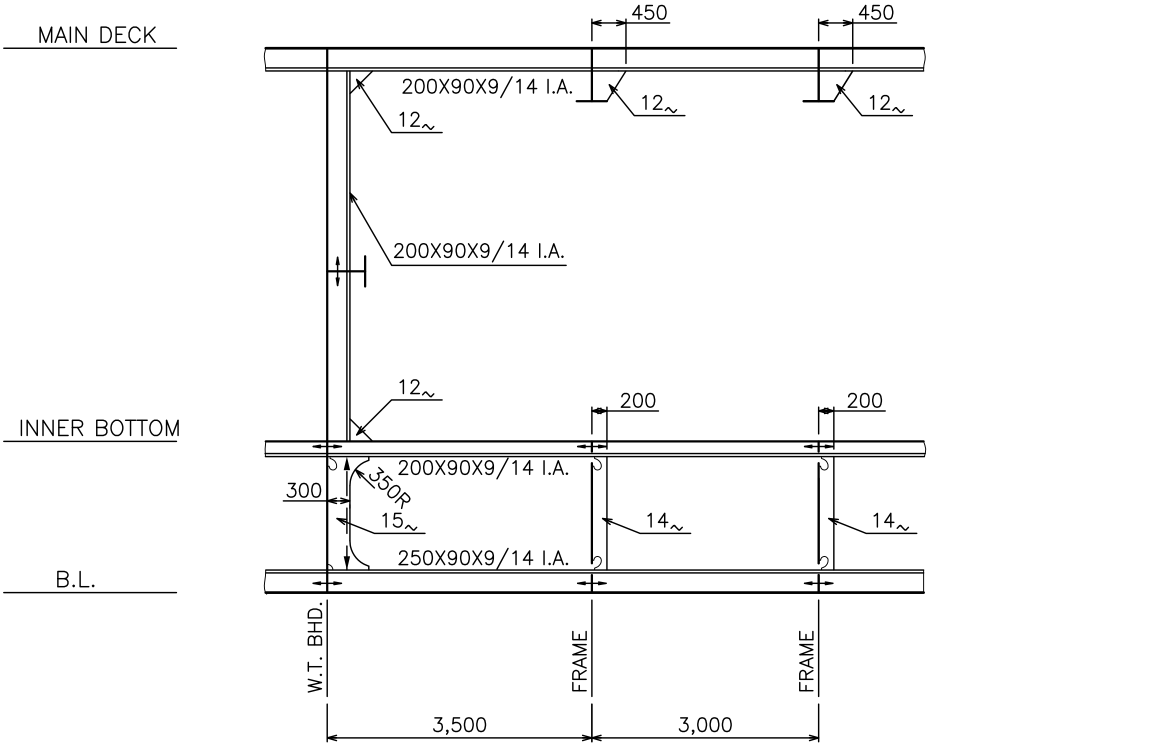
Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY
Department of Shipping and Marine Technology
Marine Design Project 2014

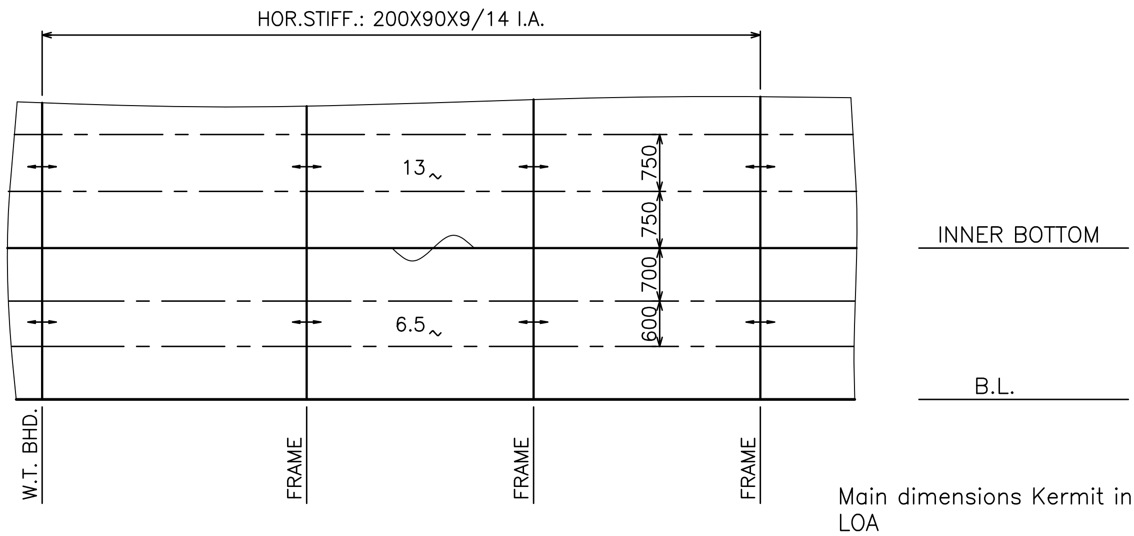
Wind Installation Vessel
Kermit
Frame section monohull

Project number	MDP2014
Owner	Blekinge Offshore
Class	DNV
Prep. by	Structure
Date	05/12/2014
Check. by	YY
Date	GG/MM/AAAA
Scale	1:125
Size	A3
Shts	1 of 3
Draw. n°	MDP2014_2-200-05

TYPICAL LOGNL. SEC.



TYPICAL GIR. SEC.



Main dimensions Kermit installation vessel

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depthto main deck	7.00 m
Design draft	4.00 m
Scantlign draught	6.20 m

NOTE:

1. BOTH SIDES ARE SYMMETRICAL UNLESS OTHERWISE SPECIFIED.

2. ALL TRANSVERSE SECTIONS ARE SHOWN IN LOOKING FORWARD.

3. STEEL QUALITIES

   A36......MARKED TO BE GRADE "NV-36" STEEL.

   A32......MARKED TO BE GRADE "NV-32" STEEL.

   UNSPECIFIED TO BE GRADE "NV-NS" STEEL.

ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY
Department of Shipping and Marine Technology
Marine Design Project 2014

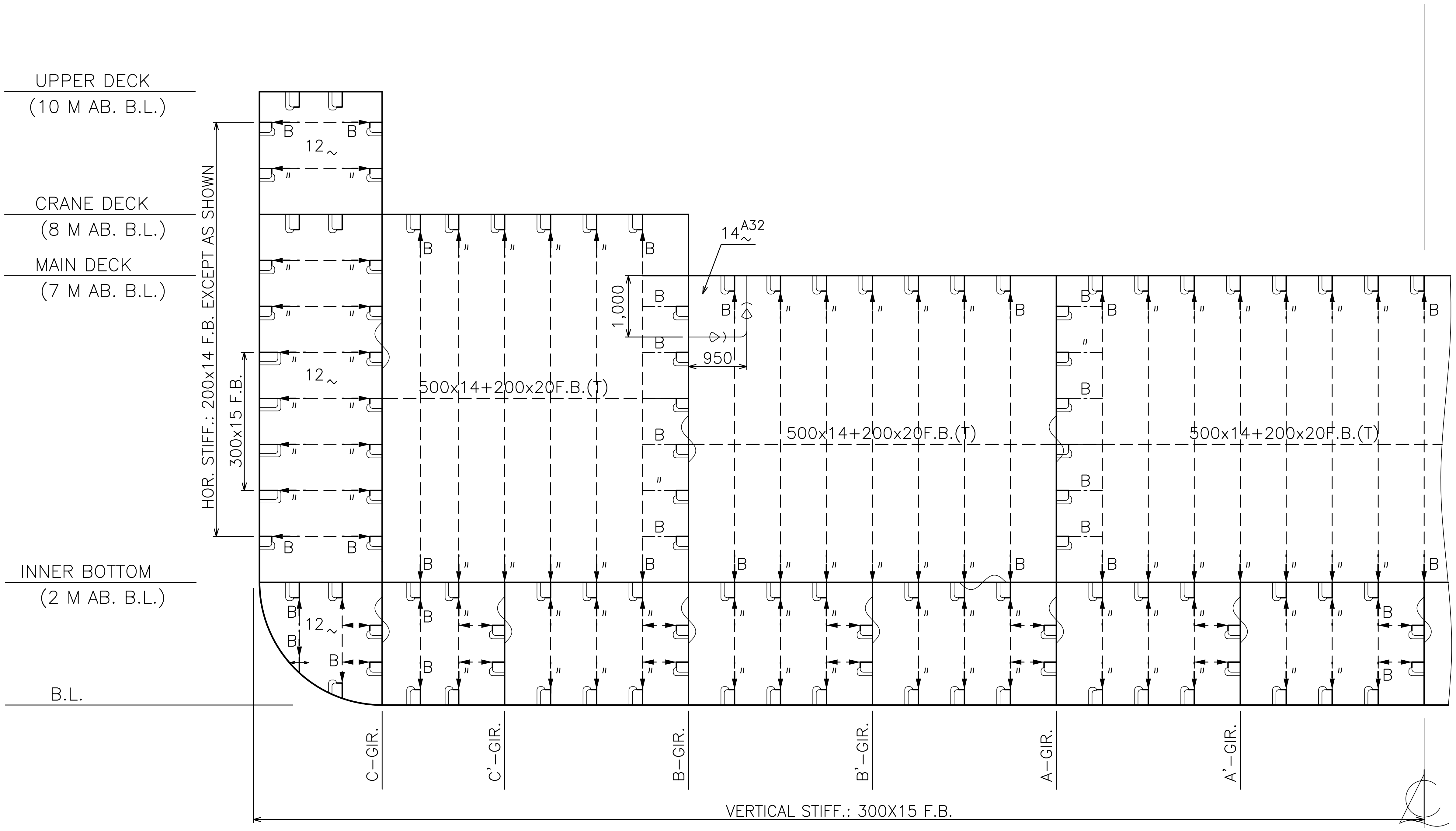
Wind Installation Vessel
Kermit
Frame section monohull

Project number	MDP2014
OWNER	Blekinge Offshore
Class	DNV
PREP. BY	Structure
DATE	05/12/2014
CHECK. BY	YY
DATE	GG/MM/AAAA
SCALE	1:100
Size	A4
SH.S	2 of 3
DRAW. N°	MDP2014_2-200-05

W.T. BHD.

PLATE THICKNESS : 9~ EXCEPT AS SHOWN

VERTICAL STIFF: 200x90x9/14 I.A. EXCEPT AS SHOWN



NOTE:

1. BOTH SIDES ARE SYMMETRICAL UNLESS OTHERWISE SPECIFIED.

2. ALL TRANSVERSE SECTIONS ARE SHOWN IN LOOKING FORWARD.

3. STEEL QUALITIES

A36......MARKED TO BE GRADE "NV–36" STEEL.

A32......MARKED TO BE GRADE "NV–32" STEEL.

UNSPECIFIED TO BE GRADE "NV–NS" STEEL.

Main dimensions Kermit installation vessel

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depthto main deck	7.00 m
Design draft	4.00 m
Scantlign draught	6.20 m

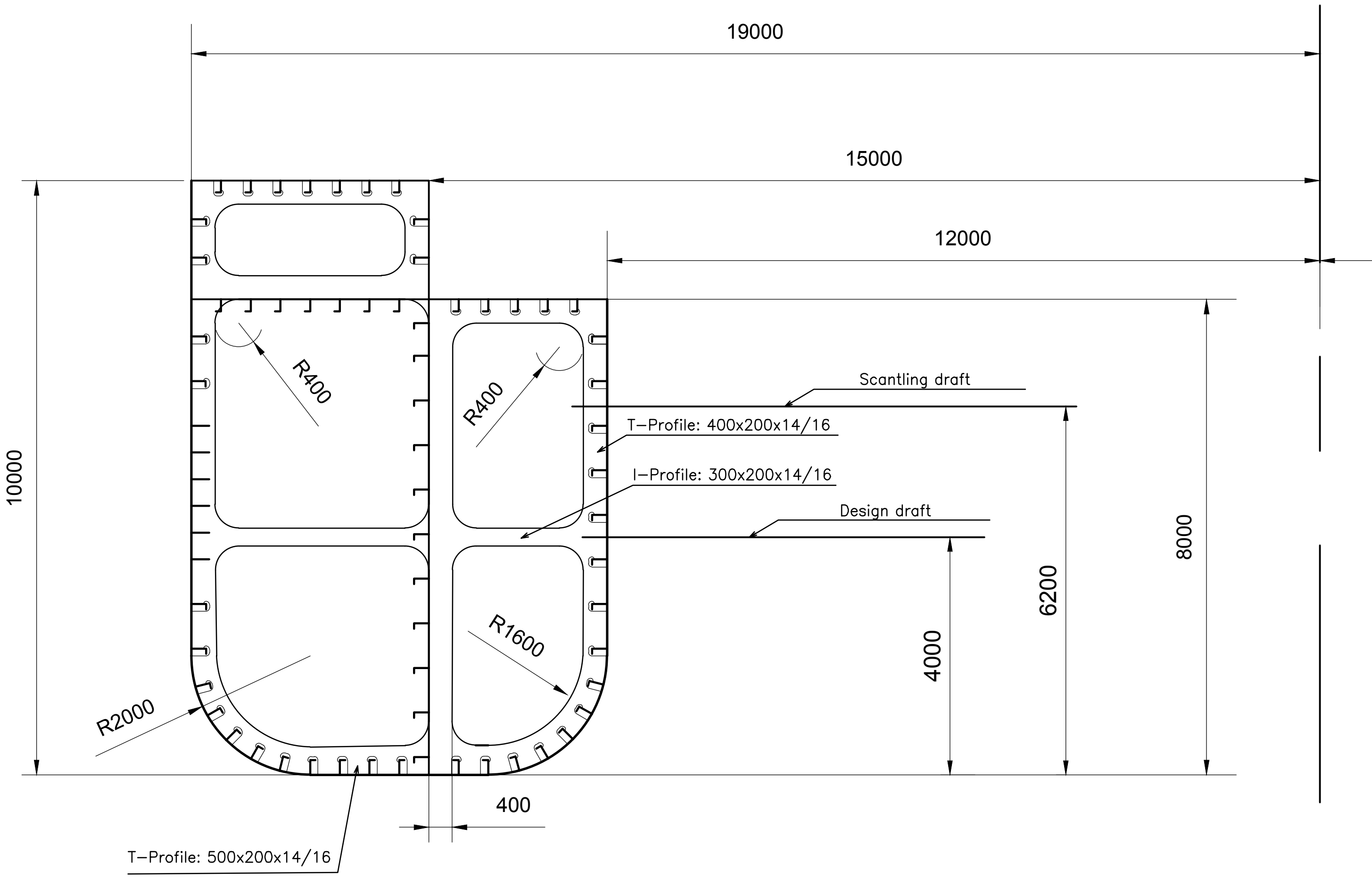
ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD
Revisions					

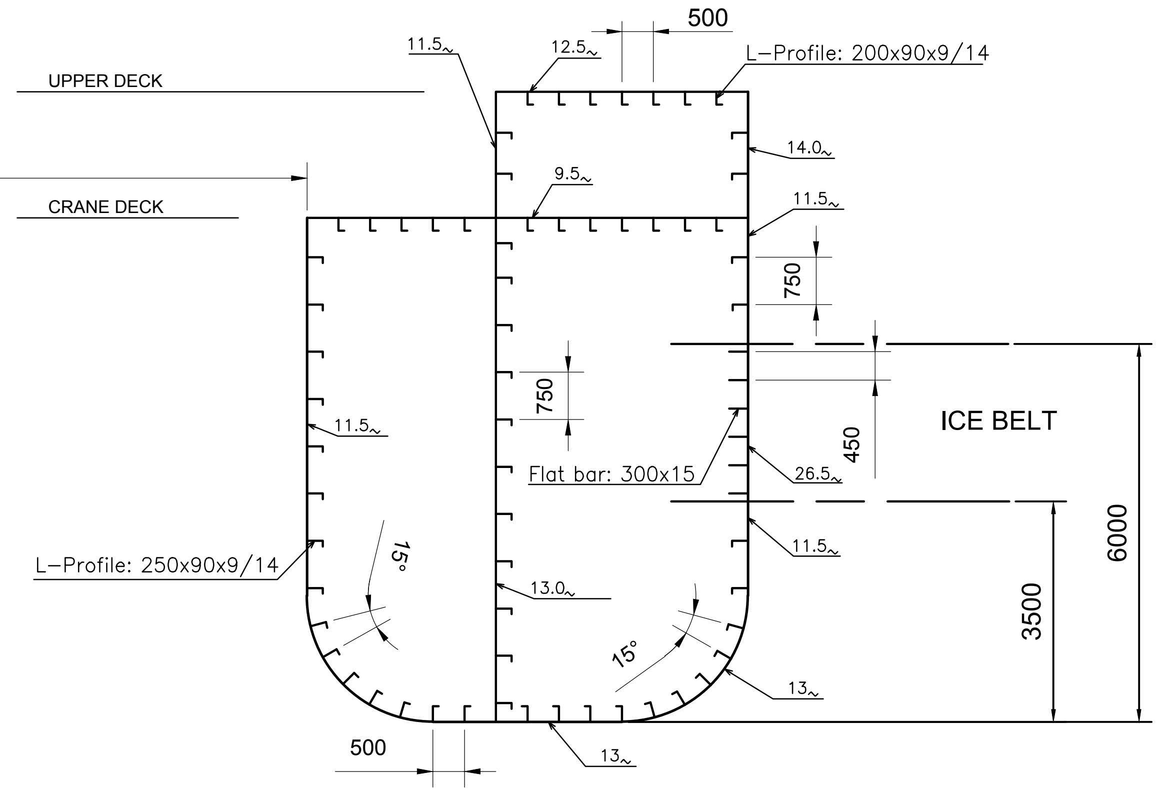
CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number MDP2014 OWNER Blekinge Offshore Class DNV	
Wind Installation Vessel Kermit Frame section monohull	PREP. BY Structure	DATE 05/12/2014
	CHECK. BY YY	DATE GG/MM/AAAA
	SCALE 1:75	Size A3 / SH.S 3 of 3
	DRAW. N° MDP2014_2-200-05	

Catamaran cross section
at web frame

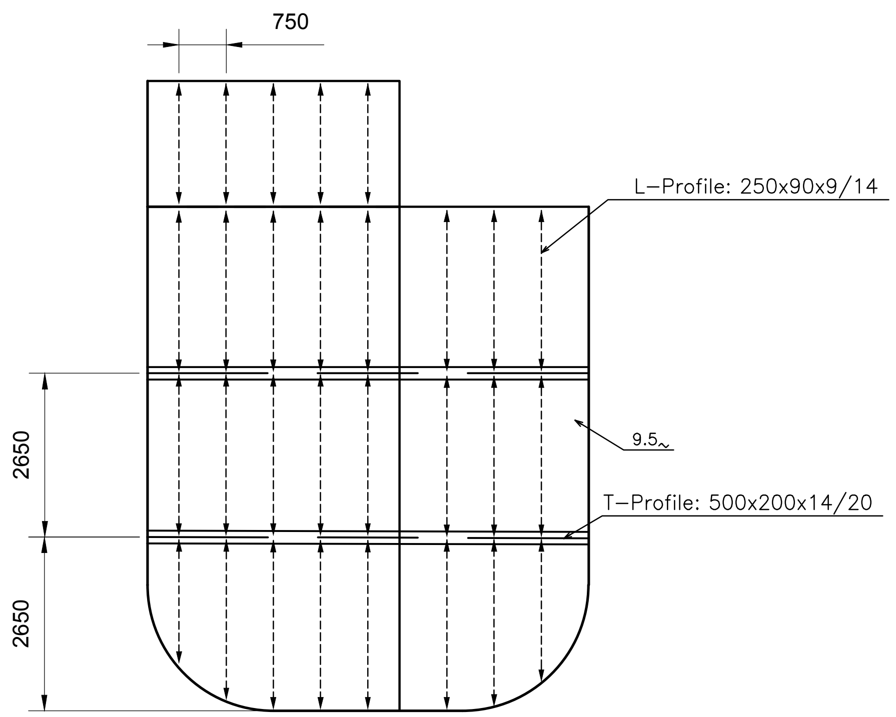
Catamaran cross section
at ordinary frame

CL





W.T. BHD.



NOTE

1. STEEL QUALITIES

UNSPECIFIED TO BE GRADE "NV–NS" STEEL.

Main dimensions Kermit

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m
Scantling draught	6.20 m

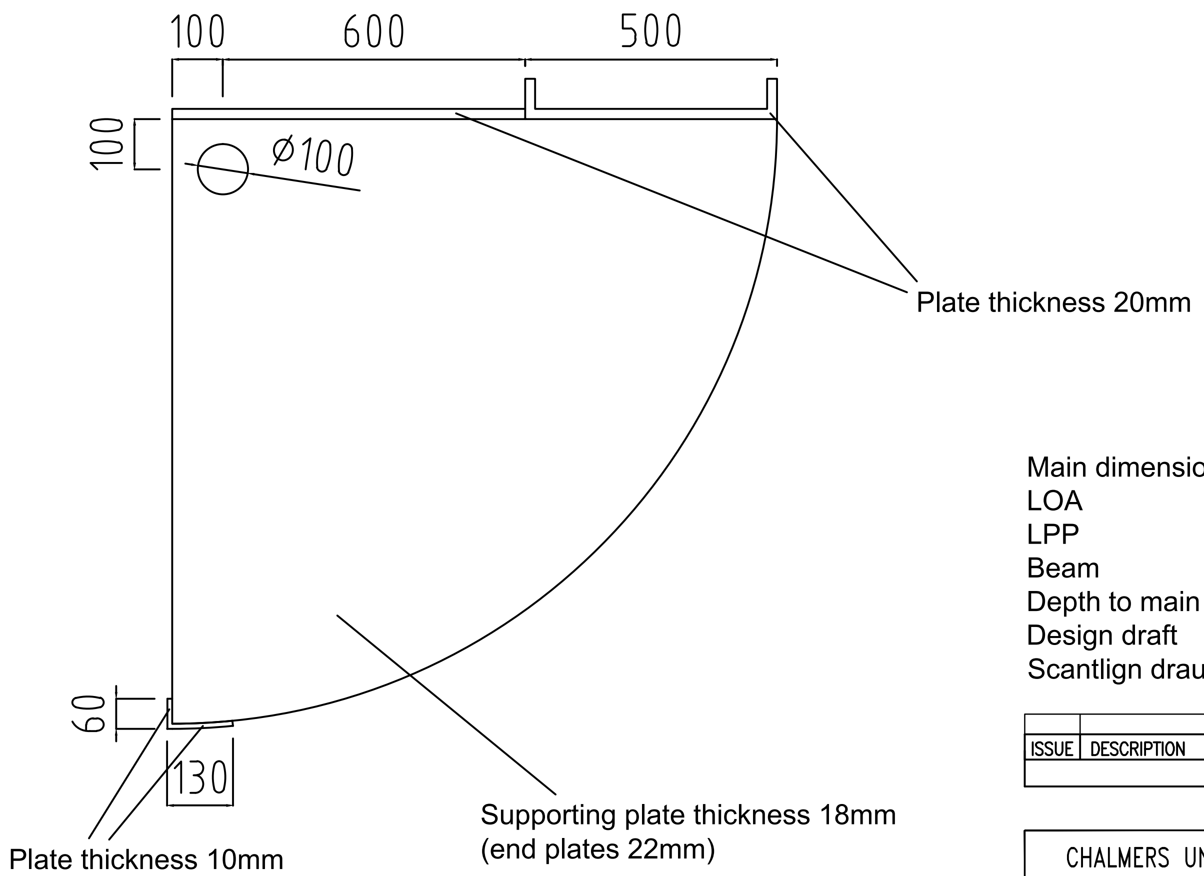
ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number: MDP2014 OWNER: Blekinge Offshore Class: DNV
Wind Installation Vessel Kermit Frame section catamaran	PREP. BY: OL — DATE: 10/12/2014 CHECK. BY: YY — DATE: GG/MM/AAAA SCALE: 1:120 — Size: A3 — SH.S: 1 of 1 DRAW. N°: MDP2014_2-200-06

Cross–section of the retractable rails

The spacing between the quarter–circle plates is 275mm.



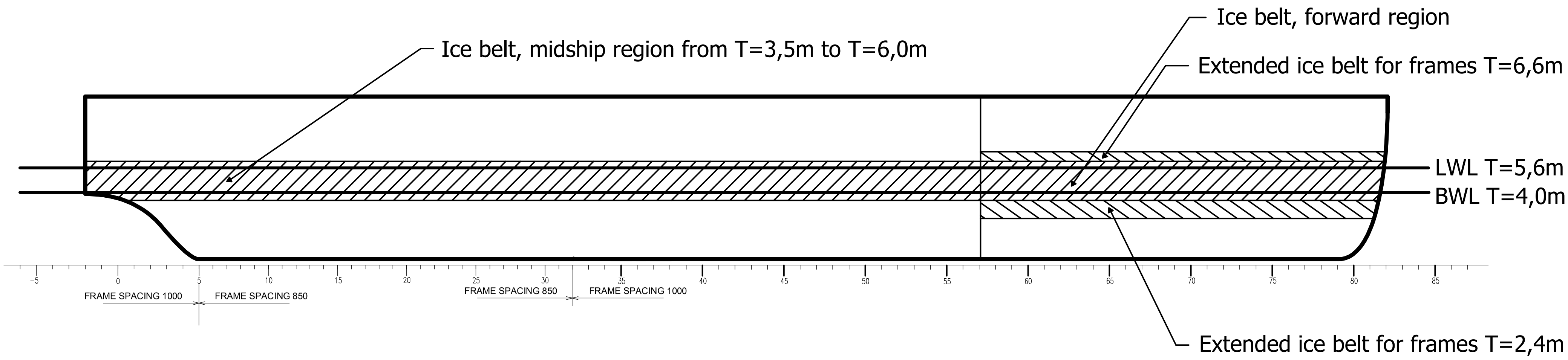
Main dimensions Kermit installation vessel

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m
Scantlign draught	6.20 m

ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number: MDP2014 OWNER: Blekinge Offshore Class: DNV	
Wind Installation Vessel Kermit Cross–section retractable rails	PREP. BY: Structure	DATE: 05/12/2014
	CHECK. BY: YY	DATE: GG/MM/AAAA
	SCALE: 1:15 · Size: A4	SH.S: 1 of 1
	DRAW. N°: MDP2014_2-200-15	



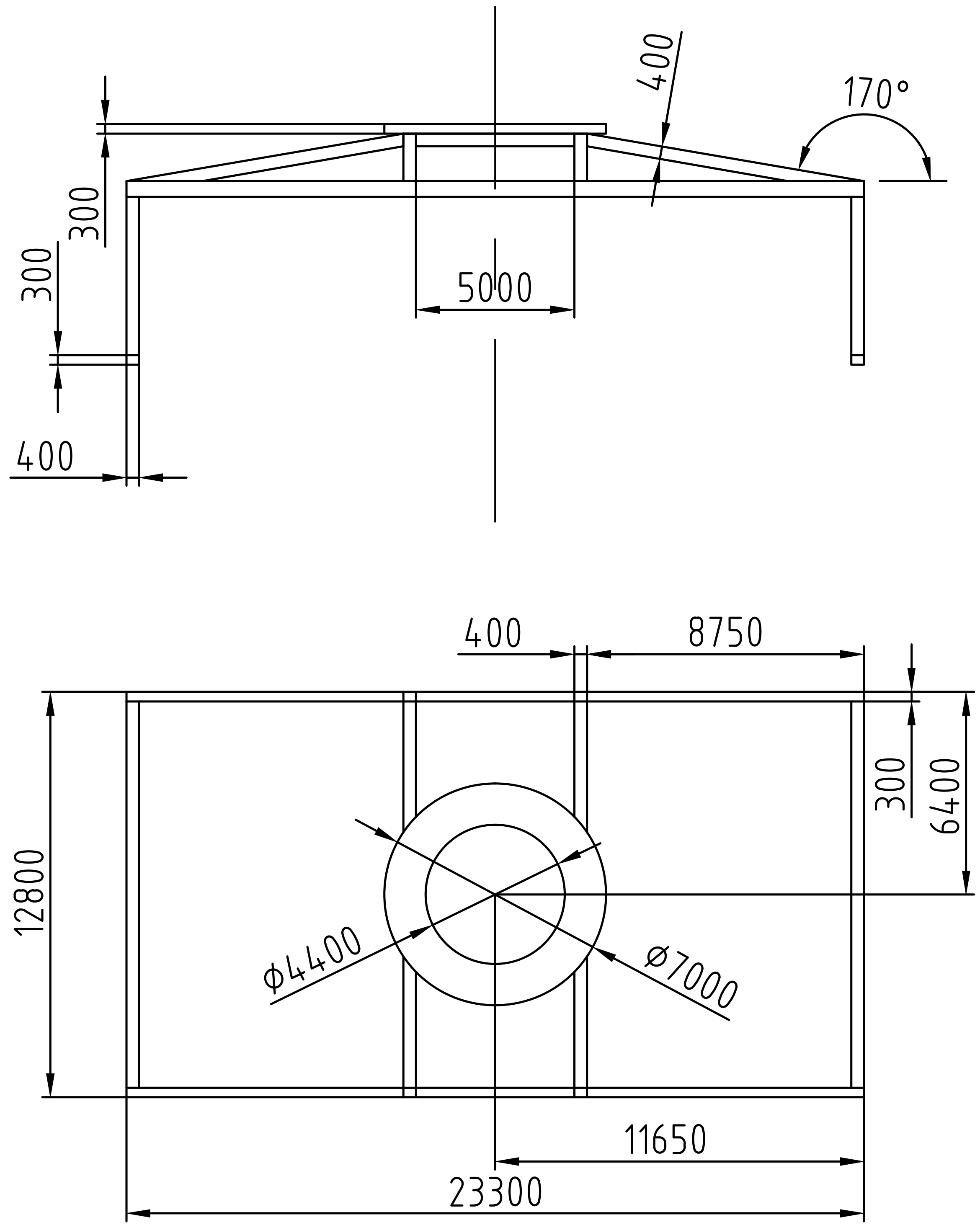
Main dimensions Kermit

LOA	80.00 m
LPP	77.50 m
Beam	38.00 m
Depth to main deck	7.00 m
Design draft	4.00 m
Scantlign draught	6.20 m

ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

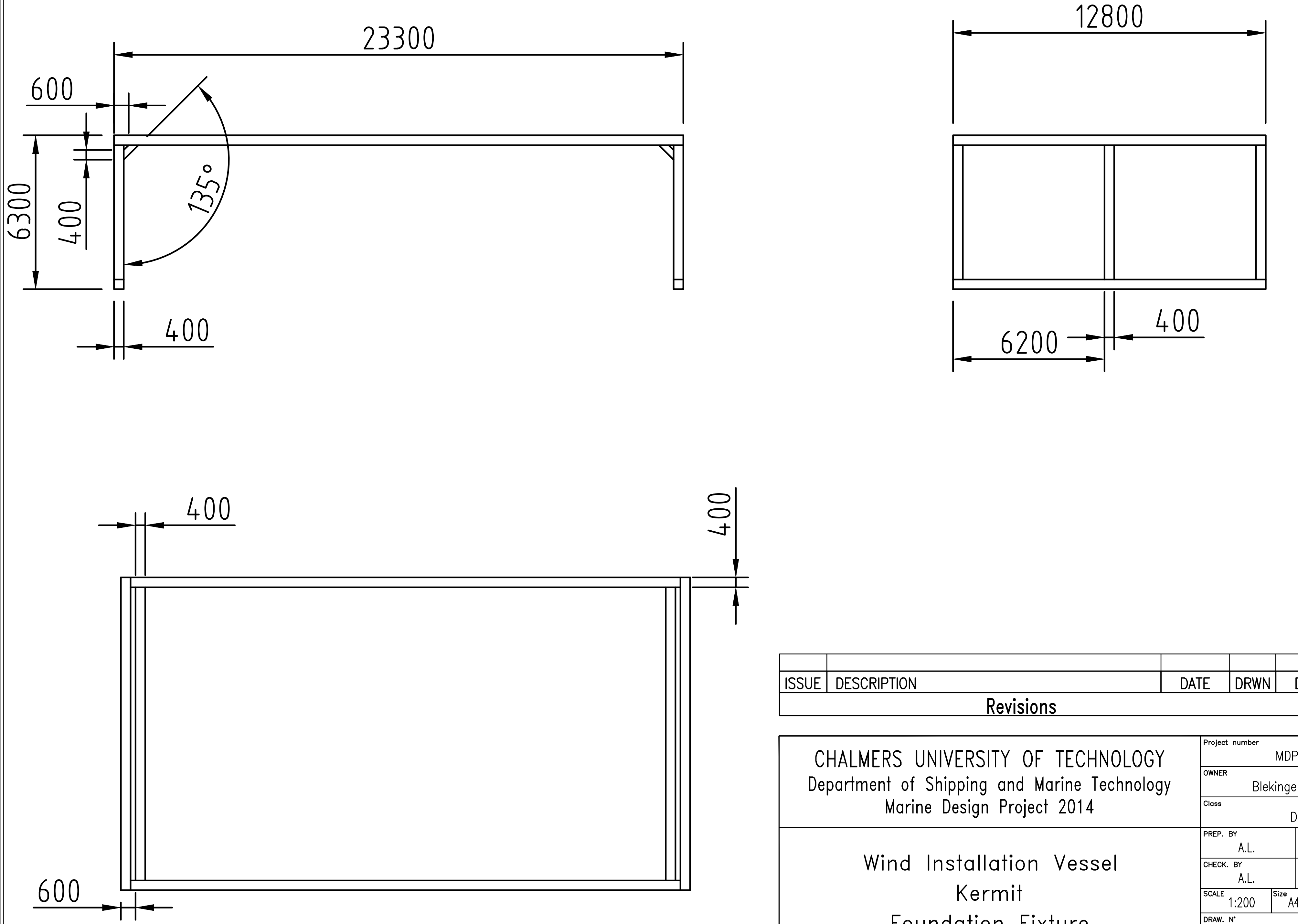
CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number: MDP2014 OWNER: Blekinge Offshore Class: DNV	
Wind Installation Vessel Kermit Side view for of belt	PREP. BY: OL	DATE: 10/12/2014
	CHECK. BY: YY	DATE: GG/MM/AAAA
	SCALE: 1:350 — Size: A4	SH.S: 1 of 1
	DRAW. N°: MDP2014_2-200-19	



ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY Department of Shipping and Marine Technology Marine Design Project 2014	Project number: MDP2014 OWNER: Blekinge Offshore Class: DNV
Wind Installation Vessel Kermit Windmill Fixture	PREP. BY: A.L. — DATE: 05/12/2014 CHECK. BY: A.L. — DATE: 05/12/2014 SCALE: 1:200 — Size: A4 — SH.S: 1 of 1 DRAW. N°: MDP2014_2-200-20



ISSUE	DESCRIPTION	DATE	DRWN	DATE	CHKD

Revisions

CHALMERS UNIVERSITY OF TECHNOLOGY
Department of Shipping and Marine Technology
Marine Design Project 2014

Wind Installation Vessel
Kermit
Foundation Fixture

Project number: MDP2014	
OWNER: Blekinge Offshore	
Class: DNV	
PREP. BY: A.L.	DATE: 05/12/2014
CHECK. BY: A.L.	DATE: 05/12/2014
SCALE: 1:200	Size: A4 / SH.S: 1 of 1
DRAW. N°: MDP2014_2-200-21	