Optimus Pråm - Semi-submersible wind farm installation vessel for Blekinge Offshore

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Marine Design Project 2014

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Abstract

In Sweden the government is investing lots of resources in order to meet the energy need with clean and renewable alternatives. Since wind is an unlimited source of energy the exploitations of wind farms is of great interest. This report describes a conceptual design of an innovative offshore wind turbine installation vessel for inland sea conditions, with highest possible energy efficiency and environmental friendly performance in every detail. The customer, Blekinge Offshore, main requirement is to receive a concept with an as low installation cost as possible.

The final concept Optimus Pråm includes one installation vessel, which is a semi-submersible barge, and one support vessel that supply the installation vessel with power and propulsion. The power is distributed from the support vessel to the installation vessel through a power cable. The installation vessel will be self-propelled during the installation phase using electrical motors and thrusters.

The installation vessel shall be able to handle gravity foundations and fully assembled wind turbines, transported vertically to the installation site. Foundations and windmills will be fully assembled in Karlshamn port before transit to site. During transit the installation and support vessel are connected to each other in all motions except pitch. This almost total fixed connection makes the two vessels acts like one, which gives the joint vessel excellent manoeuvrability. When installing foundations, the installation vessel and the support vessel disconnect. The installation vessel places the foundation at the planned location by ballasting until the foundation has reached the seabed. After de-ballasting the installation vessel and support vessel connects again and transit back to the harbour. When installing wind turbines the two vessels are kept connected and makes highly accurate positioning for the installation possible.

KEYWORDS
Semi-submersible, Offshore installation, Wind turbine, Gravity foundation, Ballast, Offshore Wind farm, Baltic Sea, Lifting appliances, Push barge

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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 an innovative offshore wind turbine installation vessel for inland sea conditions, displaying highest possible energy efficiency and environmental friendly performance in every detail, while displaying excellent operation.

The project members would like to thank our supervisors Senior Lecturer Per Hogström, Professor of the Practice Bengt Ramne, Dept. of Shipping and Marine Technology, Prof. em. Anders Ulfvarson, Chalmers. In addition the following persons have supported the project and their assistance has been highly appreciated:

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Blekinge Offshore	Anders Nilsson
Blekinge Offshore	Hans-Olof Svensson
Marine Machinery Systems	Mats Isaksson

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Executive summary

The following section briefly outlines the design work accomplished in this project.

Mission Profile

The mission is to transport gravity foundations and assembled wind turbines from the fabrication and assembly site at Stilleryd, Karlshamn (56°9'30.4"N 14°49'55.9"E) to the site situated 3 nm southeast of Hanö (55°58'38.1"N 14°55'50.7"E), a distance of approximately 20 nm. The turbines are transported assembled with pillar, nacelle and blades as one unit. The installation vessel shall be able to pick up foundations and windmills, one at the time, transit to site and install the foundation or windmill.

Main particulars for installation vessel

Type	Semi-submersible barge
Flag	Swedish
Class	DNV
Length over all, LOA	52.70 metre
Length between perpendiculars, LPP	45.06 m
Design draft, T (ballast condition)	2.06 metre
Depth to main deck, D	7.00 m
Breadth, B	32.00 m
Frame spacing	0.5 m
Design speed, largest foundation	3.5 knots
Design speed, wind turbine	5.5 knots
Max speed	7.0 knots
Displacement, full load condition	11 298 tons
Light ship	1 645 tons
Dead weight	9 653 tons
Station keeping	Dynamic positioning
Propulsor	2 x Rolls Royce US55P4 at 300 kW
	1 x Rolls Royce UL601 at 400 kW
Frequency converter	2 x ABB ACS 1013-A1-A
	1 x ABB ACS 1013-A1-C
Electrical motor	ABB HXR355LC4 and ABB HXR400LC4

Main particulars for support vessel

Fuel	Methanol, MGO
Fuel consumption	10 tons MGO per installed unit
	21 tons Methanol per installed unit
Main genset	4 x Wärtsilä 1000W6L20 at 1053 kW
Emergency genset	1 x Volvo Penta D5ATA/UCM274E at 85 kW
Ballast tanks	Ballast tanks with total capacity of 9 420 tons
Ballast pumps	3 x Hamworthy CA450
Frequency converter	2 x ABB ACS 1013-A3-Q

General overview

The aim of this project is to come up with a new solution to transport and install wind turbines and gravity foundations for offshore wind farms. The wind turbines are pre-assembled at the fabrication site and must be transported in a vertical position. Conventional installation ships today are heavy lift vessels with cranes, often chartered from the oil and gas industry in the North Sea. The goal for this project is to find a competitive, cheaper and more suitable solution for Baltic Sea conditions. The intention is to minimize the lifting and rigging operations offshore in order to shorten the installation time. The concept presented in the current report uses ballasting and de-ballasting for the heavy lifting operations, such as installing the gravity foundations and wind turbines.

The depths are varying from 10 to 35 metre and the vessel should be able to install gravity foundation and wind turbines in Sea state 3, i.e. a significant wave height up to 0.8 metres. The entire operation also needs to withstand 12 metre per second wind speed during installation. The solution developed in this project is a semi-submersible barge that uses ballast and dynamic positioning system instead of cranes to position the wind turbines and foundations. The shape of the barge is simple and consistent, which gives a low new build cost. The ship has four large vertical tanks in order to keep reserve buoyancy and avoid being fully submerged for the whole water depth range.

The design results in a vessel with good stability, intact as well as damaged. The sea-keeping abilities are good and the vessel can operate in the necessary sea states. Figure 0-1 shows the installation vessel.

Figure 0-1: View of the installation vessel – Optimus Pråm

Due to customer demands the operational time for installing one gravity foundation or wind turbine should be limited to one working day. This has been considered when designing the vessel and the timeline presented in Figure 0-2 meets all requirements submitted from the customer.

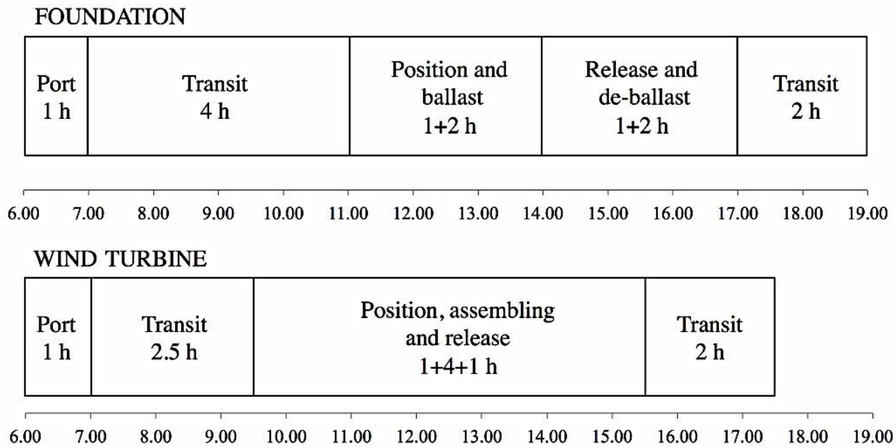

Figure 0-2: Timeline for installing one unit

The ballast system is one of the most important systems in this concept since it controls the vertical position of the cargo being lifted on-board, the stability of the ship and the wave response. Therefore almost all spaces on-board are allocated for ballasting and the dynamic positioning system. Due to the short time frame for installing; the vessel requires large pumping capacity and power support and therefore the concept also involves a support vessel. It is designed as a push-tug housing the main engines, additional machineries and storage tanks for consumables, such as fuel, lubrication oil and fresh water. The support vessel helps position the installation vessel and pushes it during transport to the wind farm site. A diesel-electric system will be used which involves a diesel-electrical generator set that supplies the installation vessel with power through an electric cable. The support vessel does also contain decks for accommodation and other necessary spaces. The connection part between the supply vessel and installation vessel is designed to function in different loading modes and has a degree of freedom for pitch motion. For all other motions the two vessels will act as a single unit.

Intact stability of the installation vessel is critical since the water plane area is low when being submerged. To increase the stability during submersion, vertical tanks are fitted to the installation vessel. For stability purposes, it is advantageous to lift cargo near the gravity centre of the vessel and try to avoid large movements of weights. Therefore, the vessel has been designed to have a large moon pool in the middle of the vessel. In front of the moon pool a pump room is located which also contains the frequency converters for the thrusters.

Main deck is covered with lifting appliances and equipment for installation and securing the cargo during transportation. Beneath is a tank top deck with mainly ballast tanks and a few machinery compartments. The aft compartments of the ship are accommodating two azimuth steerable thrusters, frequency converters and batteries. In case of an emergency, the installation vessel must be able to de-ballast and resurface. At centreline in the forward section there is one retractable azimuth thruster for improved manoeuvrability and redundancy. With this configuration, the vessel has thrust in all directions in the horizontal plane. The ship has a double bottom with a height of 1.8 metre used for additional ballast space. Shell plating is made of high strength steel with a thickness varying between 10 and 16 millimetres.

The connection part between the supply vessel and installation vessel is designed to function in different loading modes and has a degree of freedom for pitch motion. For all other motions the two vessels will act as a single unit.

List of Abbreviations

CAD – Computer Aided Design

CAE – Computer Aided Engineering

CFD – Computational Fluid Dynamics

DGPS – Differential global positioning system

DNV – Det Norske Veritas

DSC – Digital Selective Calling

ECDIS – Electronic Chart Display and Information System

EGR – Exhaust Gas Recirculation

EPIRB – Emergency Position-Indication Radio Beacon

FE – Finite Element

FEA – Finite Element Analysis

FEM – Finite Element Method

FR – Frame

GM – Metacentric height

GMDSS – Global Maritime Distress Safety System

GPS – Global Positioning System

GZ – Lever arm for restoring force

HFO – Heavy Fuel Oil

HFR – Vertical Frame

HPR – Hydroacoustic Position Reference

HS – Significant wave height

IMO – International Maritime Organisation

ITTC – International Towing Tank Committee

Iy – Moment of inertia around neutral axis

Iz – Moment of inertia around neutral axis

KB – Centre of Buoyancy above baseline

KML – Longitudinal metacentre above baseline

KMT – Transverse metacentre above baseline

KN – Cross Curves

LHV – Lower Heating Value

MGO – Marine Gas Oil

MRU – Motion reference units

MTc – Moment to trim one centimetre

MW – Megawatt

Mx – Moment around X-axis

NA – Distance to neutral axis (measured from keel)

NAVTEX – Navigational Text Messages

PA/GA – Public Address and General Alarm system

RADAR – Radio Detection and Ranging

RANS – Reynolds Average Navier-Stoke

RMS – Root Mean Square

SOLAS – Safety of Life at Sea

TPc – Tons per centimetre immersion

UHF – Ultra High Frequency

VHF – Very High Frequency

Vy – Velocity in Y-direction

Zy – Section modulus

Table of Contents

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1 Introduction

The earth's energy sources are scarce, why the interest in renewable energy production has steadily grown during the past years. In Sweden the government is investing lots of resources in order to meet the energy need with clean and renewable alternatives. Since wind is an unlimited source of energy the exploitations of wind farms is of great interest. The conditions outside the Swedish south cost are beneficial for construction of a wind farm. The southern parts of Sweden also hold a large part of the population, which makes the energy need high in this region. This need is currently saturated with energy from nuclear power plants. By the construction of a wind farm in the bay of Hanö a part of this energy will instead come from a renewable and safer source, which is in line with the Swedish environmental goals (Naturvårdsverket, 2014). The construction and maintenance of the wind farm will also create many job opportunities in the area.

1.1 Background

The European Union has set a goal on having 20 per cent of its energy from renewable sources by the year 2020. The national goal for Sweden is to have 49 per cent energy from renewable sources by the same year (Naturvårdsverket, 2014). The northern rivers cannot be further exploited which requires an expansion of primarily wind power, both land based and offshore (Edvardsson, 2014).

Existing offshore wind installation vessels are built for North Sea conditions. By using one of the existing concepts for offshore installation in the bay of Hanö, which has relatively sheltered waters, the operation will become very inefficient. Due to the high day rates for a North Sea vessel it is economically motivated to design a new concept for offshore installation in inland seas, such as the Baltic Sea.

Blekinge Offshore wants to install at least 80 complete units per year during a period of 8 years. The goal is to obtain an as low installation cost as possible per wind turbine. When completed, the wind park stands for approximately 5 per cent of Sweden's annual electricity consumption, approximately equivalent to one nuclear reactor.

1.2 Objective

The objective of the project is to develop a conceptual vessel system for installation of gravity foundations and wind turbines for inland sea conditions. The design shall include relevant technical documents and drawings.

1.3 Methodology

The work is divided between four disciplines with different areas of expertise:

- General arrangement - responsible for logistics of operation, lifting appliance, economics, life-saving appliance and general arrangement.
- Structure - responsible for the structural design of the vessel and preliminary structural calculations of lifting appliances and analysis of local critical points.
- Hydrodynamics - responsible for hull shape design, stability assessments, subdivision and power predictions.
- Machinery systems - responsible for power balance, setup of the propulsion system including engine room design and choice of engine.

All disciplines collaborate to produce an initial concept. The work then follows the process defined in the design spiral, see Figure 1-1. The project time allows two loops in the design spiral.

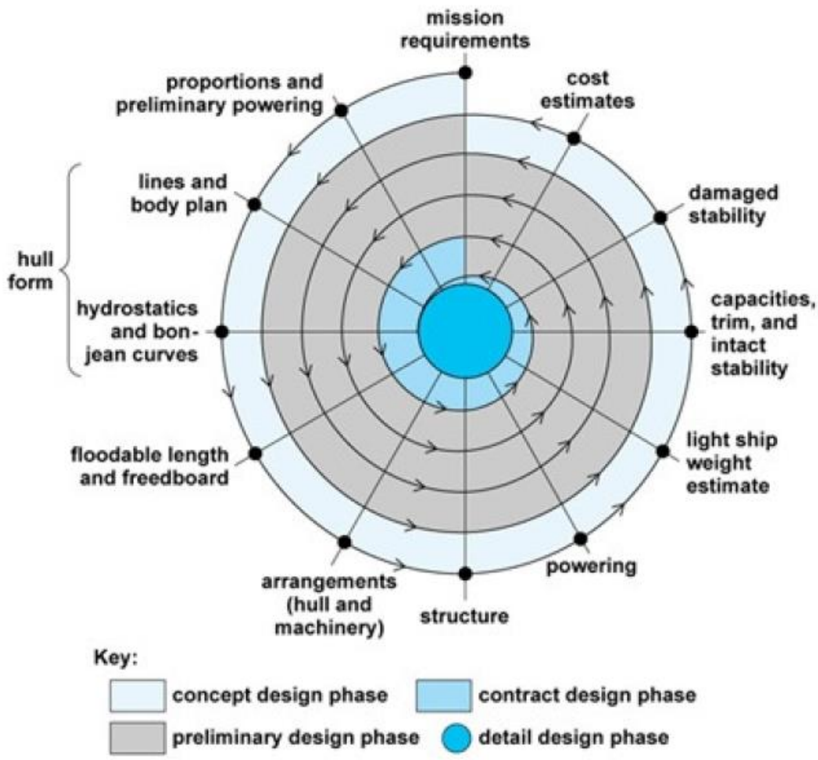

Figure 1-1: Design Spiral

1.4 Limitations and assumptions

The primary limitation for the project work is related to the limited engineering experience of the project team and the time available to solve the task. In addition the lack of existing similar designs results in the need for assumptions in order to establish an initial concept as a starting point of the evolutionary engineering work. The customer wishes for a concept with one vessel that can handle both gravity foundations and wind turbines. It is initially stated that the concept should use ballasting and de-ballasting for all lifting operations. The Optimus Pråm concept includes one installation vessel and one support vessel. In this report the focus is on the installation vessel since this is an unexploited concept for wind farms installation. The support vessel is a standard tugboat, with some modifications regarding engine power and connection parts.

The two loops of the design spiral are limited to a time frame of 16 weeks. This limits the amount of optimization and leaves room for future development. Another limitation is the lack of weather data which has led to assumptions.

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2 Design Basis

This chapter describes stakeholders and their requirements on the ship design. Major requirements specified by the client are listed.

2.1 Stake holders

Different stakeholders have different demands with which the vessels need to comply. The following stakeholders and demands have been identified:

Ship owner	Low running costs, high reliability, environmental friendly profile
Costumer	High reliability, number of working days
Crew	Safe and ergonomic working environment
Special Crew	Safe and ergonomic working environment and safe transport of special equipment
Maintenance	Proved solutions and recognized manufacture
IMO	Rules and regulations according to MARPOL and SOLAS
Swedish Transport Agency	Flag state rules
DNV	Class rules and regulations
Insurance companies	Approved vessel by class society
Second hand interest	Ability to install windmills in the Baltic Sea
Competitors	Optimized vessels for wind mill market

2.2 Specification of requirements

Major requirements specified by the client:

- The vessels necessary for the operation should be optimized for operation in Baltic Sea conditions
- At least 80 complete units, gravity foundation and wind turbine, installed per year during 8 years
- Installation of Foundations should be possible to carry out all around the year in conditions up to Sea State 3
- Installation of turbines to foundations should be possible to perform in wind speeds up to 12 m/s
- The site will be built up in steps of 500MW. The first round will start at shallow water 10-20m in depth and expand in the last round to increased depth at approximately 35m.
- Draft in the harbour is maximum 10 m
- The wind turbine installation operations have to take the regular commercial traffic in the fairway to Stilleryd in consideration.
- The acceptable tolerance in installation of foundations is 2 metre offset
- The payback time of investments for installation vessels should be limited to 350 installed turbines
- Health, safety and environment issues are of the highest priority, followed by Cost

3 General arrangement

The installation vessel as well as the support vessel will carry out various operations and have different equipment on-board for the daily activities. Because of the complexity of the semi-submersible operation; the ballasting system, high sea pressure and cargo loads, most of the focus of this project lies within the general layout of the tank compartments, machinery room and lifting appliances. This section describes how both vessels are designed and why, but a note to the reader is that most emphasis lies on the installation vessel. The support vessel has an extremely important task, since it accommodates most of the machineries and the crew, but a conventional push-tug could be modified into this support vessel.

3.1 Main dimensions

Most dimensional restrictions for this project are due to the narrow port of Stilleryd. At some places the port is only 75 metre wide with a sea depth of 10 metre and being able to manoeuvre a ship carrying a foundation of up to 2,500 tons is challenging. Since the top of the foundations is almost 10 metre wide, the moon pool must then have a minimum breadth to comply with this criterion. Table 3-1 arrays the main dimensions for both vessels.

Table 3-3-1: Main dimensions of the installation and support vessel

	Parameter	Value (metres)
Installation vessel		
	Length between perpendiculars, LPP	45.06
	Length overall, LOA	52.70
	Moulded breadth, B_M	32.00
	Moulded depth, D_M	7.00
Support vessel		
	Length between perpendiculars, LPP	27.38
	Length overall, LOA	35.00
	Moulded breadth, B_M	10.00
	Moulded depth, D_M	5.50

3.2 Deck specification

Much focus is given to designing the layout of the ballast tanks in order for the installation to submerge in a controlled manner and in different loading modes. Since the heaviest foundation weighs about 2600 tons the ballasting capacity and the overall system must be very flexible for weight changes, particularly when the foundation is released. The chapters below describe the different deck layouts for the installation vessel further. Figure 3-1 shows a profile view of the installation vessel with different deck plans. A complete set of drawings can be found in Drawings.

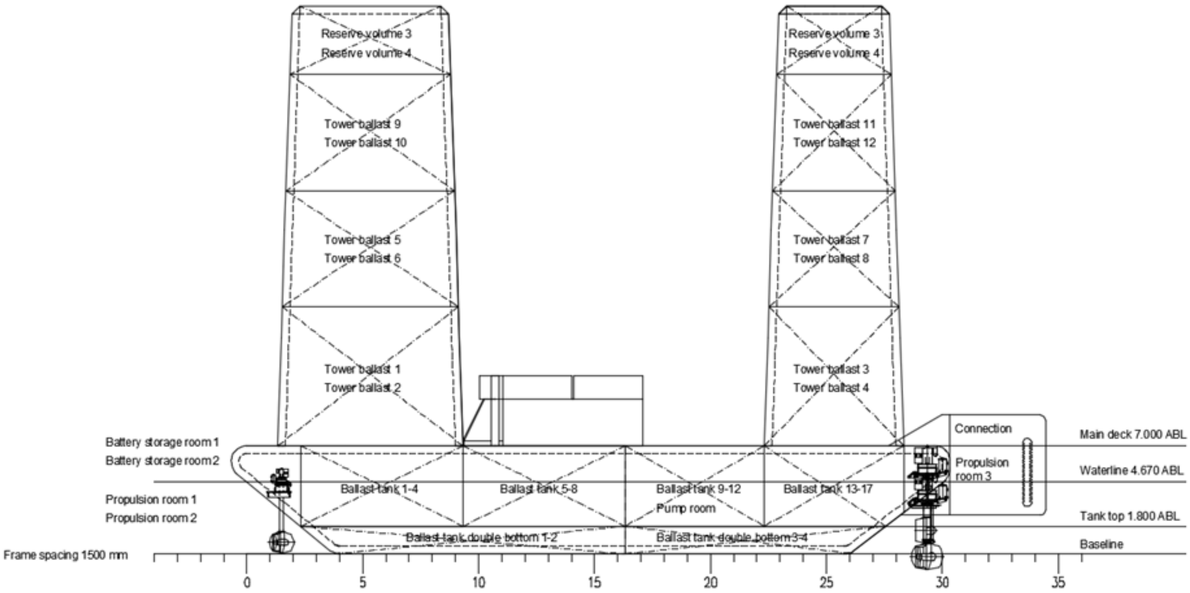

Figure 3-1: Profile view of the tank arrangement of the installation vessel

3.2.1 Double bottom

The installation vessel has a double bottom with a height of 1.8 metre used as additional space for water ballast tanks. The double bottom has a transverse girder at the end of the moon pool and a longitudinal bulkhead splitting the deck area into four sections. Having four large tanks as Figure 3-2 displays causes free surface effects which has a negative impact on intact stability. According to DNV rules permeability for tanks and storage rooms are determined to 0.97 and free surface effect is taken into account at its maximum level. For further elaboration of intact stability, see section 5.3.1.

Outside shell plating is 16 millimetres of high strength structural steel and inside structures has a plate thickness of 10 millimetres. More dimensions of the bottom structure and for various stiffeners can be found in section 7.1.3.

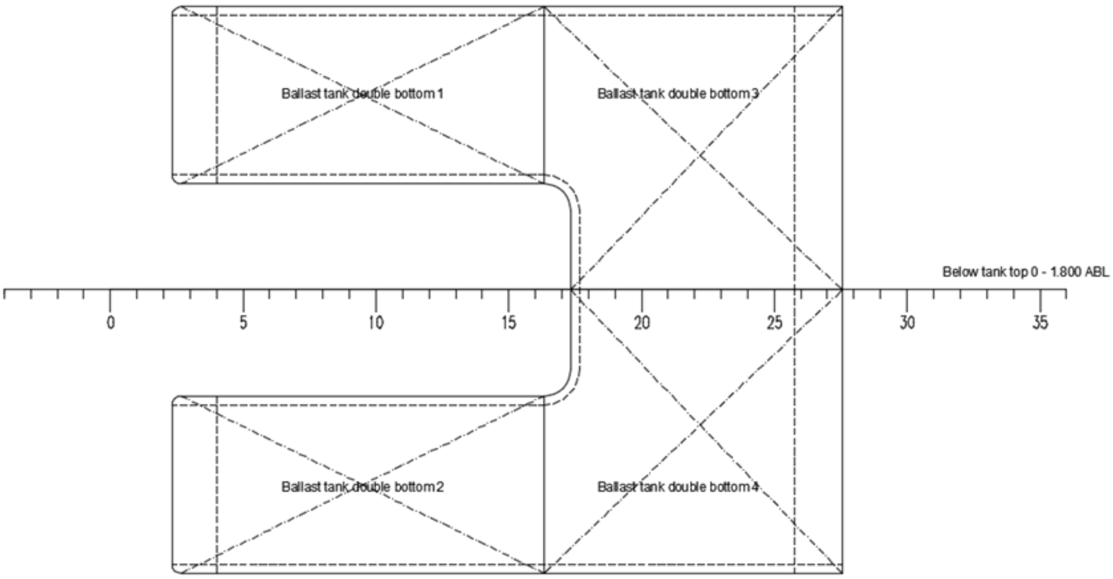

Figure 3-2: Double bottom 0 millimetre above baseline

3.2.2 Tank top

The tank top, see Figure 3-3, consists of 23 departments of which 17 are allocated for ballast tanks. Distance to the main deck is 5.2 metres making a ballast capacity of about 4825 tons of water. The reason for this large number of tanks is mainly damage stability causing unsymmetrical flooding and free surface effects. Both pontoons have a longitudinal bulkhead and the forward part of the barge is transversely divided into two sections creating space for a pump room. In addition to water pumps of centrifugal type, the pump room also contains a frequency converter for the retractable thrusters. There are seven pumps in total whereof three are for ballasting, two for bilge water and two for fire protection. The aft compartments of the ship are accommodating two azimuth steerable thrusters, two frequency converters and batteries. In case of an emergency, the installation vessel must be able to de-ballast and resurface, which motivates the need of batteries, see chapter 6.3. At the centreline in the forward section there is one retractable azimuth thruster for improved manoeuvrability and redundancy. With this configuration, the vessel has thrust in all directions in the horizontal plane.

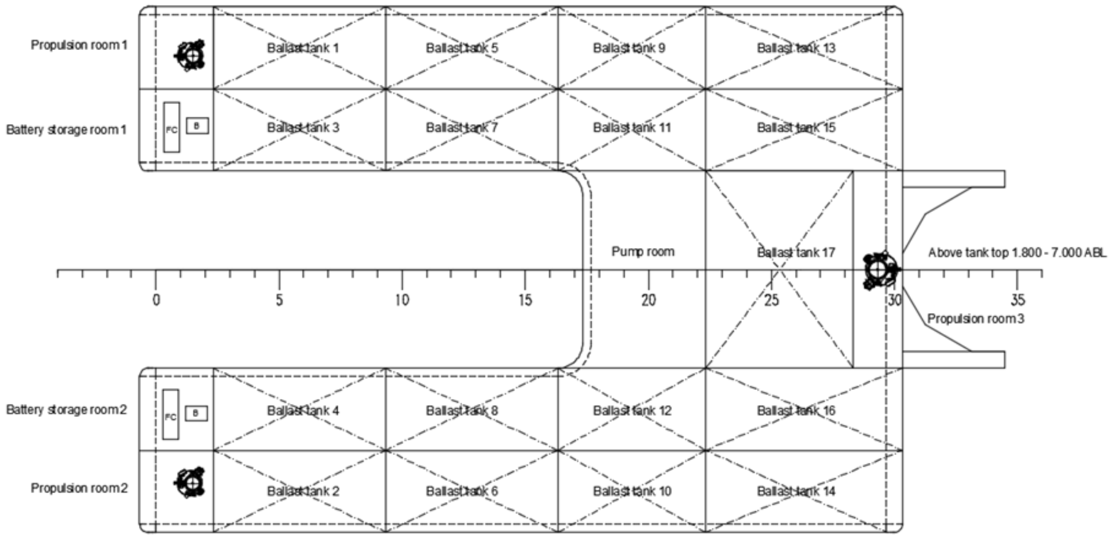

Figure 3-3: Tank top 1800 millimetres above baseline

3.2.3 Main deck and vertical tanks

The main deck, see Figure 3-4, consists of lifting appliances and equipment for cargo handling, described in Chapter 3.4 Ship equipment. Furthermore, there are four vertical tanks with ballast. Their main function is to provide reserve buoyancy and ballast capacity when the barge is submerged. According to DNV the ratio of reserve buoyancy shall not be less than 4.5 per cent for the vessel and 1.5 per cent for the forward and aft end separately (Den Norske Veritas, 2013). The vertical tanks are therefore designed to comply with these rules. Each vertical tank has three watertight horizontal bulkheads creating 16 additional ballast tanks where the top tank of each tower is a reserve volume, see Figure 3-1. They are positioned 0.85 metre inward from the sides of the ship because of damage stability, further explained in Chapter 5.6. In this way damage from the side will only be limited on the barge part of the vessel.

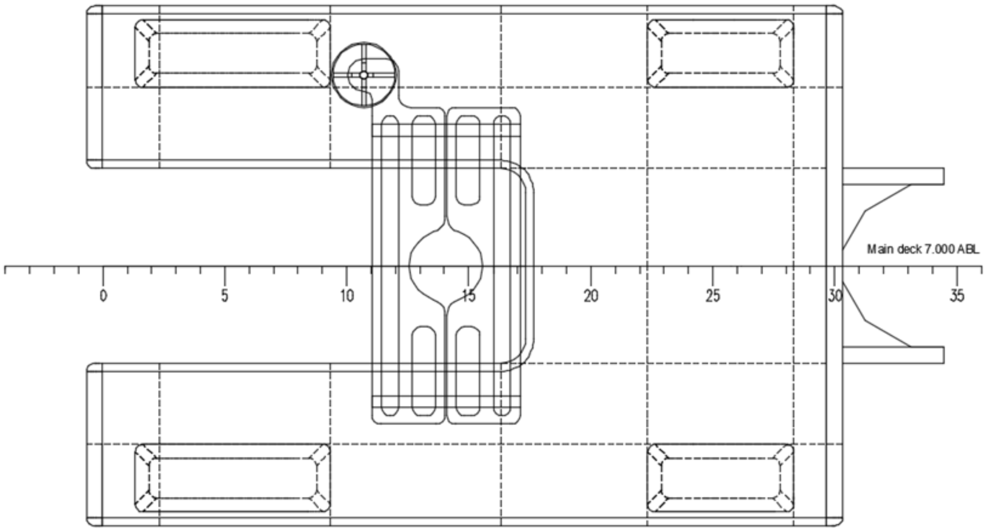

Figure 3-4: Main deck 7000 millimetres above baseline

3.3 Support vessel

The support vessel is designed as a push-tug housing the main engines, additional machineries and storage tanks for consumables, such as such as fuel oil, lubrication oil and fresh water. The support vessel helps position the installation vessel and pushes it during transit to the wind farm site. A diesel-electric system will be used which involves an electrical motor that supplies the installation vessel with power through an electric cable. The support vessel does also contain decks for accommodation and other necessary spaces. This chapter displays the general layout as a suggestion for the support vessel and most focus lies on the tank top, tween deck and main deck. The superstructure is used for navigation and accommodation, but is not presented in detail in this report. Figure 3-5 shows a profile view of the support vessel.

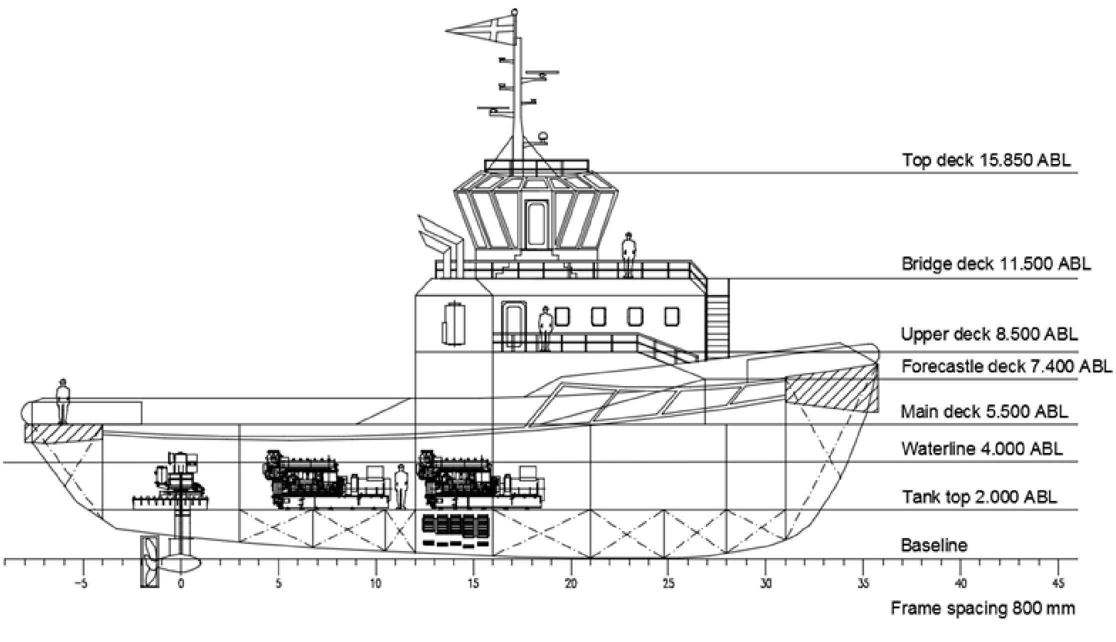

Figure 3-5: Profile view of the supply vessel

3.3.1 Tank top

The tank top deck is roughly divided into 21 compartments that could be used for tanks. Table 3-2 shows a suggestion for how the tanks could be arranged and Figure 3-6 how they are positioned. Other spaces are a small collision zone at the bow, sea chest with pumps at port and starboard side, and an aft compartment for the thrusters.

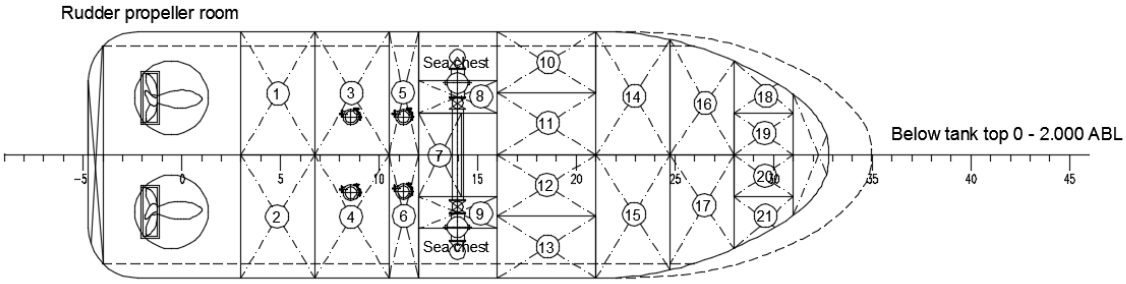

Figure 3-6: Tank top 0 millimetre above baseline

Table 3-2: Tank arrangement

No.	Tank	Capacity [m^3]
1.	Water ballast	26.8
2.	Water ballast	26.8
3.	Bunker	26.8
4.	Bunker	26.8
5.	Bilge	10.7
6.	Bilge	10.7
7.	Lub. Oil	10.0
8.	Hydr. Oil	7.0
9.	Dirty Oil	7.0
10.	Bunker	16.7
11.	Bunker	19.0

No.	Tank	Capacity [m^3]
12.	Bunker	19.0
13.	Bunker	16.7
14.	Water ballast	29.1
15.	Water ballast	29.1
16.	Water ballast	20.8
17.	Water ballast	20.8
18.	Fresh water	5.1
19.	Sewage	6.1
20	Grey water	6.1
21.	Fresh water	5.1

3.3.2 Tween deck

The aft compartment contains a rudder propeller room with two ducted Azimuth thrusters with respective electrical motor and frequency converter. Each propeller will be able to deliver 1600 kW in order to supply sufficient thrust in transit operations. The engine room consists primarily of four main genets that can produce 1000 kW each with 6 cylinders 4-stroke diesel engines. In the forward compartment there is a switchboard room and work shop. Figure 3-7 shows the tween deck layout.

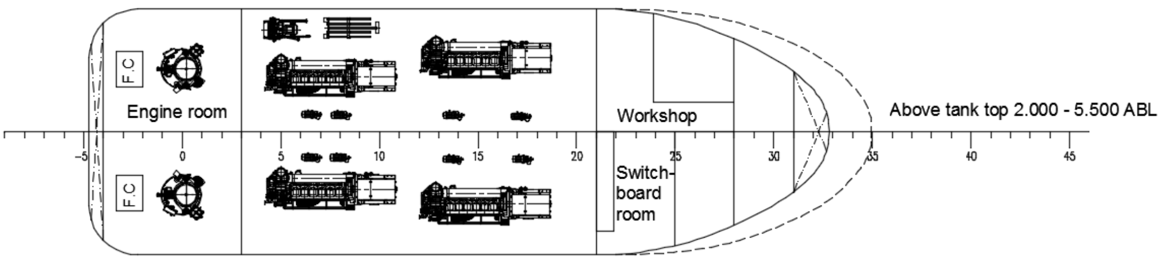

Figure 3-7: Tween deck 2000 millimetres above baseline

3.3.3 Main deck

The main deck has a large weather deck area where added equipment could be placed such as winches, small cranes and other machineries. There should also be space for life-saving appliances such as rescue crafts, lifebuoys and lifejackets. A first aid kit will be available on the main deck in case of accidents. The forecastle deck area should have sufficient space for a drum to the power cord supplying the installation vessel with electricity.

Near the bow is a machinery room for the connection shaft to the installation vessel. This concept is further explained in Chapter 3.4.3 Connection part. The shaft could either be operated by an electric motor or by hydraulic pressure and need to have an incorporated damping system to take care of the forces arising from sea motions.

The first story of the accommodation area is located at the main deck. In the accommodation area there should be toilets, day room, kitchen and a mess room available where the crew can eat and have meetings. Since the installation site is close to shore and the time for transit and installation of each unit will be limited to twelve hours, the crew will work in shifts. Because of this, there is no need for cabins for the installation personnel on the support vessel. Figure 3-8 shows the main deck layout.

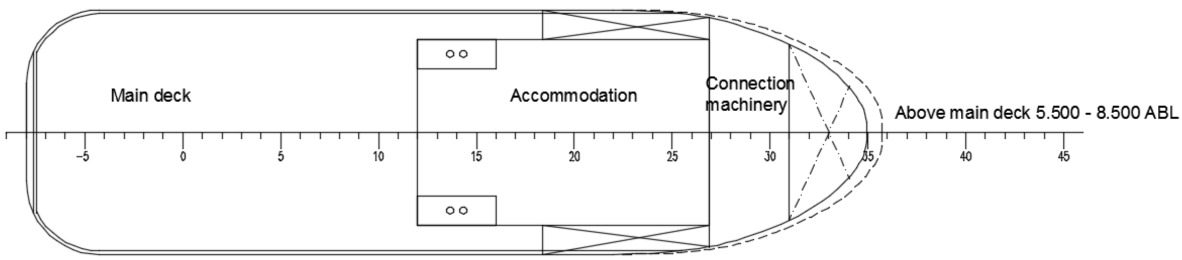

Figure 3-8: Main deck 5500 millimetres above baseline

3.3.4 Upper deck

The upper deck contains the second story of the accommodation area. It has also an emergency power supply room with an independent emergency generator of 85 kW in order to supply electric power to crucial consumers during emergencies. The location of the generator together with the emergency switchboard should be positioned in a way that it could be accessed from open deck which is why it is positioned here and not in the machinery room. Figure 3-9 shows the upper deck layout.

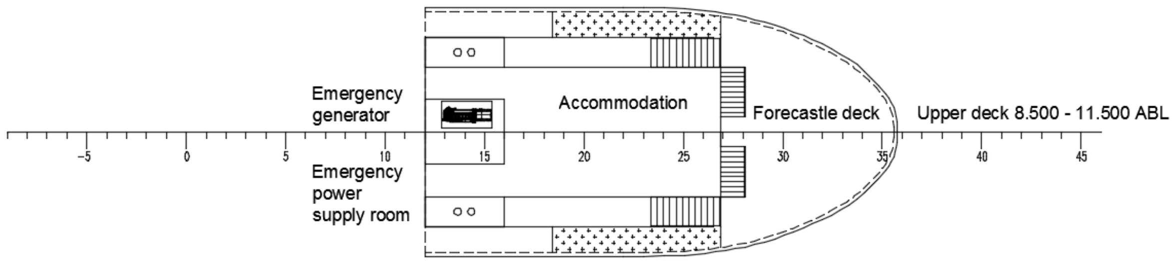

Figure 3-9: Upper deck 8500 millimetres above baseline

3.4 Ship equipment

This chapter deals with the ship equipment on-board the installation vessel. When the ship is installing the wind turbines, common practice is to have a small crane that can handle the equipment necessary for the operation and allowing for on-board access. Further, the installation vessel also needs equipment for mooring and anchoring.

3.4.1 Frame – Fixtures the foundations

In order to lift the foundation onto the installation vessel and carry it to the wind farm site a special fixture is designed. The fixture is designed as a U-shaped frame as shown in Figure 3-10. The idea is to ballast and let the installation vessel go beneath the frame and then de-ballast to lift the frame which is connected to the foundation. The frame rests on the structure surrounding the moon pool and is secured to the vessel. The frame is designed to take care of the forces arising from carrying the foundation and distributes the added weight from the foundation to the structure. The mechanical work to lift and lower the foundation is completely done by ballasting. Several lifting frames are needed at the assembly site to be efficient and able to ship the next foundations once the installation vessel revisits the harbour. The frame is connected to the bottom plate of the foundation through a truss structure of steel. When positioning the foundation, the installation vessel ballasts and the frame and truss are disconnected by a remotely operated locking device.

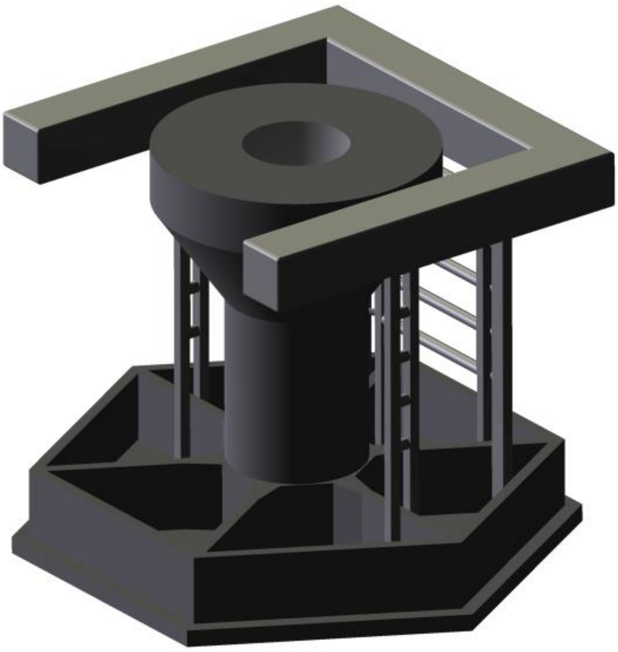

Figure 3-10: Lifting frame for gravity foundation

3.4.2 Rotatable lifting beam – Lifting appliances for wind turbines

Lifting appliances for the wind turbine is designed in two parts with one fixed structure going across the moon pool and a rotatable beam, see Figure 3-11. The rotatable beam is hinged on a pedestal and could be rotated in the horizontal plane.

Since there is no heavy-lift crane in this application the biggest issue is securing the wind turbine to the installation vessel during transport but also enable access to the holes in the bottom flange. During installation the installation vessel will position itself over the gravity foundation and carefully ballast until the bolts from the foundation pass through the holes on the flange. It is impossible to hold the wind turbine by using the bottom flange and in the meantime have access to the bolting holes. One way around this dilemma would be to hold the wind turbine by applying pressure on the pillar a few metres above the bottom flange. This solution calls for higher requirements on the rotatable beam which should provide pressure and there is also some safety aspects concerning risks due to machinery malfunction. Since the installation vessel has small roll and pitch motion and the centre of gravity for the wind turbine is very high, the lifting beam could be subjected to critical torsional loads and bending moments.

Figure 3-11: Rotatable lifting beam for holding the wind turbine

However, the best option solving above issue is by introducing a second flange with holes positioned approximately 2 metre above the bottom plate of the wind turbine. The pillar could then be bolted to the lifting appliances while the bottom flange holes are exposed with a clear height of 0.5 metre. The rotatable beam as well as the fixed beam rests on two longitudinal beams welded to the installation vessel. The load from the wind turbine is then completely distributed to the ship's hull structure and the hinge does not take any bending moment or torsional load. Figure 3-12 illustrates a close-up of the rotatable and fixed beam and bolts.

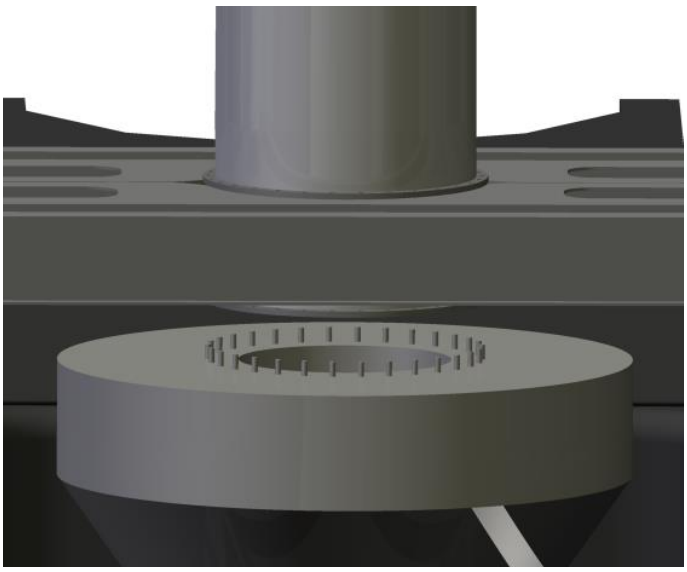

Figure 3-12: View showing the bolts and two flanges

3.4.1 Connection part

The connection between the installation vessel and the support vessel is a mechanical coupling providing efficient solution in pushing operation. By introducing this mechanical coupling, the two vessels act as a single unit with only one degree of freedom: allowing the push-tug to pitch about the transverse connection while all other motions are restrained. Benefits include abilities to operate in a wide range of sea states, complete elimination of hull contact as well as towing lines and related equipment (Intercon, 2014). The manoeuvring abilities are improved which is favourable while steering in narrow ports or installing wind turbines which require centimetre precision guidance of the lifting alliances. Since the draft of the installation vessel is larger than most tugs, a push-tug seems as the best option to avoid interference from propellers and water jet onto the object being towed. This configuration also improves redundancy and flexibility for the whole concept. Figure 3-13 illustrates the configuration.

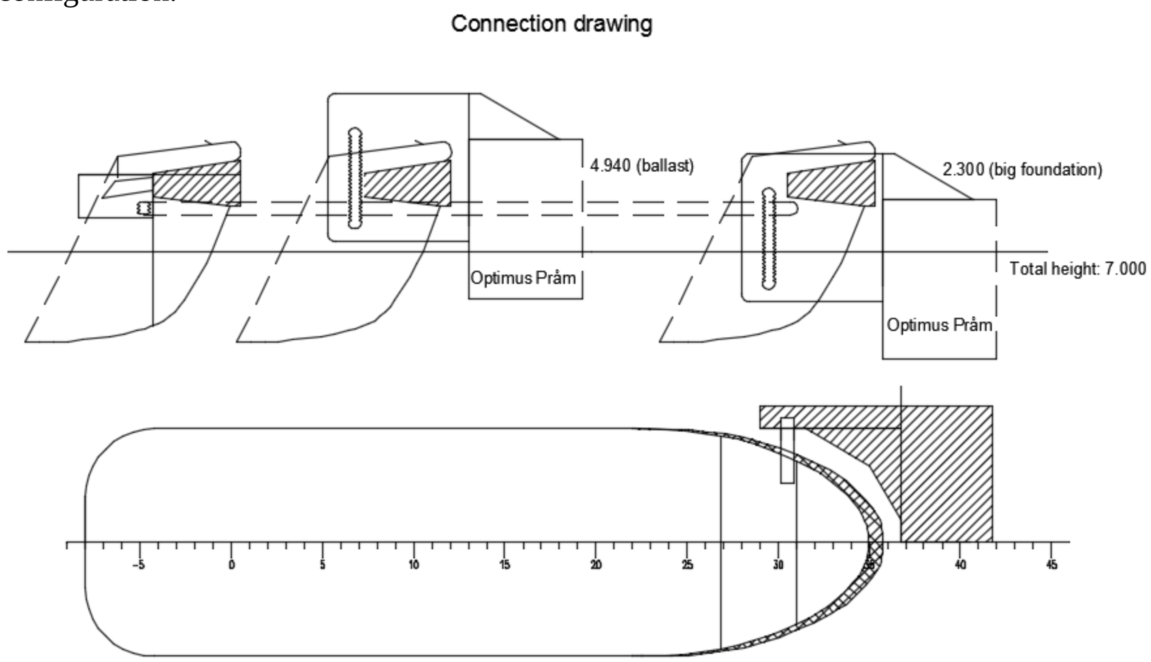

Figure 3-13: Drawing of the connection between the installation vessel and the supply vessel

The idea is a notch in the stern of the barge shaped after the tug’s bow, in which the tug enters and connects with the barge. Transverse shafts from the push-tug engage with a groove at the walls of the connection part, see Figure 3-14. The grove is designed in a jigsaw pattern in which the shafts’ head mount. This feature allows for changes in barge draft by retracting the interface heads and re-connects at a new vertical level. The shafts could be operated by electric motors or hydraulic cylinders.

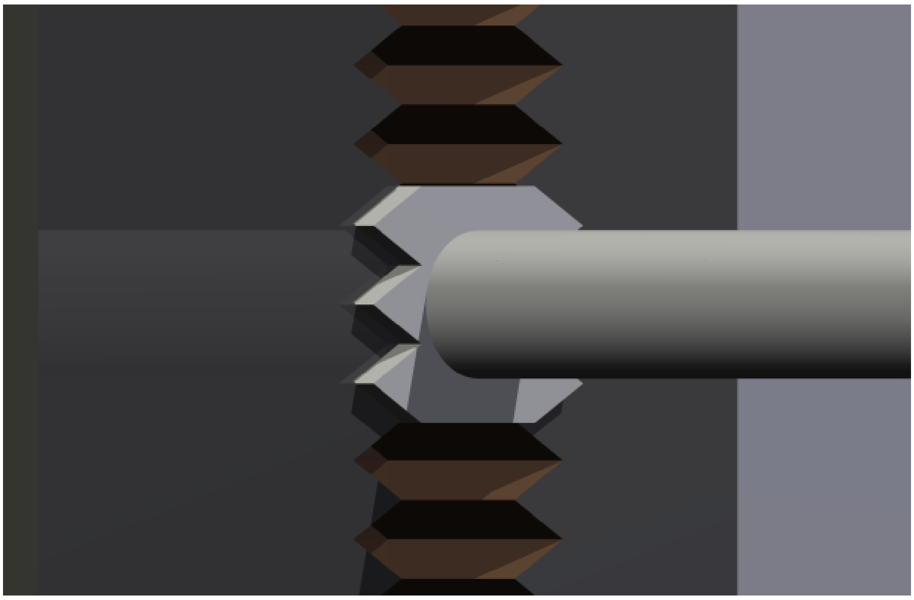

Figure 3-14: Connection interface

3.5 Risk analysis

The most likely hazards due to propulsion failure and navigation errors are identified. Navigation errors are not thoroughly investigated due to project limitations. Hazards due to connection loss between vessels and following measures are further investigated in the Chapter 6.3.

- Grounding due to navigational error
- Grounding due to propulsion failure
- Collision with port due to navigational error
- Collision with port due to propulsion failure
- Collision with other vessels due to navigational error
- Collision with other vessels due to propulsion failure
- Collision with support vessel
- Damaging cargo during operation
- Connection loss between vessels
- Malfunctioning pumps

3.6 Safety

The safety on this vessel is in accordance with the class rules provided by DNV for a push/barge vessel (Den Norske Veritas, 2013). DNV was the classification society requested by the customer. Safety enhancing equipment follows the rules described in the Safety of Life at Sea (SOLAS, 2004).

3.6.1 Life-saving appliances and arrangements

According to DNV regulations the life-saving appliances and arrangement follow Chapter III Part A and Section I of Part B in SOLAS (Den Norske Veritas, 2013). Lifebuoys shall be available on both sides of the support vessel, preferably on all open decks and at least one in the vicinity of the stern (SOLAS, 2004). None of them shall be permanently secured but be able to be rapidly cast loose in case of emergency. At least one shall have a 30 metre long buoyant lifeline. Self-ignition lights shall be attached to 50 per cent of the lifebuoys and two of those shall also have self-activating smoke signals. Lifejackets shall be placed on different locations of the vessel, easily accessible and their position shall be clearly identified. In case of emergency clear instructions shall be provided to the crew and installation staff in the language required by the ship’s flag state and in English.

3.6.2 Lifeboats

The provision and stowage of lifeboats and rescue boat is in accordance with the same rules as presented in Chapter 3.6.1. The vessel shall carry one lifeboat on each side with the possibility to accommodate 50 per cent of the crew (SOLAS, 2004). In addition, inflatable life rafts shall be provided so that there is enough survival craft on each side to accommodate all crew and installation staff on board. All lifeboats/ rafts shall be stowed in a way that they are ready for launching in not more than five minutes and protected from damage by fire and explosion.

Due to the special operations taking place when assembling the windmill to the foundation a rescue boat shall be available. During the assembling of the windmill and the foundation, technical personnel board the barge. Since there is no possibility to carry a lifeboat on the barge a rescue boat on the tug is needed. This is also stated in the classification rules. The rescue boat shall be cast loose and stay in the vicinity of the barge during the time that the personnel is conducting the assembling.

3.6.3 Fire safety

The objectives of fire safety are to the highest possible extent prevent fire and explosion and thereby reduce the risk of these damaging the ship, its cargo and crew (SOLAS, 2004). If a fire break out there shall be fire detection appliances and proper ship division, so that the risk of smoke and fire spread is prevented. The fire safety is to comply with the requirements of Ch.II-2 of SOLAS.

3.7 Navigation and communication

The rules concerning navigation and communication equipment on the support and installation vessel are in accordance with SOLAS (IMO 2012).

3.7.1 Navigation equipment

The support vessel has an integrated bridge system (IBS), which enable efficient and safe ship manoeuvring and management by the personnel (IMO, 2014). This includes RADAR, ECDIS, GPS and conning displays (IMO, 2004). For DP the following sensors and reference systems are fitted:

- Differential global positioning system
- Hydroacoustic Position Reference
- Gyros
- Motion reference units

3.7.2 Communication equipment

The vessel operates in the A1 area and has the communication equipment recommended for this area in accordance with SOLAS (IMO, 2004). All communication antennas are placed on top of the superstructure of the support vessel and on top of the vertical tanks on the installation vessel.

The ship is equipped with the following communication equipment:

- GMDSS radio station with DSC
- VHF radio system
- UHF radio system for internal communication
- Receiver capable of receiving international NAVTEX service broadcast
- Internal PA/GA alarm system
- PC network
- Wi-Fi with internet access
- EPIRB

3.8 Economical overview

One of the most motivating causes for this project, to find a new concept for offshore wind installation vessels, is the fact that all available option on the market is very costly (Edvardsson, 2014). As a reference example, the Rambiz heavy lift vessel is used. The daily rate for renting Rambiz is almost one million SEK, which points towards a great economic benefit for the costumer to construct their own vessel. In this report the economic aspects have not been thoroughly investigated just a simple analysis to get some indication if the project is of interest. The cost estimations are divided into three different parts:

- Manufacturing cost
- Running cost
- Crew cost

Maintenance cost is not included since this project only effect the cost connected to the installation. In order to estimate the cost from capital employed in the vessel in connection to the purchase the annuity model is used. The estimated daily rate for Optimus Pråm is estimated to almost 400 000 SEK. All calculations are presented in Appendix A. The result is presented in Figure 3-14.

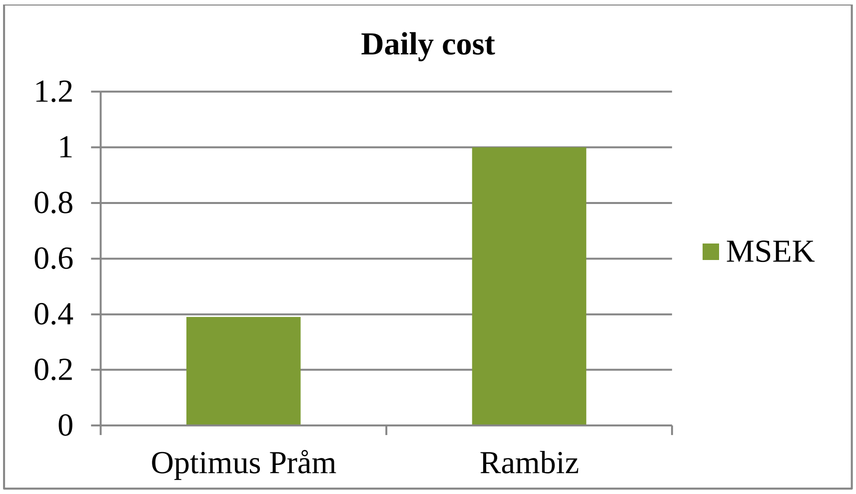

Figure 3-14: Daily cost using Optimus Pråm and Rambiz

3.9 Logistics and Operational description

The concept is designed to install gravity foundations and windmills in the Baltic Sea in the bay of Hanö, see Figure 4-1. The concept is divided as two separate vessels that will operate together. The installation vessel is a semi-submersible barge. The task for the vessel is to install gravity foundations and also install windmills onto these foundations. The dimensions of the foundations are 15-40 metres high and the weights are between 1400-2600 tons.

Figure 4-1: Area of operation in the bay of Hanö

A support vessel is needed to supply the installation vessel with power for the entire operation and propulsion during transit. The power is distributed from the support vessel to the installation vessel through a power cable. A logistic chain is carried out to find the hazardous operations in the logistics to install foundations and windmills.

3.9.1 Harbour

The harbour is located in the port area of Stilleryd, Karlshamn (56°9'25.4"N 14°49'46.1"E), see Figure 4-1. This will be the location for manufacturing and storage of the foundations and assembly of windmills before transport to installation site. The foundations are casted locally at the harbour and handled on skidding rails due to their high weights. The windmills are ordered in pieces and then assembled in the harbour. There are two different types of docksides. One is a ramp to lower the foundation to a pick-up level for the installation vessel as in Figure 4-2. The installation vessel lifts the foundation by de-ballasting the vessel, so no heavy lifting crane is required. The other dockside is also an assembly point for the windmill. By assembling and pick up the windmills at the same location, there is no unnecessary port handling.

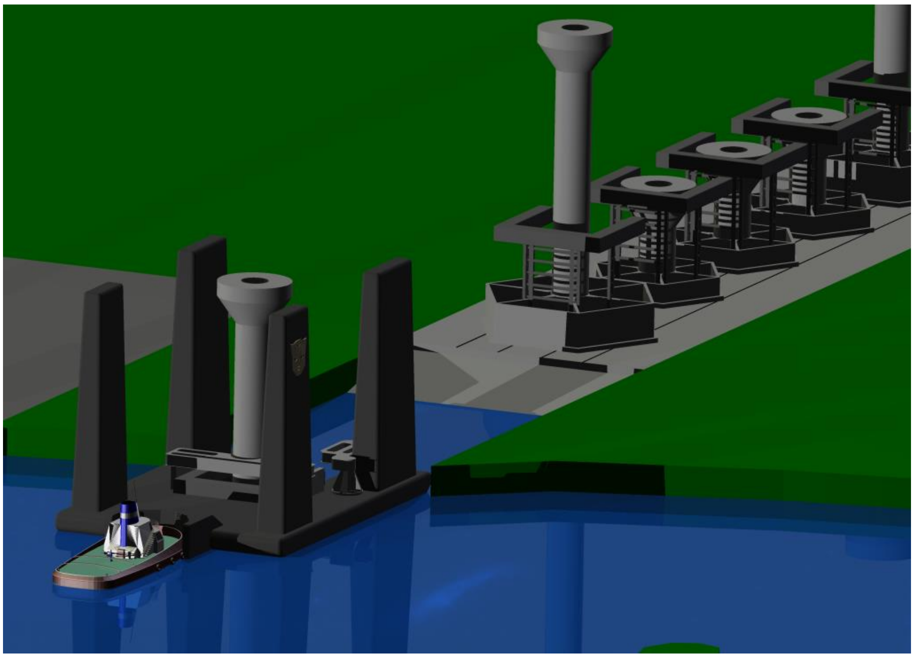

Figure 4-2: Ramp for foundation handling

The dockside for the windmill has two pick up positions, were also the windmills are assembled, see Figure 4-3. By fixing the windmill on the standard connection, no extra attachments are needed. With this solution the handling of the fully mounted windmill in port is minimized and the vessel will be able to directly pick up the windmill when it is ready for transit to the site location. Minimum two attachment points are needed to enable continuously handling of wind turbines at port.

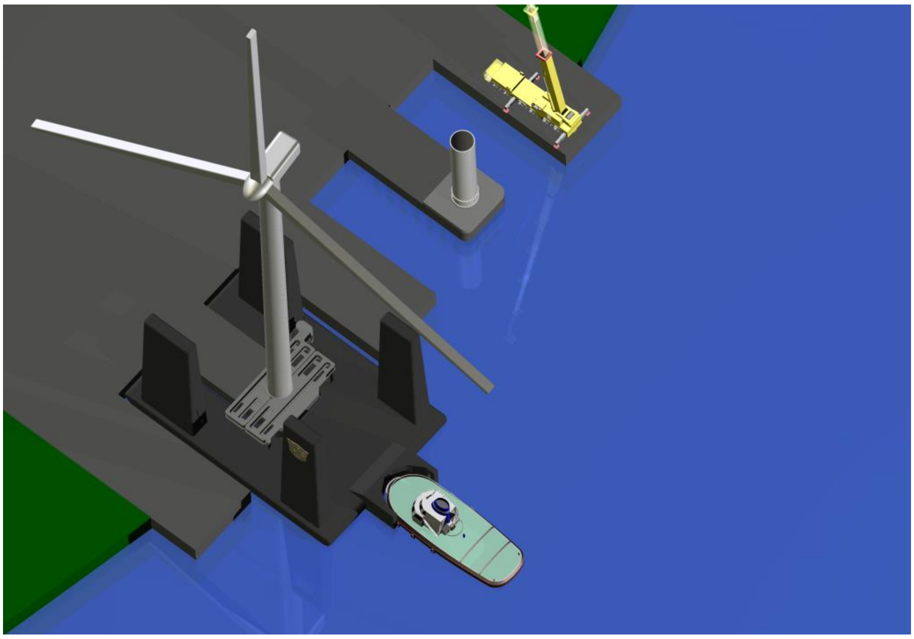

Figure 4-3: Two handling points for the windmills

3.9.2 Transit

During transit the installation vessel and support vessel is connected to each other in all motions except pitch. This type of almost total fixed connection makes the two vessels acts like one, which gives the united vessel an excellent manoeuvrability. The propulsion during transit is delivered from the support vessel and the transit speed is 3.5 knots for the largest gravity foundation. Depending on the location the transit time to the installation site is approximately four and a half hour.

3.9.3 Installation of foundation

When the vessel has reached the site location for the foundation, the installation vessel and the support vessel disconnect. The installation vessel then positions itself using thrusters while lowering the foundation by ballasting.

The installation vessel places the foundation at the planned location by ballasting the vessel until the foundation has reached the seabed. The vessel is designed for ballasting and place foundation up to a water depth of 35 metres that is demonstrated in Figure 4-3. This ballasting operation takes maximum 2 hours and the same applies for de-ballasting after realising the foundation. After de-ballasting the installation vessel and support vessel attaches again and transit home to the harbour. The foundation needs gravel ballast before installing the windmill on top of it, this operation is carried out by other vessel and is not further addressed in this report.

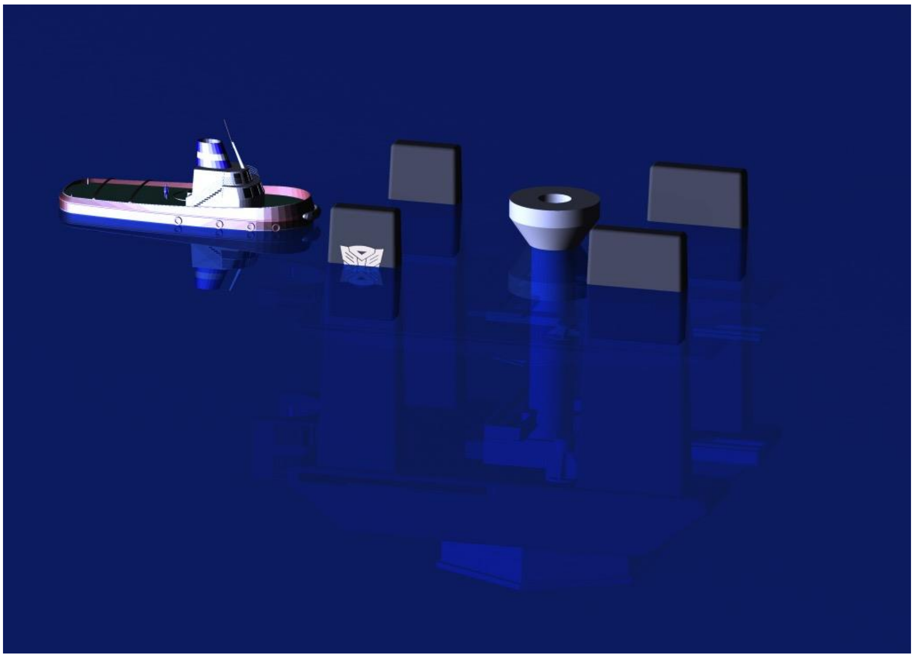

Figure 4-4: Place the foundation at sea bed by ballasting in depth of 35 metres

3.9.4 Installation of windmill

The windmill shall be placed on top of the foundation. When the vessel reaches the location of the foundation that the windmill should be mounted, the installation vessel and the support vessel stays connected as in Figure 4-5. The installation vessel does not submerge during this installation and by keeping the two vessels connected it is easier to position during the installation. Positioning during this operation has to be very precise and the installation vessel leans against the foundation to lower motions in surge and sway. A fender attached in the vessels moon pool protects the vessel from direct contact with the foundation and also damps the vessel contact motion. The major difference from installing the foundation is that to install the windmill the installation vessel has to stay put and hold the windmill safely, while personnel attaches the windmill to the foundation.

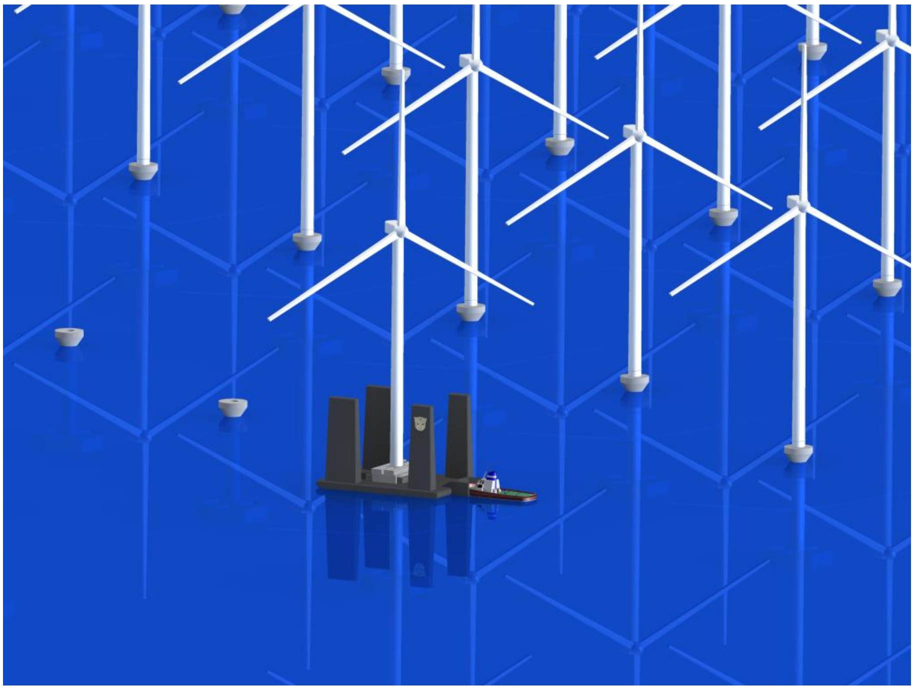

Figure 4-5: Windmill installation on site

4 Hydromechanics

In this chapter hull lines, hydrostatics, stability, sea-keeping, resistance and propeller calculations will be presented. The focus is set on the results rather than the procedure. The results are compared and discussed along with the criteria's from the client and regulations. Finally some future work is presented.

4.1 Hull Lines

The hull lines of the installation vessel can be seen in Figure 4-1, these are the result of several iterations intended to obtain sufficient stability while performing its designated tasks. The towers located on the installation vessels supply with enough stability and residual flotation in all of the operational conditions.

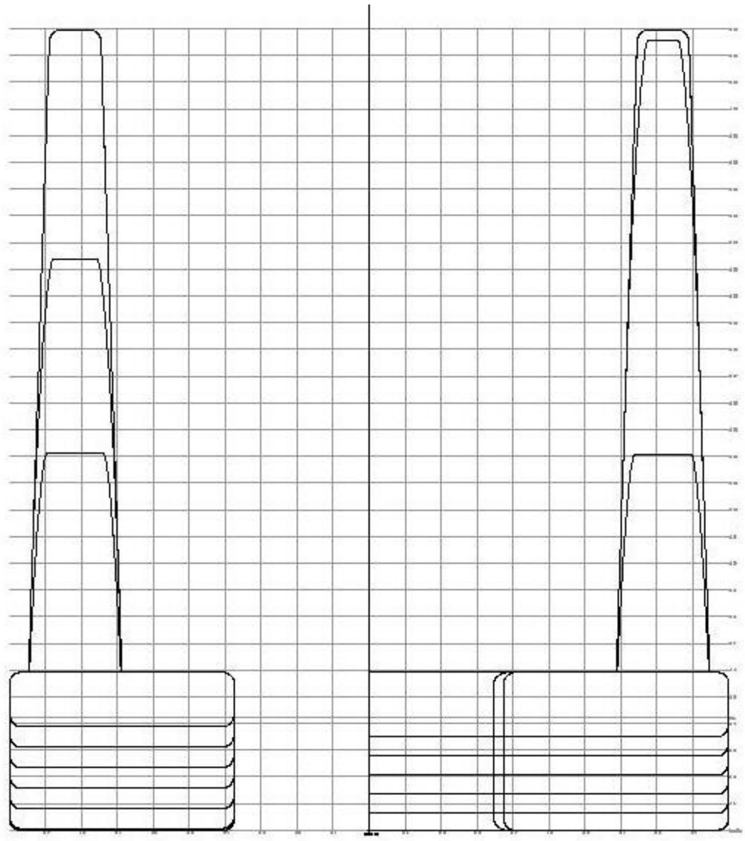

Figure 4-1: Hull lines with section spacing of 1.05 m

4.2 Hydrostatics

For modeling the installation vessel two software Rhinoceros and Maxsurf Modeler are used. Hydrostatics and stability calculations are carried out in Maxsurf Stability Enterprise software. There are two models instead of one and the reason is to have a better precision on submerged operations for gravity foundation installment and transportation. Since there is a considerable (up to 756.1 t) buoyancy contribution coming from semi-submerged gravity foundation, Model 2 is taking this buoyancy into account. Model 1 is without foundation and Model 2 is with the biggest gravity foundation. Models can be seen at Figure 4-2.

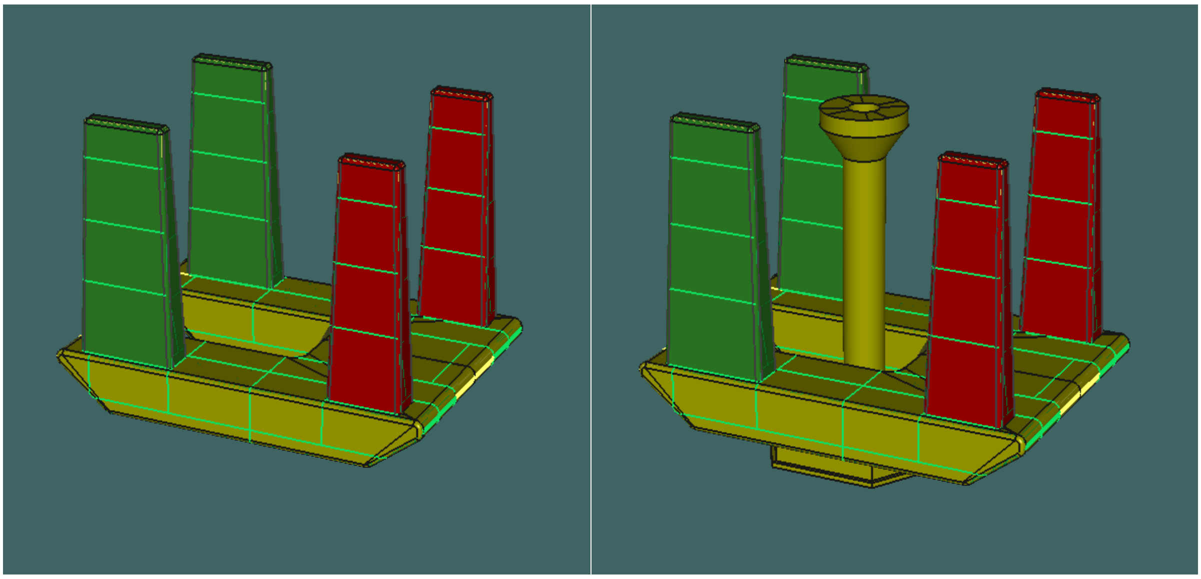

Figure 4-2: Model 1 on the left, Model 2 on the right

4.2.1 Hydrostatic Calculations

Calculations for hydrostatics are done on the model which is without gravity foundation. Since the draft is changing drastically and rapid change in the submersed geometry is observed during operations, two separate graphs will be provided. Figure 4-3 will provide information until water level reaches towers.

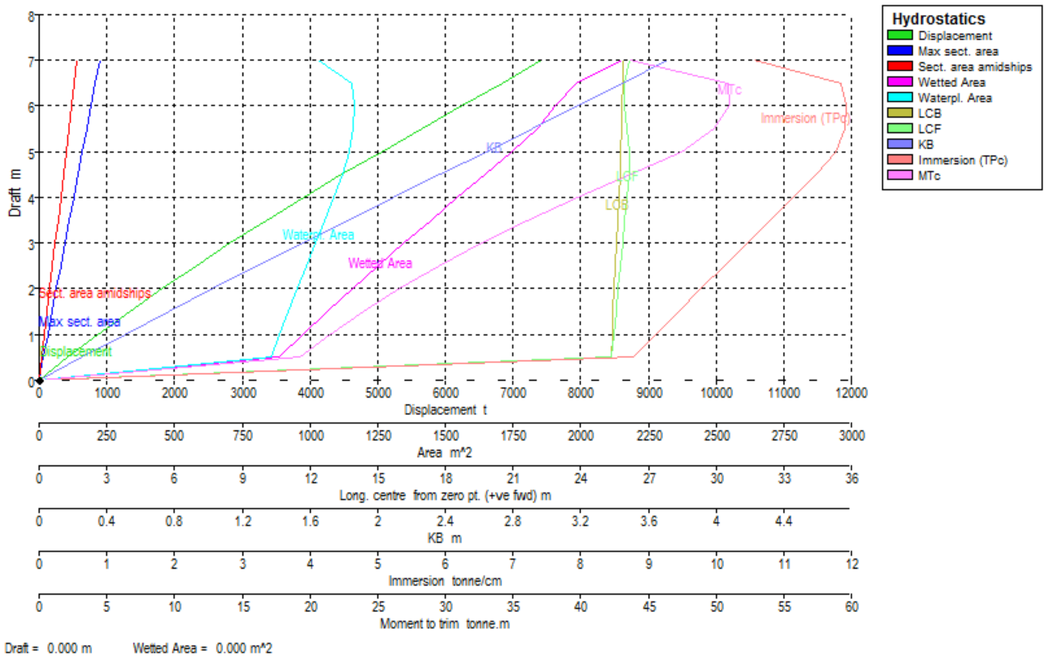

Figure 4-3: Hydrostatics between 0-7 meters of draft for Model 1

Figure 4-4 is showing hydrostatics from baseline to maximum draft of 30 meters.

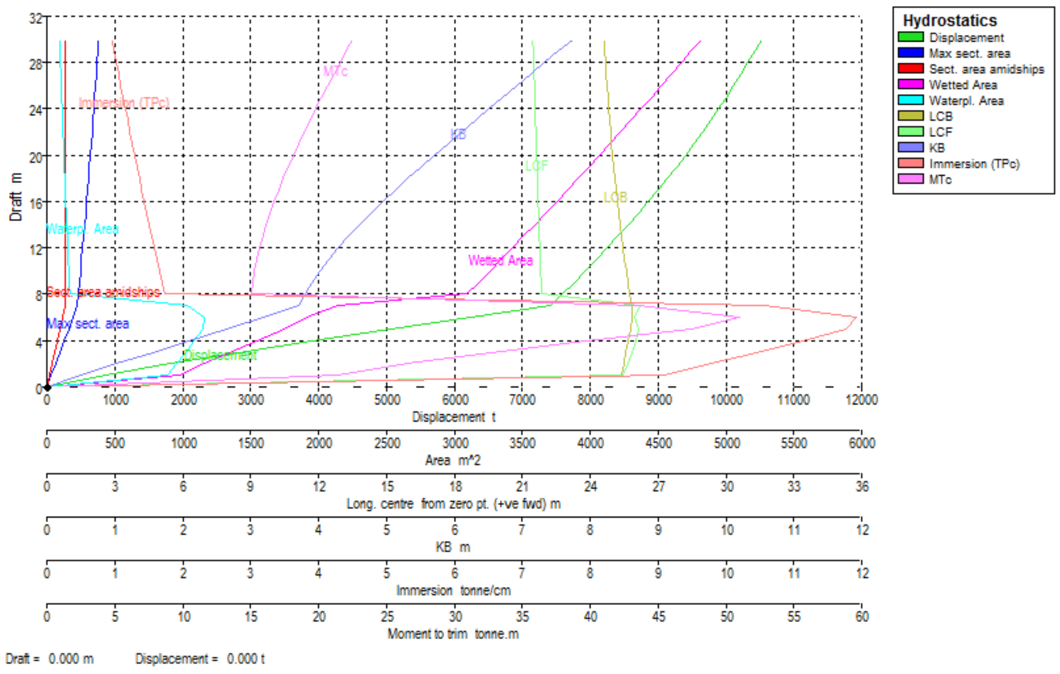

Figure 4-4: Hydrostatics between 0-30 meters of draft for Model 1

Operations with foundations will affect hydrostatics considerably; therefore the biggest gravity foundation and installation vessel will have hydrostatics as shown in Figure 4-5.

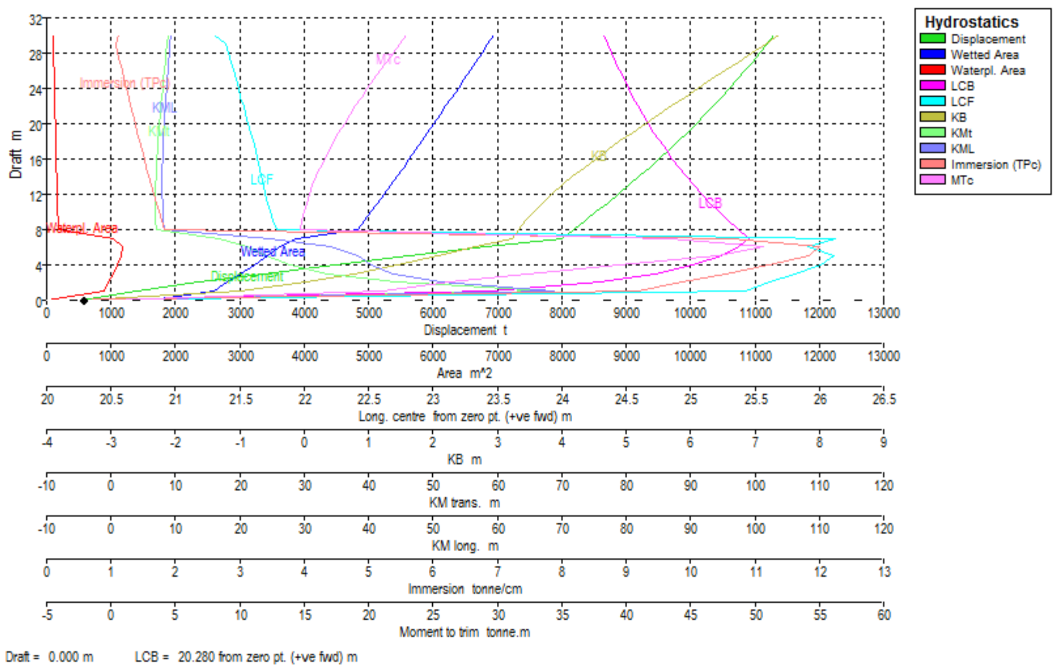

Figure 4-5: Hydrostatics between 0-30 meters of draft for Model 2

4.2.2 Cross Curves

Cross curves are calculated through Model 1 between the minimum estimated draft and maximum operation draft of the vessel. It is important to notice that cross curves are only relevant to the actual behavior of installation vessel up to some heeling angle. Excessive trim is observed during heeling due to the bow and stern geometry characteristics. The installation vessels stern has a catamaran like body, while the bow is a monohull. When one side of the vessel gets into the water during heeling, buoyancy distribution at bow and stern differs considerably. The result is trimming to the stern after certain degrees of heel. However cross curves are calculated with zero trim. Therefore one should be skeptic for the KN values at angles bigger than 40 degrees.

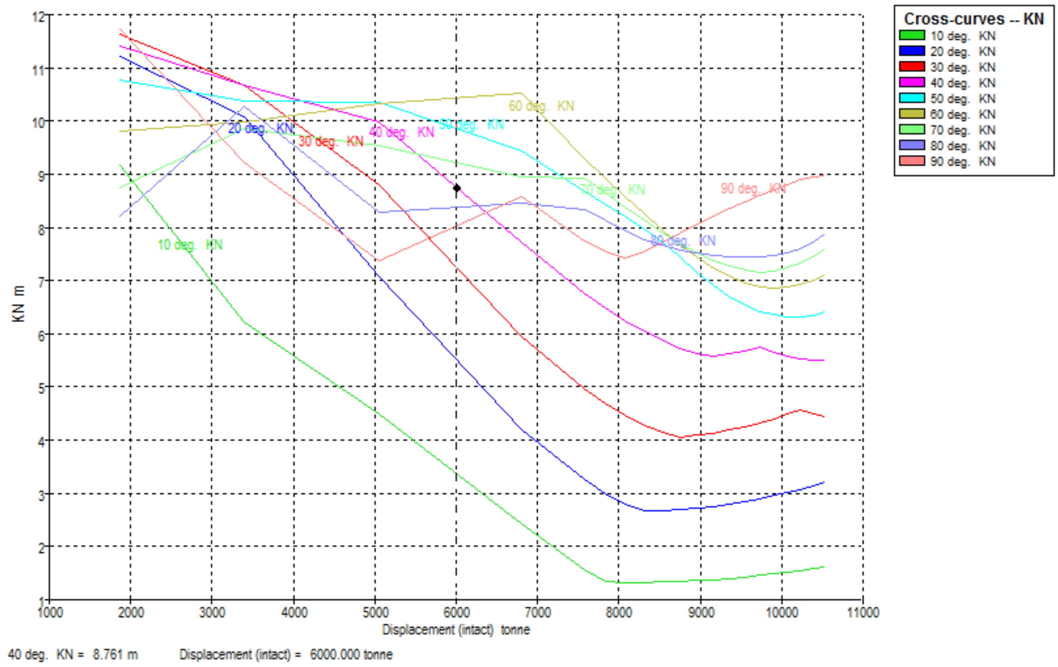

Figure 4-6: Cross Curves

4.3 Stability

According to the DNV Part 5 Chapter 7 Section 21, stability requirements in transit conditions are 2008 IS Code Part A, Ch.2.2 and 2.3. However if Part A, Ch.2.2 is impracticable, then the criteria of Part B, Ch.2.4.5 may be used. Calculations showed that except one criterion at only one loading case; Part A, Ch.2.2 is fulfilled. As stated, Part B Ch.2.4.5 is used for that criterion. Margins are measures of how much more attained values gathered as fraction of residual attained values divided by required values in.

Table 4-1: Intact Stability Criterion

	Criteria	Requirement	Unit	2008 IS Code
1	Area under GZ curve from 0^0 to 40^0 bigger than	0.055	m.rad	Part A, Ch.2.2
2	Area under GZ curve from 0^0 to 40^0 bigger than	0.09	m.rad	Part A, Ch.2.2
3	Area under GZ curve from 30^0 to 40^0 bigger than	0.03	m.rad	Part A, Ch.2.2
4	GZ at 30^0 or maximum rightening arm bigger than	0.02	m	Part A, Ch.2.2
5	Maximum of GZ shall not be smaller than	15	degree	Part B, Ch.2.4.5
6	Initial GM shall not be smaller than	0.15	m	Part A, Ch.2.2
7	Angle of steady heel shall not be greater than	16	degree	Part A, Ch.2.3
8	Angle of steady heel / Deck edge immersion angle shall not be greater than	80	%	Part A, Ch.2.3
9	Area *b* / Area *a* shall not be less than	100	%	Part A, Ch.2.3

The weather criteria, Part A Ch.2.3, is a measure of ability of the vessel to withstand the combined effect of beam wind and rolling. Installment vessel is subjected to a steady wind pressure acting perpendicular to centerline. Heeling lever caused by gust wind l_{w2}, the resultant angle of equilibrium φ_0 , roll owing to wave action φ_1 ; angle of vanishing stability, downflooding point or 50^0, whichever is less φ_2 can be as seen at Figure 4-7. Therefore vessel will have sufficient dynamic stability which can be interpreted as kinetic energy required recovering vessel from heeled condition caused by the wind or the severe roll.

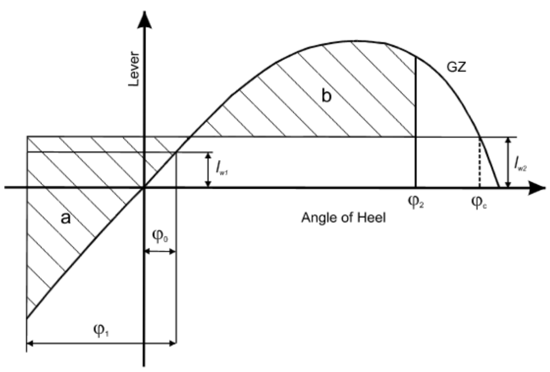

Figure 4-7: Weather Criteria Supplement

Intact stability and damage stability criteria in temporary submerged condition stated in DNV Part 5 Chapter 7 Section 21 can be seen in Table 4-2 and Table 4-3.

Table 4-2: Intact stability in temporary submerged condition

	Criteria	**Requirement**	**Unit**
1	GM at equilibrium shall not be less than	0.3	m
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.1 meters	15	degree
4	maximum righting arm shall occur at an angle of heel not less than	7	degree

Table 4-3: Damage stability in temporary submerged condition

	Criteria	**Requirement**	**Unit**
1	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.05 meters	7	degree
2	angle of heel after flooding shall not exceed	15	degree

As stated in previous DNV rules, “For the purpose of damage stability calculations, a damage extent of 5 m horizontally along the surface shall be assumed for all exposed surfaces except the cargo deck. Watertight bulkheads may be considered to remain intact provided that the distance between adjacent bulkheads exceeds 5 m. The damage penetration into the structure shall be assumed to be equal to 0.76 m and the vertical extent of damage is assumed to be from the cargo deck or its horizontal extension upwards without limit. For the cargo deck a damage extent of 5 m shall be assumed. Watertight bulkheads may be considered to remain intact provided that the distance between adjacent bulkheads exceeds 5 m. The damage penetration into the cargo deck shall be assumed to be equal to 0.76 m.” Damage exerted on hull is dimensioned and colored as red in Figure 4-8. Damage cases are created by placıng the red volume which is damage on bulkheads. Reader should note that when there is a damage on side of the vessel vertical towers are not damaged because distance between the vertical tower and side of the barge part is greater than 0.76 meters.

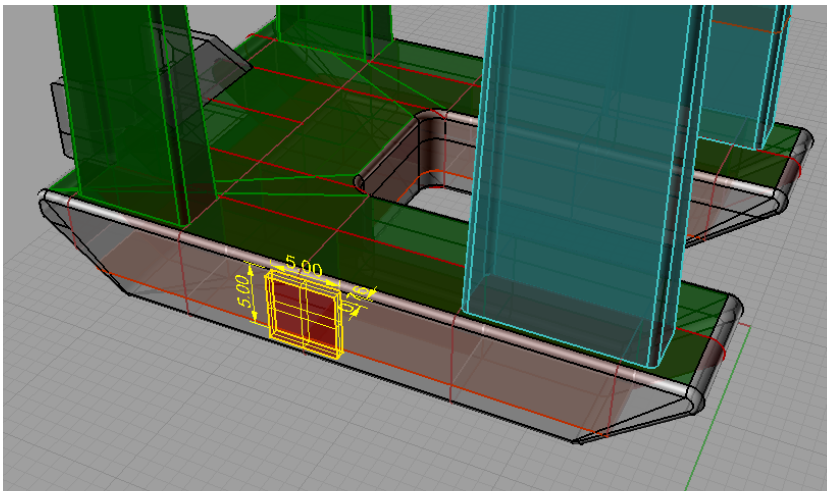

Figure 4-8: Damage Case 3 illustrated

All results for intact stability are compiled in to tables as can be seen in Table 4-4.

Table 4-4: Intact Stability Results

			Load Case 1		Load Case 2		Load Case 3		
Crit. No	2008 IS Code	Requirements	Actual	Margin %	Actual	Margin %	Actual	Margin %	Status
1	Part A, Ch.2.2	0.055	3.5247	6308.56	1.035	1781.82	1.5885	2788.1	Pass
2	Part A, Ch.2.2	0.09	4.672	5091.06	1.133	1159.74	2.2509	2400.95	Pass
3	Part A, Ch.2.2	0.03	1.1472	3724.16	0.098	229.23	0.6624	2108.01	Pass
4	Part A, Ch.2.2	0.02	7.27	36350	1.35	6650	4.02	20000	Pass
5	Part B, Ch.2.4.5	15	15	0	16.9	12.73	24.8	65.46	Pass
6	Part A, Ch.2.2	0.15	48.369	32146	13.03	8587.33	16.794	11096	Pass
7	Part A, Ch.2.3	16	0.3	98.11	0.4	97.24	0.3	97.92	Pass
8	Part A, Ch.2.3	80	2.17	97.95	5.68	92.9	10.2	87.25	Pass
9	Part A, Ch.2.3	100	147.27	47.28	110.9	10.97	168.99	68.99	Pass

Table 4-5: Intact Stability for Temporarily Submerged Conditions Results

		Load Case 4		Load Case 5		
Crit. No	Requirement	Actual	Margin %	Actual	Margin %	Status
1	0.3	1.765	488.333	1.245	315	Pass
2	15	35.8	138.667	32.5	116.667	Pass
3	7	plus 40		21.8	211.429	Pass

4.3.1 Tank Arrangement and Tank Ventilations

List of compartments and tanks can be found at Appendix B, permeability are determined according to DNV rules and for machinery spaces permeability of 0.85, for tanks and empty compartments permeability of 0.97 are assigned for intact and damaged conditions. Free surface effect is taken into account at its maximum value. This assumption is on the safe side because in some heeling angles free surface in the tanks can be lower than other angles. Density is assigned as 1.025 t/m^3. Downflooding points are determined after extensive damage stability calculations since they have to stay out of the water up to some heeling angles in order to fulfill the requirements.

4.3.2 Intact and Damage Stability Loading Conditions

Loading cases and corresponding models for intact stability calculations are shown in Table 4-6.

Table 4-6: Loading cases and corresponding model

	Loading Case	Model Name
1	Ballasted Condition	Model 1
2	5 MW wind turbine transportation	Model 1
3	Biggest gravity foundation transportation	Model 2
4	Biggest gravity foundation installment	Model 2
5	Biggest gravity foundation surfacing	Model 1

According to the description given for damage stability calculations, the 12 most severe damage cases are generated. As can be seen at Appendix B, number 1 represents damaged compartment or tank while number 0 means no damage.

4.3.3 Stability Assessment

Stability results will be presented here for intact conditions and intact temporarily submerged conditions. Full data for results can be seen in Appendix B.

4.4 Intact Stability

Intact stability of the installation vessel is investigated in this chapter. Connection between the supply vessel and installation vessel is ignored in transportation condition due to restrictions of the software.

4.4.1 Ballasted Condition

Loading condition of ballast leg is consisting of lightship and 165.45 tons of ballast at the fore tanks for fixing trim. Almost half of all the voyages done by the installation vessel will be in this condition. Draft of the vessel is 2.05 meters in this condition, which is the least draft among other loading cases.

Figure 4-9 shows the GZ curve for the ballast condition. The reason why the GZ curve starts from starboard heeling to port side is weather criteria calculations. Heeling stopped at 40 degrees since criteria for intact stability does not demand more angle of heel.

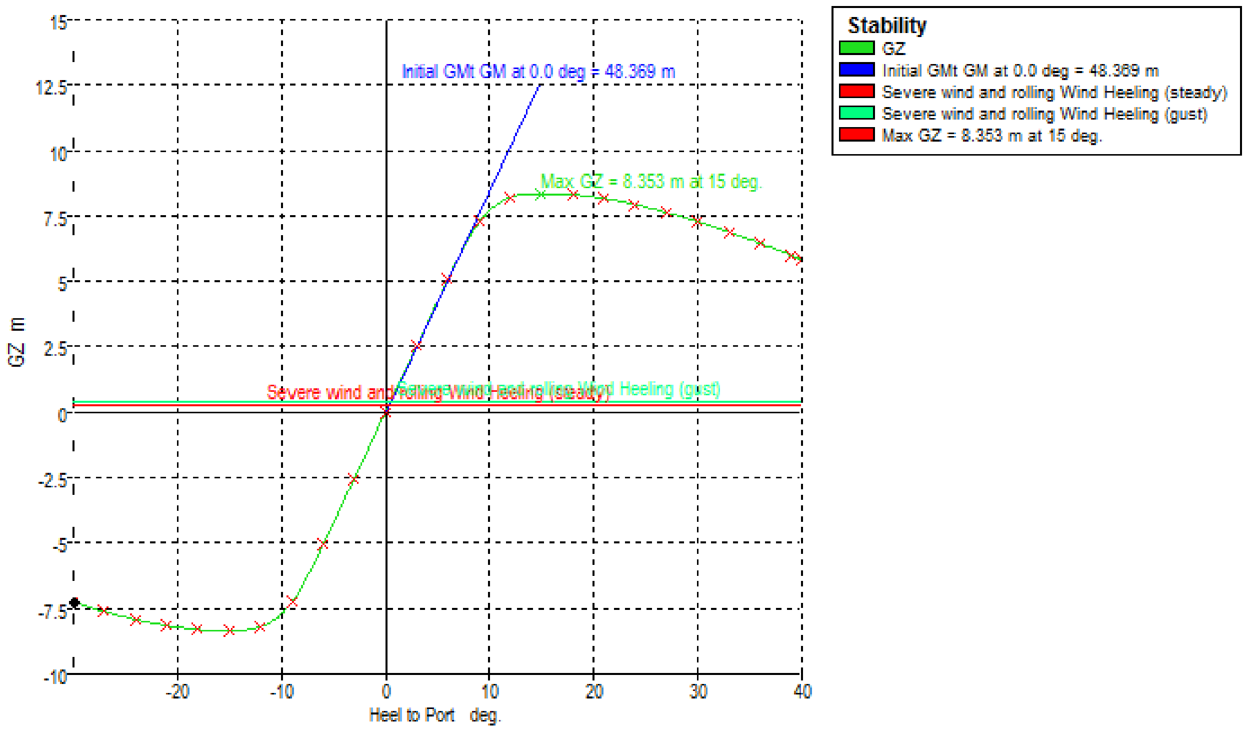

Figure 4-9: GZ curve of Ballasted Condition

Intact stability criteria shown in Table 4-7 are fulfilled.

Table 4-7: Ballasted Condition Stability Criteria

	Criteria	2008 IS Code	Requirement	Actual	Margin %	Status
1	Area under GZ curve from 0 to 40 bigger than	Part A, Ch.2.2	0.055	3.5247	6308.56	Pass
2	Area under GZ curve from 0 to 40 bigger than	Part A, Ch.2.2	0.09	4.672	5091.06	Pass
3	Area under GZ curve from 30 to 40 bigger than	Part A, Ch.2.2	0.03	1.1472	3724.16	Pass
4	GZ at 30 or maximum rightening arm bigger than	Part A, Ch.2.2	0.02	7.27	36350	Pass
5	Maximum of GZ shall not be smaller than	Part B, Ch.2.4.5	15	15	0	Pass
6	Initial GM shall not be smaller than	Part A, Ch.2.2	0.15	48.369	32146	Pass
7	Angle of steady heel shall not be greater than	Part A, Ch.2.3	16	0.3	98.11	Pass
8	Angle of steady heel / Deck edge immersion angle shall not be greater than	Part A, Ch.2.3	80	1.64	97.95	Pass
9	Area *b* / Area *a* shall not be less than	Part A, Ch.2.3	100	147.27	47.28	Pass

4.4.2 5 MW wind turbine transportation

Loading condition of biggest wind turbine consists of lightship, wind turbine, wind turbine handling unit and 2057.56 tons of ballast for lowering the center of gravity. Since the 5MW wind turbine has the biggest weight and highest center of gravity, other wind turbine sizes are assumed to be safe and not calculated.

The GZ curve for this loading condition is shown in Figure 4-10. As in the previous condition, the GZ curve starts from starboard side heeling to port side for weather criteria calculations. Heeling stopped at 40 degrees since criteria for intact stability does not demand more angle of heel.

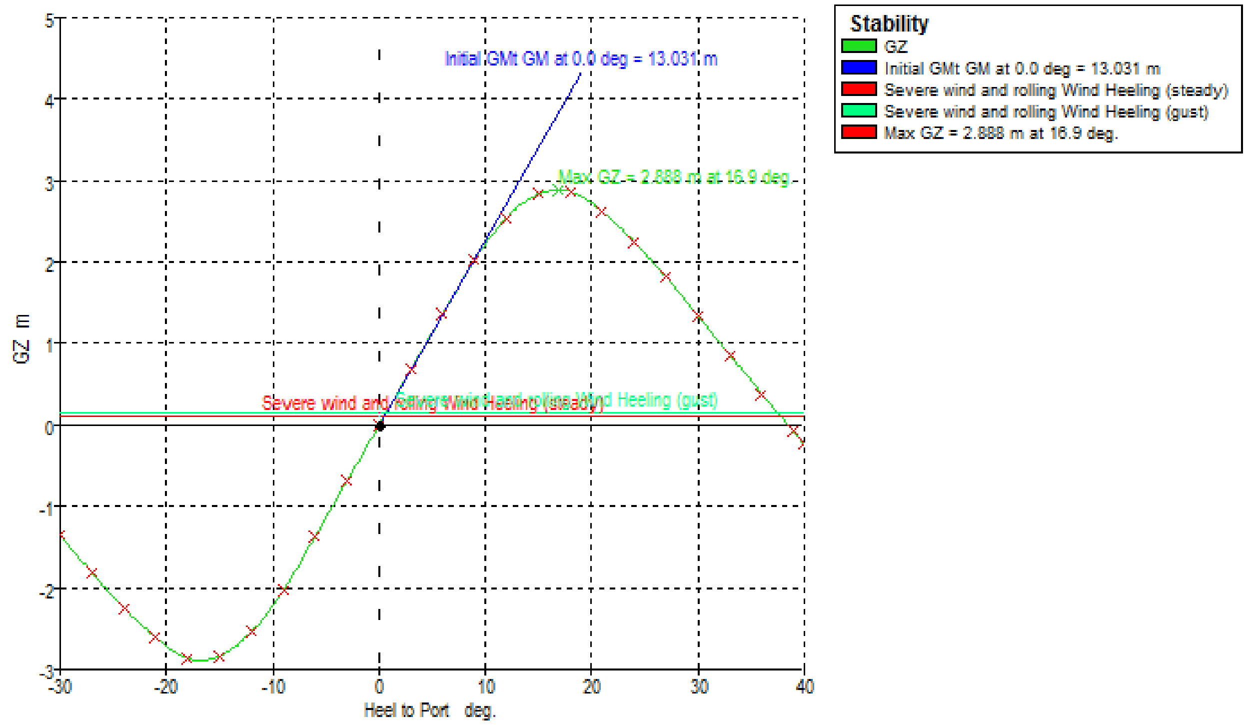

Figure 4-10: GZ curve of 5 MW wind turbine transportation

Intact stability criteria shown in Table 4-8 are fulfilled.

Table 4-8: 5 MW wind turbine transportation condition stability criteria

	Criteria	2008 IS Code	Requirement	Actual	Margin %	Status
1	Area under GZ curve from 0 to 40 bigger than	Part A, Ch.2.2	0.055	1.035	1781.82	Pass
2	Area under GZ curve from 0 to 40 bigger than	Part A, Ch.2.2	0.09	1.133	1159.74	Pass
3	Area under GZ curve from 30 to 40 bigger than	Part A, Ch.2.2	0.03	0.098	229.23	Pass
4	GZ at 30 or maximum rightening arm bigger than	Part A, Ch.2.2	0.02	1.35	6650	Pass
5	Maximum of GZ shall not be smaller than	Part B, Ch.2.4.5	15	16.9	12.73	Pass
6	Initial GM shall not be smaller than	Part A, Ch.2.2	0.15	13.03	8587.33	Pass
7	Angle of steady heel shall not be greater than	Part A, Ch.2.3	16	0.4	97.24	Pass
8	Angle of steady heel / Deck edge immersion angle shall not be greater than	Part A, Ch.2.3	80	5.68	92.9	Pass
9	Area *b* / Area *a* shall not be less than	Part A, Ch.2.3	100	110.9	10.97	Pass

4.4.3 Biggest gravity foundation transportation

Loading condition of biggest foundation consists of lightship, biggest gravity foundation, foundation handling unit and 1004.36 tons of ballast for fixing trim. As explained for wind turbines, this condition for foundations is considered to be the most extreme case therefore other sizes of foundations are thought to be safer and not calculated. However draught in this case is very important in order to be able to operate in harbor. Draft is 4.672 meters and 5 meters of foundation lower part, in total 10 meters of draught limit is not exceeded.

The GZ curve for this loading condition is shown in Figure 4-11. As in the previous conditions, GZ curve starts from starboard side heeling to port side for weather criteria calculations. Heeling stopped at 40 degrees since criteria for intact stability does not demand more angle of heel.

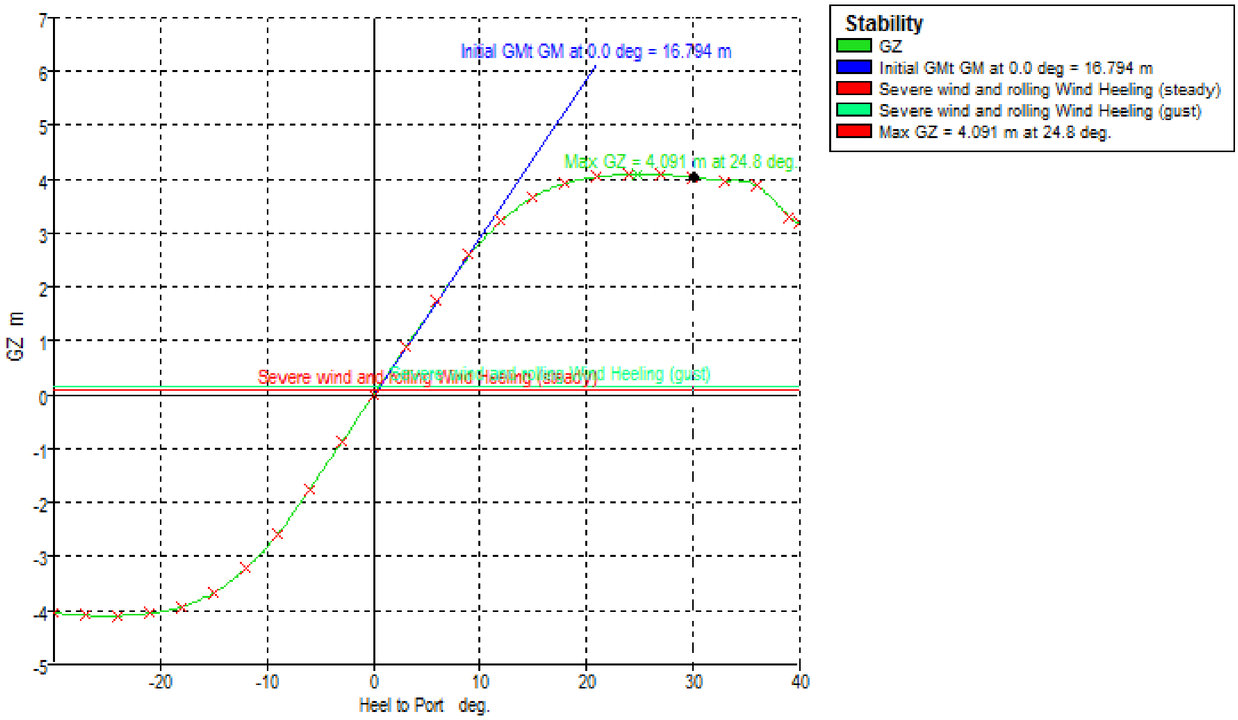

Figure 4-11: GZ curve of biggest gravity foundation transportation

Intact stability criteria shown in Table 4-9 are fulfilled.

Table 4-9: Biggest foundation transportation condition stability criteria

	Criteria	2008 IS Code	Requirement	Actual	Margin %	Status
1	Area under GZ curve from 0 to 40 bigger than	Part A, Ch.2.2	0.055	1.5885	2788.1	Pass
2	Area under GZ curve from 0 to 40 bigger than	Part A, Ch.2.2	0.09	2.2509	2400.95	Pass
3	Area under GZ curve from 30 to 40 bigger than	Part A, Ch.2.2	0.03	0.6624	2108.01	Pass
4	GZ at 30 or maximum rightening arm bigger than	Part A, Ch.2.2	0.02	4.02	20000	Pass
5	Maximum of GZ shall not be smaller than	Part B, Ch.2.4.5	15	24.8	65.46	Pass
6	Initial GM shall not be smaller than	Part A, Ch.2.2	0.15	16.794	11096	Pass
7	Angle of steady heel shall not be greater than	Part A, Ch.2.3	16	0.3	97.92	Pass
8	Angle of steady heel / Deck edge immersion angle shall not be greater than	Part A, Ch.2.3	80	10.2	87.25	Pass
9	Area *b* / Area *a* shall not be less than	Part A, Ch.2.3	100	168.99	68.99	Pass

4.4.4 Biggest Foundation Installment

Loading condition of biggest foundation installment consists of lightship, biggest gravity foundation, foundation handling unit and 7068.83 tons of ballast for fixing trim. Maximum draught 30 meters and displacement 11297.48 tons are reached in this condition.

The GZ curve for this loading condition is shown in Figure 4-12. Heeling between 0 to 40 degrees is considered to be enough since ventilation of the tanks (downflooding points) gets into water before 40 degrees.

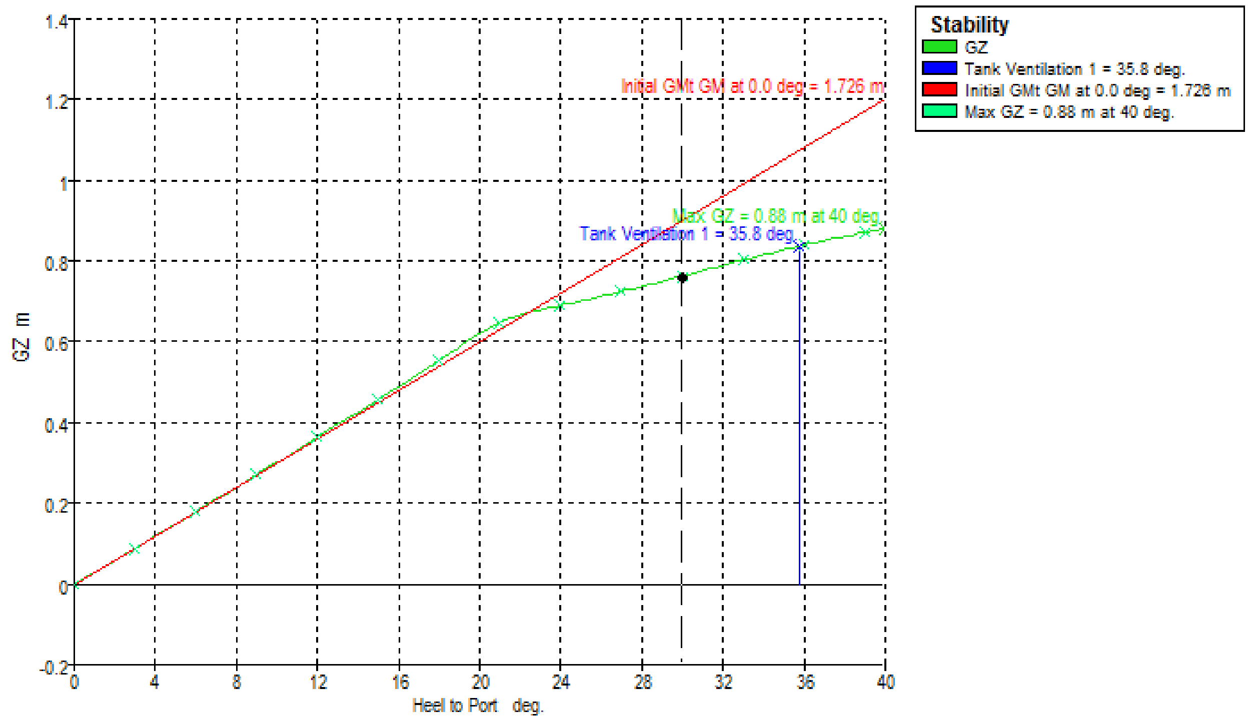

Figure 4-12 GZ curve of Biggest Foundation Installment Condition

Intact stability criteria shown in Table 4-10 are fulfilled.

Table 4-10: Biggest Foundation Installment Condition Stability Criteria

	Criteria	**Requirement**	**Actual**	**Margin %**	**Status**
1	GM at equilibrium shall not be less than	0.3	1.765	488.333	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.1	15	35.8	138.667	Pass
3	maximum righting arm shall occur at an angle of heel not less than	7	plus 40	plus 471	Pass

4.4.5 Biggest gravity foundation surfacing

Loading condition of biggest foundation installment is consisting of lightship, foundation handling unit and 8829.40 tons of ballast for fixing trim. In order to keep installation vessel at the same draught before releasing the foundation, 1760.57 tons of additional ballast it taken and then foundation is released.

The GZ curve for this loading condition is shown in Figure 4-13. Similar to previous loading case heeling between 0 to 40 degrees is considered to be enough since ventilation of the tanks (downflooding points) gets into water before 40 degrees.

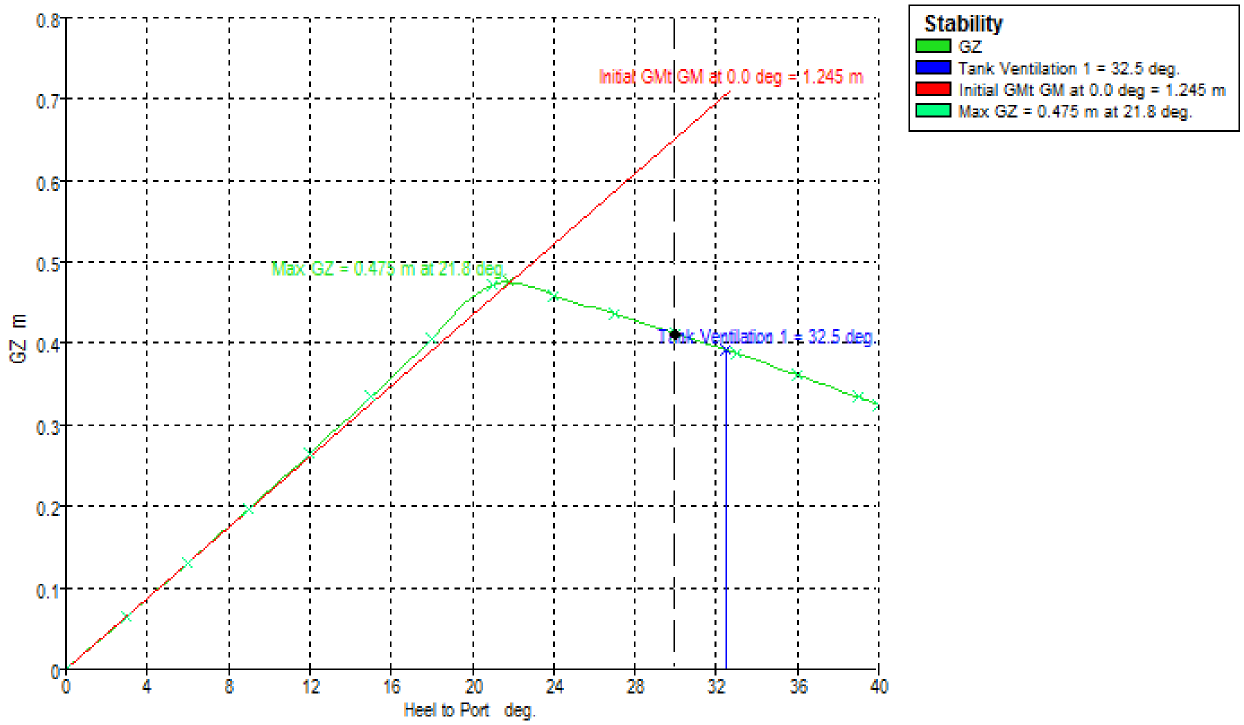

Figure 4-13 GZ curve of biggest gravity foundation surfacing

Intact stability criteria shown in Table 4-11 are fulfilled.

Table 4-11: Biggest Gravity Foundation Surfacing Condition Stability Criteria

	Criteria	Requirement	Actual	Margin %	Status
1	GM at equilibrium shall not be less than	0.3	1.245	315.000	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.1	15	32.5	116.667	Pass
3	maximum righting arm shall occur at an angle of heel not less than	7	21.8	211.429	Pass

4.5 Damage Stability

Damage case calculations are carried out in order to check DNV requirements. As indicated before, 12 cases are generated by damaging not only one or two compartments but bulkheads. According to damage length (5x5x0.76 meters) in some cases sometimes 4 even 5 compartments are damaged at the same time. Position of the vertical tanks is 0.85 meters inward of side of the vessel because of damage length stated before. In this way a damage from the side will be only limited on the barge part of the vessel. Damage stability is also the reason of this many number of ballast tanks and this high ventilation points. Unfortunately number of the tanks could not possible be less than current design because damages are causing unsymmetrical flooding almost all the cases. Unsymmetrical flooding cannot be handled easily neither at 5 MW wind turbine transportation nor semi submerged conditions. It is not possible to present all 60 cases in the report, therefore only the most critical damage case for each loading will be presented. However all results and damage cases can be seen at Appendix B.

Since there is no exact criteria in Maxsurf to check our criteria, one heeling arm which is equal to 0.05 meter constant, range above the heeling is checked. That’s the reason of presence of heeling arm at GZ curve plots.

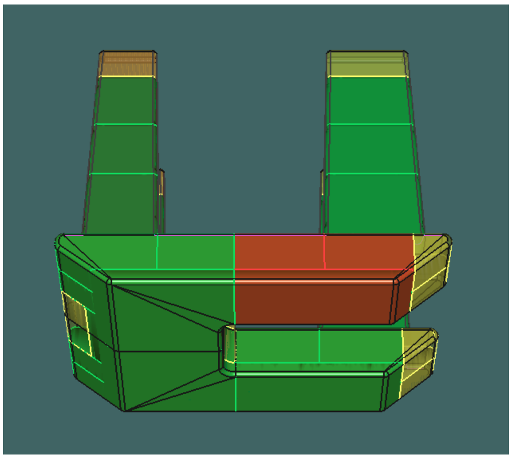

Figure 4-14: Damage Case 2

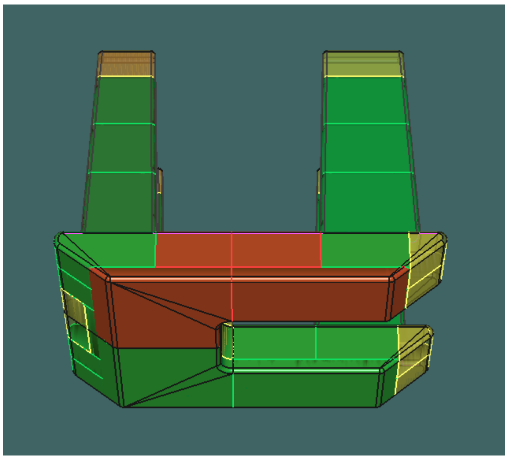

Figure 4-15: Damage Case 3

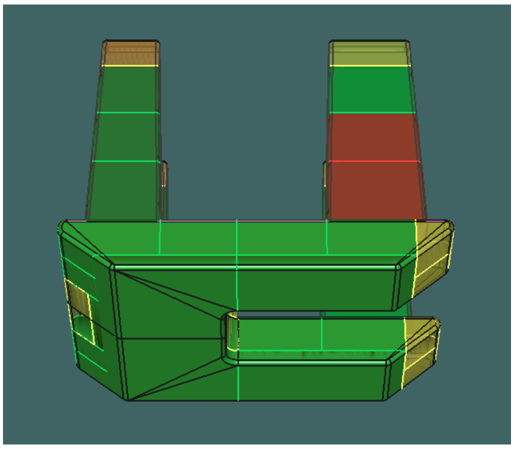

Figure 4-16: Damage Case 7

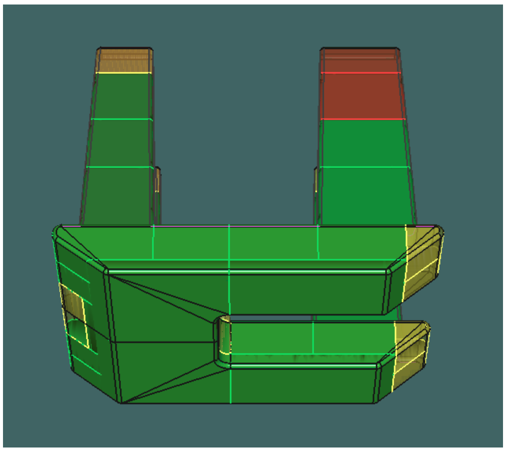

Figure 4-17: Damage Case 9

Table 4-12: Critical damage cases for each load case

	Loading Case	Critical Damage Case
1	Ballasted Condition	Damage Case 3
2	5 MW wind turbine transportation	Damage Case 2
3	Biggest gravity foundation transportation	Damage Case 3
4	Biggest gravity foundation installment	Damage Case 7
5	Biggest gravity foundation surfacing	Damage Case 9

4.5.1 Ballasted Condition

Damage cases for ballasted condition are having biggest margins in comparison to all others. Here damage case 3 will be presented because the smallest margins are observed in this case. 4 tanks are flooded as can be seen at Figure 4-15 in this condition because of the damage on the one transverse and one horizontal bulkhead.

Table 4-13: Ballasted Condition Damage Case 3 Stability Criterion

	Criteria	Requirement	Actual	Margin %	Status
1	angle of heel after flooding shall not exceed	15	6.4	57.59	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.05 meters	7	33.6	379.64	Pass

4.5.2 5 MW wind turbine transportation

Biggest wind turbine is the most critical transportation compared to foundation transportation and ballast leg. The reason is obviously extreme high center of gravity of wind turbine. As stated before unsymmetrical flooding is problematic because of the extremely unsymmetrical bow and stern. The GZ curve can be seen at Figure 4-18.

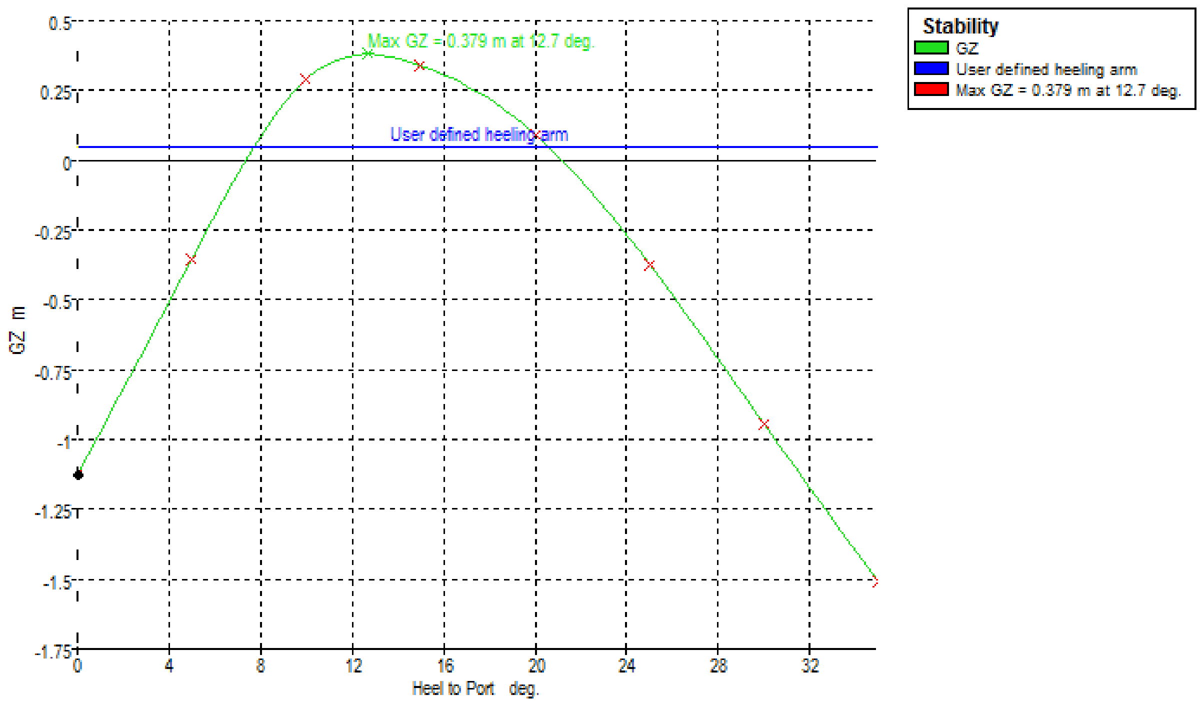

Figure 4-18: 5 MW wind turbine transportation, Damage Case 2 GZ Curve

The most critical case is damage case 2 because almost one half of the catamaran body is flooded. This means significant heel and trim, however installation vessel can handle all damage cases.

Table 4-14: 5 MW wind turbine transportation, Damage Case 2 Stability Criterias

	Criteria	Requirement	Actual	Margin %	Status
1	angle of heel after flooding shall not exceed	15	7.4	50.84	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.05 meters	7	12.8	82.26	Pass

4.5.3 Biggest gravity foundation transportation

Biggest foundation transportation is almost as safe as ballasted condition in sense of positive GZ range which can be seen at Figure 4-19. However angle of equilibrium in some cases, maybe dangerously high.

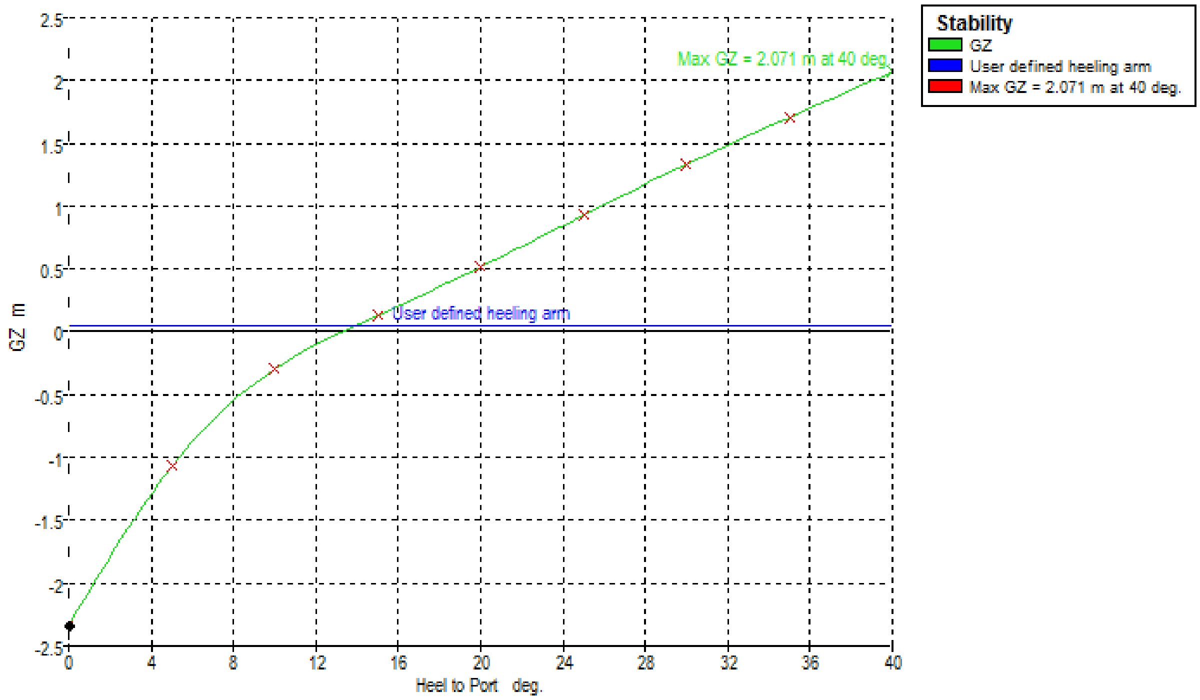

Figure 4-19: Biggest gravity foundation transportation, Damage Case 3 GZ Curve

All damage cases satisfied in this loading case. Margins for damage stability at damage case 3 can be seen at Table 4-15.

Table 4-15: Biggest gravity foundation transportation, Damage Case 3 Stability Criteria

	Criteria	Requirement	Actual	Margin %	Status
1	angle of heel after flooding shall not exceed	15	13.3	11.33	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.05 meters	7	26.0	272.00	Pass

4.5.4 Biggest gravity foundation installment

Semi-submerged conditions are observed to be more vulnerable to vertical tank (tower) damages. As can be seen from damaging cases bulkheads between two neighboring tanks or compartments are damaged at a time. GZ values which can be seen at Figure 4-20 are much smaller compared to ballasted condition or biggest foundation transport because of the much smaller surface area. After many different vertical tank designs, with current design buoyancy center is successfully kept above center of gravity of the installation vessel. This means small but continuous positive rightening lever during heel in many occasions.

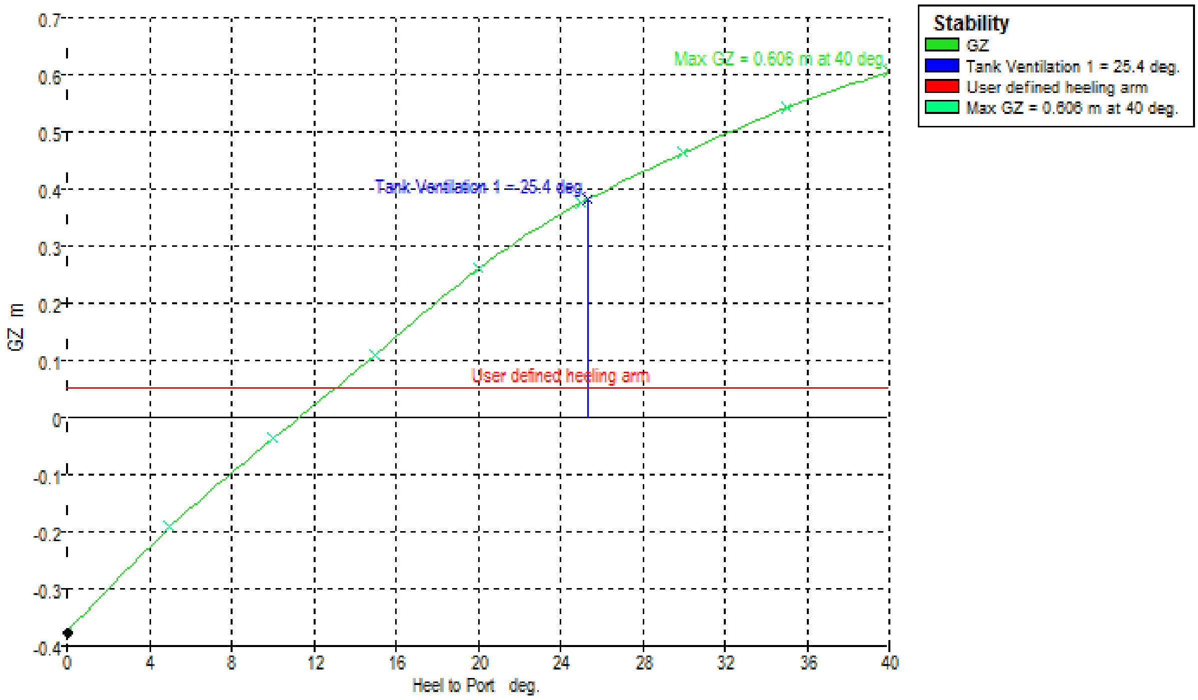

Figure 4-20: Biggest gravity foundation installment, Damage Case 7 GZ Curve

Damage stability margins can be seen at Table 4-16.

Table 4-16: Biggest gravity foundation installment, Damage Case 7

	Criteria	Requirement	Actual	Margin %	Status
1	angle of heel after flooding shall not exceed	15	11.3	24.74	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.05 meters	7	12.4	77.00	Pass

4.5.5 Biggest gravity foundation surfacing

This loading condition is observed as the most severe one among all cases. There are several critical cases but one of them will be presented here. GZ curve can be seen at Figure 4-21 and margins for damage stability criterions are at Table 4-17.

Table 4-17: Biggest gravity foundation surfacing Damage Case 9 GZ Curve

	Criteria	Requirement	Actual	Margin %	Status
1	angle of heel after flooding shall not exceed	15	6.9	53.86	Pass
2	positive range of the GZ curve shall be minimum within this range, in conjunction with a height of not less than 0.05 meters	7	8.8	26.19	Pass

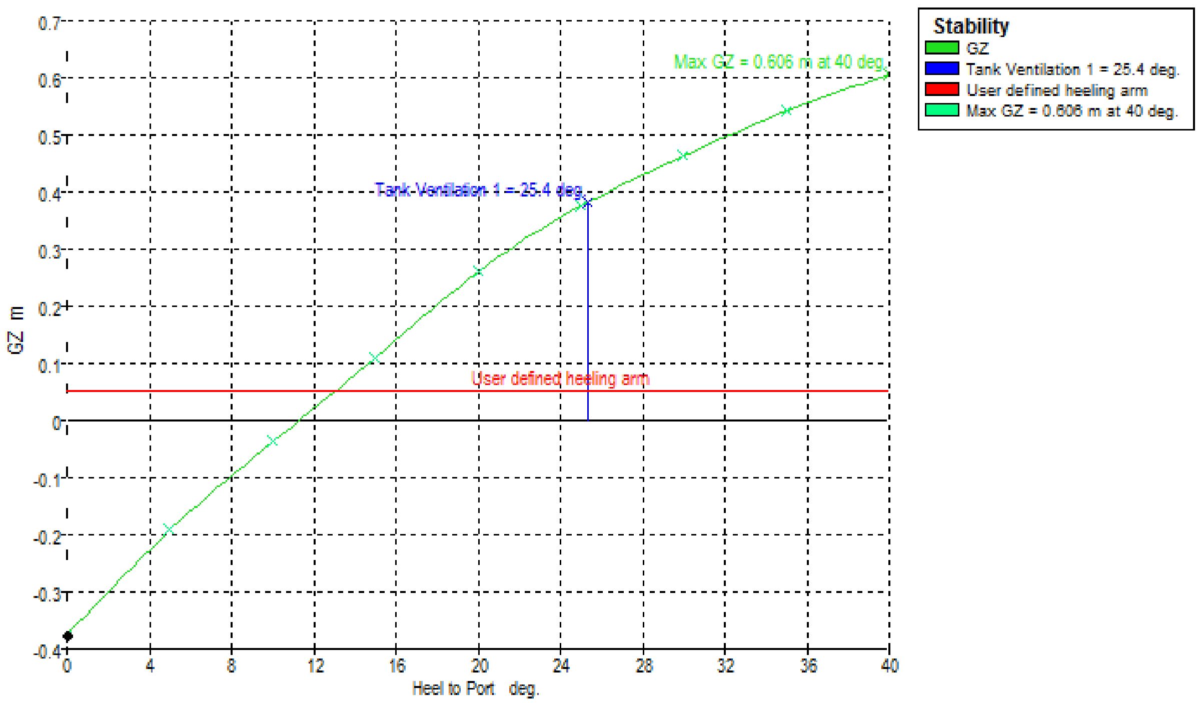

Figure 4-21: Biggest gravity foundation surfacing Damage Case 9 GZ Curve

4.6 Resistance

Resistance calculations are performed in two steps; the first one is a rough estimation based on the hull shape and the second one is a more detailed CFD simulation of different load cases.

4.6.1 Preliminary Estimation

Normal approximation procedures such as ITTC-78 (Larsson & Raven (2010)) usually work well for ship-shaped structures. However, the hull shape of the installation vessel cannot be regarded as ship-shaped, especially in submerged condition. Therefore this approximation has to be based on assumed drag coefficients for the structure and the resistance are then calculated based on the projected area, velocity and density of the fluid. This gives a preliminary approximation of the resistance and the required power.

In Figure 4-22 to Figure 4-24, the results from the preliminary resistance approximation are shown. These are a very rough estimation which considers the influence of wind-, wave- and viscous resistance for the hull, foundation and windmill. Comparison of Figure 4-22 and Figure 4-23 reveal that the resistance is larger for the windmill transit condition than for the foundation transit condition. This result can be questioned and motivates a further, more detailed investigation. This approximation does not consider the very large wake region behind the foundation when it is transported through the water, which should change the result.

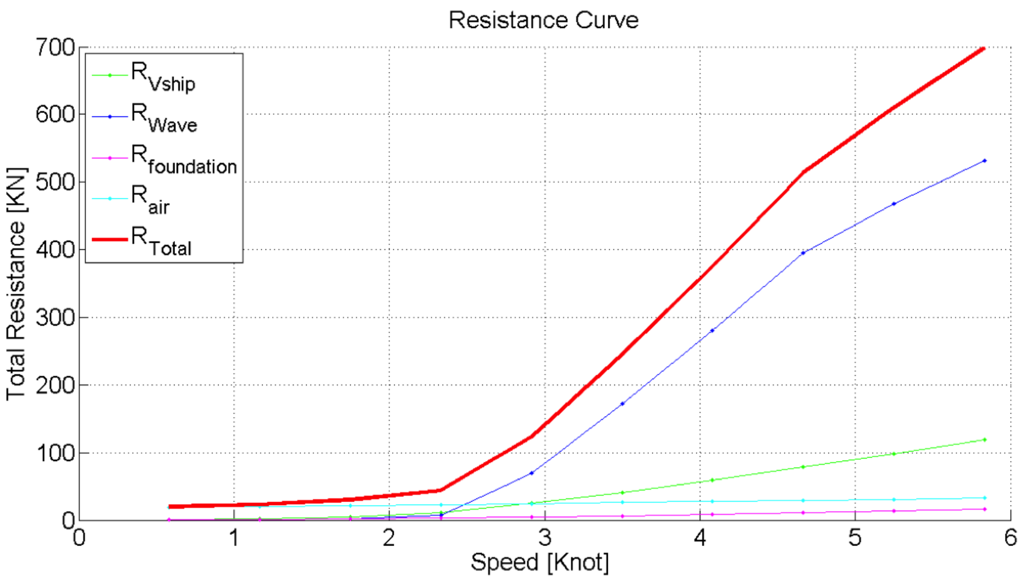

Figure 4-22: Resistance approximation for transit with foundation

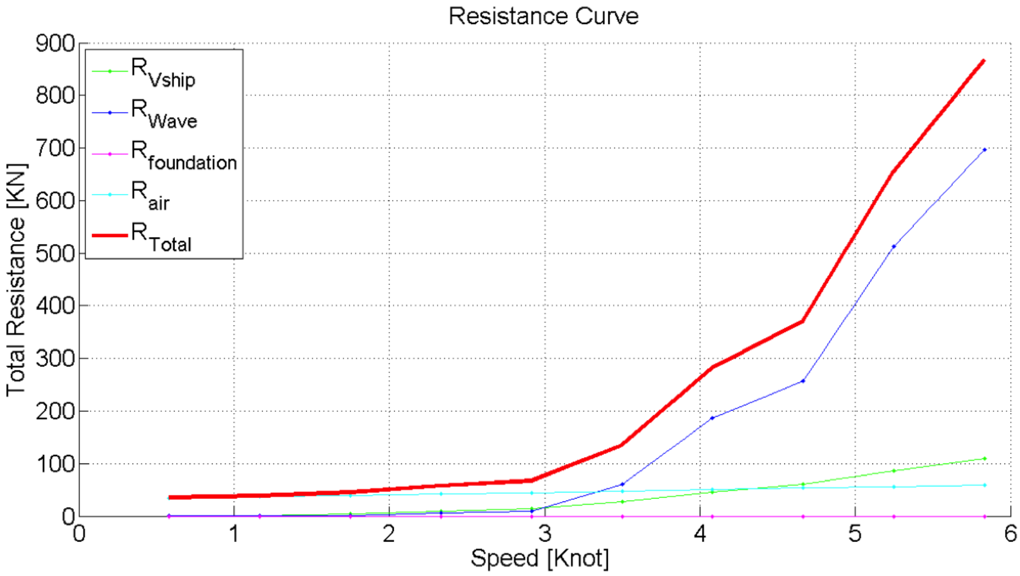

Figure 4-23: Resistance approximation for transit with windmill

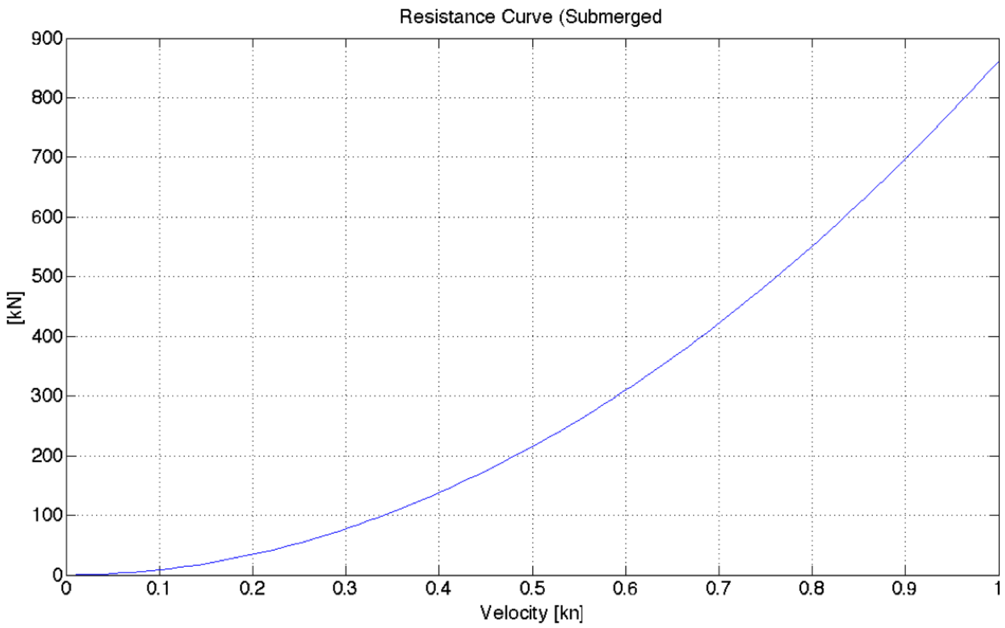

Figure 4-24: Resistance approximation for submerged condition

4.6.2 CFD

The CFD calculations are carried out using Ansys workbench as pre-processor (ANSYS inc., (2014)) and Fluent software as solver and post-processor (ANSYS inc., (2014)). In these simulations the drag coefficient for the part of the hull that is in water and the part that is in air are calculated separately and then the contributions are added together. According to Larsson & Raven (2010), wave resistance does not have a significant impact at these Froude numbers and is therefore neglected in the simulations. To compensate for the small wave resistance that however is present, the resulting drag force and effective power was increased by 10 %.

In the CFD simulations, a fluid domain is defined 100 m in front of the installation vessel, 100 m behind and 40 m to the side. Since the vessel is symmetric around its centreline only half of the body is modelled. To save computational time but still obtain good results, the mesh is refined on the surface of the hull and then it gradually becomes coarser further away from the hull. Mesh sizing on the hull is set to 0.23 m. All conditions are simulated with the same flow direction, which is the negative z-direction of Figure 4-31. The reason for this choice of flow directions is that the installation vessel is considered to be positioned with the bow against the wind during installation and during transit this gives the highest loading. To model the turbulence of the flow a realizable k-epsilon model is used. This is a commonly used model that works well on bluff bodies that do not give a lot of rotation in the turbulent boundary layer. Fluent is based on Reynolds Averaged Navier-Stokes equations (RANS) for the laminar flow computation and the realizable k-epsilon model is also based on RANS and the Reynolds shear stresses to model the turbulent regions and this makes it a good choice for this simulation (Davidson (2011)).

Three load conditions are considered in the CFD simulations; transit condition with the largest foundation, submerged condition with the largest foundation and the windmill installation operation. Resistance and effective power for each condition are presented in Figure 4-25 - Figure 4-30.

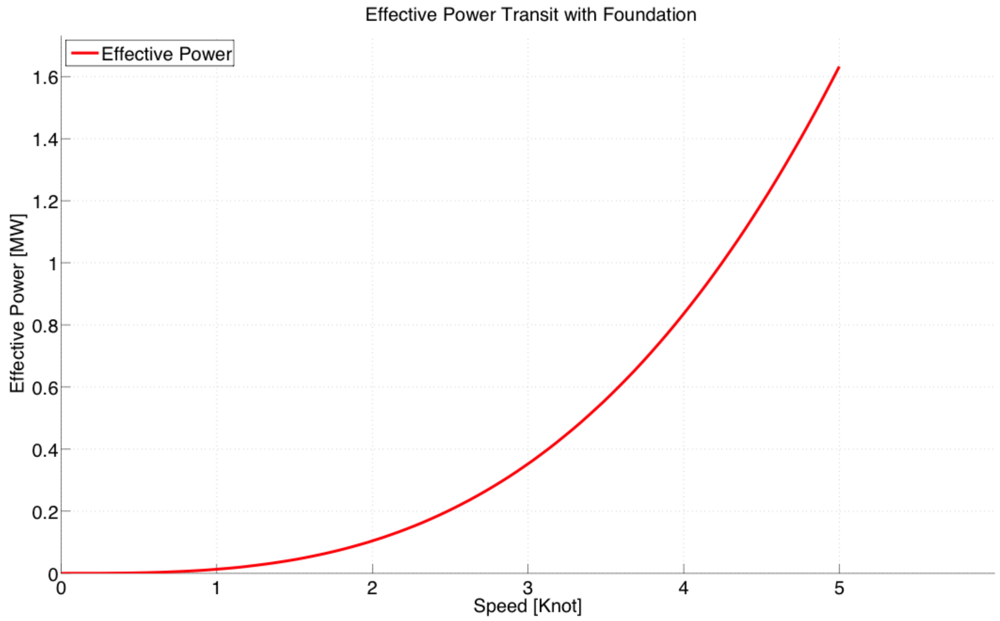

Figure 4-25: Effective power in transit with foundation

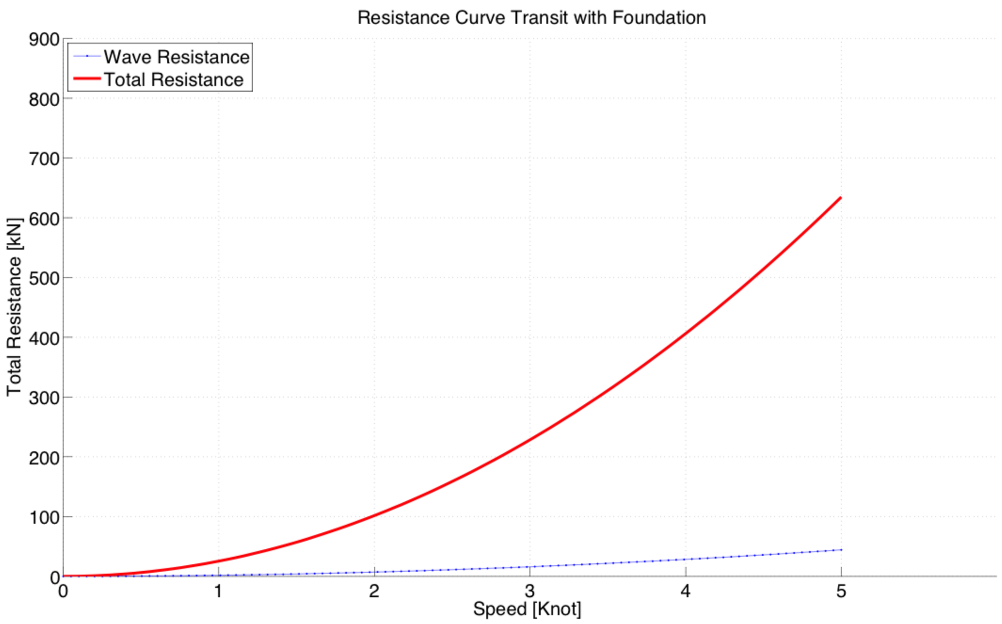

Figure 4-26: Resistance in transit with foundation

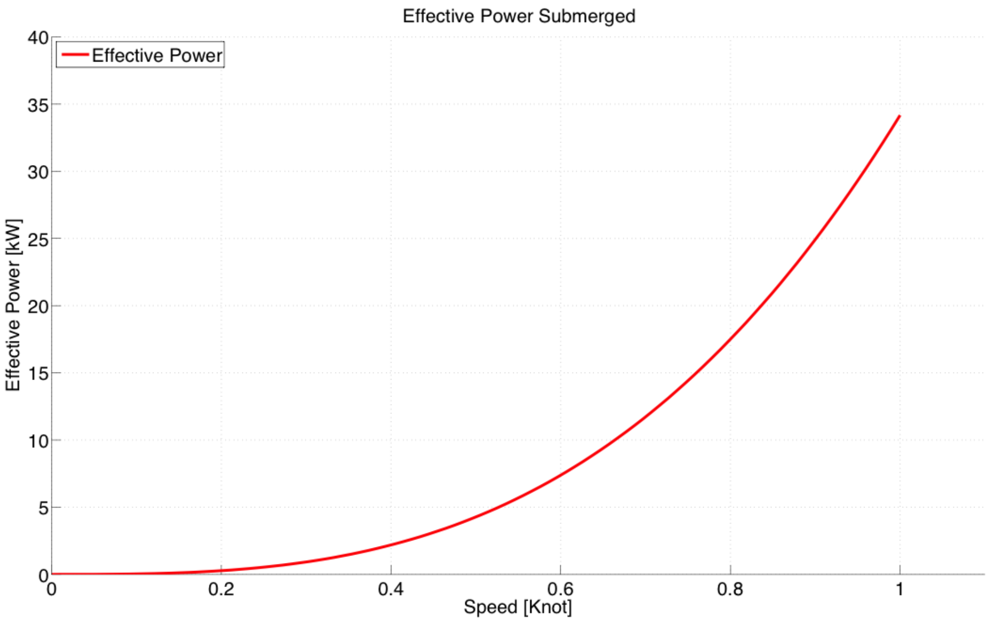

Figure 4-27: Effective power in submerged condition

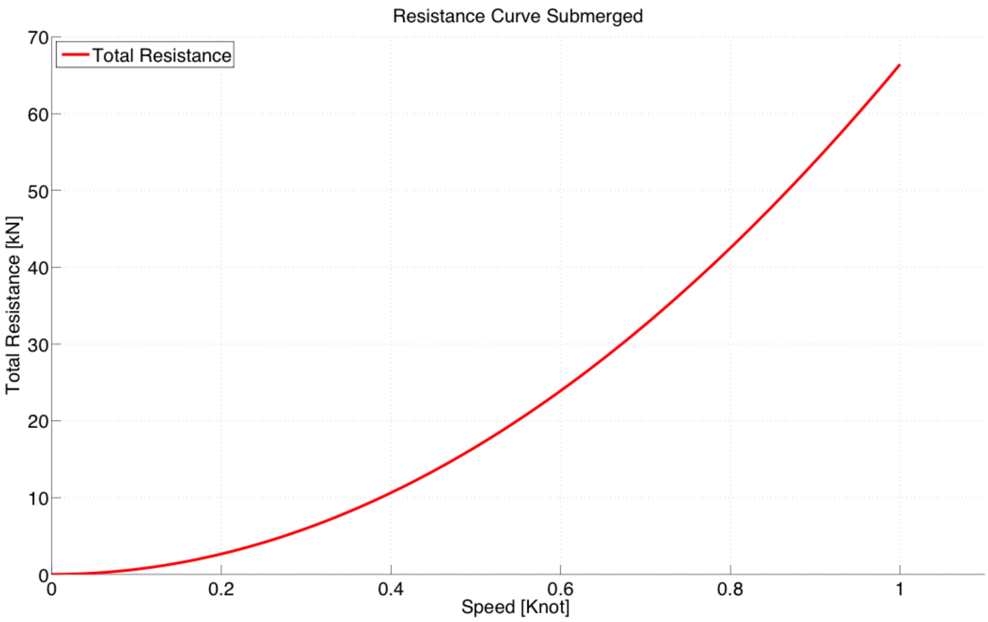

Figure 4-28: Resistance in submerged condition

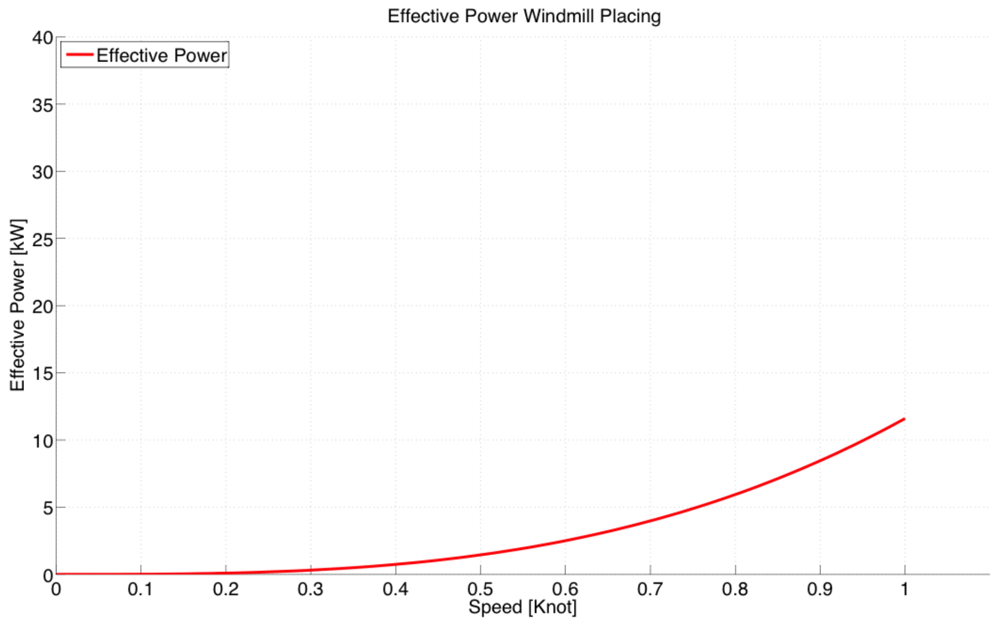

Figure 4-29: Effective power in windmill installation condition

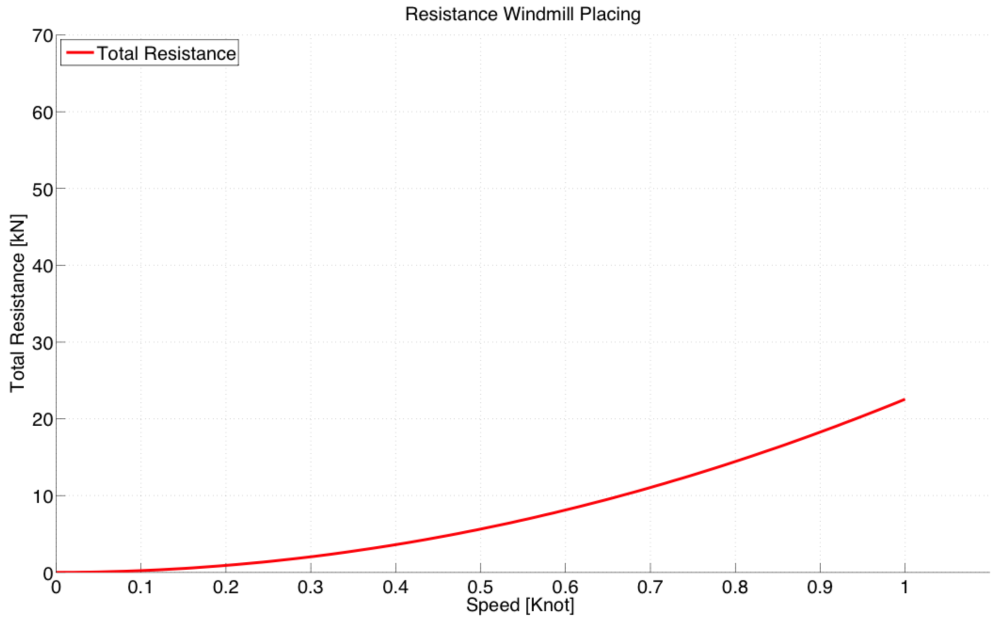

Figure 4-30 - Resistance in windmill installation condition

As can be seen in Figure 4-25, the resistance for transit with the largest foundation is approximately 700 [kN] which is close to the preliminary approximations as can be seen in Figure 4-22. The difference was expected to be larger and the conclusion of this is that the wave resistance in the CFD simulation probably is underestimated, especially at the higher Froude numbers.

During both the windmill installation and the submerged condition the installation vessel is supposed to keep a fixed position or doing some shorter moves to position the foundation or windmill. Hence, the resistance of interest is that induced by wind and current acting on the vessel. In Figure 4-27 and Figure 4-28 a current velocity of 0.5 knots is assumed and the installation vessel should be able to move at a speed of 0.5 knots, which makes the total relative velocity 1 knot. This is then the velocity used for the approximation, and the wind resistance is neglected for the submerged condition. For the windmill installation condition a similar assumption is used for the relative velocity between the water and the installation vessel but since the wind resistance is a significant part of this condition, a wind speed of 12 [m/s] is assumed. The submerged condition is the worst case when considering the positioning and since the support vessel not can be used for positioning of the installation vessel, the thrusters on the installation vessel must be dimensioned to deliver the effective power in Figure 4-27. Installation of the windmill is less demanding and the support vessel can be docked to the installation vessel to provide with extra manoeuvrability.

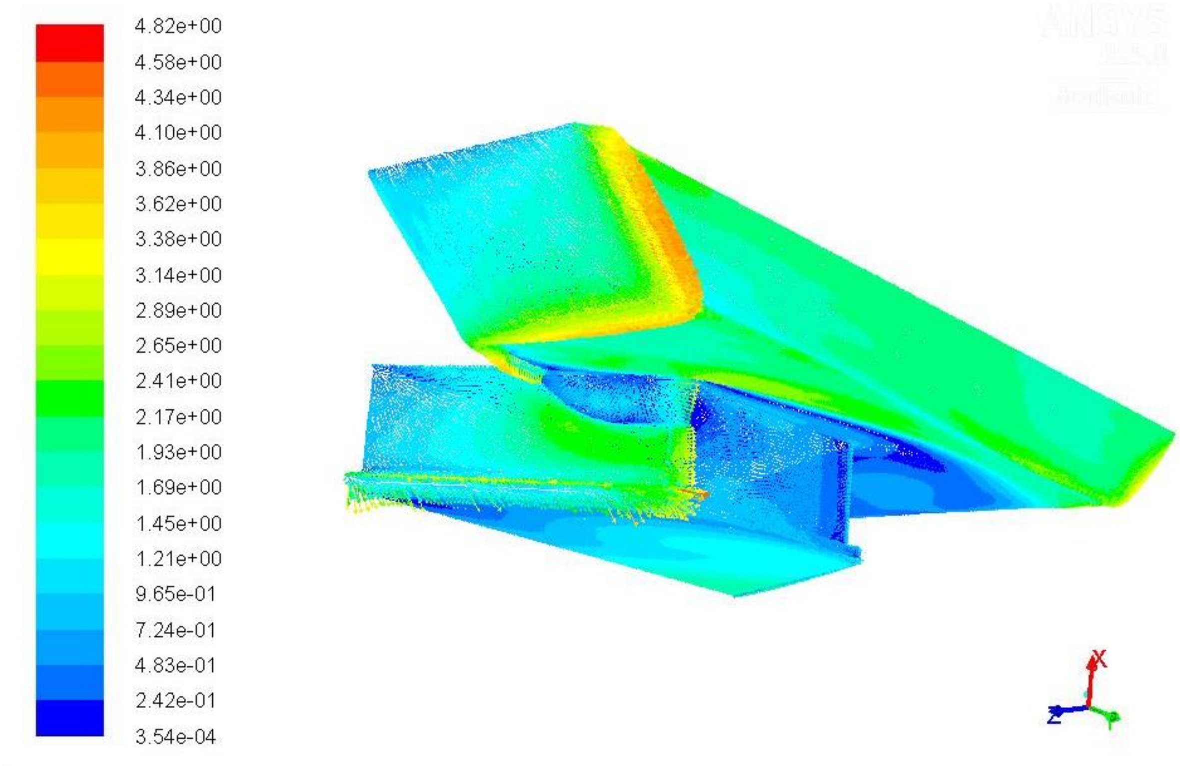

Figure 4-31: Velocity distribution

Figure 4-31 shows the velocity distribution over the hull and foundation in the worst loading case. As expected, the foundation creates a large re-circulatory region behind when transported through the water. The flow direction of this simulation is in the negative z-direction of Figure 4-31, which is the flow direction in transit. Figure 4-32 shows the same thing but from a view that more clearly show the wake area of the foundation, flow direction in this figure is from the lower edge of the figure and up.

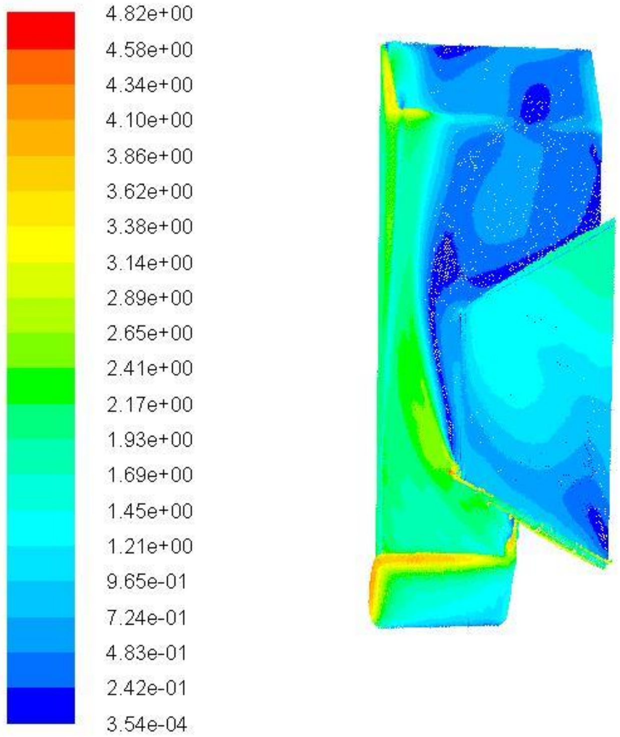

Figure 4-32: Velocity distribution over the bottom of the hull and foundation

In the submerged condition, the flow creates a wake between the vertical tanks, as can be seen in Figure 4-33. Flow direction is from the upper edge of the figure and down.

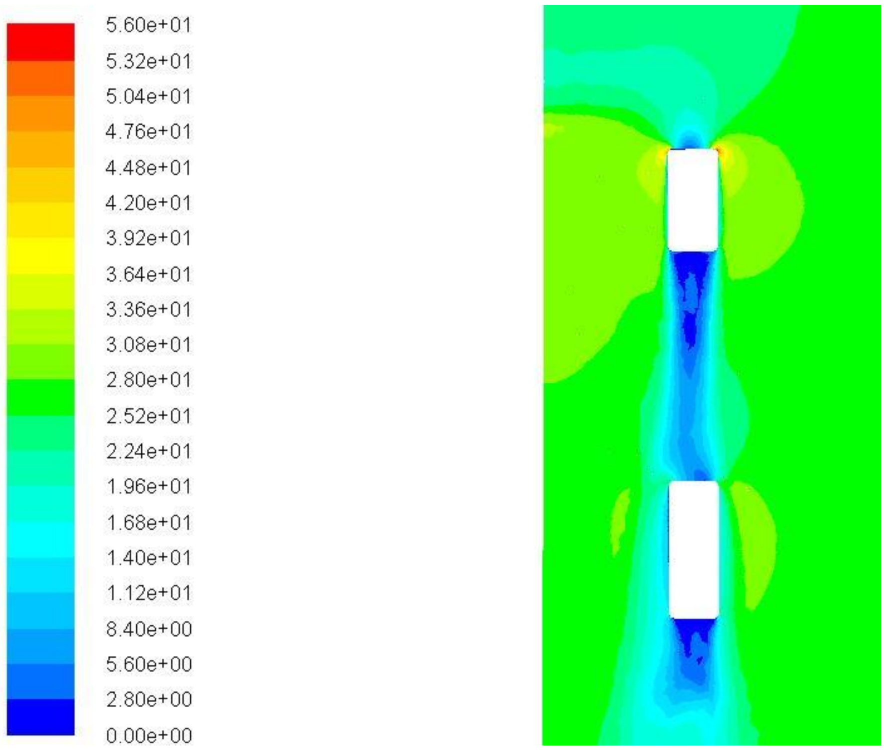

Figure 4-33: Velocity distribution around towers from above

4.7 Propeller design

Based on the resistance calculations in the above section, the propeller performance is also estimated. The evaluation was initially started using the openProp code and the inputs are optimized according to the problem statement. This evaluation resulted in a propeller with low efficiency and this code is incapable of evaluating many essential parameters. Therefore an alternative approach was chosen for the propeller evaluation by referring the $K_{T,}$ K_Q and efficiency curves assuming the propeller to be one of the Wageningen B-series propellers. These are calculated based on number of blades, blade area ratio, pitch to diameter ratio and the advance coefficient. The propeller is a 4 bladed propeller having a blade area ratio of 0.7 and pitch to diameter ratio ranging from 0.5 to 1.40. The open water characteristics like advance ratio, thrust coefficient and efficiency of the propeller for a propeller with a suitable diameter were estimated using the curves in (M.M.Bernitsas, 1981) . The other inputs like the resistance, wave making coefficient, Taylor wake fraction, thrust deduction at corresponding speed, form factor etc. are taken as inputs from the resistance calculations. First the hull efficiency is estimated based on the Taylor wake fraction. The thrust deduction and the thrust required on a single thruster are calculated. From the values of thrust coefficient and advance ratio the RPM, power delivered and torque required are estimated. The following plots show the various evaluation parameters which are used for the performance prediction.

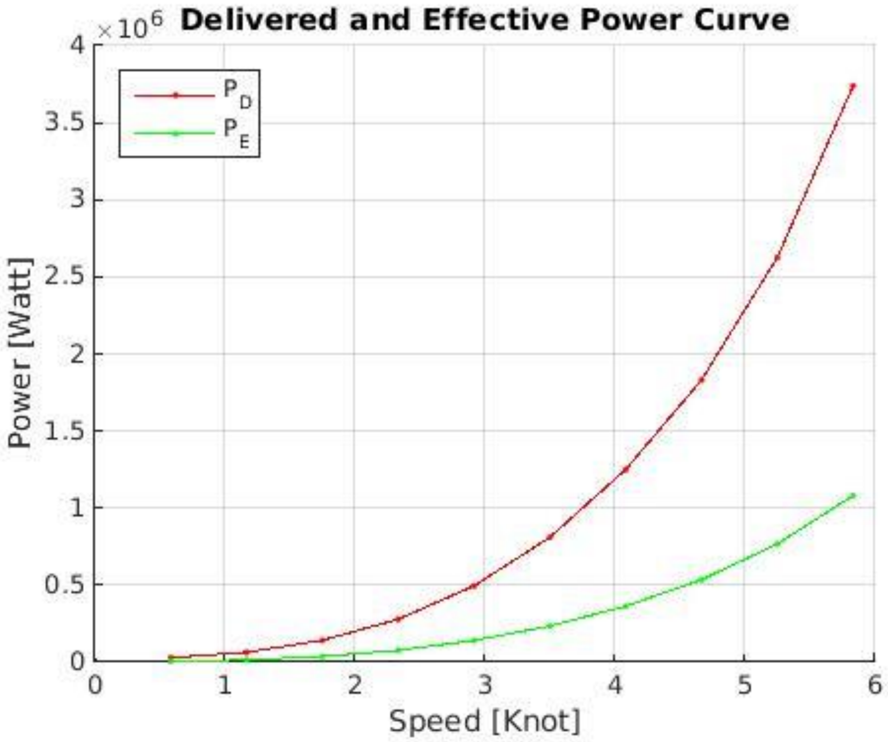

Figure 4-34: Delivered and Effective power curve for the loading case of windmill

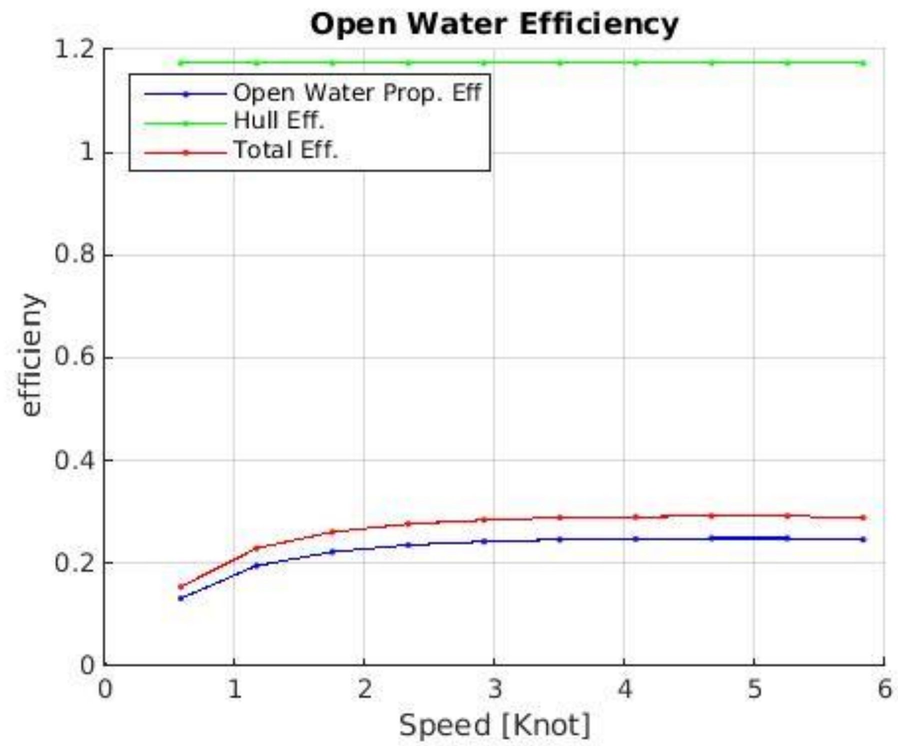

Figure 4-35: Curves of open water efficiency, hull efficiency and total efficiency with varying ship speed for the worst loading case

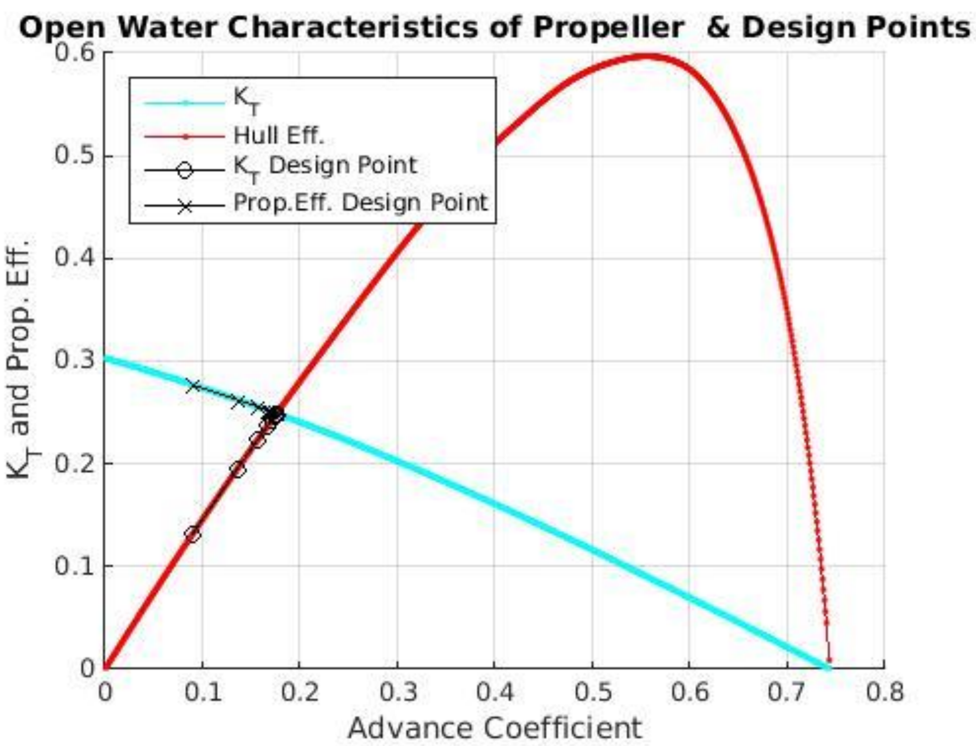

Figure 4-36: Open water characteristics versus the advance coefficient for the propeller with worst loading case

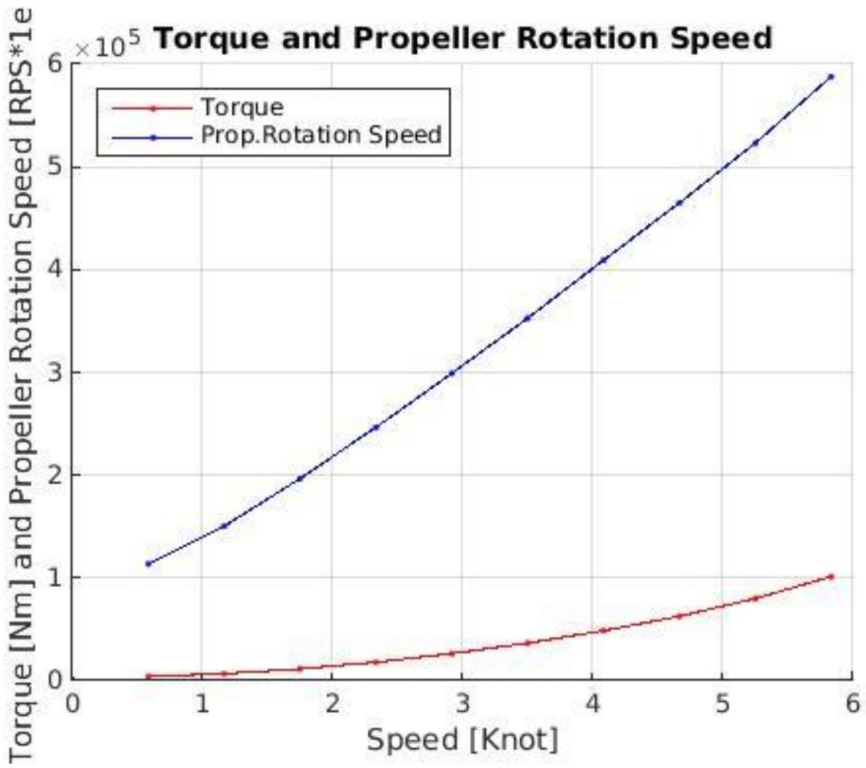

Figure 4-37: Torque and RPM of the propeller versus the ship speed for the worst loading case

The propeller for the support vessel in the case of an installation vessel carrying a windmill is selected for the evaluation. The propeller is having a pitch of 0.7 and should be able to deliver a required thrust from resistance calculations. Other parameters vary with the ship speed as shown in Figure 4-34, Figure 4-35 and Figure 4-37.

Cavitation

There are two main factors which govern the cavitation of the propeller:

1. Thrust factor

$$\tau = \frac{thrust}{(0.5 * \rho * V^2 * A_p)}$$

2. Cavitation number

$$\sigma = \frac{(P_0 - P_y)}{(0.5 * \rho * V^2)}$$

(Korkut, 2008)

Where:
ρ is the density of water

V is the total inflow velocity to the propeller

A_p is the projected area

P_0 is the static pressure at the propeller shaft line

P_v is the vapor pressure of water

The thrust factor and cavitation number are calculated for a velocity of 6 knots and at rotational speed of 5.2 rad/sec for the heavily loaded propeller having different suitable diameters. For a propeller diameter of 2.8 meter the thrust factor has a value of 0.25 and cavitation number of 1.02. These values are compared with the Burril's curve (Korkut, 2008) which predicted a cavitation risk of 2.5%. This is the minimum possible size one can consider for the design. However this prediction is for a non-ducted propeller. In case of a ducted propeller the projected area will be more which reduces the thrust factor and thereby reduces the risk of cavitation and furthermore it increases the efficiency of the propeller for a limiting diameter. Therefore the propeller with a duct is considered.

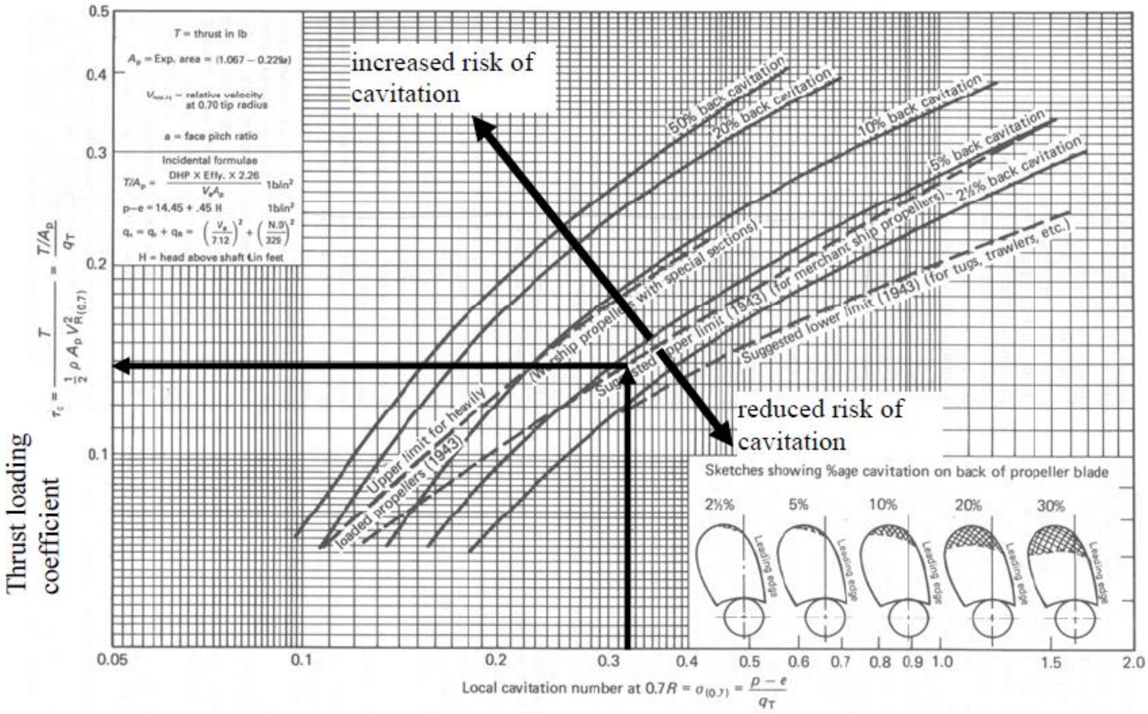

Figure 4-38: Burril's curve for estimating the risk of cavitation (Korkut, 2008)

4.8 Sea-keeping

Calculation for sea-keeping is performed by using Maxsurf Motion Advanced V8i. The software provides two method for calculations, panel and strip theory method. Since the panel method is proven to work poorly for this case the strip method was selected for this task. Also the hull itself provides some problems because of its non-conventional shape. In simplified terms the software has problems accepting the design.

The analysis is conducted by looking at the most motion sensitive case during operation. This case is stated to be when the windmill is carried and installed by the installation vessel. Also for this case the margins for positioning are the smallest.

The JONSWAP wave spectra are chosen for the analyses. This wave spectrum is originally based on North Sea conditions. This selection of wave spectrum can be considered as a conservative choice since the North Sea is known to be harsher than the bay of Hanö. Although the limiting operational sea state is set to sea state 3 analyses up to sea state 5 have been conducted.

In order to use the JONSWAP wave spectra for the analysis three parameters are needed. Significant wave height Tz, the modal wave period Tp and the peakedness factor γ. The first two parameters are gathered from the report "Theoretical Manual of Strip Theory Program "SEAWAY for Windows"". This report has been used before for similar task during the Naval Architecture master program at Chalmers University of Technology and the data is considered to fulfill the requirements of sea state parameters for this analysis. (Journée & Adegeest (2003)).

The peakedness factor γ is set to 3.3 which is the standard value. Finally the modal period Tp is calculated with following equation.

$$T_p = 1.199 * Tz$$

The parameters for the first 5 sea states are presented in Table 4-18.

Table 4-18: Sea State wave spectrum parameters

	Significant Wave Heigh,t Hs [m]	Average wave period, Tz	Modal wave period, Tp	Gamma, γ
Sea State 1	0.5	4,186 s	5.019 s	3.3
Sea State 2	0.65	4,546 s	5.450 s	3.3
Sea State 3	0.8	5,025 s	6.025 s	3.3
Sea State 4	1.1	5,505 s	6.601 s	3.3
Sea State 5	1.65	6,104 s	7.319 s	3.3

Influences from damping were not taken into account during the analysis. For this reason the result should be seen as conservative. Also the support vessel has not been taken into account when performing the analysis. The result is presented as polar plot for the heave, roll and pitch motions during sea state 3. Five different headings relative wave motions are equally distributed between 0° and 180° is taken into consideration during the analysis. In the polar plots 0° heading is at the upper part of the circle, 90° to the right and 180° at the bottom. The heading indicates from which direction the waves are acting on the vessel. The heading is also based on the direction of travel during transit. This means that the catamaran part is considered as bow (pointing at 0° of heading) and the solid end is considered as aft (pointing at 180° of heading). The numbers within the circle indicates the speed of the vessel in knots.

Figure 4-39 shows the heave motion (vertical movement) for the vessel. As can be seen the motion of heave is peaked for heading of 0° at a speed of 5 knots. However when installing the windmill the speed is reduced to zero, therefore the heave motions will be 0.078 m during sea state 3 and windmill installation. At worst the heavy motion is 0,247 m at 5 knots with 0° heading. The result also reveals that it is more beneficial to have the waves coming from behind in order to reduce heave motions.

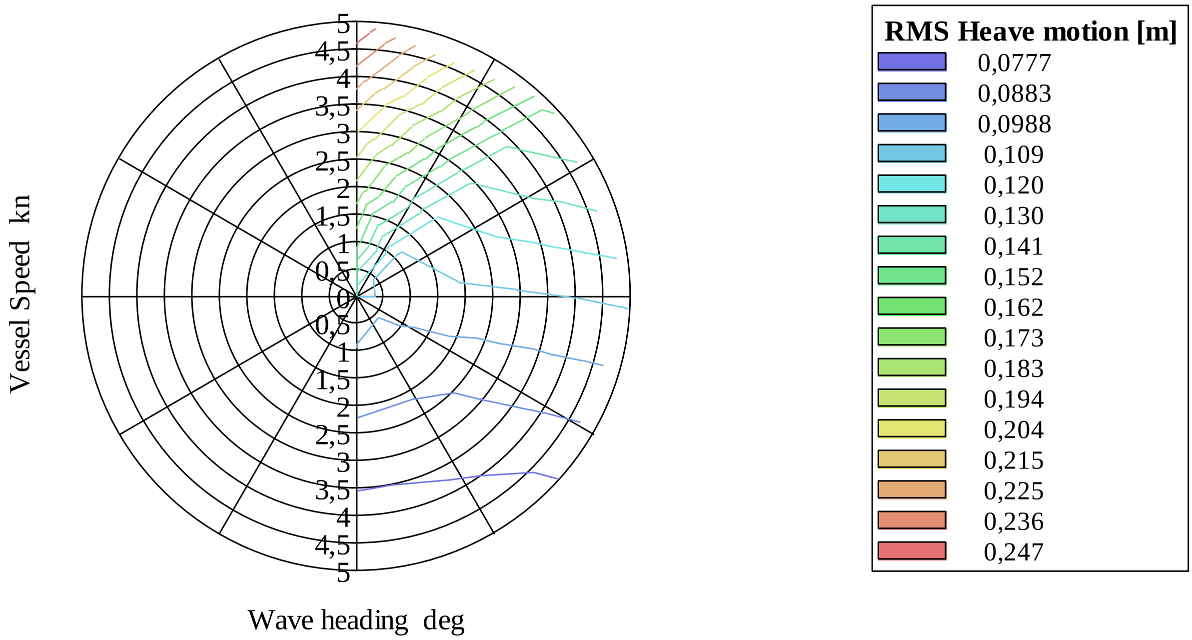

Figure 4-39: Heave motion polar plot for sea state 3

The second polar plot (Figure 4-40) regards the roll motion (turning motion around the vessel longitudinal axis) for the vessel with the biggest windmill attached. In sea state 3 with zero speed the rolling motions is 0.205°. For other speeds the motion tends to be small until the reach of 4 knots in speed for headings between 0° and 45°. Worst case is at 45° heading with 5 knots of speed.

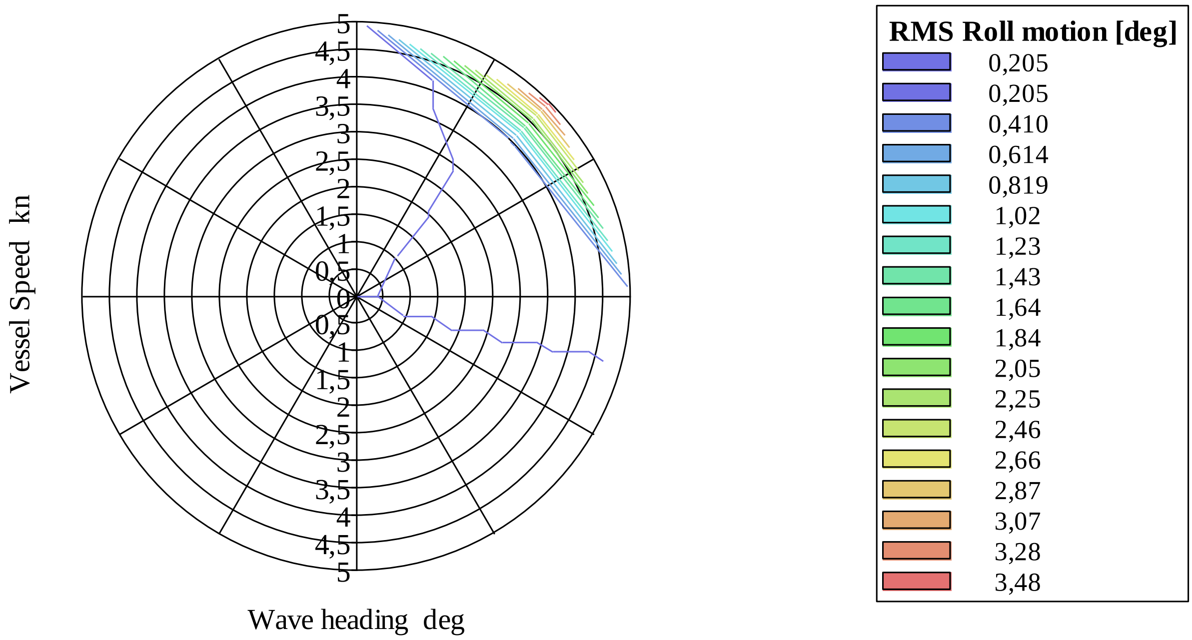

Figure 4-40: Roll motion polar plot for sea state 3

Third plot (Figure 4-41) is the polar plot of pitch motions (turning motion around the vessel transverse axis). Compared to roll motions the worst case here is when heading 0° at 5 knots. The pitch motion for sea state 3 with zero speed is 0.210° which is similar to the rolling motion. An interesting observation is that pitch motions are more sensitive against change in speed. Compared to roll motion, where the result is rather constant up to 4 knots, pitch motions gradually increases with speed for headings between 0° and 45°.

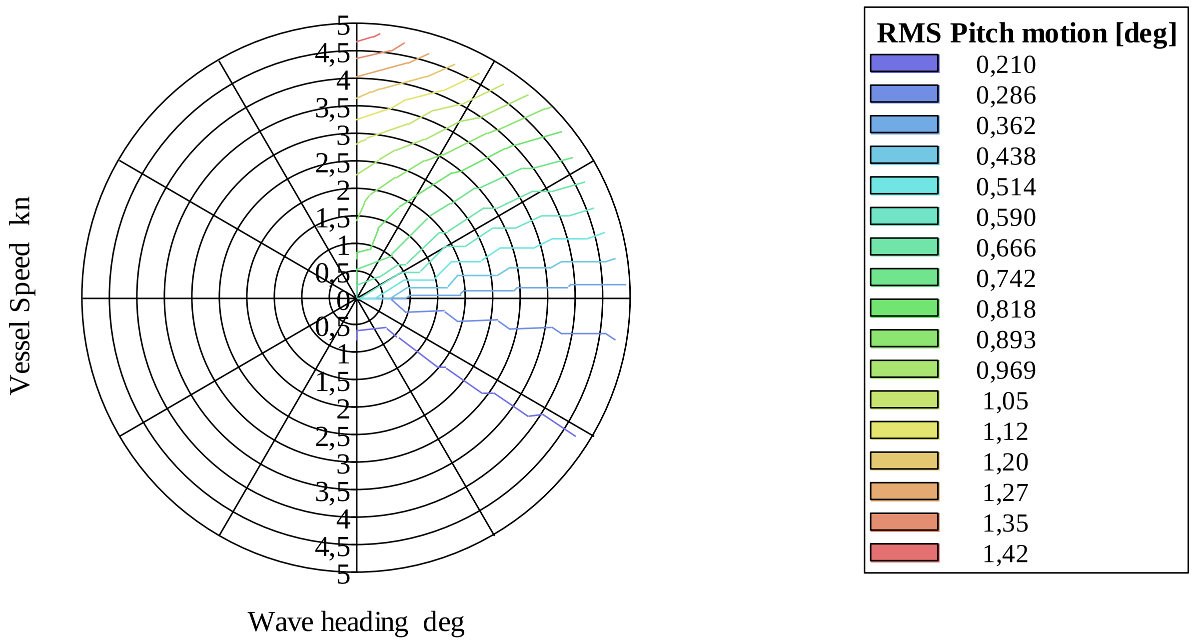

Figure 4-41: Pitch motion polar plot for sea state 3

4.9 Conclusion

The conclusion drawn based on hydromechanics work is presented in the following chapter.

4.9.1 Stability

As can be seen from the results of intact and damaged stability conditions for installation vessel, it can be concluded that all criterions are satisfied. Beam of the vessel plays a key role for the extensive intact stability. However beam kept as small as possible otherwise a large GM would have been a big issue for sea keeping characteristics. After many iterated setups for vertical tanks, the current design is determined for best intact and damage stability. It is important to enlighten extreme big and small margins in loading cases. As mentioned before breadth of the vessel is providing an enormous contribution to moment of inertia that causes the excessive GM. Large GM means a very steep GZ curve for starting heeling angles. However deck of the vessel gets into water around 15^0 of heel. Result of submerged deck is loss of waterplane area and stability. Therefore for example margins for maximum GZ shall occur after 15^0 and weather criteria for deck immersion has small margins. On the other hand area under GZ curve is much greater than requirements. That’s because of great breath of the vessel as stated. One can be suspicious for some loading cases vessel is far more stable than required. That is in fact true but installation vessel is supposed to handle all loading cases including biggest wind turbine transportation which was for more challenging case.

4.9.2 Resistance

As can be seen in Figure 4-25 toFigure 4-30, the most demanding load condition is the transit with the largest foundation (Figure 4-25). In this case the support vessel deliver the majority of the propulsive power and therefore the thrusters on the installation vessel can be dimensioned to cope with the worst load condition when installing. Figure 4-27 show the worst load condition when the support vessel is not connected to the installation vessel. This condition demands an effective power of 35 [kW] and this is what the thrusters should be dimensioned for.

4.9.3 Sea-keeping

As can be seen from the result of the analysis the motion of the installation vessel seems to be rather small. Since no measurable criteria values have been given from the client it is hard to say whether these movements are satisfyingly small or not. Motions affecting the sensitive windmill tower have not been investigated. The reason for this is that the windmill manufacturer is highly secretive regarding their products. Therefore there is no sufficient data for comparing. The results are also very conservative since the analysis did not include the support vessel. When this is connected to the installations vessel the motions should be further suppressed and smaller.

5 Machinery

As Optimus Pråm is a semisubmersible barge it has a machinery arrangement that is divided between an installation vessel and a supply vessel. This chapter describes the arrangement and presents different concepts and ideas, as well as their advantages and disadvantages. The installation vessel requires a large pump capacity to be able to ballast and de-ballast quickly, some propulsion in order to install the foundation and windmills accurately and power supply to manage all installation operations described in this section.

Due to time limitation, the main focus is to identify the required engine size on the support vessel and the machinery system on installation vessel, i.e. pump room arrangement, thrusters, and frequency converters.

5.1 Stakeholders and requirements

The following bullets list the requirements from the stakeholder that have been accounted for when arranging the machinery systems on the installation and supply vessel respectively;

- *The vessel should be optimized for operation in Baltic Sea conditions*. In the area where the wind farm is located, a current of up to 0.5 knots is expected and designed for. The propulsion system on the installation vessel is required to produce enough thrust to stay in position during the installation operations.
- *Health, safety and environment (HSE) is of highest priority.* To meet the emission regulations in the Baltic Sea, both vessels shall be designed to be environmentally friendly.
- *The acceptable tolerance in installation of foundations is 2m offset.* The thrusters will have to produce enough thrust to stay in position to meet this requirement when installing the foundations.

5.2 Concept evaluation of power systems

The installation vessel has to be kept in position during the installation phase of foundations and windmills. One of the requirements from the client is a maximum offset of 2 meters during installation of foundations. However, for the windmills the requirements are much higher. During the transit to site, the support vessel will require power to push the installation vessel. The power to both vessels is delivered from the support vessel and this section describes the concepts that are evaluated to meet these requirements and ensure a satisfactory result, along with conclusions of a final decision.

5.2.1 Several supply vessels

More than one tug can be used to keep the installation vessel in position during the installation operation by towing it. This concept would not require any propulsion on the installation vessel itself, which would reduce the complexity of the arrangement. The pumps would still need a source of power but the main issue with this arrangement is positioning and cost.

5.2.2 Mechanical arrangement

A mechanical arrangement on the installation vessel, i.e. having an engine on board, requires shafts, fuel tanks, and more advanced ventilation systems. The installation vessel would keep itself in position during the installation and thus be self-propelled. A mechanical arrangement on the installation vessel to supply power for the ballast pumps and propulsion systems would take too much space from the ballast tanks and is not deemed fit as a solution.

5.2.3 Electrical arrangement

The installation vessel is self-propelled during the installation phase using electrical motors and thrusters. A power cable is attached to a support vessel that delivers all the power needed for the installation vessel. Using electrical motors on the installation vessel saves a lot of space and delivers short response time when manoeuvring.

5.2.4 Conclusion

Having at least three supply vessels operating simultaneously is very expensive since this involves three crews and three vessels binding capital and consuming fuel. In addition, the positioning is not satisfyingly accurate.

The electrical arrangement is most fit to cover all needs in a suitable manner and is therefore chosen for further development. Since there is limited space and a thruster with attached electrical motor can be fitted far away from the power source, the arrangement can be divided and placed to meet the limitations. There is also a large non-propulsion electrical load needed on the installation vessel, e.g. pumps that are large electrical consumers, an electrical arrangement is very favourable.

5.3 Concept evaluation of propulsion systems

A number of propulsion systems were considered as possible solutions for the installation vessel and after some discussions three were investigated further. The propulsion systems subjected to further investigation are presented below, along with a weighted evaluation matrix.

5.3.1 Tunnel thrusters

Tunnel thrusters are built inside the hull and are therefore very well protected in case of grounding or impact with the foundation. Two tunnel thrusters would be installed in each stern corner, and one at the bow. Since the tunnel thrusters at stern are angled they have to be over dimensioned in order to produce the same forward thrust as a straight thruster. The angled thrust can be thought of as two vectors, one going forward and one going sideways. The size of the vector producing forward thrust is depending on the angle of the tunnel thruster.

5.3.2 Voith-Schneider

Voith-Schneider propellers deliver very high manoeuvrability, propulsion power and are used more on tugboats and fireboats by day. As the Voith-Schneider propellers require a big draft they might be prone to grounding or damage during foundation installation.

5.3.3 Azimuth thrusters

Azimuth thrusters deliver great manoeuvrability and dynamic positioning. As the installation vessel is slanting at stern, the propellers will be dimensioned to never be below the keel, rendering an acceptable draft for the system. Two azimuth thrusters in the aft of the installation vessel and a retractable thruster at the bow will ensure good manoeuvrability during operations in the anticipated conditions. The azimuth thrusters are able to rotate 360 degrees in the horizontal plane which provide the installation vessel with high manoeuvrability i.e. is able to generate thrust in any desired direction.

5.3.4 Conclusion

Tunnel thrusters would give a high safety as they are well protected within the hull but if one is lost during operation the propulsion required from the remaining thrusters is too big. The poor manoeuvrability of the tunnel thrusters is also an issue. The great draft of the Voith-Schneider thrusters cannot be disregarded, it is crucial that the propulsion system is not damaged during operation.

Table 5-1 shows the weighted evaluation matrix used to come to a final decision and it can be seen that the azimuth thrusters have the best result and will be the chosen propulsion system for the installation vessel. In the following section a further explanation and also selection of azimuth thrusters are presented. The alternatives show different attributes of the propulsion system and the weight represents the importance of said alternative. The weight ranges from 1 to 5, where 1 represents the lowest and 5 highest level of importance. The total value of each propulsion system is then compared to the total value of the ideal solution, and the closest value represents the most desirable solution.

Table 5-1: Weighted evaluation matrix of propulsion systems

Alternative		**Ideal**		**Tunnel thrusters**		**Azimuth thrusters**		**Voith-Schneider**	
	Weight:	w	t	w	t	w	t	w	t
Redundancy	4	5	20	2	8	5	20	2	8
Complexity	4	5	20	4	16	3	12	2	8
Manoeuvrability	5	5	25	3	15	5	25	5	25
Reliability	5	5	25	4	20	4	20	3	15
	Total value		90		59		77		56

5.4 Propulsor

In this section, a suggestion for design of the propulsion arrangement for the installation and support vessel is described. Suggestions for drivers for the propulsors are presented together with additional systems such as frequency converter. The information and specification that is presented is used for further design of the system such as the electrical balance and power requirements.

5.4.1 Installation Vessel

The main propulsors for the installation vessel will consist of two azimuth thrusters from Rolls-Royce and are together able to produce approximately 10 tons in bollard pull condition to meet the requirements, see Chapter 4.6. The value for bollard pull is based on preliminary information and is only used for rough estimation regarding power requirements. The thrusters are located in the area below propulsion room 1 and 2 respectively, see drawing MDP2014_1_101_01.

In addition, a retractable azimuth thruster from Rolls-Royce is added for improved manoeuvrability and redundancy. It will be able to produce a thrust of 7.4 tons in bollard pull conditions. The thruster is located in the area below propulsion room 3, see drawing MDP2014_1_101_01.

Figure 5-1 and Table 5-2 present the specification and principle sketch of the selected propulsors.

Table 5-2: Detailed information on thrusters used on the installation vessel

Installation Vessel	Retractable thruster	Azimuth thruster
Manufacturer	Rolls-Royce	Rolls-Royce
Type	UL601	US55P4
Number Installed	1	2
Power Input [kW]	400	315
Input Speed [rpm]	1500	1500
Diameter [mm]	1300	1050
BP [tonnes/each]	7.4	5

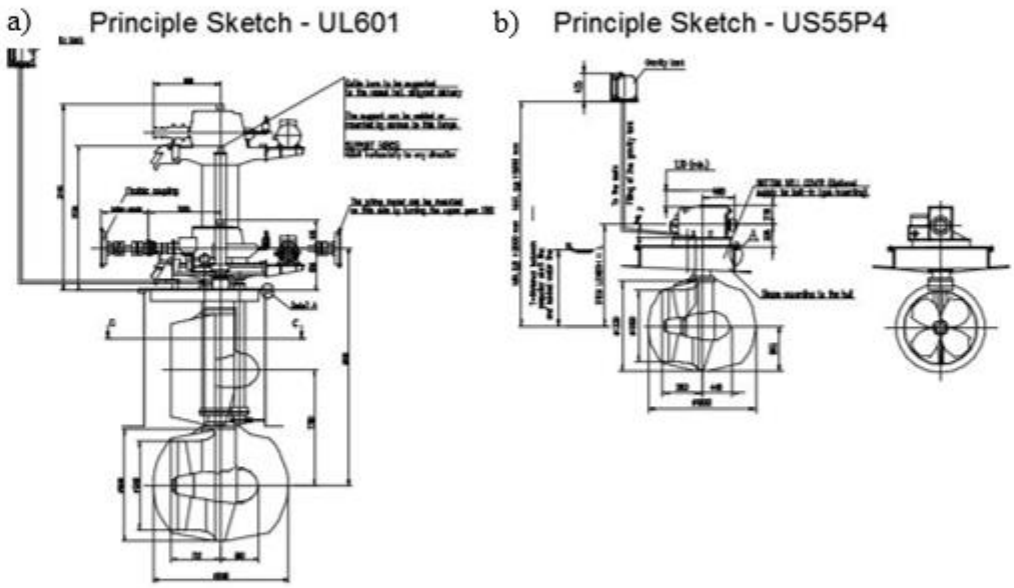

Figure 5-1 a) Principle sketch of retractable azimuth thruster and b) azimuth thruster on the installation vessel respectively (courtesy of Rolls-Royce)

5.4.2 Support Vessel

The design of the propulsion arrangement for the support vessel is driven by the requirement regarding thrust in transit operations, which considers pushing the installation vessel to site. To be able to meet the requirement, two ducted azimuth thrusters are used. Delivered power of 1600kW per driver to each propeller will result in a propeller diameter of 2.8 meters according to calculations, see Chapter 4.7. A principle sketch can be seen in Figure 5-2.

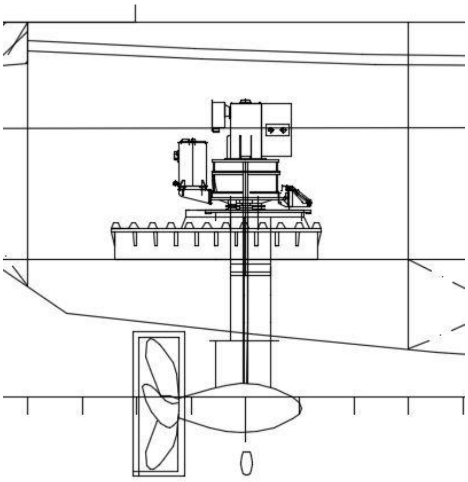

Figure 5-2: Principle sketch of ducted azimuth thruster with electrical motor

5.4.3 Electrical motors

A diesel-electric system will be used for the installation and support vessel. The drivers for the two azimuth thrusters and the retractable thruster of the installation vessel consist of three electrical induction motors from ABB. For the two azimuth thrusters, the output of the electrical motor is 315kW and for the retractable thruster the electrical motor has an output of 400kW. For more details see Figure 5-3 and Table 5-3. The electrical motors for all thrusters on the installation vessel are located vertical of the propulsors as an L-configuration.

For the support vessel, the two azimuth thrusters will be provided by mechanical power through two electrical induction motors from ABB. The electrical motors will be located vertically on top of the propulsors, in the same manner as for the installation vessel as can be seen in Figure 5-2. Dimensions and principle sketches of the electrical motors are presented in Figure 5-3 and Table 5-3.

Table 5-3: Specification of electrical motors

Electrical Motor	Manufacturer	Number	Type	Power Input [kW]	Input Speed [rpm]
Support Vessel	ABB	2	AMI 450L6L	1600	1000
Installation Vessel	ABB	2	HXR 355LC4	315	1500
Installation Vessel	ABB	1	HXR 400LC4	400	1500

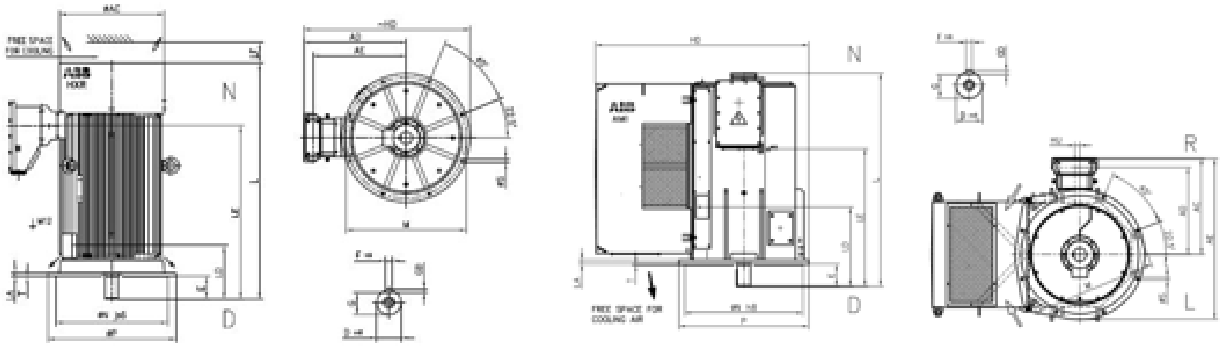

Figure 5-3: Principal sketch of electrical motors (ABB 2011)

5.4.4 Frequency converter

To be able to change the electrical motor speed to any desired condition, a frequency converter is needed. The function of the frequency converter is to adjust the frequency and voltage from the electric power supply to conditions required from the electrical motor at a certain speed. The frequency converters are chosen to fit the power size of each motor respectively.

For the support vessel, two frequency converters support both azimuth thrusters, and they will be located in the rudder propeller room, see drawing MDP2014_1_101_01. For the installation vessel, three frequency converters are added. Two of the converters are located in battery storage room 1 and 2 due to space limitations, and serve the azimuth thrusters. For protection, the converters are located in enclosed space. The third converter is located in the pump room and serves the retractable thruster, see drawing MDP2014_1_101_01.

The reason why the converters are installed on the installation vessel is due to the increased flexibility that this arrangement offers. If the support vessel has a breakdown, it could be replaced by an existing vessel on the market that complies with the requirements needed or by retrofitting a vessel from the second hand market.

5.5 Electrical Balance

To be able to design the power plant in the most efficient configuration on the support vessel, the power requirement for the total system with installation vessel included needs to be investigated. The investigation considers all different operational modes for both vessels. The operational modes under consideration are:

- Port loading – Loading of foundation and windmill in port
- Transit to site – Pushing operation to site
- Positioning – Positioning at site
- Installation of windmill
- Installation of foundation
- Transit to port – Pushing operation to port
- Harbour mode – The installation and support vessel laying in port, no operation
- Emergency – Only considering power supply to support vessel and equipment that is essential for safety

The electrical balance for this project is based on both calculations and assumptions. The assumptions are mainly concerning the support vessel where similar ships are studied. It is primarily tugboats that are investigated further when considering the support vessel, where a similar electrical balance (Ma, 2012) acts as a reference for the project. The electrical balance is used and additional equipment is added for this specific project requirements. The electrical balance serves as a rough overview of consumers that are included for a typical tugboat in the range needed for this project. For more details about the electrical balance, see Appendix C.

The estimation of power needed at certain operational modes is done by investigating when a specific consumer is required for the system. The values presented in Table 5-4 are calculated in a conservative manner which means that load factor and simultaneity factor of the consumers are not included. Instead, the values of the equipment and electrical motors are assumed to be used continuously with a load factor of 1. This approach is used to avoid designing a power plant that is underestimated. It is also assumed that the transmission losses are 10% within the system, a value that is common for diesel-electric systems (MAN Diesel & Turbo, 2014).

In Table 5-4 the total power requirements for certain operational modes are presented, for more details see Appendix C.

Table 5-4: Power requirements for certain operational modes

Power Distribution [kW]						
Operational Mode	Port Loading	Transit to site	Positioning	Inst. Windmill	Inst. Foundation	Transit Port
Support Vessel	172	3265	1677	1729	1729	3265
Installation Vessel	704	29	744	1734	1734	29
Total Requirement ~	1000	4000	3000	4000	4000	4000

When considering power supply during harbour mode, a shore connection can be used. The cable will feed the system directly via the main switchboard. The advantage of this configuration is that there is no need to run any generator on board the support vessel, which will result in saved energy since no auxiliary equipment needs to be running.

5.6 Prime Movers

When the power requirement and distribution over operational time is decided according to the electrical balance, it is possible to decide an engine configuration. The number and size that is proposed regarding engines in this project is decided in a conservative manner. This is due to the uncertainties in the power balance and consequently to avoid designing a power plant that is not able to meet the requirements.

The support vessel acts as the power supply of the installation vessel. The advantage of this configuration is that the system is able to be more compact since it is collected as one system instead of several. This results in a more efficient operation of the engine system, both when considering maintenance and operation, and also due to the fact that the installation and support vessel have different demands at certain operational modes see Appendix C. In other words, the installation and support vessel is considered as one unit when considering power balance over time.

The configuration of the power system on the support vessel will consist of four generator sets able to produce 1000 kW each, with 6 cylinder 4-stroke medium diesel engines.

Table 5-5: Specification of generator sets

Main Gensets	Manufacturer	Number	Type	Power Output [kW]
Support Vessel	Wärtsilä	4	1000W6L20	1000

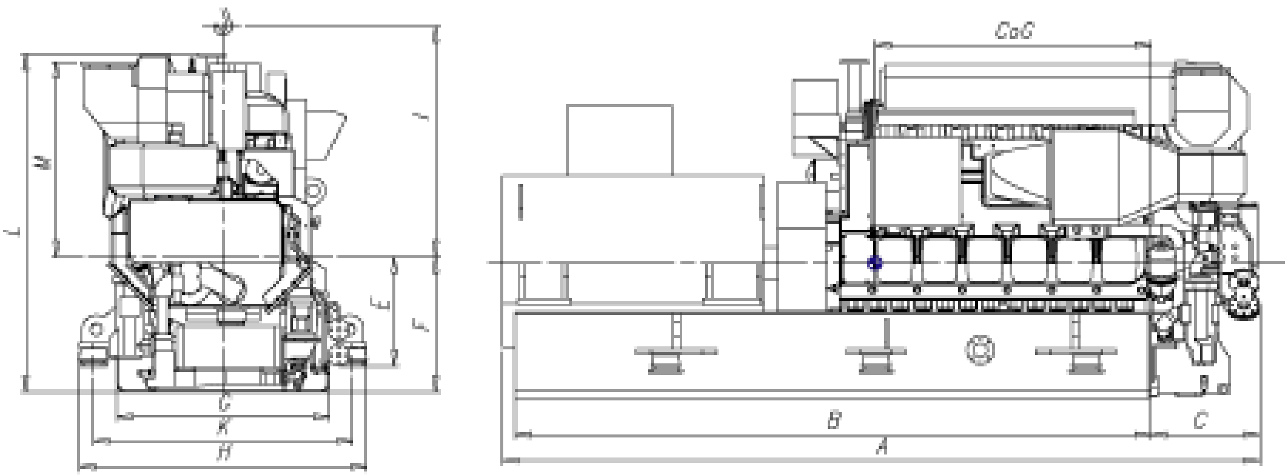

Figure 5-4: Principle sketch of generator sets (Wärtsilä, 2014)

5.6.1 Emergency Generator

The support vessel will be equipped with an independent emergency power supply system. The system shall support electric power to crucial equipment concerning safety such as emergency and navigation lights, communication equipment, fire systems and etc. The location of the emergency power supply system should be above the uppermost continuous deck, where it also should be possible to enter from open deck according to Transportstyrelsen (TSFS 2014:1).

For the support vessel, the emergency power supply will be a diesel generator located on upper deck together with the emergency switchboard see Appendix C. The genset is from Volvo Penta and is able to produce 85kW to fulfil the required need established in the electrical balance.

Table 5-6: Specifications of emergency generator

Emergency Genset	**Manufacturer**	**Number**	**Type**	**Power Output [kW]**
Support Vessel	Volvo Penta	1	D5A TA/UCM274E	85

5.7 Ballast system

The ballast system is one of the most important systems on the installation vessel. It helps control the stability and work condition of the installation vessel. During the installation operations, the ballast system must be stable and reliable. It also has to be able to deliver enough pumping capacity to fit the time frame of an installation.

5.7.1 Type selection

There are three main typical types of ballast system, which are water pump type, compressed air type and hydrostatic pressure type. The water pump type uses a pump to transport the water, while the compressed air type uses air pressure difference, and the hydrostatic pressure type uses gravity. Each of them has advantages and disadvantages. In Table 5-7 an evaluation matrix of the different pump types is presented. The pumps are graded from 1 to 5, where 5 is the most favoured. According to the summation results, water pump type is the best alternative for this vessel.

Table 5-7: Evaluation matrix of pump types

	Water Pump	**Compressed Air**	**Hydrostatic pressure**
Position limitation of ballast tank	5	4	2
Piping arrangements	4	3	5
Power consumption	3	4	4
Cost	3	4	2
Ballast time	5	4	2
Reliability	4	4	5
Flexibility	5	1	1
Structural reinforcement	5	1	5
	34	**25**	**26**

As there is no limitation of the placing of the water pump there is more freedom of component arrangement in comparison with the other types. The hydrostatic pressure type requires the ballast tanks to be positioned at the bottom of the vessel. Although the hydrostatic pressure type is simplified with less pipe lines and consumes less energy, it is not easy to be controlled compared with water pump type. It is required to combine with one of the other two systems to de-ballast.

During the installation operation, the vessel is expected to move accurately, it needs to be able to ballast and de-ballast at any time to compensate the wave response and keep the vertical position. Compared with the two types above, the compressed air type is moderate in every factor. When it works, it needs to inject compressed air into the tank. When de-ballasting, the compressed air will give additional load to structure. However, compressed air type and hydrostatic pressure type are usually combined in one vessel to undertake de-ballasting and ballasting respectively.

As seen in Table 5-7, the advantages of the water pump outweigh the disadvantages of the compressed air and hydrostatic pressure pumps. In conclusion, the water pump type can be applied in this installation vessel. This system will be a remotely controlled system.

5.7.2 Concept design

According to the classification rules (DNV, 2011), the ballast systems shall be designed so that if the equipment has a failure or is mal-operated, the liquid in the tank will not move without control. The solution is to apply individual valves for each tank. If one tank fails the valve for that tank can be closed remotely. If one valve fails, the other valves will be kept closed. Then there will be no movement of water in the pipelines. Another requirement is that the ballast system should be arranged so that any tank can be controlled by at least two independently driven pumps, meaning that three pumps are needed on the vessel. The conceptual design of the ballast system can be seen in Figure 5-5.

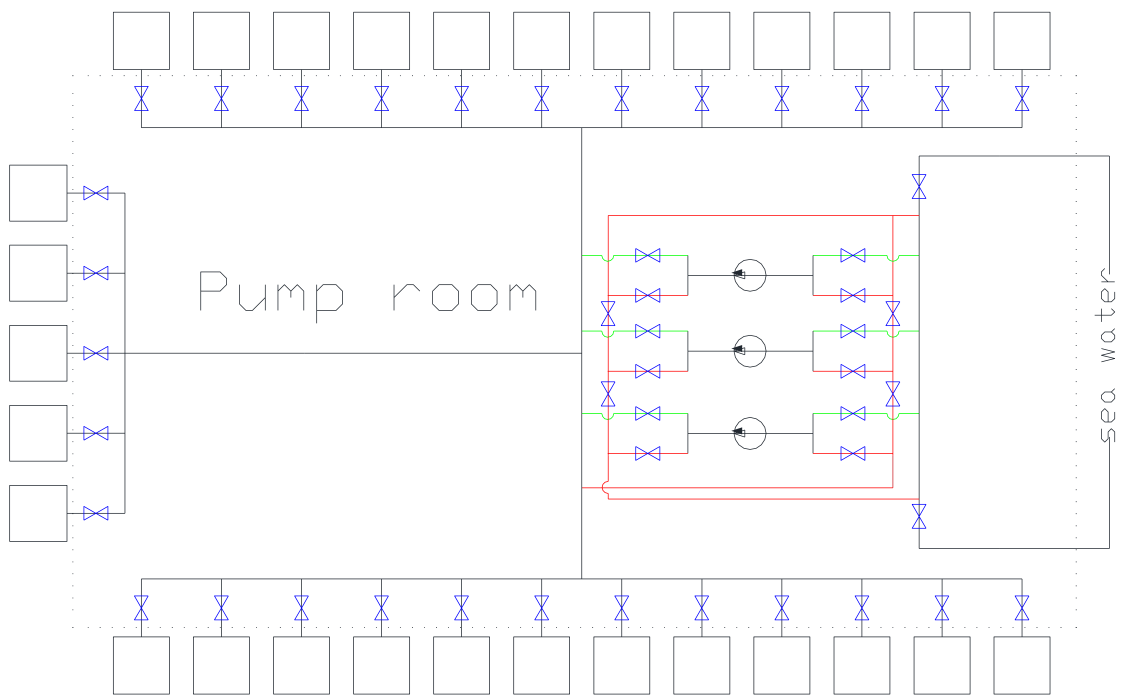

Figure 5-5: Conceptual design of ballast water pipe arrangement on installation vessel

The green pipe lines that can be seen in Figure 5-5 are set for ballast, while the red pipe lines are set for de-ballast. There are three pumps installed in the ballast system totally. Two of them set for daily work, another is for redundancy. As the rules said, any two of the pumps can control all the tanks and the individual valves can separate each tank from the entire system.

5.7.3 Pump selection

According to the stability calculation in Chapter 4.3 the maximum ballast volume is approximately 9000 m^3. From the work time schedule, the ballast time is expected to be around two hours with two pumps, which means that the capacity of each pump should be around 3000 m^3/h, including a high redundancy.

From Figure 5-6 it can be seen that the largest distance from the sea level to the highest ballast tank top is 26.2 m which could be the maximum pump head. When water is pumped into this tank, the distance will be shortened. The actual required head should be in the range from 18.7m to 26.2m.

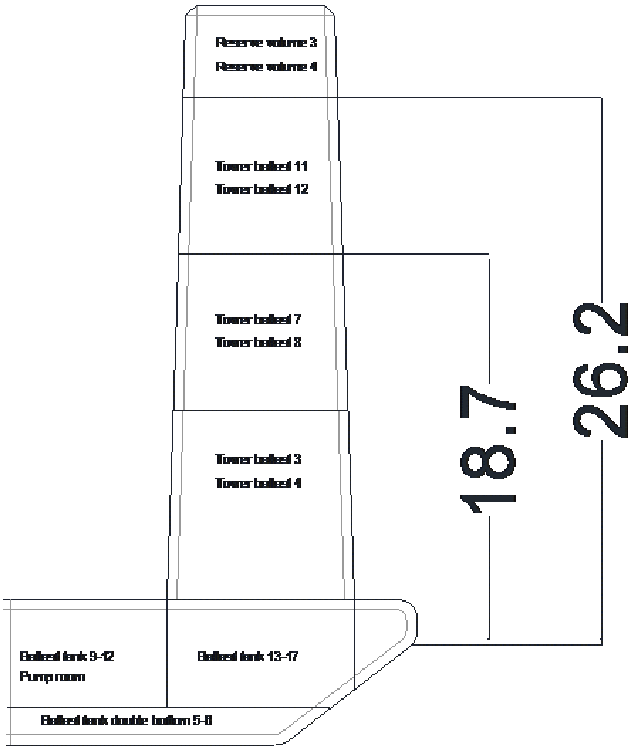

Figure 5-6: Pump head height to the highest ballast tank

According to the estimation above, the ballast pump is supposed to have a 3000 m^3/h capacity and 30 m head (including head loss).

Figure 5-7: The selected pump to be used on the installation vessel (*Wärtsilä Hamworthy, 2014)*

Figure 5-7 shows the selected pump; a Hamworthy model CA single stage centrifugal pump which is manufactured in both vertical and horizontal designs for capacities ranging from 380 m^3/h to 6000 m^3/h. The flexible design can make it easier to install the pump in the narrow pump room. Furthermore, this type of pump has fluid barrier protection of the mechanical seals which can extend its life and reduce maintenance time, as well as costs.

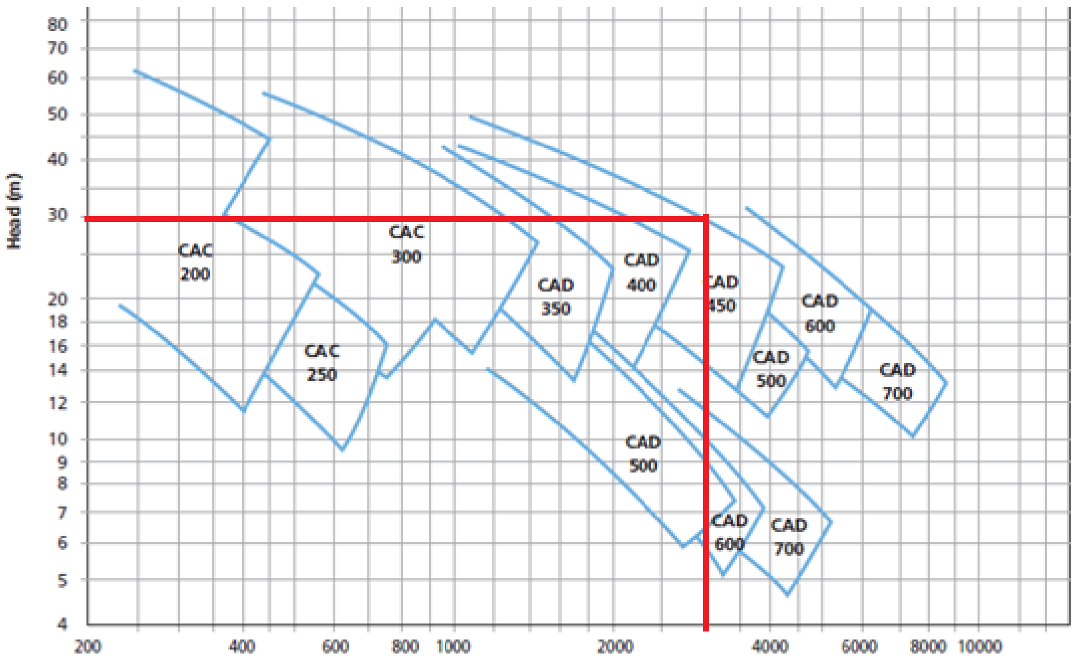

Figure 5-8: Pump capacity and head range for various models (*Wärtsilä Hamworthy, 2014)*

Figure 5-8 illustrates the capacity and head range of model CA in different type. According to the task requirement, CAD 450 should be selected.

5.7.4 Pipe size estimation

It is assumed that the flow speed in the pipe is 2.5m/s. This value is received from experience. Because the high speed inside the pipe may cause higher pressure drop, 2.5m/s is the limitation and will be used as a design speed.

Afterwards, the diameter of the main pipe can be estimated as below:

$$D_m = 2\sqrt{\frac{nQ}{3600v\pi}}\ (m)$$

Where:

Q = the pump capacity, 3000 m^3/h

n = the amount of pump in work, 2

v = flow speed, 2.5m/s

$$D_m \approx 0.95\ m$$

The combined cross-sectional area of the branch pipe lines can not to be less than that of the main pipe. In this ballast system, there are 29 ballast tanks and each tank has an individual branch pipe connecting to the main pipe. The diameter of branch pipes can be estimated as below:

$$D_b = D_m\sqrt{\frac{1}{29}}\ (m) \approx 0.20\ m$$

According to the rules (DNV, 2011), the main ventilation pipe should be at least 25% larger than the filling pipe. So the diameter of ventilation pipe can be estimated as:

$$D_v = 1.25D_m \approx 1.20\ m$$

The size of each individual tank can be estimated in the same way. Its diameter cannot less than 0.25m. These branch ventilation pipes should connect to the nearest funnel in the tower. If the main ventilation pipe is divided into 4 branch pipes which are installed in 4 towers respectively, each pipe cannot be less than 0.60 m.

5.7.5 Head loss (Pipe resistance)

Due to the friction of the pipes inner wall and energy loss at elbows, the pump head cannot reach the supposed value. So the loss of head needs to be applied to make sure that the actual pump head is still in the range of design point.

There is a specific equation called Darcy-Weisbach equation (Darcy-Weisbach, 2002), used to estimate the head loss in straight pipes. Here the head loss in one branch pipe can be calculated. Compared with the length of main pipe, branch pipes give higher resistance. Use diameter of branch pipe to do the resistance estimation.

$$h_l = f\frac{L}{D}\frac{V^2}{2g}\ (m)$$

Where:

h_l = the main head loss (m)

f = the coefficient of friction, new pipe 0.02, old pipe 0.04 (*Darcy-Weisbach, 2002)*

L = the length of straight pipes (m), roughly 25 m (Hooper & McKetta, 1991)

V = the average flow speed in the pipe (m/s), 2.5m/s

D = the diameter of straight pipes (m), 0.2m

g = the gravity acceleration (m/s^2), 9.81 m/s^2

It can be estimated that each branch pipe has approximate 25 m straight pipe, including the 10 m equivalent length from two 90 degree elbows and one valve. The equivalent length of a regular 90 degree elbow is approximately 8 m, and the equivalent length of a valve is approximately 1.6 m.

Then the result of Darcy-Weisbach equation is 0.3 m, times 29 ballast tanks the result will be 9.6 m which is the total head loss. The actual head is about 20.4 m which is in an allowable range.

5.7.6 Pump room arrangement

Figure 5-9 and Figure 5-10 show the pump room arrangement. In the pump room of the installation vessel there are seven pumps total; three ballast pumps, two bilge pumps, and two fire pumps. In some emergency situations, the function of these three kinds of pumps can replace each other. There is also a frequency converter arranged in the pump room which is used to support the retractable thruster. According to the side view, the frequency converter is lifted by a platform above the ballast pipe lines. As the air in the pump room is humid, the platform could be established as a separate room to protect the frequency converter from wet air. The other space in the pump room can be occupied by pipe lines and valves. It can be seen from the top view that there is adequate space on both sides of frequency converter.

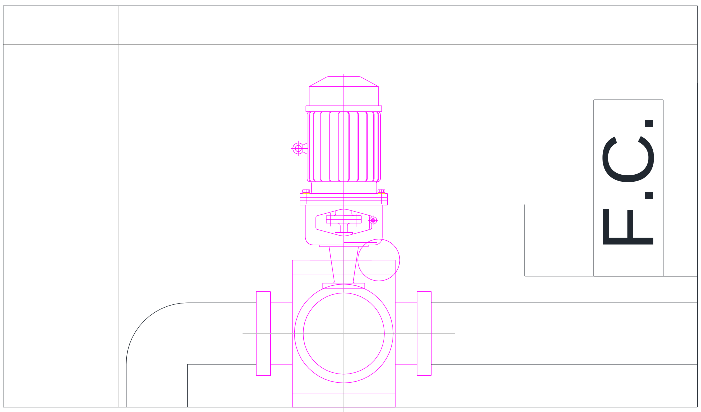

Figure 5-9: Side view of pump room

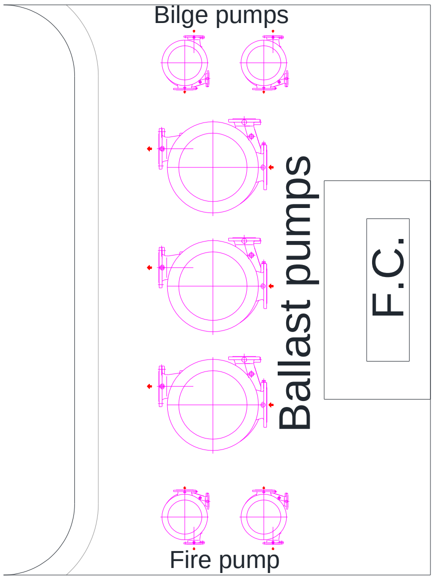

Figure 5-10: Top view of pump room

5.7.7 Power consumption of ballast system

The power consumption is a significant value to join the electrical balance calculation in the stage of engine selection. According to the equation for calculating the pump power below, the power requirement of the ballast system can be calculated.

$$P = \frac{nQH\rho g}{3600 \cdot 1000e}\ (kW)$$

Where:

Q = the pump capacity, 3000 m^3/h

n = the amount of pump in work, 2

H = the head of pump (m), 30 m

ρ = the seawater density (kg/m^3), 1025 kg/m^3

g = the gravity acceleration (m/s^2), 9.81 m/s^2

e = the efficiency of pump (m), 0.85

$$P \approx 600\ (kW)$$

This is the power consumption at highest head and lowest flow. Applying the same process to the lowest tank which height is 2 m. At the 2m head the pump capacity is approximately 8000 m^3/h, and the efficiency is about 0.13.

$$P \approx 690\ (kW)$$

So the ultimate condition is at lowest head and highest flow.

5.8 Bilge water system

The task of the bilge system is to discharge bilge water overboard. It includes bilge pump, bilge water main pipes, branch pipes, suction filter, distribution valve box, sludge tank and sewage oil separator. The leakage of hull plate and pipeline, as well as condensate results from temperature differences can generate bilge water that has to be discharged in time. Otherwise, the structure may corrode or other equipment be damaged. In this section, the dimensioning of the bilge pump and bilge pipe size is presented.

5.8.1 Bilge pipe

According to the rules (DNV, 2011), the internal diameter of the main bilge line shall not be less than given by the following formula:

$$d = 1.68\sqrt{L(B\ +\ D)} + 25\ (mm)$$

L = length of ship (m)

B = breadth of ship (m)

D = depth of ship to bulkhead deck (m)

$$d \approx 95\ (mm)$$

The value of the internal diameter of the main bilge line is same as the estimation of internal diameter of main ballast pipe line.

5.8.2 Bilge pump

The pump capacity Q in m^3/hour may also be determined from the following formula, according to DNV rules (DNV, 2011):

$$Q = \frac{5.75d^2}{10^3}$$

d = bore of bilge pipe in mm according to the result above

$$Q \approx 52\ (m^3/h)$$

The result of required capacity can also be found in DNV rules (DNV, 2011), which shows pipe diameter and corresponding bilge pump capacity. For the value of $Q \approx 52\ (m^3/h)$, calculated above, the corresponding bore of bilge pipe is 95 mm. In order to save power, the head should be as low as possible.

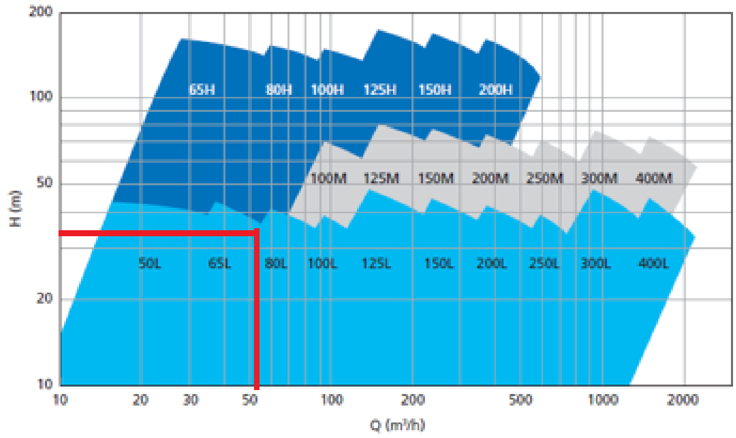

Figure 5-11: Selected pump and diagram to show pump capacity (*Wärtsilä Hamworthy, 2014)*

According to the pump requirement, Hamworthy C2G - 80LA can be selected as can be seen in Figure 5-11.

5.9 Fire system

Fire system includes fire detection system, smoke control system and fire extinguishing system. In case of fire on board any of the vessels it has to be detected and dealt with quickly. To ensure the safety of the vessel and crew, a good fire system is of highest priority

5.9.1 Fire detection and alarming

The fire detection and alarming includes automatic sprinkler alarm system, fixed fire detection and fire alarm systems. Automatic sprinkler is supported by a fresh water pressure tank which is used to maintain the work pressure for the automatic sprinklers. It also needs to connect with the main fire pipe.

Fixed fire detection and fire alarm systems are combined with sensitive thermal and smoke detectors. It can detect the density of suspended particles in the early stage of fire.

This installation vessel, although it is totally remote controlled, the pump room and motor room should be well protected. In pump room and motor room, both fire systems are installed, as well as humidity detector. Because the installation vessel is driven by electricity, humid air inside tank could make short circuit of electric circuit and electrical equipment.

5.9.2 Smoke propagation control

To ensure the smoke can be discharged from fire position, ventilation is significant. All the pump room and motor rooms should have cross and separate ventilation. So the ventilation pipes in the installation vessel should be separated in four towers.

In the smoke, all the machinery and equipment should be monitored and efficient operated. So in the motor room and pump room, far infrared cameras should be installed. They can monitor the work condition and temperature anytime, also inside smoke.

5.9.3 Fire pump

In fire control and extinguishing system, the fire pump is the heart of the fire system. The calculation of the fire pump shall follow IMO rules (IMO, 2009). The capacity of fire pump shall be 4/3 times of one bilge pump, but does not need to exceed 180m^3/h (IMO, 2009).

$$Q_f = \frac{4}{3} Q_b \text{ (IMO, 2009)}$$

Where:

Q_b is the capacity of one bilge pump

$$Q_f \approx 70\ m^3/h$$

Then the type of fire pump can select the same as bilge pump as can be seen in Figure 5-11.

5.9.4 Fire main pipe

The diameter of fire main pipe should meet the maximum capacity of two fire pumps.

$$D_f = 2\sqrt{\frac{nQ}{3600 v\pi}}\ (m) \text{ (IMO, 2009)}$$

Where:

Q is the pump capacity, 70 m^3/h;

n is the amount of pump in work, 2;

v is flow speed, 2.5m/s .

$$D_f \geq 0.15\ (m)$$

5.10 Risk control options

If the power supply from the support vessel to the installation vessel is lost during operation, the installation vessel will need enough power on board to de-ballast and surface by itself. The amount of power that has to be available is calculated for the worst case scenario, when the installation vessel is fully submerged and has just released the biggest foundation. At this depth an approximate of 3000 tons of ballast water will have to be pumped out, the installation vessel will have to move away from the foundation, and the releasing mechanism may also have to be supplied for. If one pump is used it will require about 300 kWh to de-ballast completely and surface the installation vessel. For redundancy, a total of 500 kWh on board the installation vessel is sufficient to supply every need in case of an emergency. Some solutions to ensure that the installation vessel can de-ballast and surface in case the power is lost are described in the following sections along with a conclusion to come to a final decision.

5.10.1 Batteries

Batteries can be installed on board the installation vessel to ensure enough power for the pumps to de-ballast the tanks. Recently an electric public transportation ferry was presented in Stockholm to reduce costs, noise, and emissions from their services (Sjövägen, 2014). The batteries on board the ferry have a total storage capacity of 500 kWh, with a very compact volume, low mass, and a long lifetime of more than 10 years. These are very similar specifications to what is needed on the installation vessel.

5.10.2 Emergency generator

An emergency generator can be installed on board the installation vessel to produce power for the pumps. If the power from the support vessel is lost, the installation vessel will have sufficient fuel and an emergency generator big enough to produce all the power needed to surface. This solution can deliver reliable power but, more equipment on the installation vessel is needed to ensure safe operation, e.g. fire protection equipment and ventilation systems. A fuel tank will also have to be installed on board.

5.10.3 Compressed air

On submarines, compressed air is used to empty the ballast tanks of water to create positive buoyancy. In the same way, compressed air on board the installation vessel can be used to quickly empty enough water from the ballast tanks to create positive buoyancy and ensure a safe surfacing.

5.10.4 Location of attachment

The location of the attachment between the support vessel and installation vessel can be either somewhere on the deck of the installation vessel or in one of the vertical tanks. If the attachment is located on the deck of the installation vessel and the connection is broken during submerging, another emergency solution has to be used in order to surface. If the connection is located on one of the vertical tanks, above water, it might be possible to reattach a new cable from the support vessel.

5.10.5 Conclusion

Based on discussions with experts, the solution using batteries to power the pumps is deemed to be the most advantageous. The batteries have enough power for one pump to de-ballast enough water for the installation vessel to surface and even move it to some extent in any direction. Using compressed air stored on the installation vessel would require too much space in order to de-ballast enough tanks. An emergency generator would require a lot of additional equipment on board with fuel tanks needed, ventilations and so on. The arrangement of an emergency generator is considered too complex compared to using batteries and also means loss of ballast tanks.

5.11 Fuel selection

During the last few decades, the fuel price has kept increasing while the requirement to limit the emissions has become more and more strict. Many alternative fuels have started to be considered for marine engines such as LNG, biodiesel and alcohol fuels.

For this design project, the main focus has been on low emissions, high efficiency and robust solution. In Table 5-8, the data of three different combustion concepts is presented.

Table 5-8: Fuel characteristics

Property	Methanol	Natural Gas	Diesel
Density (kg/l)	0.79	0.44(as LNG)	0.85
Boiling point (°C)	65	-162	150-370
Flash point (°C)	11	-188	60
Auto ignition (°C)	464	540	240
Viscosity at 20°C (cSt)	0.6	N/A	13.5
Octane RON/MON	109/89	120/120	-
Cetane No.	3	-	45-55
LHV (MJ/kg)	20	50	42
LHV (MJ/l)	16	22	36
Flammability Limits, Vol%	7-36	5-15	1-6
Flame Speed (cm/s)	52	37	37
Heat of Evaporation (kJ/kg)	1178	N/A	233
Stoichiometric Air-Fuel Ratio	6.4	17.2	14.7
Adiabatic flame temp. (°C)	1910	1950	2100

The regulations on control of diesel engine NOx emissions are mandatory to follow. In addition the fuel oil sulphur limits are becoming stricter. The changing of limitations of NOx and SOx are shown in Figure 5-12 and Figure 5-13.

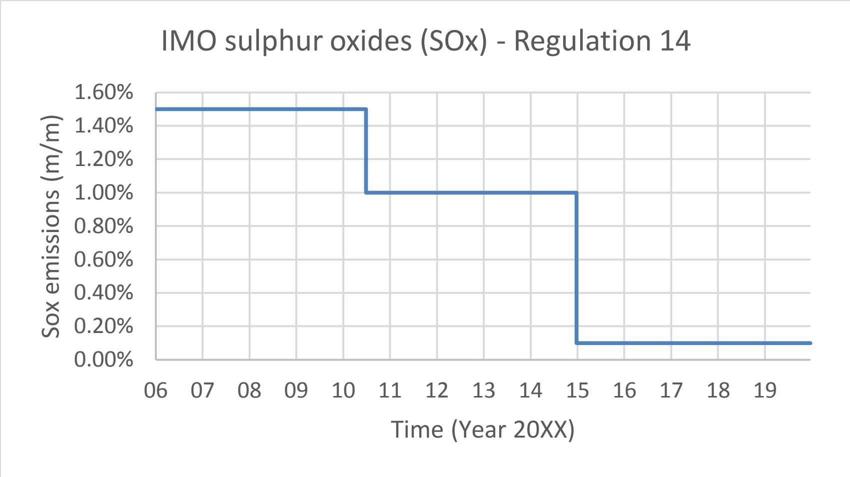

Figure 5-12: Sulphur oxides emission limitations

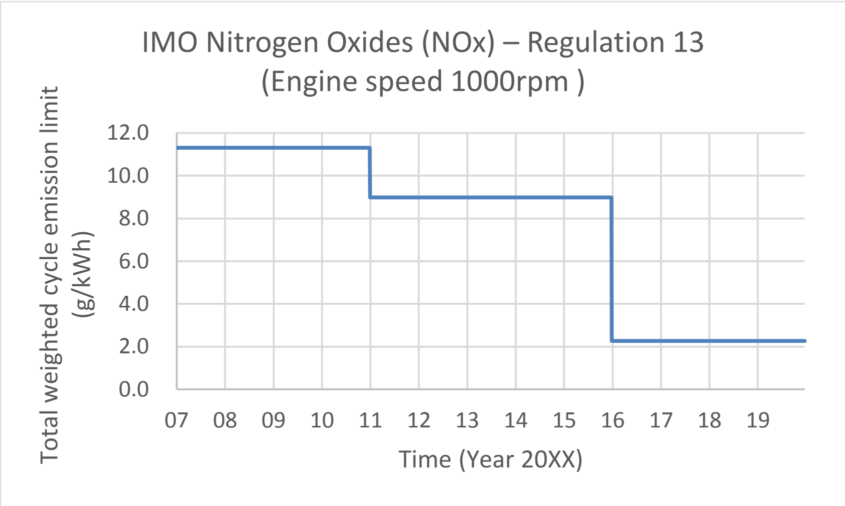

Figure 5-13: Nitrogen Oxides emission limitations (at engine speed 1000 rpm)

According to the International Convention for the Prevention of Pollution from Ships (MARPOL), the Regulations for the Prevention of Air Pollution from Ships (Annex VI, 2008) seek to minimize airborne emissions from ships. As the installation and support vessels will be built after 2016, they have to comply with the strictest environmental rules which is regulation 13: The NOx emission must be lower than 2.3 g/kWh and the fuel oil sulphur must be limited below 0.1% (expressed in terms of % m/m – that is by weight) or an exhaust gas cleaning system need to be installed to reach a similar SOx emission level.

LNG has been promoted as a marine fuel and has seen some significant development during the last decade. However, the installing of a LNG supply system is expensive, e.g. the whole fuel storage and distribution system need to be designed to maintain the LNG at -163°C in order to keep it liquid. The distribution system of LNG also needs to be designed to maintain the LNG at -163°C. LNG can be handled efficiently in large volumes but for smaller volumes the cryogenic equipment becomes too costly.

Methanol is a liquid at ambient conditions and can be handled similar to other liquid fuels. Considering the distribution and the required on-board system the cost of methanol is much lower than LNG. Methanol is a low flashpoint fuel (flashpoint <60°C) which requires some extra considerations and features for the storage and distribution system on-board which makes it somewhat more expensive than an MGO installation. Compared to an HFO installation, the methanol alternative is about similar in cost since the fuel purification and separation system is eliminated.

Similar to LNG, methanol does not contain any sulphur and for IMO emission regulations, methanol could be a good solution to meet the sulphur oxides emission requirements. By nature, LPG, LNG and methanol generate less CO2 emissions during combustion than oil based fuels. The majority of the methanol produced uses natural gas as feedstock, but methanol can also be produced from a vast variety of fossil freed feedstock such as biomass which is then called bio-methanol.

Preliminary results from methanol tests on medium speed marine engines show that the NOx emissions almost reach Tier III levels without modifications to the combustion. To be sure to meet Tier III, the engine supplier could either fit an EGR system or fit an SCR (Selective Catalytic Reactor) in the exhaust system. The EGR will have some fuel penalty since the combustion of the engine needs to be tuned for low NOx emissions. With after treatment with an SCR the engine combustion can be tuned for lowest possible fuel consumption and the SCR will ensure the NOx reduction.

The Figure 5-14 shows the posted price history for different fuels. The cost per energy content kWh/kg is quite similar between methanol and MGO. The fuel methanol market is much less liquid and transparent compared to the MGO market and methanol suppliers has indicated that there will be an approx. 20% discount for the methanol price to the client which will make the methanol competitive.

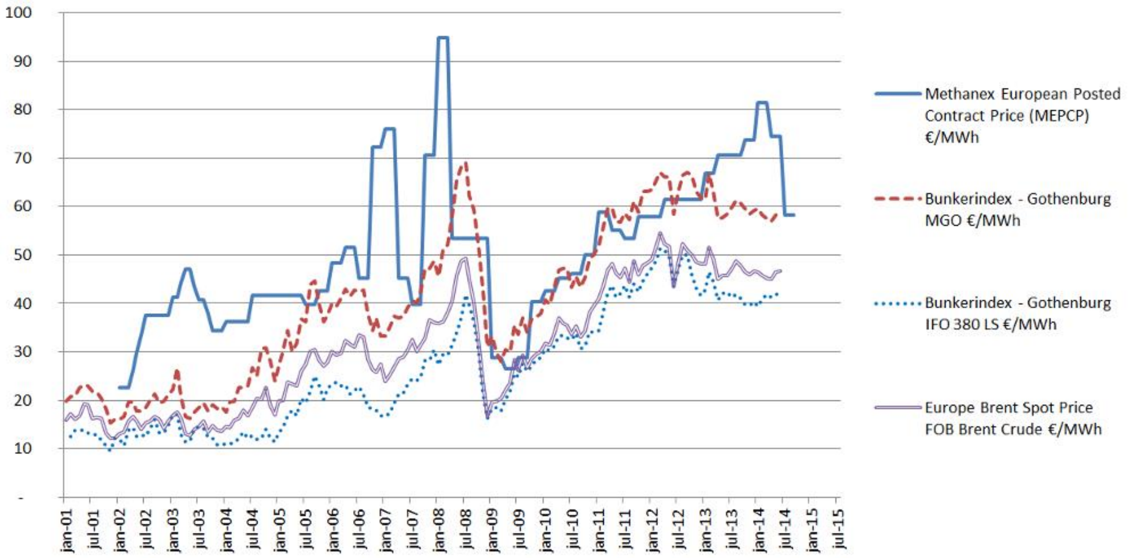

Figure 5-14: Fuel price fluxuations the last two decades

Both methanol, LNG and MGO will provide alternatives to HFO in order to fulfil the upcoming SOx requirements. Methanol is expected to be the least expensive alternative. There are also added benefits to use methanol. With methanol operation it will be easier to fulfil the NOx tier III requirements and the possibility to seamless blend in bio-methanol will open up for significant reduction of greenhouse gas emissions.

In conclusion, methanol is a good alternative fuel to diesel. In this project, dual-fuel engine is used for four engines, which is diesel and methanol fuels. It will meet stringent emission rules without too much extra cost.

5.12 Methanol Adoption

When using methanol as fuel in a standard diesel engine, several systems need to be adopted for methanol operation (Marine methanol, 2014). These systems are ignition system, injection system and supply system.

5.12.1 Ignition system

From Table 5-8 it can be seen that the cetane number (indicator of the combustion speed of fuel) of methanol is much lower than diesel fuel, meaning that methanol is difficult to ignite by heat from compression. The use of methanol in diesel engines requires some auxiliary equipment to overcome the low ignition quality of methanol. There are three ways to solve this problem (Tsuchiya, K., & Seko, T, 1995):

1. Glow assist: The ignition is accomplished by a combination of heating from compression, heating from a glow plug and the catalytic effect of the platinum within the glow plug on the methanol within the fuel.
2. Spark assist: It is commonly used in gasoline engines and LNG engines, the combustion process of the air-fuel mixture is ignited by a spark from a spark plug.
3. Pilot ignition (Pilot injection): The methanol is injected close to TDC (top dead centre) and ignited by a small amount of pilot fuel (in our case it's diesel oil).

Glow-assisted or spark-assisted methanol engines, however, have lower thermal efficiency under low load operating conditions because the flame does not propagate sufficiently throughout the whole mixtures. This may be because the amount of lean mixture increases over the lean flammable limits. A pilot ignition DI methanol engine has been developed to improve the thermal efficiency under low load condition.

5.12.2 Injection system

For a methanol engine with pilot injection system some extra systems are needed;

Fuel supply system: Additional piping for supply of methanol, a high pressure methanol injector, and control systems are needed.

Cylinder heads modification: The cylinder heads require an added inlet entrance for supply of methanol. The exhaust valves are modified to resist excess wear because the exhaust gases from combustion of methanol have a much lower concentration of lubricating particulates than exhaust gases from traditional diesel fuel or heavy fuel oil.

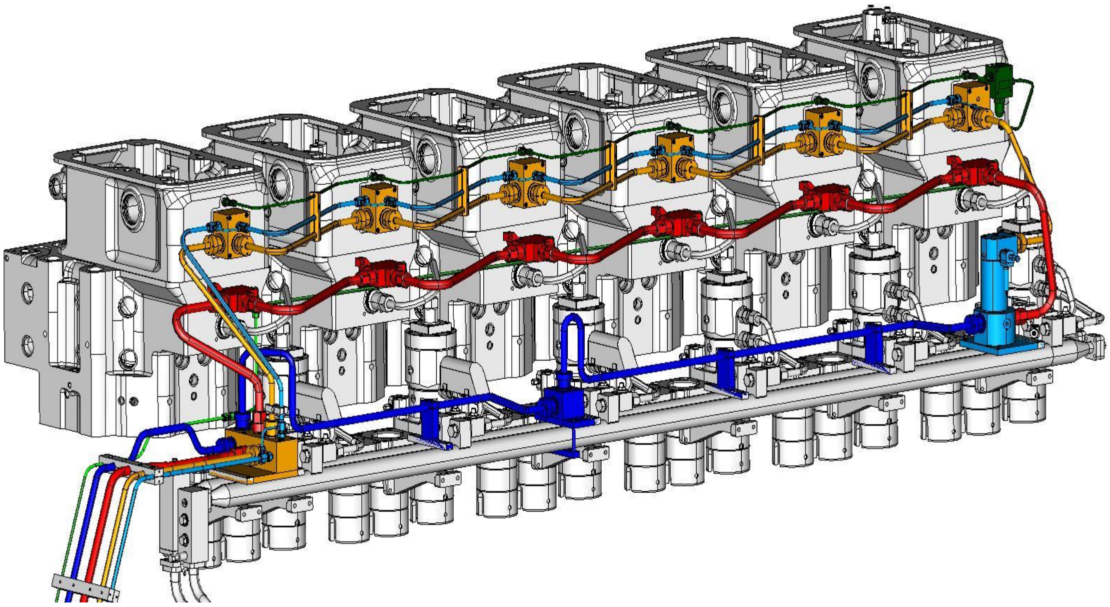

Figure 5-15: Wärtsilä Methanol-Diesel retrofit solution on-engine piping (courtesy of Wärtsilä)

5.12.3 Supply system

Methanol fuel tank: The structure of methanol fuel tank is the same as the diesel tank, but methanol is corrosive to some metals including aluminum, zinc and manganese. Similar to ethanol, compatible material for fuel tanks, gasket and engine intake have to be used. In this case, tanks for methanol will use special paints to protect against corrosion.

High pressure methanol pump: To meet the pressure requirement of the engine injection system.

5.13 Fuel consumption and CO_2 emissions

In the operation there are six stages; port loading, transit to site, positioning, installation of wind mill, installation of foundation, and transit to port. Different stages have different demand of power. Some engines may be deactivated during some stages to save fuel.

Meanwhile, methanol burns in open air, forming carbon dioxide and water. The formula is shown below:

$$2\mathrm{CH}_3OH + 3O_2 \rightarrow 2CO_2 + 4H_2O$$

This means that 1 kg of Methanol will produce 1.375 kg of carbon dioxide.

Table 5-9: Fuel consumption and carbon dioxide emissions for each stage

Operation modes	**Port Loading**	**Transit to site**	**Positioning**	**Installation**	**Transit Port**	**Harbor**
1000W6L20 FC(g/h)	203229	203229	203229	203229	203229	38850
Number of 1000W6L20 working	1	4	3	4	4	1
Duration(Hours)	1	4	1	5	2	11
MGO Consumption(kg/h)	203.2	812.9	609.7	812.9	812.9	38.9
Total MGO Consumption(kg)	203.2	3251.7	609.7	4064.6	1625.8	427.4
Methanol Consumption(KG)	426.8	6828.5	1280.3	8535.6	3414.2	897.4
CO2 Emissions(kg)(methanol)	586.8	9389.2	1760.5	11736.5	4694.6	1234.0
CO2 Emissions(kg)(MGO)	650.3	10405.3	1951.0	13006.7	5202.7	1367.5
Cost for Methanol(SEK)	1263.3	20212.3	3789.8	25265.4	10106.2	2656.4
Cost for Methanol with discount	1010.6	16169.9	3031.9	20212.3	8084.9	2125.1
Cost for MGO(SEK)	1161.2	18578.9	3483.6	23223.7	9289.5	2441.7

Where:

LHV of MGO is 42 MJ/kg

LHV of Methanol is 20 MJ/kg

1 kg of MGO will produce 3.2 kg of carbon dioxide

1 kg of Methanol will produce 1.375 kg of carbon dioxide

Methanex European Posted Contract Price is 320 €/mt

MGO price at Gothenburg is 769 $/mt (2014-11-21)

EUR/SEK is 1:9.283 (2014-11-24)

USD/SEK is 1:7.535 (2014-11-24)

5.14 Emissions

According to International Convention for the Prevention of Pollution from Ships (MARPOL),the Regulations for the Prevention of Air Pollution from Ships (Annex VI) are used to regulate to airborne emissions from ships (SOx, NOx, ODS, VOC). This chapter describes many of the environmental issues that have to be managed and how they are solved for this task.

5.14.1 NOx Emissions

According to IMO (IMO Nitrogen Oxides (NOx) – Regulation 13), after 1 January 2016 the governed limits are stated by Tier III, the highest requirement. This vessel has to fulfil Tier III requirement as it will operate in an emission controlled area, so called ECA. The engines of this vessel run under 1000 rpm, so the nitrogen oxides should be lower than 2.3 g/kWh. When the engines run on diesel fuel a catalytic reduction unit (SCR unit) manufactured by Wärtsilä is used. It will also gain benefits from the SCR-unit to reduce the nitrogen oxides emissions while burning methanol.

5.14.2 SOx Emissions

According to IMO Regulation 14, after 1 January 2015 Sulphur Oxide (SO_x) emissions should be less than 0.1% m/m. The vessel fulfils the requirements with both methanol and diesel fuel. The fuel will be low sulphur content, and when using methanol, there is no sulphur during the combustion, i.e. no sulphur oxides will be produced.

5.14.3 Ballast water treatment

The ballast water treatment should obey the IMO International Convention for the Control and Management of Ships' Ballast Water and Sediments (BWM Convention), since this vessel is only operating in one place, there is no need to add any ballast water treatment system.

5.15 Electric arrangement

Figure 5-16 shows the electric arrangement of both the installation vessel and supply vessel. Since the installation vessel does not have generators on its own, electric power has to be supplied from the support vessel. In the transmission from the supply vessel to installation vessel, a transformer is used to increase the voltage, as high voltage requires a smaller diameter of the power cable. The theoretical calculation is shown below.

The diameter of the power line will influence the connection, copper wire is used, and the current density of copper wire is 2.5 A/mm^2. For the generator produce alternating current, the current is calculated by the formula below:

$$I = \frac{P}{\sqrt{3} \times U}$$

Where:

P is the power needed by the installation vessel.

U is the voltage in the cable.

The cable diameter is calculated by formula below:

$$d = 2\sqrt{\frac{I}{J * \pi}} = 21mm$$

Where:

J is the current density.

According to these two formulas, increasing the voltage is the only way of reducing the diameter of the power cable. Higher voltage has less transmission losses, so a transformer to increase the voltage from 400V to 3000V is added.

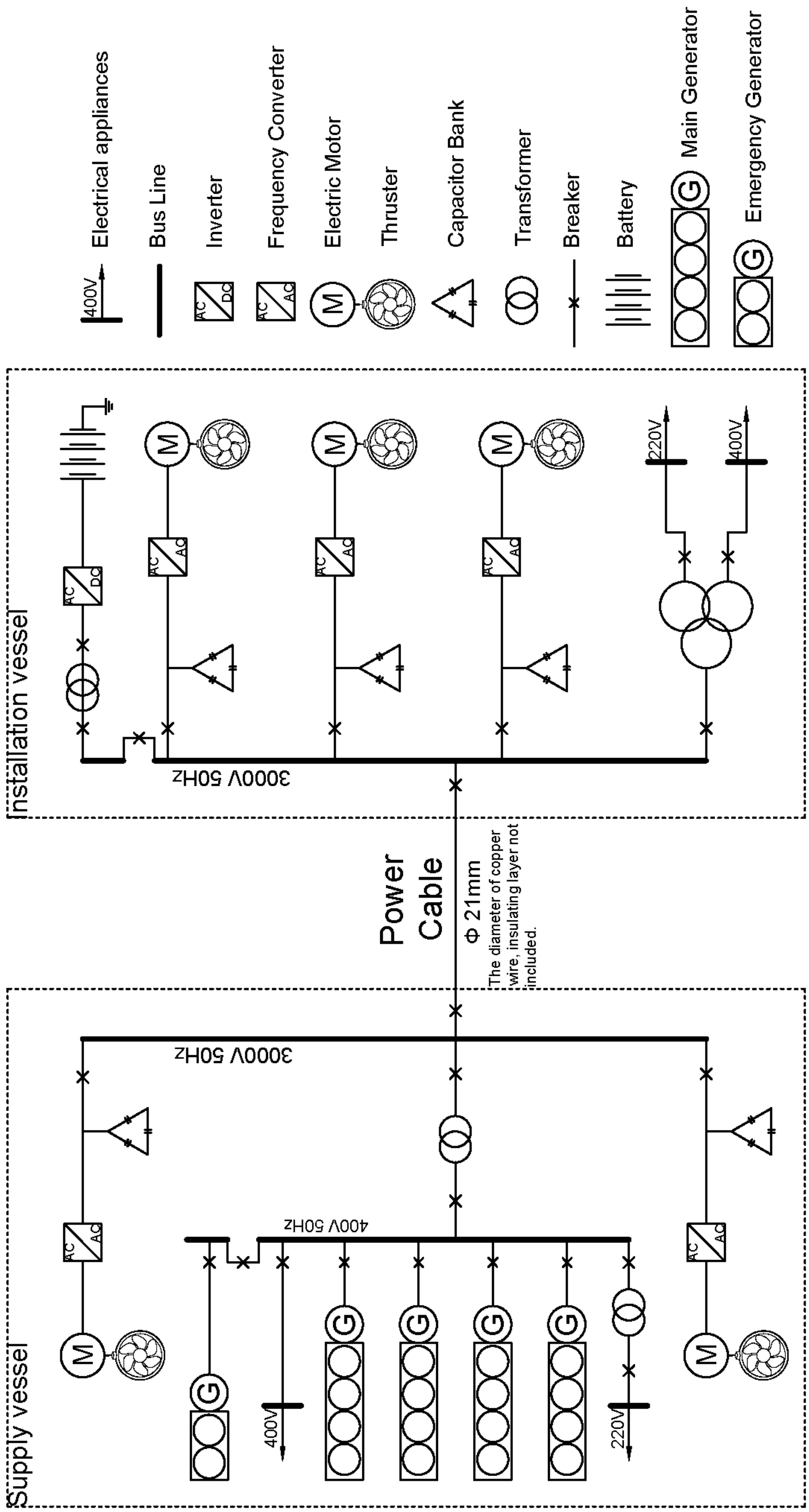

Figure 5-16: Power cable arrangement

6 Structure

The installation vessel is designed using DNV structural rules for barges, which are given in Section 14 of DNV rules for “Offshore service vessels, tugs and special ships” (DNV, 2011). These rules refer to DNV rules for “Hull Structural Design, Ships with Length Less than 100 metres” (DNV, 2012). This report focuses on structural arrangement of the installation vessel. Due to time limitation structural arrangement of the support vessel is left for future work.

6.1 Structural arrangement

Structural arrangement of the installation vessel is done in two parts; it is differentiated between a barge and vertical tanks (Figure 6-1). The installation vessel is longitudinally stiffened with spacing between longitudinal stiffeners of 500 mm. Due to high lateral sea pressure and high cargo load frame spacing is 500 mm, of which every third is a web frame (web frame spacing of 1500 mm). Web frame spacing only changes where the aft vertical tanks are connected to the barge.

To optimize weight of the structure cross-ties connecting web frames and bulkhead structures are introduced in side and bottom structures. Introducing cross-ties reduces the beam span of the frames to a half of it, which results in 4-times smaller required section modulus for frames. Smaller required section modulus leads to smaller frame dimension and weight optimization since the weight of the cross-ties is relatively small compared to the frames. The vessel’s structural arrangement, i.e. scantlings dimensions, is driven by local strength requirements due to high lateral sea pressure when submerged to maximum design depth of 30 m, which is the dimensioning loading condition.

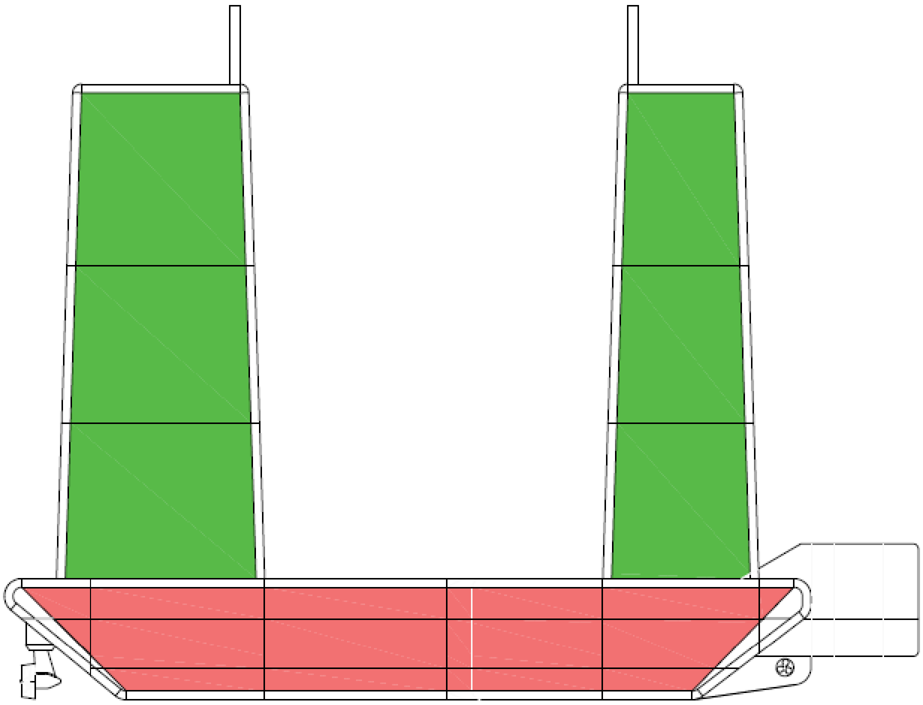

Figure 6-1: Differentiation between the barge (red color) and the vertical tanks (green color)

6.1.1 Material

Material used in the installation vessel is high strength structural steel with yield stress of not less than 355 MPa. In DNV rules it is denoted as NV-36. High strength steel is chosen due to the weight optimization reasons, which are driven by satisfying draft in port requirements, see Chapter 2.2, and to minimize the price of the structure. In comparison with using normal strength steel with yield point not less than 235 MPa a weight reduction of 29% is achieved (26% for the barge and 33% for the vertical tanks).

6.1.2 Structural cross-sections

As mentioned before the installation vessel is divided into two parts. In the barge there are 3 different cross-sections: A-A, B-B and C-C. Section A-A is positioned at the catamaran hull body, B-B and C-C are at the mono hull body. Section B-B is where the pump room is positioned (no longitudinal bulkhead in the middle).

In the vertical tanks there are 6 different cross-sections (D-D, E-E and F-F and G-G, H-H and I-I), which are positioned at different vertical positions. Reason for relatively high number of structural cross-sections in the vertical tanks is due to the different dimensioning hydrostatic pressures and weight optimization. From the hydrostatic point of view a low vertical center of gravity is desired, which results in designing the vertical tanks as light as possible. Having large number of structural cross-section increases complexity of a building process on the other hand.

All mentioned cross-sections are shown in Figure 6-2 and Figure 6-3 with their calculated properties shown in Table 6-1 and Table 6-2.

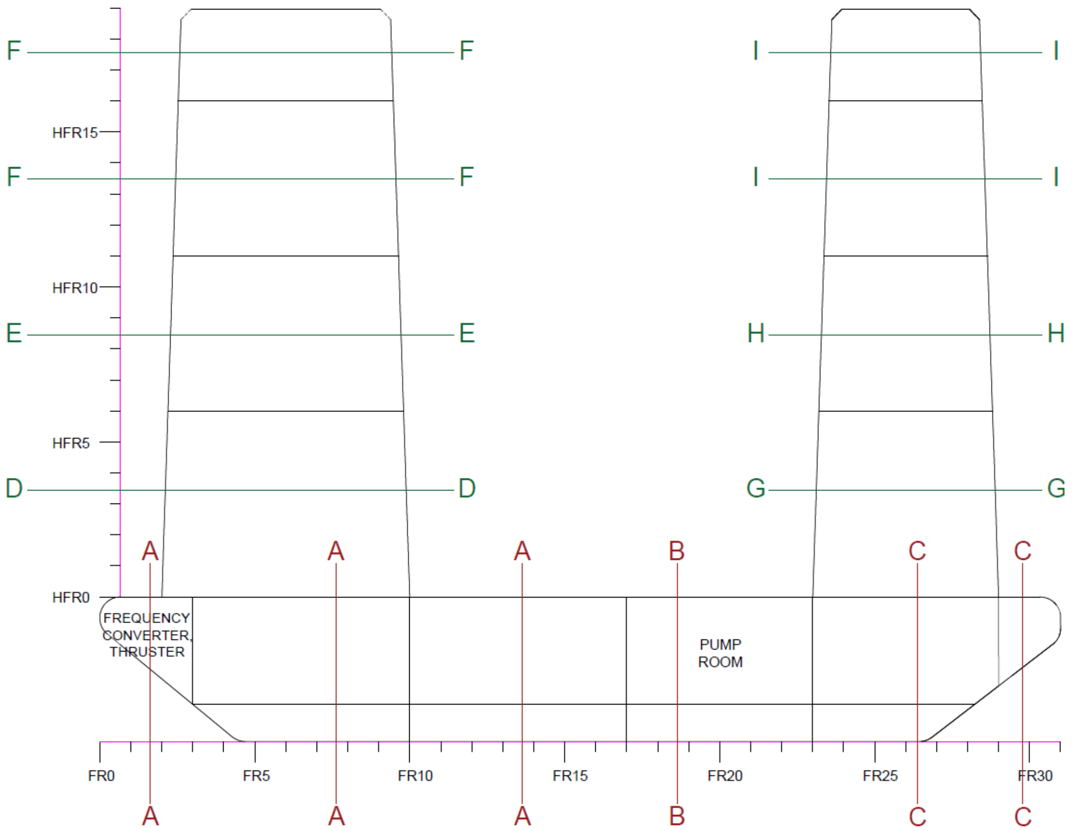

Figure 6-2: Structural cross-sections in the installation vessel (side view)

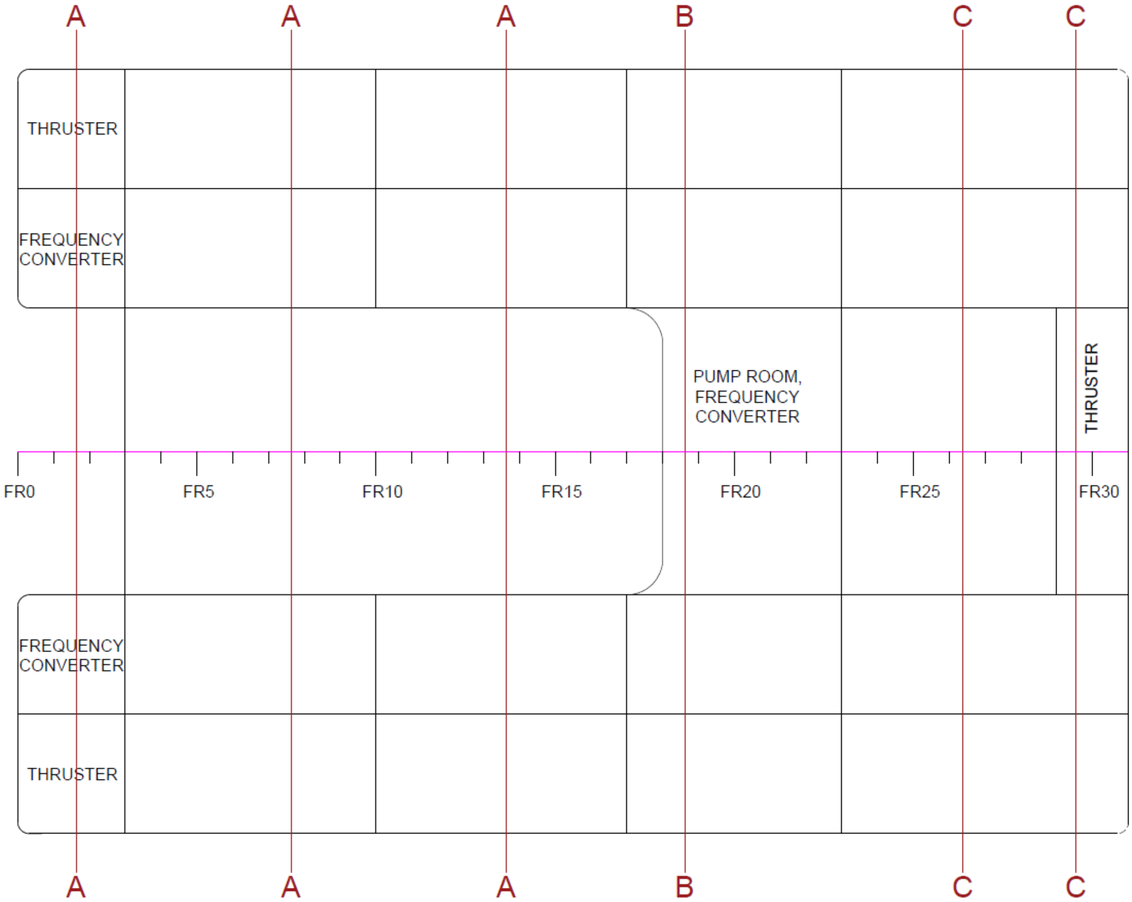

Figure 6-3: Structural cross-sections in the installation vessel (barge only) (top view)

Table 6-1: Structural cross-section properties (barge)

Section	Frame number (from – to)	I_y [m^4]	NA [m]	$Z_{y,deck}$ [m^3]	$Z_{y,keel}$ [m^3]
A-A	FR0 – FR17	15.07	3.58	4.21	4.40
B-B	FR17 – FR23	23.55	3.64	6.48	7.00
C-C	FR23 – FR31	23.45	3.56	6.58	6.82

Table 6-2: Structural cross-section properties (vertical tanks)

Section	Vertical frame number (from – to)	I_y [m^4]	NA [m]	Z_y [m^3]
D-D	HFR0 – HFR6	2.78	2.075	1.34
E-E	HFR6 – HFR11	2.00	2.075	0.96
F-F	HFR11 – HFR19	1.50	2.075	0.72
G-G	HFR0 – HFR6	1.91	2.075	0.92
H-H	HFR6 – HFR11	1.43	2.075	0.69
I-I	HFR11 – HFR19	1.07	2.075	0.52

Note: Position 0 m is at deck and going upwards

6.1.3 Scantlings

Profiles used for longitudinal stiffeners are chosen to be bulb flats profiles in different dimensions. Dimensioning criteria for chosen scantlings is the hydrostatic pressure when the vessel is submerged to the design depth of 30 meters. Dimensioning is done in a way that scantlings are utilized close to 100%. All calculations and utilization factors for structural elements are shown in Appendix D – Structure.

Transverse web frame stiffeners, cross-ties and deck girders are chosen to be T-profiles (two plates welded together) in different dimensions. Dimensions for girder profiles are chosen according to DNV rules (DNV, 2011: Section 3: C602) so that no stiffeners are needed on girders' webs. Profiles used in bottom and side structures are almost the same since they were all dimensioned for the same lateral sea pressure.

6.1.3.1 Barge

In Figure 6-4 the structural concept of one pontoon is shown for better understanding of scantlings and their positioning. Since the local strength requirements are driving scantling dimensions the chosen structural concept is the same in the mono hull body, which results in relatively similar structural concept around the whole barge. Detailed drawings of all sections are attached in Drawings.

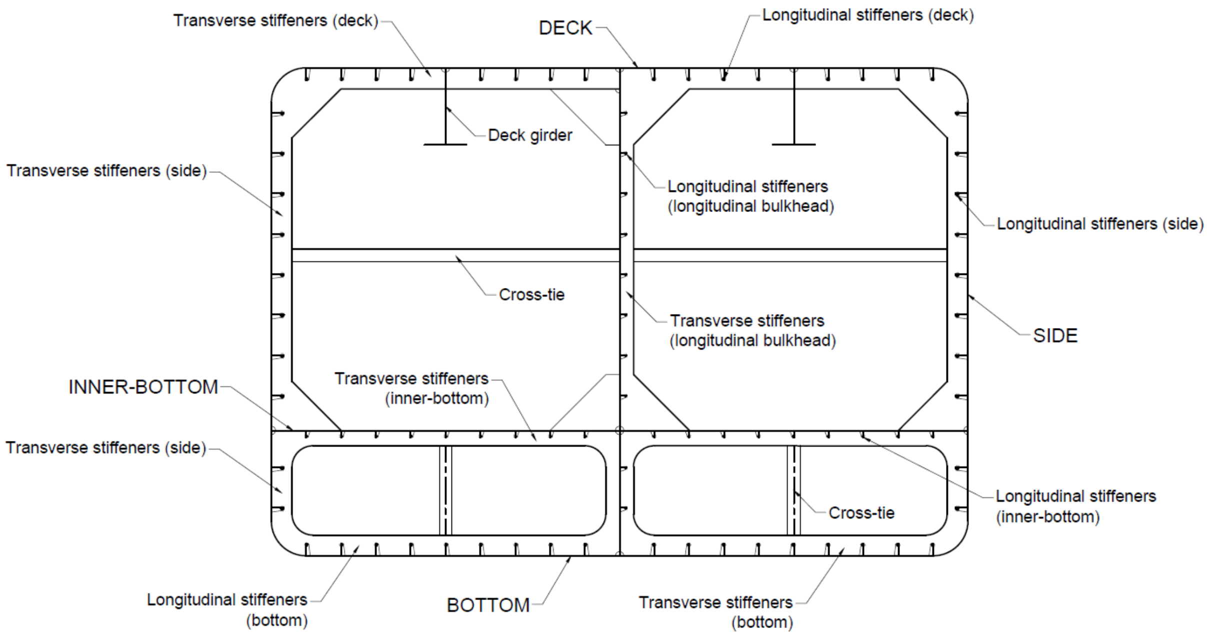

Figure 6-4: Structural concept of one pontoon part of the catamaran hull body

Bottom structures:

- Plate thickness: 16 mm
- Longitudinal stiffeners: Bulb flats: HP 180x9
- Transverse stiffeners: T-profiles: 300x14/180x20
- Cross-ties: T-profiles: 180x10/180x20

Inner-bottom structures:

- Plate thickness: 10 mm
- Longitudinal stiffeners: Bulb flats: HP 100x6
- Transverse stiffeners: T-profiles: 200x10/100x20

Side structures:

- Plate thickness: 16 mm
- Longitudinal stiffeners: Bulb flats: HP 180x9
- Transverse stiffeners: T-profiles: 300x14/200x20
- Cross-ties: T-profiles: 180x10/180x20

Deck structures:

- Plate thickness: 14 mm
- Longitudinal stiffeners: Bulb flats: HP 180x8
- Transverse stiffeners: T-profiles: 300x14/200x22
- Deck girders:
    - A-A, B-B (sides) and C-C (sides): T-profiles: 1100x25/600x40
    - A-A (frequency converter room): T-profiles: 500x14/250x40
    - B-B (in the middle: pump room): T-profiles: 700x20/550x40
    - C-C (in the middle): T-profiles: 1000x25/600x40

Longitudinal bulkheads:

- Plate thickness: 10 mm
- Longitudinal stiffeners: Bulb flats: HP 100x6
- Transverse stiffeners: T-profiles: 200x10/100x18

Transverse bulkheads:

- Plate thickness: 10 mm
- Horizontal stiffeners: Bulb flats: HP 100x6
- Vertical girders: T-profiles: 400x25/600x40

6.1.3.2 Vertical tanks

In Figure 6-5 the structural concept of the vertical tanks is shown for better understanding of scantlings and their positioning. Detailed drawings of all sections are attached in Drawings.

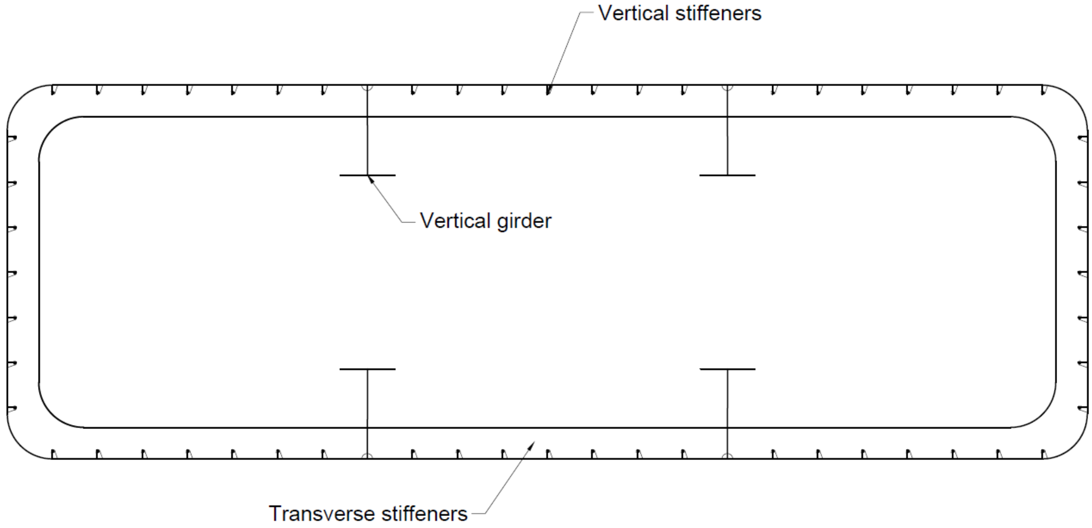

Figure 6-5: Structural concept of the vertical tanks

Section D-D:

- Plate thickness: 13 mm
- Vertical stiffeners: Bulb flats: HP 140x6.5
- Transverse stiffeners: T-profiles: 400x10/200x30
- Vertical girders: T-profiles: 1200x16/550x50

Section E-E:

- Plate thickness: 11 mm
- Vertical stiffeners: Bulb flats: HP 160x8
- Transverse stiffeners: T-profiles: 320x13/150x30
- Vertical girders: T-profiles: 800x12/450x40

Section F-F:

- Plate thickness: 9 mm
- Vertical stiffeners: Bulb flats: HP 100x6
- Transverse stiffeners: T-profiles: 260x10/140x20
- Vertical girders: T-profiles: 700x12/320x30

Section G-G:

- Plate thickness: 13 mm
- Vertical stiffeners: Bulb flats: HP 160x8
- Transverse stiffeners: T-profiles: 400x10/200x30
- Vertical girders: T-profiles: 1200x16/620x50

Section H-H:

- Plate thickness: 11 mm
- Vertical stiffeners: Bulb flats: HP 140x6.5
- Transverse stiffeners: T-profiles: 320x13/150x30
- Vertical girders: T-profiles: 800x14/500x40

Section I-I:

- Plate thickness: 9 mm
- Vertical stiffeners: Bulb flats: HP 100x6
- Transverse stiffeners: T-profiles: 260x10/140x20
- Vertical girders: T-profiles: 700x10/450x25

Horizontal bulkheads (at $Z = 9\ m$ and $Z = 16.5$ m up from deck):

- Plate thickness: 10 mm
- Horizontal stiffeners: Bulb flats: HP 240x9.5

Horizontal bulkheads (at $Z = 24\ m$ up from deck):

- Plate thickness: 10 mm
- Horizontal stiffeners: Bulb flats: HP 200x8.5

6.1.4 Bulkhead requirements

Bulkheads for the installation vessel are positioned according to ballasting requirements and according to the rules for damage stability described in Chapter 4.3.2.

In the barge there are 5 longitudinal and 8 transverse bulkheads in total. Three of the transverse bulkheads are designed and positioned as collision bulkheads according to rules (DNV, 2012). Two of them are positioned at a distance of 4.5 m from the aft perpendicular (catamaran hull part), at the other is positioned at a distance of 3.0 m from the forward perpendicular (mono hull part).

Each vertical tower has 3 horizontal bulkheads. Two of them are positioned in between different cross-sections and there is one more added to divide sections F-F and I-I (the top ones) due to reserve buoyancy requirements.

All of the bulkheads are dimensioned as watertight bulkheads.

6.2 Loading conditions

When dimensioning the structure, only critical loading conditions are considered. The most critical loading condition for both parts used for designing the structure is when the installation vessel is fully submerged. This is when the installation vessel is subjected to the maximum value of the hydrostatic pressure. The hydrostatic pressure depends on the depth, where the side structures are designed to withstand the same hydrostatic pressure as the bottom structures.

Dimensioning hydrostatic pressure values for the barge are equal to:

$$p_{B,deck\ structures} = 237.4\ kN/m^2$$

$$p_{B,side\ structures} = 306.0\ kN/m^2$$

$$p_{B,bottom\ structures} = 306.0\ kN/m^2$$

And for the vertical tanks:

$$p_{VT,D-D} = p_{VT,G-G} = 237.4\ kN/m^2$$

$$p_{VT,E-E} = p_{VT,H-H} = 149.2\ kN/m^2$$

$$p_{VT,F-F} = p_{VT,I-I} = 75.7\ kN/m^2$$

Furthermore, assumption that all tanks are empty and full lateral sea pressure is acting on the outer plates is made. Even though assuming there are no dynamic loads, i.e. quasi-static condition applies, the submerged loading condition is taken conservative as tanks will be filled with ballast when submerged.

6.2.1 Barge

Five different loading conditions are considered:

1. Ballast leg,
2. Transit mode with the largest/heaviest foundation,
3. Transit mode with the largest windmill,
4. Submerged to maximum design depth, foundation still connected (sinking),
5. Submerged to maximum design depth, foundation released, fully ballasted (surfacing).

Due to the shape of the installation vessel, i.e. mono hull combined with catamaran hull, the transition area between the two hull shapes, where the foundation load is acting, becomes critical with possible high stress concentrations. Therefore FE analyses for the transit loading condition with the foundation and for submerged loading condition are carried out in order to investigate the possible stress concentrations, see section 6.4.

6.2.2 Vertical tanks

For the vertical tanks the fully submerged condition is used for dimensioning. Three different structural cross-sections along vertical direction are identified for each vertical tank since sea pressure varies with depth.

Dynamic loads are considered as the area of the tanks exposed to significant wind and sea current loads.

The wind load should be taken into consideration due to the large size of the vertical tanks. Given their height of 28.5 m the moments caused by wind loads on the vertical tanks may have a significant influence on the scantling of the intersection with the barge structure. These moments are distributed further down to the barge's deck and bottom structures. Therefore, the vertical tanks are positioned so they coincide with barge's transverse bulkheads and transverse frames.

The wind load is calculated according to ABS rules for environmental loadings (ABS, 2011), since corresponding rules from DNV does not exist. The following assumptions are made:

1. Wind conditions are the same for all sections of the vertical tanks,
2. Wind velocity is uniform and the wind load acts horizontally,
3. The wind load effect can be calculated separately for the direction along and perpendicular to the vessel's length (X- and Y- axes).

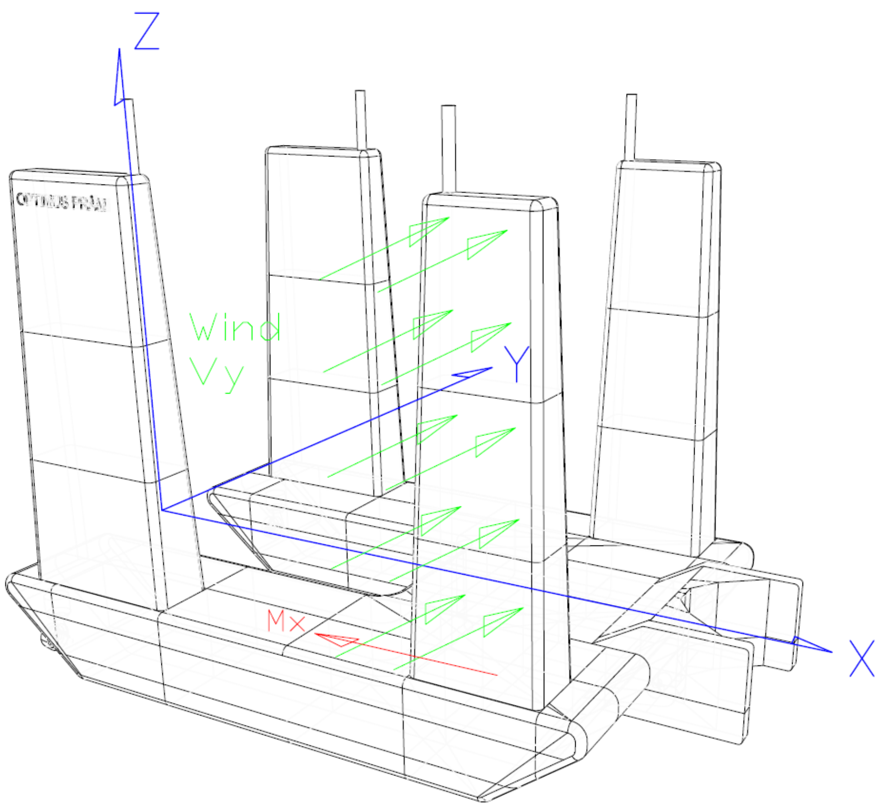

Figure 6-6: Wind velocity can be decomposed into two orthodox directions

The wind velocity is taken as a conservative value. Even though the installation vessel is to operate in sea state 3 (with wind speeds around 10 knots) the wind velocity is set to 50 knots, which is the minimum wind velocity according to ABS rules (ABS, 2011).

As mentioned before the bending moments act on the bottom of the vertical tanks and they are obtained as the integration of the wind loads on the vertical tanks. Therefore, since the wind velocity has been decomposed into two directions perpendicular to each other (Figure 6-6), the resulting bending moments can also be decomposed to moments acting around these two axes.

The largest bending moment due to wind pressure occurs when wind direction is perpendicular to the vessel's length (Y-axis), which results in the biggest moment around the ship's length direction (X-axis, i.e. M_x) (Figure 6-6).

Sea current load is calculated in similar approach as for the wind load with the same assumptions. Sea current velocity is conservatively taken as 5 knots (actual sea current velocity in the area is less than 0.5 knots). The biggest bending moment occurs when the sea current is acting perpendicular to the vessel.

Calculated moments and resulting stresses due to wind pressure and sea current pressure are shown in Table 6-3.

6.3 Longitudinal strength

In this section hull bending strength calculations for all described loading conditions are presented and hull buckling strength is considered.

6.3.1 Hull bending strength

To check hull bending strength still water bending moments and wave bending moments need to be calculated for different loading conditions. Still water bending moments are obtained from hydrostatic calculations (see Appendix B - Hydromechanics), while wave bending moments are calculated from DNV rules (DNV, 2011).

Calculated wave bending moments are equal to:

- Sagging: $M_w = -42.01\ MNm$
- Hogging: $M_w = 35.76\ MNm$

According to DNV rules the maximum allowable stress used in longitudinal strength calculation is equal to:

$$\sigma_l = 140 \cdot f_1 = 194.6\ MPa$$

Where special conditions (the installation vessel is being pushed) apply.

Longitudinal strength calculations (Table 6-3) for ballast, transit with foundation and transit with windmill loading conditions are calculated with still water and wave bending moments super-positioned. For other loading cases wave bending moments are neglected.

Table 6-3: Longitudinal strength calculations

Loading condition	Maximum bending moment [MNm]	Mode	Longitudinal position [m]	σ_{deck} [MPa]	σ_{keel} [MPa]	% of allowed stress (deck)	% of allowed stress (keel)
Lightship	32.82	hogging	26.44	4.69	5.07	2.4%	2.6%
Ballast	72.69	hogging	27.35	10.38	11.23	5.3%	5.8%
Transit, foundation	76.30	sagging	19.15	17.33	18.11	8.9%	9.3%
Transit, foundation	64.27	hogging	32.21	9.18	9.92	4.7%	5.1%
Transit, wind mill	57.32	hogging	30.55	8.19	8.85	4.2%	4.5%
Sinking, foundation	99.28	sagging	20.52	22.54	23.57	11.6%	12.1%
Surfacing, foundation	13.61	sagging	15.5	3.09	3.23	1.6%	1.7%
Surfacing, foundation	6.49	hogging	32.37	0.93	1.00	0.5%	0.5%
Wind load, D-D	2.94	N.A.	N.A.	2.20	2.20	1.1%	1.1%
Wind load, G-G	2.06	N.A.	N.A.	2.24	2.24	1.2%	1.2%
Sea current load, D-D	10.35	N.A.	N.A.	7.74	7.74	4.0%	4.0%
Sea current load, G-G	7.77	N.A.	N.A.	8.45	8.45	4.3%	4.3%

As it can be seen from Table 6-3 the longitudinal strength calculations are satisfied with quite a large margin. This safety margin is expected since the structural arrangement is driven by the local strength requirements (sea pressure when submerged). Since the installation vessel has rather unusual shape and a small length vs. breadth ratio, it is questionable whether beam theory still applies. This further motivates to study the structures using FE-Analysis.

Longitudinal strength calculations are taking only bending moments into consideration without torsion. Large torsional loadings in the transition area between catamaran and mono hull are expected when the vessel is introduced to wave environment. Therefore, the longitudinal strength should be further investigated and the calculated safety margin is not a fully dependable measure.

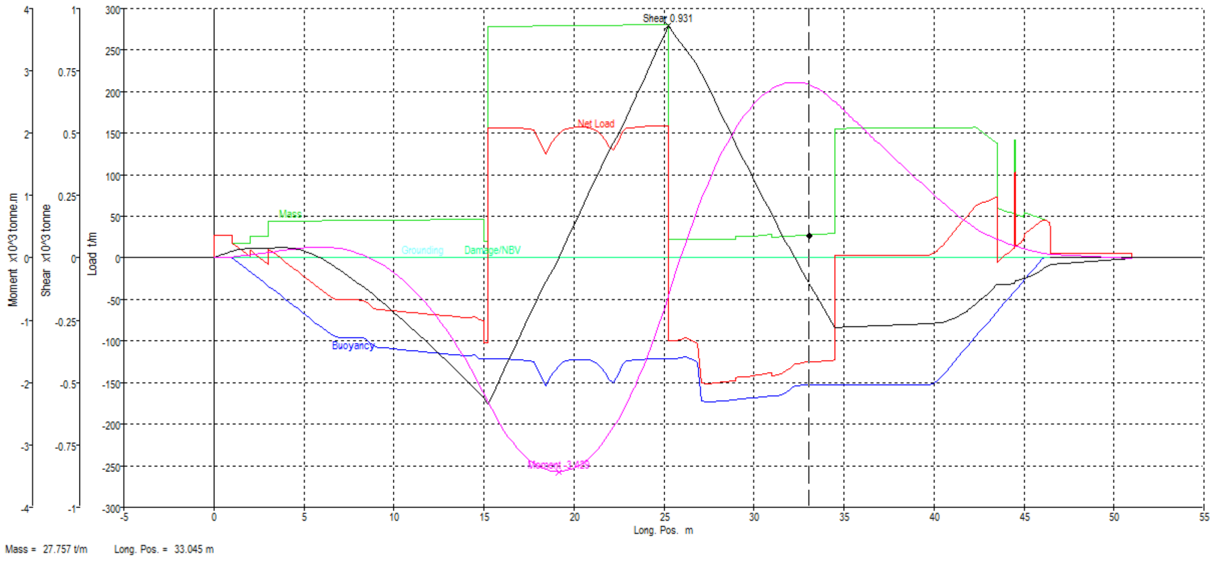

Figure 6-7: Longitudinal strength for transit mode with the heaviest foundation (still water)

6.3.2 Hull buckling strength

DNV rules for barges do not address any required buckling controls to be performed. Additionally, stresses on deck and keel due to the longitudinal bending moments are relatively small (i.e. below 15% of the maximum allowable stress). Therefore, buckling controls are not performed for the installation vessel.

6.4 FE Analyses

The point of interest in the FE Analyses is the structural behaviour of the recognized hot spots, i.e. the moon pool corners. In the present case there are a few complications that is clarified in the following sections.

The driving factor for the structural design of the installation vessel is the class society rules and calculations. Most of such calculations are based on deterministic approaches of classical mechanics as such, the beam theory. Considering the dimensions of the installation vessel, the global design cannot be considered viable to be approached as a beam structure. Disregarding this aspect might cause problematic structural reactions. To achieve results of as accurate results as possible, a full-body model is studied under most excessive loading cases and boundary conditions.

The FE geometry model includes all major load carrying structural components including hull plating, bulkheads, girders, longitudinal, transverse and bulkhead stiffeners. The only negligence is in case of the cross-ties which is explained below. Although the hull shares the same main dimensions with the original design, it is simplified to a model with no bilges or curvature. Simply put, a box shaped structure, in order to avoid complications and longer calculation time. It should be pointed out that although different in shape the structural components are fully consistent with the main design.

Loads included in the FE Analyses are:

- The largest/heaviest foundation,
- The filled ballast tanks,
- The buoyant forces and
- The structure's weight.

Although it is preferred to undertake the analysis with more details and a variety of boundary conditions, given the time constraints, only two cases with most extreme loading conditions are considered:

1. Transit of the largest foundation,
2. Fully submerged to maximum design depth with the largest foundation.

As for boundary conditions two different boundary conditions are considered. One case includes extreme loading of the structure by simply supporting the vessel on both ends. It is worthy to point out that all simulation attempts include buoyant forces which provide more realistic results and structural responses. The more realistic boundary condition is when only 2 nodes are fixed in degrees of freedom as follows: one is fixed, while the other is only fixed in the vertical translation direction. In both cases negligible effects of boundary conditions is desired. To satisfy the symmetry of the vessel, symmetrical boundary conditions are applied to the relevant region.

6.4.1 Limitations

The most limiting factor in the FE modelling process of installation vessel is time. This factor affects many aspects of the modelling in the following ways.

- The applied approach limited the use of bulb flats in the structure which caused the use of plates instead. To lessen the effect of this geometrical inconsistency, the plates are defined so that they have the same cross-sectional moment of inertia.
- To simplify the geometry modelling process the body was designed and assembled in a CAD software and then imported to the CAE software. In this approach and due to limited time it was decided to use default connections between the structural elements. Doing so the nature of this connection is not investigated to see if they fulfil the known boundaries between the structural components. This may be a source of errors in the results.

6.4.2 Assumptions

The following assumptions are made:

- The box-shaped bow and stern and sharp cornered geometry instead of bilges may cause stress concentration in these areas.
- As the hull shape is symmetric here only half the hull is modelled.
- Considering the fact that the vessel is only designed for approximately 700 foundation transport operations and the fact that only a few of these 700 are the biggest ones the chances of low cycle fatigue occurrence is considered to be low and therefore neglected.
- In order to provide a more realistic result set and to simplify the model, the buoyant forces are assumed to be equally distributed on the bottom plating.
- The foundation load is evenly distributed over the area where the frame fixture will be placed.
- The simulation is undertaken for calm sea state and under quasi-static condition, i.e. no dynamic loads.
- All components but major structural are disregarded.
- The yield strength of the used steel is 355 MPa and modelled as a linear material.
- It is assumed that the cross-ties do not make much contribution in structural integrity in transit mode but have much effect while submerged.

6.4.3 Results

Here the results of the following three cases will be mentioned:

- Structure, simply supported on both ends in transit mode of the largest foundation
- Structure, supported on two nodes aft and fore in transit mode of the largest foundation
- Submerged to maximum design depth, installation mode of the largest foundation

All relative figures not included in this section are available in the Appendix D - Structure.

6.4.3.1 Simply supported structure on both ends in transit mode of the largest foundation

In this case the structure is simply supported at both ends (limited translation and free rotation). This case represents an extreme condition of a wave with half-length equal to length overall.

The deformation, maximum principal stresses, maximum shear stress and stress safety factors are well in the expected margin for both plating and interior components.

As can be seen an acceptable maximum deformation of 10.216 mm is achieved from the analysis. It should be kept in mind the said max deformation is acting on some of the internal components and bottom plating. By observing the two following figures the position for maximum deformation can be found.

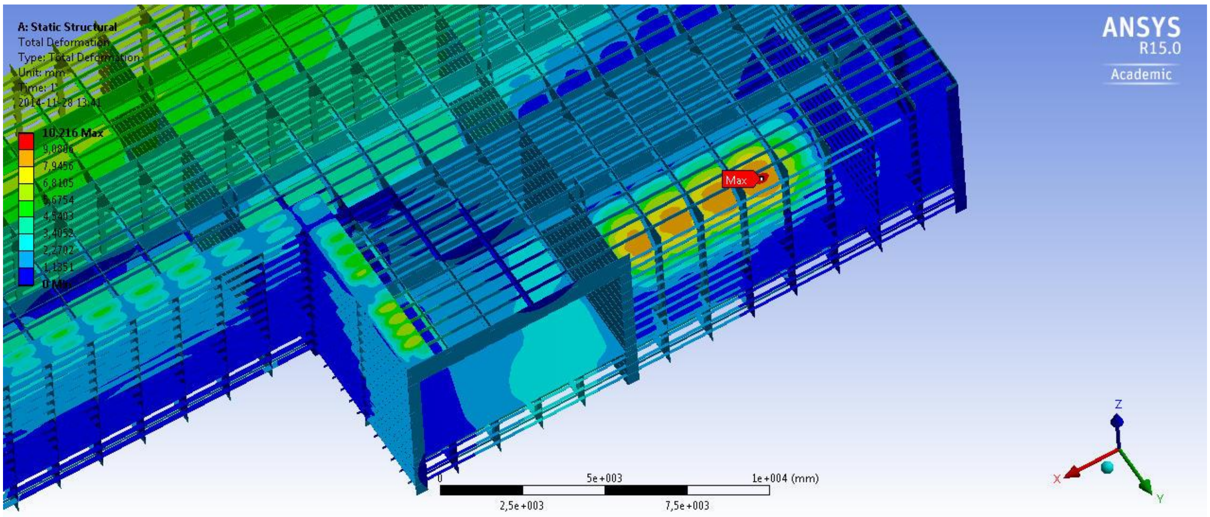

Figure 6-8: Maximum deformation on the inner bottom just fore of the pump room bulkhead

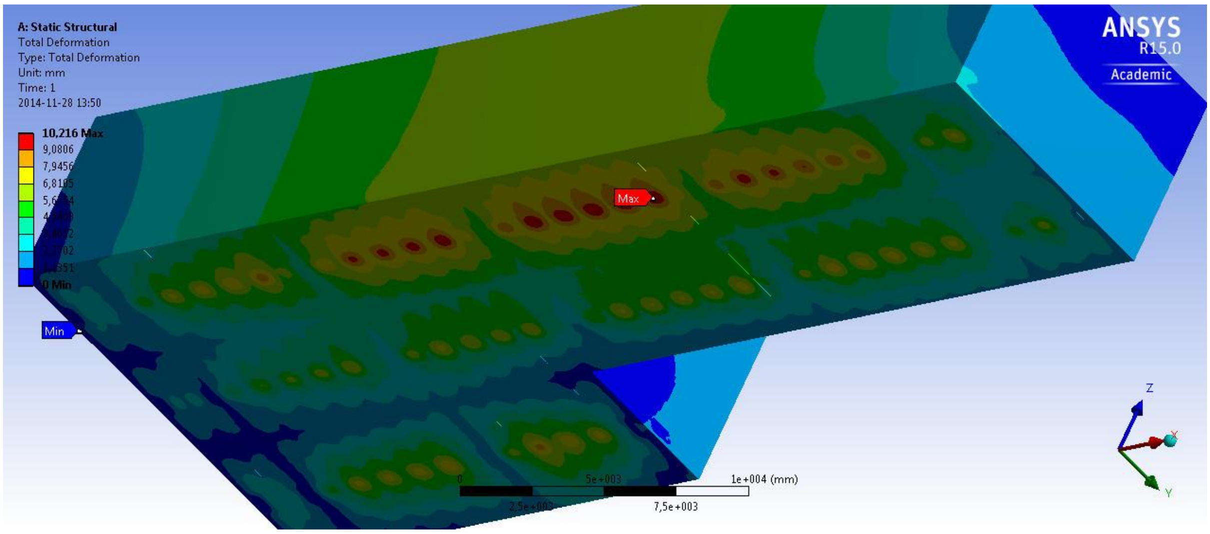

Figure 6-9: Maximum deformation of the bottom plating due to hydrostatic pressure

The only anomaly from this loading condition is the failure of the one longitudinal stiffener on the inner bottom just fore of the pump room bulkhead. Due to extreme loading and support conditions and the small margin to failure of the said component it is decided that in this case failure is negligible.

6.4.3.2 Structure supported on two nodes aft and fore in transit mode of the largest foundation

The reasoning behind changing the boundary conditions of the model with the previous analysis is to let the structure have more freedom of movement as a hole hence experiencing a more realistic support condition.

The only matter here as will be illustrated is the stress concentration in the fixed nodes which is the global maxima. The appearance of such loads in almost stress free fields shows the inconsistency of the model with reality. Therefore it can be neglected as the stress development field around these nodes does not interfere with hot spots and points of interest.

Here, a more deformed body is the result of the analysis.

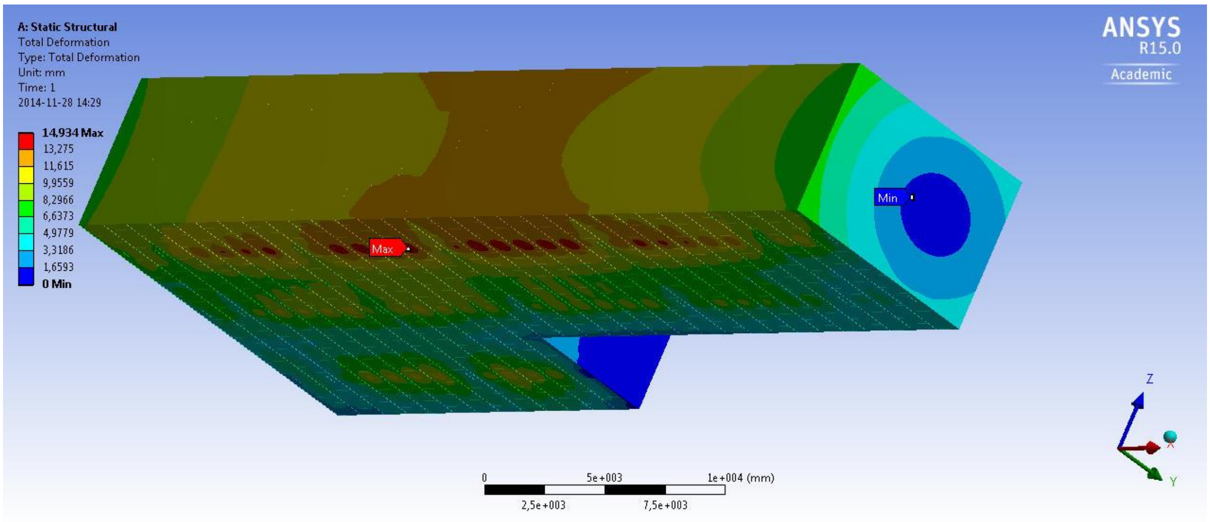

Figure 6-10: The side view of the structure deformation

As no specific and concerning deformations or stresses are measured in this point of the study, further visual representation was not deemed necessary. The only point of some importance is the extreme high local stresses and extreme deformation fluctuation around the fixed nodes which can be disregarded. The reason behind such decision is the limited development of the said stress and deformation fields.

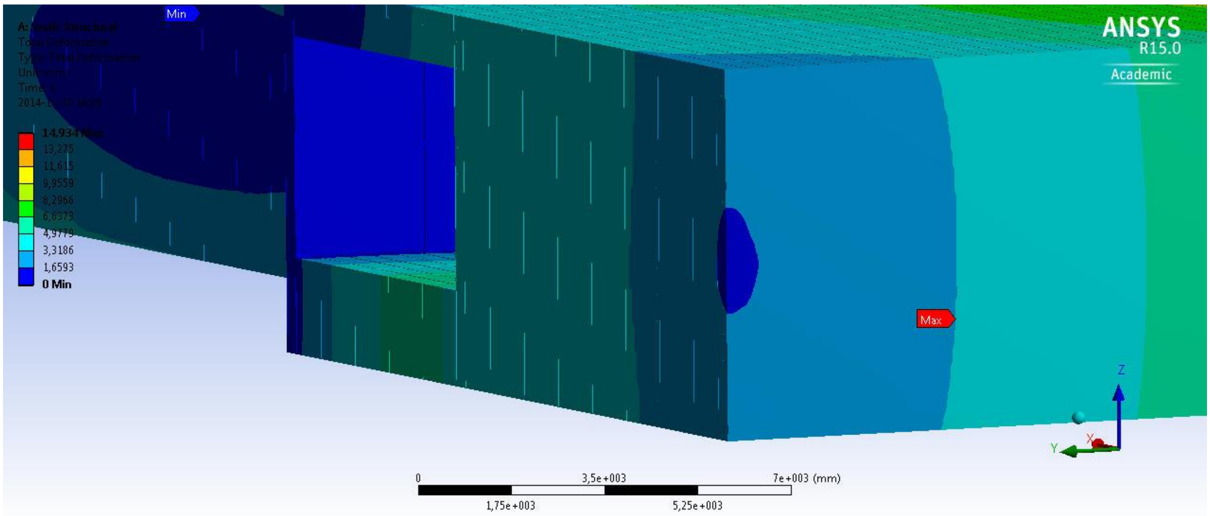

Figure 6-11: The fixed node foremost of the hull

6.4.3.3 *Structure submerged to the maximum design depth*

In this analysis the driving factor for the design of plate thicknesses, the hydrostatic pressure on the submerged body, is on trial. Considering the overlapping schedule for design of the cross ties and modelling the geometry the cross-ties are not included in the model, which is the reason for structural failures. As can be concluded from the previous cases the structure's strength is higher than needed in transfer mode. This case is to illustrate the structural response in absence of cross-ties.

As can be seen in Figure 6-14 higher than yield strength stresses on frames and longitudinal stiffeners are present. The reason why is the misrepresentation of the model by disregarding the cross-ties. These failures are well within expectation and the aim here is to make sure of the integrity of hull plating.

The highest deformations are experienced for this load case, especially on the hull plating in this task, which is due to the oversight of the cross-ties in the model.

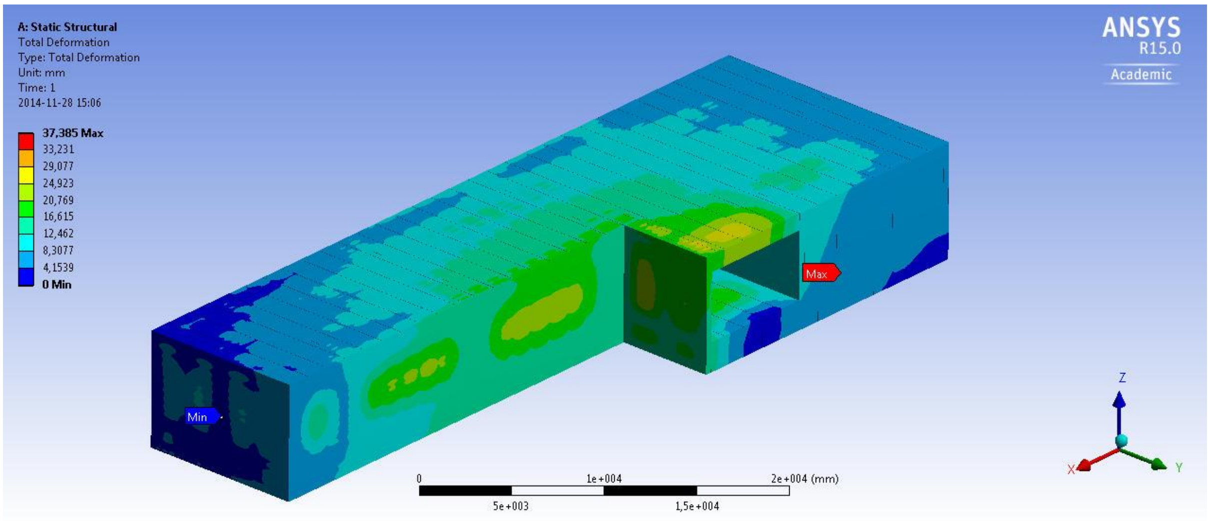

Figure 6-12: Deformation of the hull plating under hydrostatic pressure

From Figure 6-12 and Figure 6-13 it can be seen that a similar deformation pattern compared to the previous analysis (6.4.3.2) occurs. This is the result of using the same boundary conditions in these two cases.

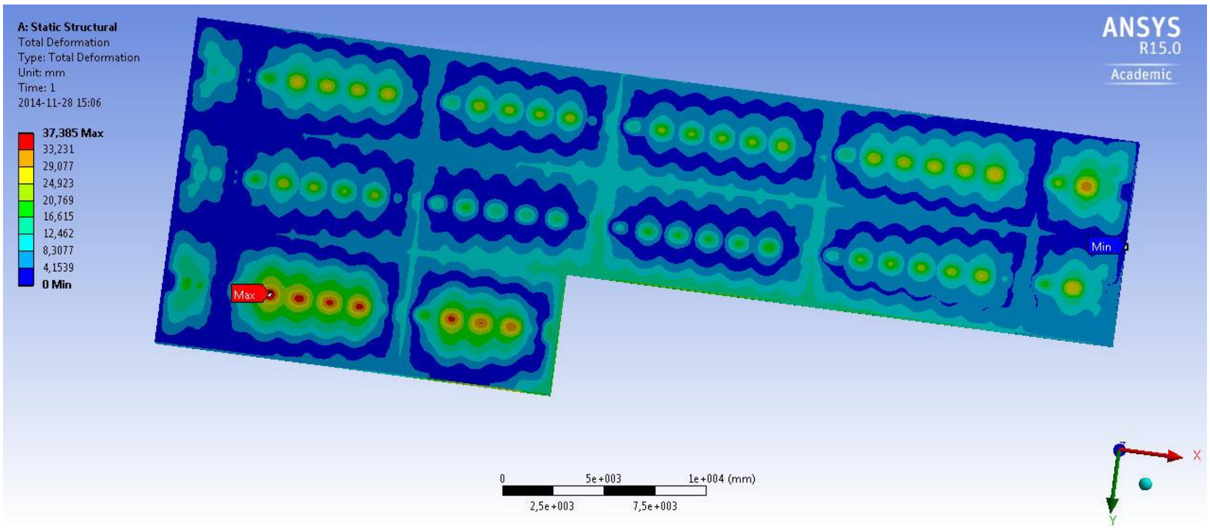

Figure 6-13: Deformation of the bottom plating under hydrostatic pressure

Figure 6-14 shows the deformed frames and interior structural components, which go through excessive deformation and over yield strength stresses.

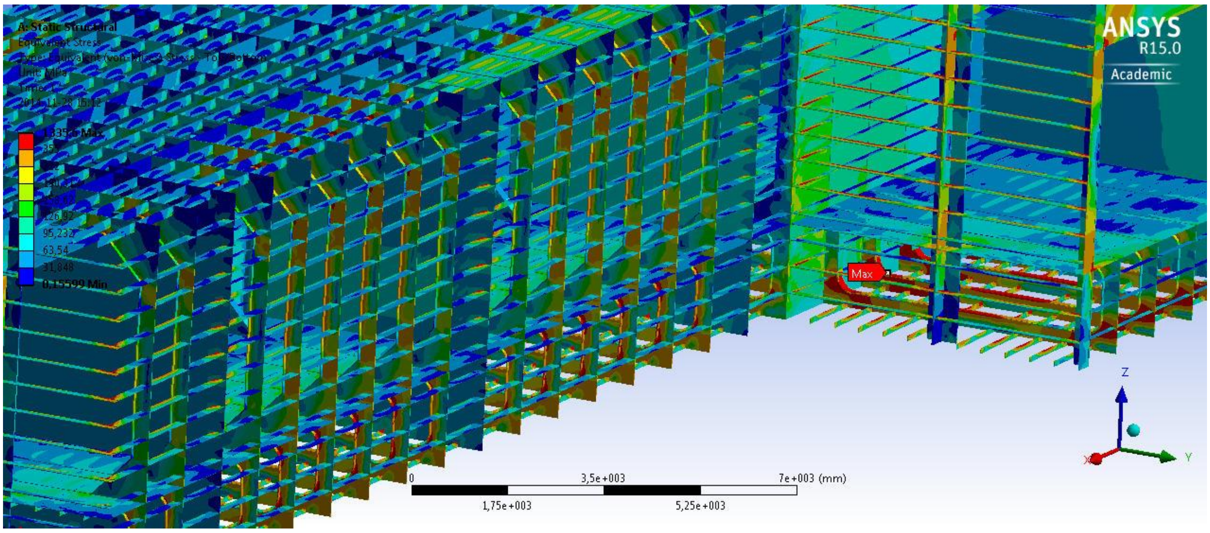

Figure 6-14: A closer representation of the mentioned components

6.4.4 Discussion

As a rule of thumb, the element length should not be higher than 5 times the thickness (Hogström, 2012). For most components this criteria is satisfied except for the flange of the web frame girder. In regards to this matter to be sure of the result consistency the deformations and stresses in the area are investigated. The achieved results show no anomalies or out of ordinary behaviours. Another point of interest relevant to a proper mesh is the element shape. As a rough guideline the element aspect ratio of higher than 3 should be treated with caution and aspect ratios higher than 10 should cause alarm (Felippa, 2004). In this model such elements are rare and the results show no inconsistencies with the circumferential elements.

As brought to attention before, the modelled hull geometry has inconsistencies with the original design. Although, based on assumptions, this should not be considered a flaw as it is a conservative approach.

To simplify the modelling and analysis process, only the major load carrying structural components are considered to contribute to the structural integrity of the vessel. Doing so the effects of some components such as tank structures, weight of machinery components, the vertical ballast tanks' structure weight etc. are neglected. This might cause inaccuracy of results compared to a more detailed model.

As presented in the results section 7.4.3, in order to fix the vessel in its place a few nodes are used. This is done to prevent the model from moving in co-ordinate system due to unbalanced loads on top and bottom. This approach causes the visual presentation of hot spots on those nodes. Considering that these effects are small and do not reach much into the structure they are negligible.

As mentioned in the results section of the present text in case of hydrostatic pressure while submerged to maximum depth some failures appear this event is due to misrepresentation of the model by not including the cross-ties. The cross ties strengthen the mid-section between longitudinal bulkheads, in both horizontal and vertical directions which in the real structures avoids occurrence of such failures.

In the present case, the analysis is quasi-static, which does not address the dynamic loading and structural response of the structure in case of e.g. wave loads. But considering the results of the present FE model and the fact that the driving factor of the structure design is the class regulations it is safe to state that the structure will respond well to any loading conditions.

6.4.5 Conclusion

Based on the achieved results and including the calculated safety factors in points it is proven that:

- In the moon pool corners (structural hot spots), the main point of concern in the design process, no extreme deformation or stresses are found
- Considering very low deformation rate in all components the next step is weight optimization of the structure in agreement with class rules and regulations

6.5 Lightship weight distribution

Lightship weight distribution (Figure 6-16) shows distribution of steel weight, machinery and fixture weights in longitudinal direction. Weights of frequency converters, thrusters, electrical motors, pumps and batteries are taken into account; weight of piping is not included in the distribution.

Steel weight distribution (Figure 6-15) is calculated separately for the barge (Table 6-4) and for the vertical tanks (Table 6-5). Weights of welds and connecting brackets (intersection of elements) are not included. On the other hand, weights of structural elements are calculated in a conservative matter. Weight of overlapping elements (e.g. frames in corners) is not deducted and box-shaped cross-section is assumed in the calculation.

Density of steel used in the weight calculation is equal to:

$$\rho_{steel} = 7850\ kg/m^3$$

Total calculated weight is shown in Table 6-6.

Table 6-4: Weight calculation by structural elements (barge)

	Longitudinal stiffeners		Frames	Bulkheads
Section	Weight [kg/m]	Weight [kg]	Weight [kg]	Weight [kg]
A-A	14,563	371,353	81,365.96	31,322
B-B	19,076	171,688	39,512.19	21,101
C-C	19,129	229,548	47,003.76	47,477
	Total [kg]:	772,590	167,882	99,899

Table 6-5: Weight calculation by structural elements (vertical tanks)

	Longitudinal stiffeners		Frames	Bulkheads
Section	Weight [kg/m]	Weight [kg]	Weight [kg]	Weight [kg]
D-D, E-E, F-F	19,629	235,545	63,152	28,147
G-G, H-H, I-I	19,656	176,905	51,311	21,110
	Total [kg]:	412,451	114,462	49,257

Table 6-6: Total weight

Barge [t]:	1,040.4
Vertical tanks [t]:	576.2
Total [t]:	1,616.5

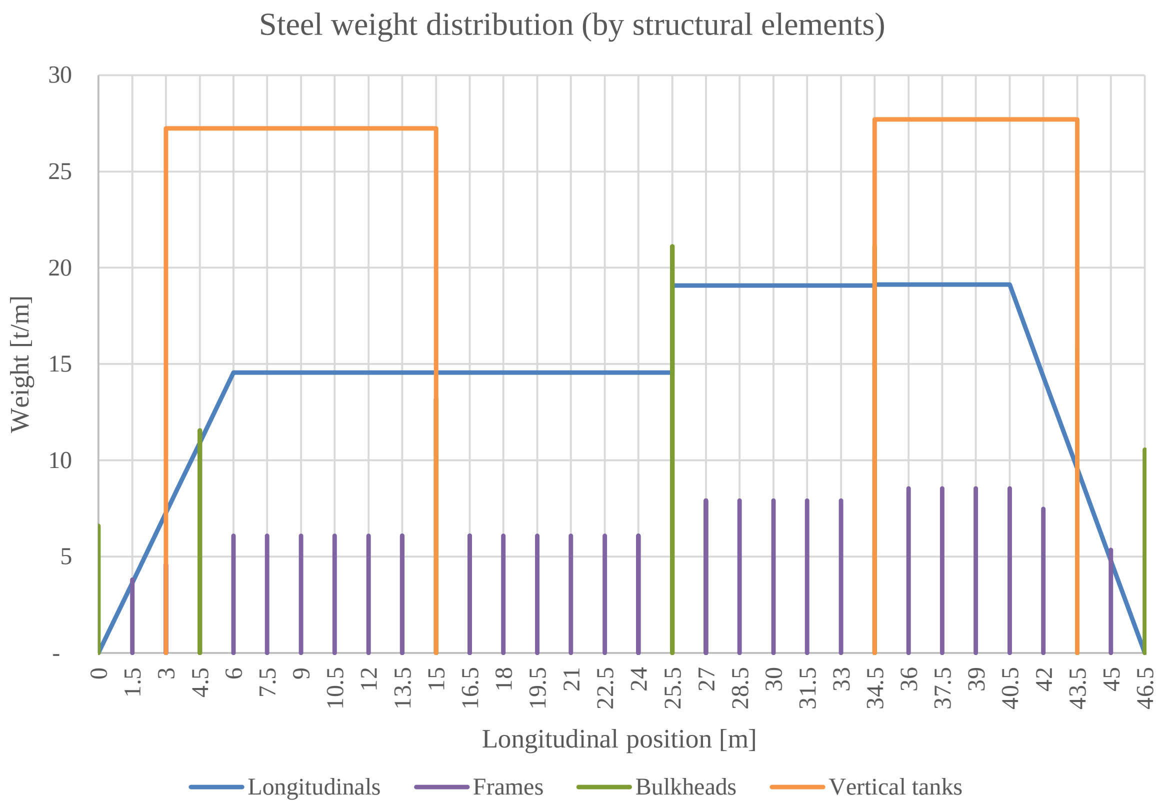

Figure 6-15: Steel weight distribution (by structural elements)

Figure 6-16: Lightship weight distribution

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7 Future work

7.1 General Arrangement

The size and design of the gravity foundations are not specified explicitly from the customer. Thus, the foundation design from a similar project, a wind farm outside Lillgrund, was used. Since the size of the foundation is essential for the vessel design this estimation will have great impact on the end performance.

Due to time limitations most focus is put on designing the installation vessel why the support vessel needs to be further investigated. Engines, lifting appliances and connection part between installation and support vessel are analysed and defined but to optimize the performance they must be more thoroughly evaluated.

The size of the crew is based on assumptions and may change, why the deck plan and accommodation lay out is likely to change as well. Equipment for navigation and communication is chosen in accordance with class rules and additional equipment may be required to optimize the navigation. Only a simple risk analysis with the most basic risks is made and further investigation can result in a need of updated deck plans and additional life-saving appliances. Life rafts and life boat locations and attachments are not looked into in detail.

The current time schedule is based on a number of assumptions; the vessel is able to maintain a certain speed during transit, installation and assembly. One customer requirements is that the installation time for one foundation or wind turbine is limited to twelve hours and the design is adapted to this. If this time overrun there will be a need for sleeping cabins on the support vessel.

One of the main motivators for this project is the economic aspect and the potential investment saving of a new concept for wind farm constructions. A brief estimation is made, based on material cost, crew cost and running cost. In order to motivate the importance of this project, more thorough calculations have to be made based on a larger number of cost parameters. Also, one key advantage compared to other wind farms is the calmer weather at the Baltic Sea, but since there was no wind and wave data available, it was difficult to take advantage of this fact.

7.2 Hydromechanics

Damage stability should be investigated further because it was not possible to cover all damage cases possible with this limited time and resources. Probabilistic damage calculations can be done for an extensive assessment.

Since in transit mode, two vessels are coupled but not fixed to each other for 6 degree of freedom. If two vessels were modelled in a way that only heave and pitch motions are allowed, stability and sea keeping assessments would have been more accurately calculated.

Since the shape of the installation vessel is different from a normal “ship-shape” it will be necessary to perform model tests to determine the resistance of the installation and support vessel. Especially the interaction between the two vessels needs to be investigated further but also the influence of the wave resistance at higher Froude numbers is of interest. The wave resistance at high Froude numbers is probably underestimated as it is now.

Model test is highly recommended as a future extension of this work. Also analysis with software's that can handle this odd hull shape is preferable. Damping coefficients also needs to be sought for in order to perform a more true analysis. It is also necessary to investigate how the sea keeping will be affected when the support vessel is connected to the installation vessel.

There are also other operational cases that could be investigated. For example transit with foundation, installation of foundation, cases with different sizes of windmills/foundations.

7.3 Machinery

The engine arrangement on the support vessel has to be investigated further from a human factors perspective, to make sure that noise and vibrations do not exceed allowed limits.

The maintenance of both vessels has to be defined so it can be performed as efficiently and safely as possible. As stated in section 6.5, the electrical balance serves as a rough overview of electrical consumers of both vessels and needs to be corrected for actual values.

The estimation of head loss throughout the system on the installation vessel only serves as a rough overview since an accurate number of components still are unknown, which implies that further investigation is needed. The pump system with auxiliary system also needs to be tested, due to the accurate amount of pipe components are unknown at this time, in the later stages of design, additional details of pipe line can be determined, rendering a more accurate result.

An active ballast system on the installation vessel can reduce the wave motions and ensure safe installation operations and transit to site. Although it has not been studied during this project, it is something that should be looked in to during the next design loops.

7.4 Structure

For future work the structural design of the installation vessel can be optimized regarding both weight and set up of structural elements. The weight of the structure can be optimized by investigation of alternative element spacing and dimensions.

For now the installation vessel is assumed to use steel with yield strength of not less than 355 MPa in all structural elements in the barge. By using a combination of high strength and normal steel there might be a possibility for slight improvement in steel costs.

Our structural work has mainly focused on the installation vessel. Future work should include a structural analysis of the rotating lifting arms as well as the interface for the support vessel. Although some preliminary design and analysis efforts to develop the lifting appliance for foundation was carried out further improvements are necessary.

There are still some aspects of the structure which have not yet been investigated such as fatigue and vibrations. Vibrations are assumed to impact the windmill most of all, which is why the rotating lifting arm should also be investigated for installation of vibration compensating devices. Further on it should be made sure that eigenvalue frequencies of the cross-ties and wind mills are not resonant with the installation vessel.

For further more reliable FE analysis a more accurate buoyant force distribution and magnitude is necessary. Considering the time window only two most extreme cases were analysed. Simulating other cases may make further weight optimization possible.

The quasi-static approach of the FE analysis neglects the motions caused by dynamic loading of the vessel. Knowing that these loading conditions have their most effect on the pontoons it could be a point of interest to study the effect of such loads.

To achieve more realistic results of the FE analyses the effort should be put on minimizing the reaction force of the supports. If time is given there are solutions to this matter. The first step is a comparison between the magnitudes of the forces around the supports with shear forces away from them. Doing so, the shear force magnitude caused by introducing unrealistic support can be determined. Then it is possible to decide if these forces are negligible or not on points of interest in the study.

In case the effect of these misrepresented conditions is more than negligible the following solution can be implemented:

- Adding a linear shear load equal to the shear forces caused by support,
- Introducing a linear load causing a shear distribution equal to shear load of supports.

Finally the finite element analysis needs to be improved by implementing a full model including all the structural details. The FE analysis is showing expected results of stress concentrations in the transition area between the mono hull and the pontoons, which should also be the focus of an extended analysis.

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