Aalto University School of Engineering
Department of Mechanical Engineering
Marine Technology
MEC-E2011 – Ship Design Portfolio

Khione

Final report

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Executive Summary

The purpose of this design project was to design a supply and research vessel for use in the arctic. The ship must be able to sail through 1.65 m thick ice at 3 knots, have 500 m^2 of space available for laboratories and offices and feature 2 medium size helicopters. The propulsion must be diesel electric with fixed pitch propellers and provide a propulsive power of around 31 MW and the ship must be able to sail semi-autonomously. The semi-autonomous operation is what sets this ship apart from other vessels. There are autonomous ships under development, like the MV Yara Birkeland, but there are no (semi-) autonomous ships currently in operation. With this autonomy, crew size and with the cost of shipping can be decreased. It was also decided that the ship will use hydrogen as an auxiliary power source.

At the start of this project, the design requirements, parameters and limits were defined based on the ship's mission. With this information, the overall dimensions of the ship were determined using refence data and Normand's method, the results being an overall length of 127 m, a breadth of 22 m, a draft of 7 m and a displacement of 11896 tonnes. Given the overall dimensions, line drawings were created as well as a CAD model using Delftship. These figures were then revised using NAPA. In the new iteration the displacement was slightly lowered and the corresponding design draft became 6.4m. The structural design is based on regulations from the Polar Code and DNV.

The financial analysis revealed that this ship will require government support in order to operate, as the cargo being carried is limited. This is however normal for a scientific vessel and the ship is in itself a prototype for semi-autonomous and hydrogen technology.

Future stages of the design will have to focus more detailed structural and hydrostatic design. Hopefully this project can function as a baseline for future special purpose icebreakers.

Project Team

Marcus Fagerlund

I officially started my master's studies this autumn even though I took some master's level courses the year prior. During my master's studies I want to focus on the structural side of shipbuilding, but when possible, I also want to take courses that widen my understanding of shipbuilding.

My previous experience comes from my bachelor's studies here at Aalto and working experience from different summer jobs. My bachelor's thesis tried to predict changes in the stresses ships experience because of climate change. I would not say I was entirely successful, but it opened my eyes to the challenges faced by engineers working with related questions. My working experience comes from two of my summer jobs. One was as a ticket salesman onboard a small archipelago ferry and the other one was in the hull design department of a shipyard. These two jobs have given me a good perspective on the differences between design intention and actual usage of ships.

My personal development target for my professional life is simply to be the best engineer I can be. For me this means being good at something, in my case the structural side, and knowing enough about a wider subject to know how it may impact your specialty.

Sanna Granqvist

I started my master's studies in autumn and before that my experience in the field was from my bachelor's thesis and summer work at Helsinki shipyard. The courses in the autumn gave good basic knowledge of shipbuilding which will be helpful for both studies and work. My bachelor's thesis was about ''future energy sources for shipping'' which gave a good insight to used energy sources and opportunities for new ones. Continuing on the energy theme, I did a study on energy saving devices at the summer work.

As I'm not sure where in the shipbuilding process I want to work in the future I try to get different courses to get good knowledge on ships overall, and courses good for the project engineer path. At the moment I could see myself working with different machinery or propulsion solutions or a ship design department. Interlinked courses for this course are PNA, Ship stability and buoyancy, Marine and ship systems engineering (report left to spring) done in autumn, and at the moment Ship hydrodynamics.

As my personal development targets, is to get more knowledge on the overall ship design process. I hope to learn much from the NAPA modelling since I enjoyed 3D modelling courses in bachelor's studies.

Oskar Veltheim

I have experience on ships and shipbuilding from school, work and LRK activities. I did my bachelor thesis about calculation methods for ice loads on ships on last summer and started master studies in this autumn. I have now worked over a year in Helsinki Shipyard and I worked one summer in Meyer Turku. In both shipyards I worked and still work in project departments. I have been active in LRK and now is my third year in board of LRK. In LRK I have learned about marine field in excursions and other events with students and experts of the field.

I am interested in ice-ship interactions. I chose Arctic Marine Technology -study path and I have chosen courses accordingly. I have planned taking courses about ship structures and Arctic Technology -courses (MEC-E400X). In future I want to work with icebreaking vessel design.

I want to gain more knowledge over ice-ship interactions. I am especially interested about ice loads and damages to ships due ice and arctic marine safety. More I learn about ice as material I get more interested about it. Also, I want to get better using Matlab. It is used in many courses and I think while studying is great opportunity to train such skills. Finally, I want to learn time and personal resource management as I have a lot of intensive courses, work and LRK board duties. These skills are extremely useful in work life.

Juhan Voutilainen

I completed my bachelor's degree in mechanical engineering, so I have basic knowledge of the physical phenomena in the field of marine technology. My bachelor's thesis was about environmental regulations of ships operating in the area of the northern sea route. I have gained additional maritime knowledge especially about ice-going vessels by working for past year in the company with focus in research and design of arctic vessels. Last autumn I started my master's studies in marine engineering. Since that I have gained more in-depth knowledge about basic concepts in marine engineering.

I am interested in hydrodynamics of different marine vessels. Because of that interest, I am aiming to achieve expertise in the field of hydrodynamics. My plan to reach the expertise is more or less by following the hydrodynamic expert -study path. Study path includes courses like Fluid dynamics, Computational Fluid Dynamics, Computational Ship Hydrodynamics, etc. Additionally, I have been planning to further diversify my future expertise by studying the topic also at exchange university.

Main personal development target kind of already came up in previous paragraph. The target is to become expert in my own field. In smaller scale, one of my targets is to become more fluent with computer aided tools. During all my studying career, for some reason, I have been avoiding all computer aided tools. Now, I am starting to notice that avoiding them is not anymore possible, so mastering them is one of my intermediate personal developing targets on the way to the expertise in my field.

Contents

Nomenclature

Abbreviations	
AB	Arctic Bridge
AUV	Autonomous Underwater Vehicles
DWT	Dead Weight Tonnes
DWTC	Cargo Weight
DWTC&E	Weight of crew and their effects
DWTH	Hydrogen weight
DWTPR	Weight of provisions
EN	Equipment number
HFO	Heavy Fuel Oil
MDO	Marine Diesel Oil
MHS	Required number of Man Hours
NPV	Net Present Value
NWP	North West Passage
NSR	Northern Sea Route
PC4	Polar Class 4
PSV	Platform Supply Vessel
ROV	Remotely Operated Underwater Vehicles
SAR	Search and Rescue
SFI	
SOLAS	International Convention for the Safety of Life at Sea
TSR	Transpolar Sea Route
WPA	Water plate area

Symbols	
A	Area of profile view of the hull
B	Greatest breadth
C_B	Block Coefficient
C_M	Midship section area coefficient
C_P	Prismatic coefficient
C_{ST}	Hull structure cost
D	Depth
H	Height from the summer load waterline to the top of the uppermost deckhouse
K_{MAN}	Man hour cost
K_{STEEL}	Cost of steel per ton
L_{PP}	Length between perpendiculars
P	Power
T	Draft
t_{net}	Shell thickness
W_E	Lightship Weight
W_M	Machinery Weight
W_O	Outfitting weight
W_S	Structural Weight
Δ	Displacement
μ	Mean value
ρ	Density

1 Design Context and Mission

1.1 Design Mission and Objectives

The mission of the team is to design a safe, reliable, and efficient research and re-supply vessel for use in Arctic waters. The ship will operate independently in extreme conditions and will provide supplies for people who completely rely on it. Thus, the reliability and safety of the ship are highly prioritised in this project.

The efficiency of the research and re-supply operations will be guaranteed by providing top level laboratories, underwater research capability, helicopter hangars and most importantly by making the ship capable of operating in thick ice. Additionally, the ship will be designed to have capability for semi-autonomous operations. Our ambition is to create an innovative design which will bring new solutions to the industry and show the way for future ships performing operations in ice-covered waters.

1.2 Design Variables, Innovations and Boundaries

The vessel must be able to operate in arctic conditions. The route that the vessel will travel may vary, but it will be in the Arctic. The extreme environment sets requirements for the vessel such as icebreaking capability up to 1.65 m thick ice while maintaining the sailing speed of 3 knots. The propulsion system should be able to produce a propulsive power of about 30 MW, divided over two diesel electric propellers with fixed pitch. Furthermore, the ship should feature bow thrusters for maneuverability. The advantage of diesel-electric propulsion over a diesel engine is that diesel-electric propulsion can provide maximum torque at all speeds (Wärtsilä, 2016), which is ideal for an icebreaker. Icebreakers generally sail at slow speeds, but they still need to apply a lot of force to get through the ice.

On the vessel, there must be science laboratories and offices, which are 500 m^2 in area combined. About half of this space (250 m^2) will be offices, each office being 10 square meters, shared by 2 people. There will be 80 researchers. As for the crew, an average crew size is 25 people (Deloitte, 2011), but because this ship will be semi-autonomous, the ship crew is expected to only be 20 people, 6 of which are licensed helicopter pilots, 3 helicopter mechanics and 12 perform the ship's operations. Every 2 crew members share one 10 m^2 room, with the captain having his/her own room.

Research Onboard

The laboratory will have the latest technology that is needed for high-quality research on biological, chemical, or physical oceanography as well as paleoceanography. For underwater research, there will be different sensor arrangements for hydrographic survey and oceanographic research. The vessel will have both multibeam and single beam echosounders for determining the water depth and mapping the seabed. There will also be a towed side scan sonar, i.e. a towed vehicle that is equipped with a sonar system that can create an image of the sea floor from a large area at once. There will be a CTD system to measure the conductivity, temperature, and pressure of the seawater. The vessel also has supporting technology for remotely operated underwater vehicles (ROV's) and autonomous underwater vehicles (AUV's). The vessel will have ultra-short baseline, an acoustic positioning system, that can communicate with subsea transponders on the sea floor or on a ROV. For the AUVs, the vessel will also have an inverted ultra-short baseline, which will make it possible for AUVs to autonomously dock or track the vessel.

Resupply Operations

As a resupply vessel, the two main innovations are also the helicopter hangar and the supply storage implementation. The hangar facility must fit two medium-size helicopters and it must provide shelter from the weather and suitable conditions for maintenance checks and refueling between flights. The helideck is located at the bow, from where the helicopters are pulled to hangar.

For the helicopter hangar, there is a boundary condition that there is no need for converting options, for example, fitting one large-size helicopter instead of two medium-size. This is because two helicopters make it possible to execute a large re-supply operation with various goods efficiently. In addition, if one of the helicopters is not operatable, the operation is not instantly doomed to failure.

The storage implementation for supply materials must be multifunctional as the material may vary from food, water or fuels to instruments and vehicles. The storage implementation must enable innovative ways for loading and unloading both to and from the vessel as well as from the vessel to the helicopters and vice versa. It is notable that the loading to the helicopters can be both via loading the cargo into the helicopter or using a cargo hook.

Semi-Autonomous

The ship can perform semi-autonomous operations, meaning that the ship can sail by herself, but the crew can still take control at any point. In this case, the ship will not be able to start its own engine for the purpose of safety, nor can the ship drop her anchors on her own. To make a ship able to sail by itself, the ship must gather data about its surroundings. Starting with the obvious, for a ship to know its position on the world, it needs to be connected to a system like the US GPS or the future European Galileo system. However, GPS cannot see obstacles in real time, so in order to avoid collisions with underwater obstacles like icebergs, the sonar system that is used for ocean research, will be expanded to send sound waves towards the front, back and sides of the ship. Moreover, a radar system will of course be used to scan the ship's surroundings above water. The sonar system and radar will have to be very precise, because the ship must be aware of both big and small obstacles.

The ship's position also needs to be known, so that the controlling computer knows what direction the ship is pointing at. An example where this is necessary is determining the ship's orientation towards the waves, so that the ship can be rotated accordingly.

All these functions must be controllable with a computer, which means that every navigation system, be it the generators or the rudder, must somehow be connected to the control computer using cables.

Energy Consumption, Generation and Heating Onboard

Although the propulsion system is set, being diesel electric, the rest of the ship must also be provided with energy. Especially the autonomous systems and the before mentioned research equipment will require large amounts of energy. Then there is the energy required for passenger/crew accommodation. The latter is in total the biggest energy consumer. Each passenger/crew is estimated to use the per person electricity use of Iceland, which is 5.777 kW (Orkustofnun, 2020). This number can include everything from heating to cooking and thus this is considered the worst-case scenario, since there is separate system for heating the ship and things like cooking can be done more energy efficiently in bulk.

The energy consumption by the vessel's onboard operative functions (research, air transportation etc.) increase the amount of electricity needed onboard. Laboratories consume from 300 to 1000 kWh/m² annually (FriendlyPower, n.d.). Based on this, we approximated that the laboratory facilities would need a 30 kW power source, as much of the laboratory equipment and processes need power continuously, such as refrigeration and ventilation. The helicopter hangar will need approximately 10 kW, mainly for lighting and ventilation. The power for starting the helicopters will be provided with s eparate ground power units that use diesel.

Four sources of sustainable energy were considered: wind energy, solar energy, and liquid hydrogen. Wind energy was discarded because any turbines added to the ship are a dangerous obstacle for the helicopters. Solar energy would only work during summer because the sun does not rise during arctic winter. In the end, liquid hydrogen has been selected as energy source. Hydrogen is made by electrolysis of water, which can be done with any energy source on land. It is then liquified by cooling the gas down to 20 K, which is the liquefaction temperature at atmospheric pressure (Rossini, 1970). Liquid hydrogen has an energy density of 8.5 MJ/L (US Department of Energy, 2001),

Lastly, the hot air from the generators will be used to heat the ship by heating water, like is done in the RRS Sir David Attenborough. There will also be an electric heating system powered by hydrogen, for when the generators are turned off.

1.3 Design Parameters

Design parameters that affect the performance of vessels and can't be controlled by the design is environmental, economic and operational. The vessel operates in arctic or Antarctic conditions which is an uncertainty both due to icy conditions and the climate change. The ship needs to be designed for certain conditions for example to break up to 1,65 m thick ice and to operate in arctic conditions in general. The weather conditions change very likely from the conditions used in the design and for example the ice thickness varies during the year and by location. Difficulties from environmental parameters can appear in manoeuvring and control of the ship. Due to varying conditions the hull resistance also changes which can lead to loss of speed and can't be easily estimated.

The vessel will be equipped with diesel-electric propulsion for which technology exists. The fuel price is an important parameter that shipping relies on. Changes in fuel price due to different matters can't be easily predicted. Different regulations and rules can for example affect the fuel price, but also other matters related to vessels and operating. New regulations are also hard to predict and the vessel could only be designed to meet the regulations and goals that are now in use (for example emissions).

The current pandemic situation is a good example on economic uncertainty. It has affected the industry in many ways both shipbuilding and operating ships. A situation like this changes many economic factors that couldn't be predicted in the design phase. Economic parameters also include operational costs like repair and maintenance, some of them can be scheduled but a vessel can also need repairs that necessary wasn't planned. An operational uncertainty are the long operating times in harsh conditions where the arrival time to next port can be unknown, which affects also loading and unloading of cargo.

1.4 Design Constraints

The ship is designed for operating in the Arctic area, thus constraints on ship's dimensions caused by the routes and ports of the Arctic are taken into examination. Also followed by constraints regarding to Polar class and ships operating in cold climate.

Constraints Set by Routes

From the current shipping routes in Arctic Ocean the following routes are considered: Northwest Passage (NWP), Northern Sea Route (NSR), and Arctic Bridge (AB). Routes are shown in picture below.

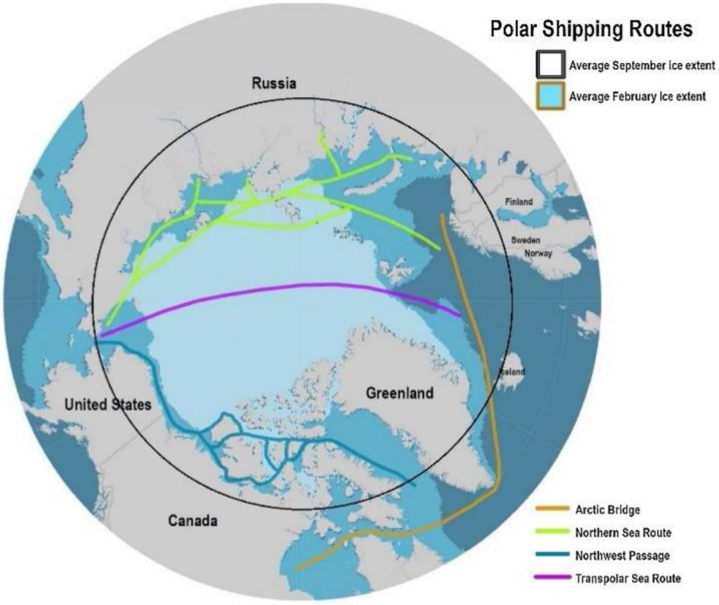

Figure 1. Arctic shipping routes (Rodriguez, 2010)

Northwest Passage includes seven different routes through the Canadian Arctic Archipelago. Two of the routes can be navigated with maximum draft of 6.4 m. Rest of the routes are suitable for drafts from 14m and more. Parts of the Northwest Passage which require draft of less than 6.4 m can be avoided while navigating through the Northwest Passage, thus constrain for draft set by the (NWP) can be considered as 14 m (Headland, 2020).

The Northern Sea Route has multiple paths. Including straits with depth less than 10 m overall with depths varying between 8-250 m. However, the shallowest areas can be avoided by taking an alternate route. As a result we can say that the maximum draft for ships operating on NSR increases to 13 m (Arctic Council, 2009).

The Arctic Bridge lays in relatively deep waters. Shallowest part of the Arctic Bridge is Hudson's Bay (NOAA, 2020). The average depth in the bay is 125 m (W. Burt, 2016). The Arctic Bridge route doesn't set any realistic constrains on the draft of our vessel. In the future ships will be able to navigate through the Arctic Ocean using The Transpolar Sea Route (TSR). Currently, only heavy icebreakers are able to sail through the TSR.

Our

project vessel is a research and re-supply vessel, meaning it will also operate away from the typical shipping routes. Thus, examination of the surrounding waters is sensible. Depths of polar seas are presented in the figure below.

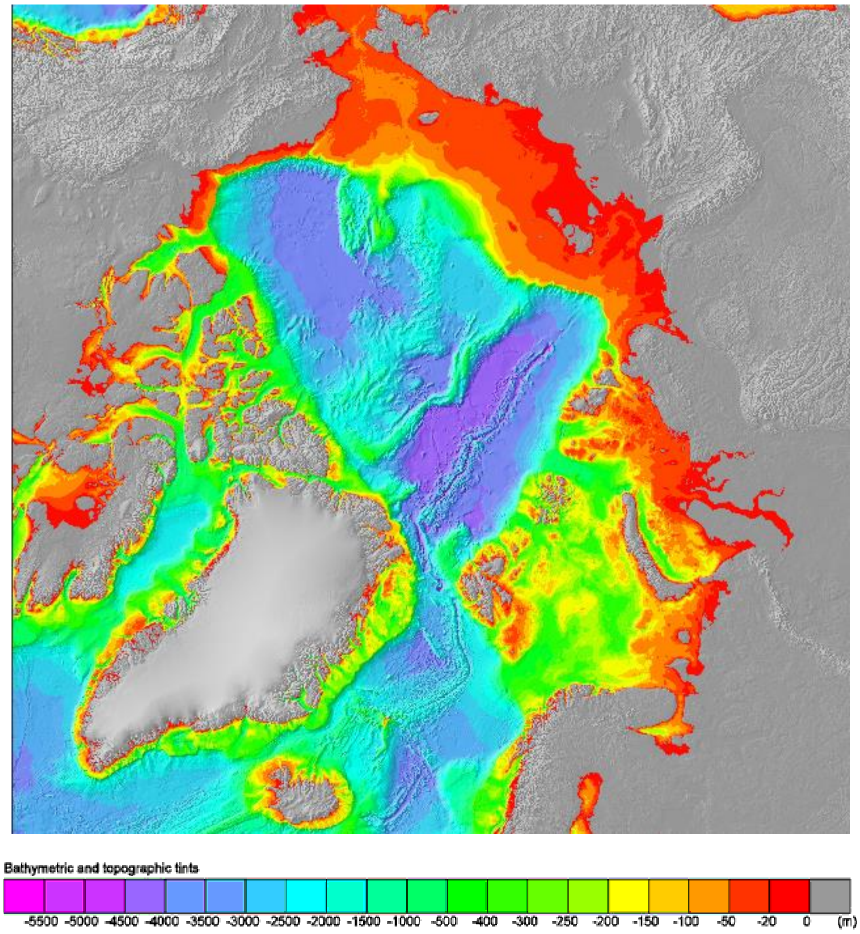

Figure 2 Bathymetric chart of Arctic area (NOAA, 2020)

As can be seen from the figure combined with the constraints presented earlier, it can be concluded that the operational area doesn’t cause significant constraints on the dimensions of the ship. The only constrains are set by the depths of the shipping routes. The constraints are affecting the maximum possible draft of the ship. Based on the constraints, the draft of the ship should be less than 13 m.

Constraints Set by Ports

Table 1 shows the depths of ports in arctic waters. Many ports are less than 8 m deep, which can be too shallow for our project ship. Fortunately, there are also ports where the water depth is deep enough for our project ship. Especially in West coast of Greenland there are ports deeper than 8 m. Ports deeper than 8 m are 9.1 m or 10 m deep and so the depths of ports limit the draft of our project ship to 9.1 m, which should be enough when studying refence ships. The maximum vessel length in most ports is up to 152 m and in some ports the maximum length is more than 152 m, this limits our project ship to 152 m. Which means that the depth of likely ports limits the main dimensions of the ship more than the depths of routes.

Table 1 Ports with cargo pier depth more than 8 m are marked with blue. (DP World, 2020)

Depths and lengths of some Arctic ports				
Country	Port	max depth [m]		max length [m]
		Gargo pier	Anchorage	
US	Nome	6.9		57.9
CA	Tuktoyaktuk	4.6		
CA	Churchill	10	21.3	>152
GL	North Star Bugt	7.6	16	>152
GL	Aasiaat	9.1	23.2	>152
GL	Qeqertarsuaq	7.6	13.7	<152
GL	Ilulissat	7.6	23.2	<152
GL	Qasigiannguit	7.6	23.2	<152
GL	Sisimiut	10	23.2	<152
GL	Maniitsoq	7.6	23.2	<152
GL	Faeringehavn	9.1	23.2	<152
GL	Paamiut	9.1	21.3	<152
GL	Kangilinnguit	10	23.2	<152
GL	Qaqortoq	7.6	23.2	<152
IS	Keflavik	7.6	23.2	
IS	Hafnarfjordur	9.1	12.2	>152
IS	Reykjavik	9.1	9.1	>152
IS	Djupivogur	3	13.7	
IS	Neskaupstadur	6.1	23.2	
IS	Seydisfjordur	7.6	23.2	
IS	Vopnafjordur	6.1	13.7	
IS	Raufarhofn	6.1	10	
IS	Akureyri	7.6	23.2	
IS	Skagastrond	4.6	13.7	
RU	Murmansk	7.6	21.3	>152
RU	Arkhangelsk	6.1	7.6	>152
RU	Dikson	6.1	7.6	<152
RU	Dudinka	7.6	9.1	
RU	Igarka	9.1	12.2	<152
RU	Tiksi	7.6	7.6	
RU	Pevek	6.1	12.2	>152

Other Design Constraints

Polar classes are described via ice descriptions. PC4 ships must be able to operate in thick first-year ice which may include old ice inclusions (IMO, 2015). Thick first-year-ice is 1.2-2 m thick (Headland, 2020). The Polar re-supply and research vessel must be capable to break 1.65 m thick ice sailing at 3 knots. So PC4 is a suitable ice class for The Polar re-supply and research vessel. The polar class sets requirements on the strength of the hull, the propeller properties and machinery onboard. There are also rules and requirements to make the ship capable to operate in cold weather. These rules apply to for example ventilation, de-icing and life-saving appliances.

2 Reference Ship and Data

2.1 Ship Category

Our project ship is categorized as a special purpose ship. Its main tasks as resupply and research vessel are resupplying Arctic research stations and do research in Arctic waters. To be able resupply research stations the vessel needs to have sufficient cargo space. Cranes mounted on deck enable loading and unloading of supplies even outside of port facilities. The ship is also equipped with a helicopter hangar, large enough to facilitate two medium size helicopters. The helicopter can be used to transport supplies when the ship is unable to go close enough shore or ice fields to unload supplies. The helicopter hangar can also be used to fix and maintain other equipment than the ship's own helicopters.

To carry out its second main task, research, the ship has scientific laboratories and offices spread across an area of 500 m². When needed helicopters are used to transport scientist to shore or ice fields to do research. In addition to laboratories, the ship has a sensor system for underwater research. The hull of the ship is instrumented with pressure gauges to collect ice load data. Ice load data can be then used to create models and methods to calculate ice loads. Laboratories and sensors enable study of such fields as biological oceanography, chemical oceanography, palaeoceanography, physical oceanography. Onboard science equipment requires great amount of electricity.

The ship has capability for semi-autonomous operations. Semi-autonomous operating requires hi-tech navigation systems and sensors to observe surroundings, but less crew is needed.

A resupply and research vessel would enable precious research in Artic areas. Arctic research stations are dependent on supply vessels having great ice breaking capability. And with so many scientists from different fields onboard there is great opportunity for diverse and cross-scientific research.

2.2 General Characteristics

Since our ship is an icebreaking special purpose ship, we need to consider characteristics for both icebreaking vessels and research vessels.

Some characteristics of ice-strengthened ships are; double hull with a gap filled with air or water ballast, special hull polymer paints that can withstand loads and has a low friction coefficient when in contact with ice, engine cooling arrangements so that water inlets and outlets don't get blocked with ice, to help manoeuvring in different ice conditions powerful bow thrusters are needed, thicker steel at the bow and at the water-line level and the rudder and propeller should be protected by the shape of the hull to prevent damage from ice moving .

The characteristics for research vessels vary with the research disciplines. It should be equipped with all necessary equipment for research including helipads, helicopter hangar, laboratories, and spaces for personnel. The main purposes for research vessels and our vessel can be for example hydrographic survey, oceanographic research, polar research, or oil exploration.

The strength of the hull should be capable of navigation in ice-covered waters, where the resistance is greater than in open water. For manoeuvrability in ice the features of the hull's shape that are important are length-to-breadth ratio, flare, mid-body and the shape of the bow and stern. Ice conditions like thickness, coverage and pressure also influence the manoeuvrability. The vessel's hull structure should be capable of different impact forces from ice. The ice class PC4 we use should be able to operate year-round in thick first-year ice, which may include old ice inclusions (Canadian Coast Guard, 2012).

To reduce power for propulsion and increase the ship's manoeuvrability in ice some performance enhancing systems can be used. For instance, to reduce drag forces and to aid manoeuvrability low friction coatings and different air bubble systems or water jet/air injection systems could be possible (Canadian Coast Guard, 2012).

2.3 Requirements

There are several types of regulations that creates requirements for our ship type e.g. requirements based on operational area (arctic and Antarctic), regulations from classification society (DNV), the International Code for Ships Operating in Polar Waters (Polar Code) and SOLAS.

The Polar Code was adopted in 2014 and applies to ships operating in arctic and Antarctic waters, it includes both mandatory and recommended provisions for measures for ship safety and pollution prevention. Since the vessel is going to operate in low air temperature the materials used should be suitable for operation at polar service temperature and for ice strengthened ships the structure of the ship should be designed to resist both global and local structural loads from ice conditions. The polar code includes functional requirements for e.g. stability in intact conditions and in damaged conditions. The ship shall have sufficient stability intact conditions when subject to ice accretion. (IMO, 2015)

SOLAS specifies minimum standards for the construction, equipment and operation of ships, compatible with their safety (IMO, 1974). Also, a code of safety for special purpose ships exists. The code is for special purpose ships that are not less than 500 gross tonnage and carries more than 12 special personnel (persons needed for the operational duties of the ship and are carried in addition to those persons required for navigation, engineering and maintenance of the ships or those that provide services for persons onboard). The special personnel in our case are scientist for research. (MSC, 2008)

We choose DNV as our classification society and they have classifications and requirements for both vessels for arctic and ice breaking service and for special purpose ships. For the vessels in arctic service the classification covers e.g. materials used in structures, strength for longitudinal and transverse hull girder, rudders and steering gears, propellers and propulsion machinery, and stability for subdivision, intact and damage (DNV GL, 2016). Since our ship is a special purpose ship and carries personnel that are neither crew members nor passengers the class sets requirements. The additional class notation adds additional level of safety in providing reference to design criteria, construction standards and other safety measures concerning special purpose ships. DNV also sets requirements for helicopter refuelling and hangar facilities e.g. how the helicopter fuel storage tanks shall be constructed, which materials are compatible and what safety equipment should be available. For the hangar e.g. structural restrictions are set (DNV GL, 2016).

Some technical requirements for ships operating in ice are that the propulsion plant and steering gear must be reliable and capable of responding fast to manoeuvring orders. It is also important for the safety that navigational and communications equipment are reliable. Ice and snow should be easily removable from the engine room and other necessary places where it can cause danger. And for visibility during night good searchlights should be available. For uncertainties with the condition's ships navigating in ice should carry fuel for manoeuvring and fresh water and other supplies if delays occur (Canadian Coast Guard, 2012).

2.4 Challenges

The challenges that affect our ship type are mostly related to the climate. The temperature, winds, different ice conditions and icing of the superstructure for example. Considering the condition and regulations one challenge is to make the hull withstand changing ice and harsh conditions. A challenge is also to get the design of the vessel to fit everything needed for our purpose, research equipment and enough space for helicopters and cargo.

2.5 Reference Ships

Two ships have been chosen as a reference ships for our design project. The ships are British RRS Sir David Attenborough and Chinese MV Xue Long 2. Both ships are designed for Artic and Antarctic research and re-supply operations. The ships are chosen as references because they are designed for similar operations in the same operation area as the ship of the design project. Similar technical features like e.g. helicopter hangars, underwater research capability, moonpools and dynamic positioning can be found from the reference ships as from our design plan. Additionally, both ships are deployed into service in 2019, which ensures us that both selected references are up to date.

Table 2 Reference ships

	RRS Sir David Attenborough	MV Xue Long 2
Length overall [m]	128	122.5
Beam [m]	24	22.3
Draft [m]	7	8.3
Gross tonnage	15000	12769
Deadweight [t]	4475	4530
Machinery	Diesel powered, Bergen B33:45 engines (2x 9-cylinder and 2x 6-cylinder)	Diesel powered, Wärtsilä 32 engines (2x 16-cylinder and 2x 12-cylinder)
Power of machinery [kW]	2x 5400 + 2x 3600	2x 9280 + 2x 6960
Cargo volume [m³]	2100+660 of aviation fuel	Not published
Passengers	30 crew + 60 scientists	90
Year built	2019	2019

3 Hull Form and Hydrostatics

3.1 Main Dimensions

As discussed in Section 1.4, the constrains regarding our project vessel's main dimensions are set by the ports. From those comes the maximum draft of 9.1 m and maximum length of 152 m. As the main dimensions are constrained by the ports, our vessel's main dimensions are limited by the dimensions. We gave a couple of methods a try as we determined our ships main dimension:

Statistical Method

We tried the statistical method with the similar initial values as our reference ships have. The table from the course material, which had reference values for the ratio of deadweight tonnage and displacement tonnage (DWT/Δ), did not have a good reference vessel type regarding our project. Thus, we calculated the displacement in tonnes from the reference ships and divided the deadweight with that. This gave us a value for DWT/Δ, which was around 0.35. We compered this with the course material's reference table. The value landed close to RoPax vessels, but our value was a bit higher than the usual value of them. Thus, the value we calculated for the DWT/Δ was quite good represent of our vessel type, as it is somewhat close to the typical passenger vessels, but still is not one, as it has more complex mission.

In Table 3 Statistical method, there is one example with the output of statistical method with the values of our reference ship. The results of this method were problematic no matter which reference values we used.

Table 3 Statistical method

Inputs	
Deadweight	4575
DWT/Δ	0.345544
V (Knots)	13
Density of water (t/m3)	1.025
Hull Section type	U-section

Ship's Main Characteristics	
Δ	13239.99
LPP	138.69
B	21.34
T	5.39
D	10.27
C_B	0.810
C_M	0.994
C_P	0.815
C_{WP}	0.878

The ships length was over 10 m longer than our reference ship, which alone would not be a problem, but the draft differed also a lot. The drafts of our reference ships are 7 m (RRS Sir David Attenborough) or 8.3 m (MV Xue Long 2). The statistical method, however, proposed that our draft should be only around 5.4 m. This would mean that our draft would be 23-35 % smaller than the reference ships'. Considering the ice breaking capability and the alike missions with the reference ships, we determined that this would be too big of a change in the main dimension, and thus decided not to use the statistical method to define the main dimensions.

Normand's Number

We used data of our reference ship RRS Sir David Attenborough to determine our vessel's main dimensions using Normand's Number. The data about the length, beam, draft, block coefficient, deadweight and displacement were available, but we used different approximations to determine the hull, machinery, outfitting and fuel weight for the calculation of the Normand's Number. The Normand's Number ended up being 2.45. Requiring at least a draft of 7 m, the length becomes 119 m, and the beam becomes 22 m.

Thus, the ship's main dimensions based on this method were:

Table 4 Normand's Number

Length	119 m
Beam	22 m
Draft	7 m
Displacement	12912 tonnes

Final Main Dimensions

The results of the Normand's Number were the starting point of defining further our hull form. The main dimensions got changed by a little during the hull forming process and the final values of our main dimensions are presented in Table 5:

Table 5 Main dimensions

Length between perpendiculars	117.1 m
Length overall	132.92 m
Beam	22 m
Draft	7 m
Displacement	11895.7 tons

3.2 Hull Form

The design process of the hull form for ice going vessels usually takes in consideration to minimize the ice resistance by optimal shapes of the beam and bow, to ensure good operational characteristics, enables the ship to go astern as much as required and minimizes the ice impact on the propellers (Riska, 2010). The design for ice-going vessels is primary based on the vessel's intended use, since our vessel is a research vessel (not a typical icebreaker) and is also going to operate in open water the hull is designed to have good manoeuvring (Quinton & Lau, 2005). While designing the hull for a polar class vessel the ice loads on different areas needs to be taken in consideration mainly when choosing the bow. An icebreaking bow (without a bulbous bow) enables the vessel to ride over the ice and exerts downward force to break the ice, compared to non-icebreaking bows which has a more crushing behaviour for the ice (Dolny, 2018). The hull form is typically optimized to clear the ice away from the propellers and other underwater appendages and to reduce the surface drag of the ice on the aft section (Dolny, 2018).

Sectional Area Curve

There are some typical characteristics related to the hull of ice-going ships. Hulls are designed to break, bend, and push the ice away by using its own weight. Form of the bow of ship has big effect on this capability. Usually, icebreakers are built with full bows. Full bow results in more displacement in the front of the ship which improves manoeuvring in the ice. Longitudinal centre of buoyancy (LCB) and the station of maximum beam is thus often shifted forward of the amidship (Quinton & Lau, 2005). Parallel mid-bodies on the ice breakers tend to be avoided because ice breaking capability of parallel mid-body is weak (Moton, 1991). These facts have effect on sectional area curves of ice breakers, which are often tilted to the forward of amidship.

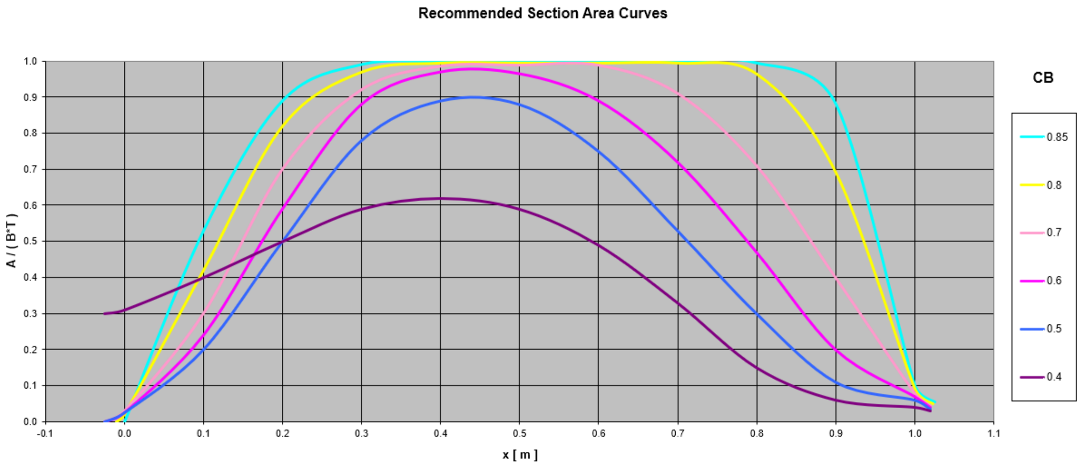

Figure 3 Recommended Section Area Curves

The sectional area curve gives the displaced volume of the vessel. While determining our vessels main dimensions we used the value for the block coefficient as 0.7 and looking at available data of icebreakers the block coefficient is on average around 0.62. Our value for the sectional area curve came to correspond the value earlier used. The curve matches the characteristics of a short mid-body described earlier and the form of the bow grows sharper than the stern. The centre of the area under the section area curve gives the LCB for the vessel, which in our case is backward of the amidship. Since our vessel is not designed as a typical icebreaker and to keep performance for open water the LCB does not need to be that fore as for an icebreaker.

Bow Design

The bow shape that has been selected for this ship is the spoon bow, seen in Figure 4.This bow shape is commonly used on icebreakers, including on both the RRS Sir David Attenborough and the MV Xuen Long 2. Furthermore, a spoon bow with a small stem angle (between 20-25 degrees) is considered to have very low ice resistance (Riska, 2010), (Quinton & Lau, 2005). The reason this stem angle must be small, is because a smaller stem angle increases the vertical component of the pushing force from the bow onto the ice. This in turn increases the downward bending load onto the ice. Thus, thicker ice can be broken with the same pushing force (Riska, 2010).

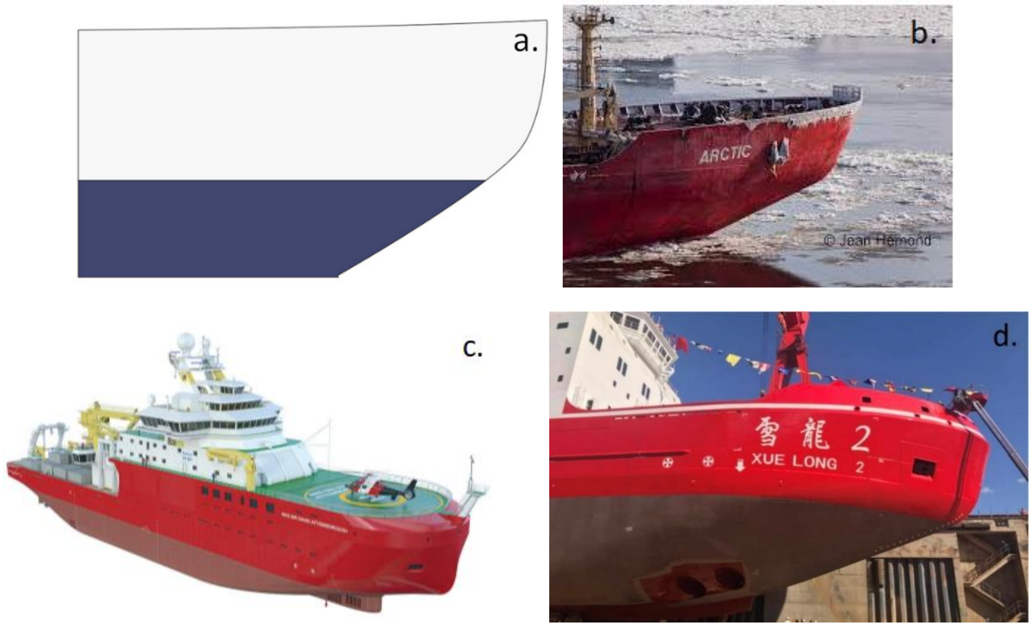

Figure 4 Ice breaking ships with spoon bows: a. schematic of a spoon bow, b. bow of the a Fednav Arctic ship (Hémond, 2014), c. Bow of the RRS Sir David Attenborough (Ingenia, 2018), d. Bow of the MV Xuen Long 2 (Aker Arctic, n.d.).

Furthermore, at the very bottom of the hull, from the bow to the shoulders, there will be a wedge that allows ice to flow beneath the bow and be pushed to the side, preventing ice from getting to the propellers (Czimmek, 1991). This design is illustrated in Figure 3. This adjustment was made at a late stage, so it is not shown in the line drawings at the end of this document.

Figure 5 Schematic of the wedge used to push ice away from the hull, adjusted from (Czimmek, 1991).

The bow should also prevent shoulder crushing, which is the piling up of ice that is in contact with the ship's shoulder. Shoulder crushing can create an increase in ice resistance; however, scale model tests are currently the only way shoulder crushing can be predicted and so shoulder crushing will not be considered in this ship's design (Riska, 2010).

Stern Design

Only little sources were found about stern design of icebreaking vessels that were not behind paywall. Viewing pictures of icebreaking vessels one can notice that they have long and gentle stern (Figure 6). Stern should have large enough clearances between tip of propeller blades and stern frames and bottom of the level ice sheet. Clearances must be large enough to avoid loads that can occur when ice floes are forced between the propeller and the stern frame and when propeller can hit large ice floes (Traficom, 2019). Number of propellers affects greatly to the stern design. Stern design must such that it protects rudders and propellers (Canadian Coast Guard, 2012).

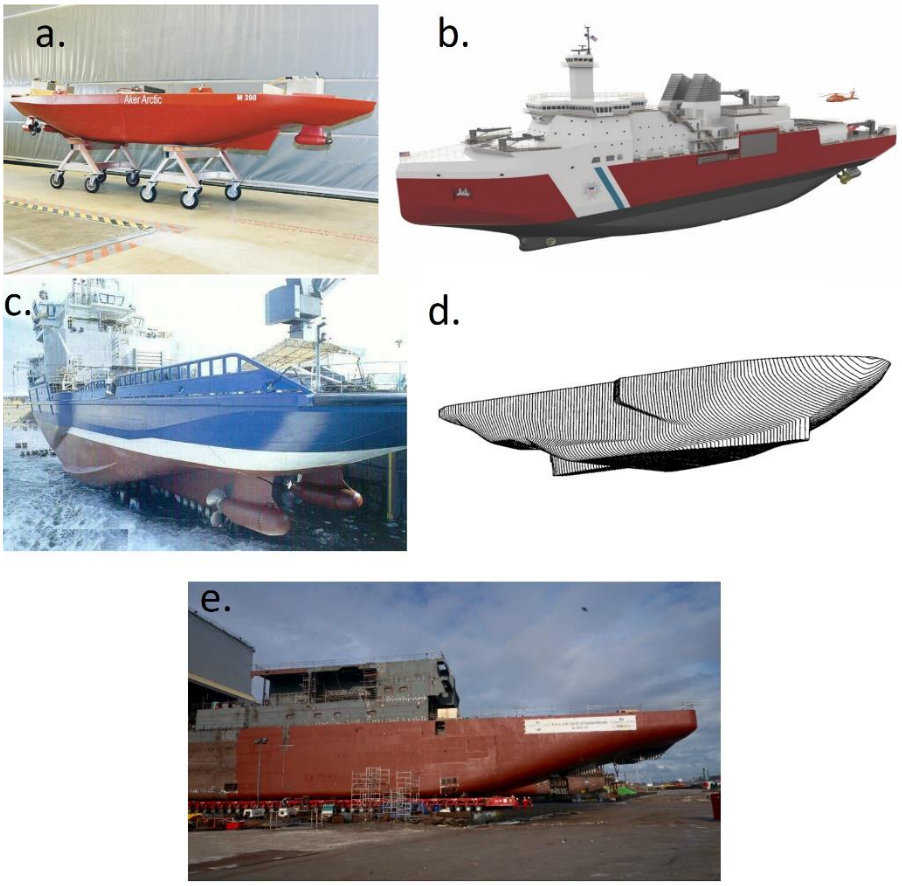

Figure 6. Sterns of icebreaking ships: a. hull model of icebreaker Polaris (Riska, 2010), b. planned U.S. Coast Guard Polar Security Cutter (Werner, 2019), c. stern of multipurpose icebreaker Botnica (Riska, 2010), d. hull shape of icebreakers Finnica and Nordica (Sodhi, 1995), e. stern of RSS David Attenborough (British Antarctic Survey, 2017).

Draft Sketch

The draft sketch was done with the provided *hull form* -excel file. The sketch was done by varying the non-dimensional values and keeping an eye on the Section Area Curves and on the values of the coefficients of fineness.

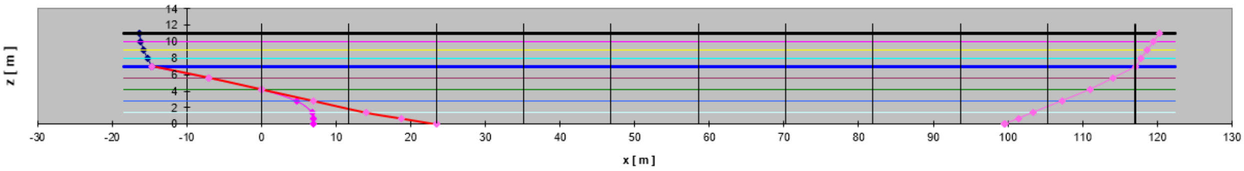

Figure 7 Profile sketch

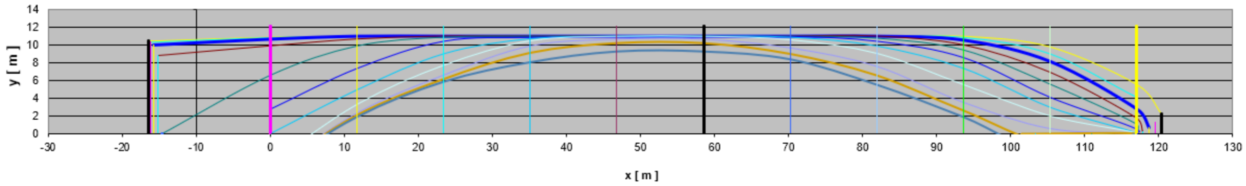

Figure 8 Waterline sketch

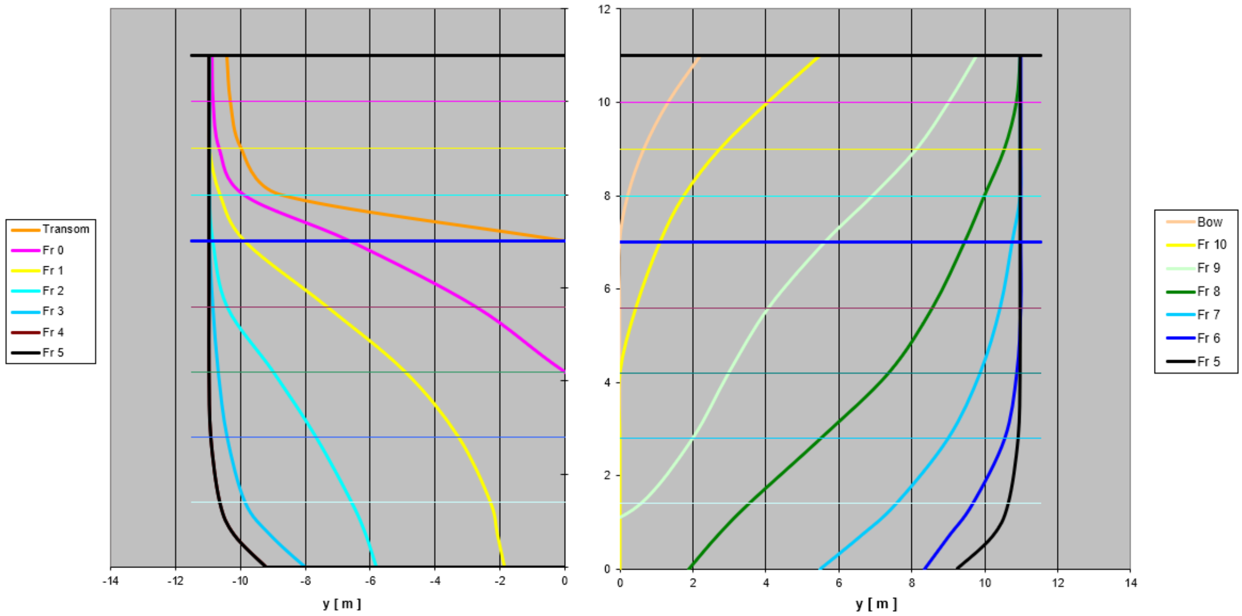

Figure 9 Frame Section sketches

3.3 Hydrostatics

Hull Model

The above presented draft sketch was the basis for further shaping the hull model. This was implemented by importing the draft sketch to Delftship and shaping the hull form based on them. The result of this process can be seen in Figure 10.

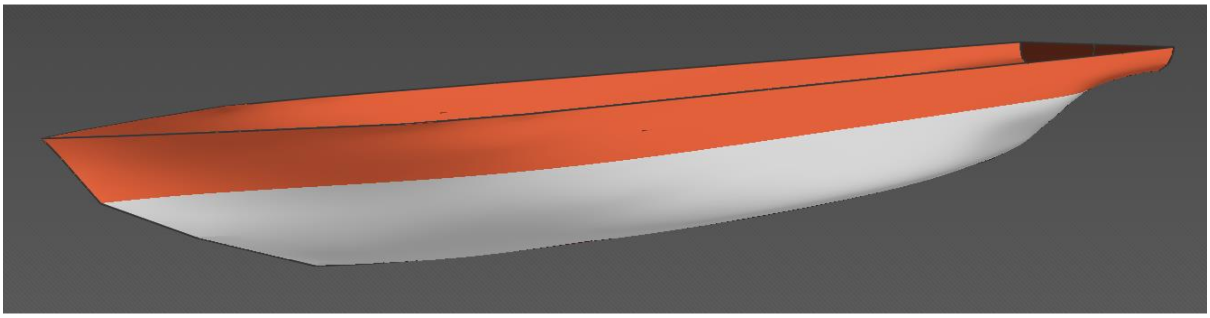

Figure 10 The hull model

After the model was ready, the waterlines and frame sections were determined to create the lines plan picture below.

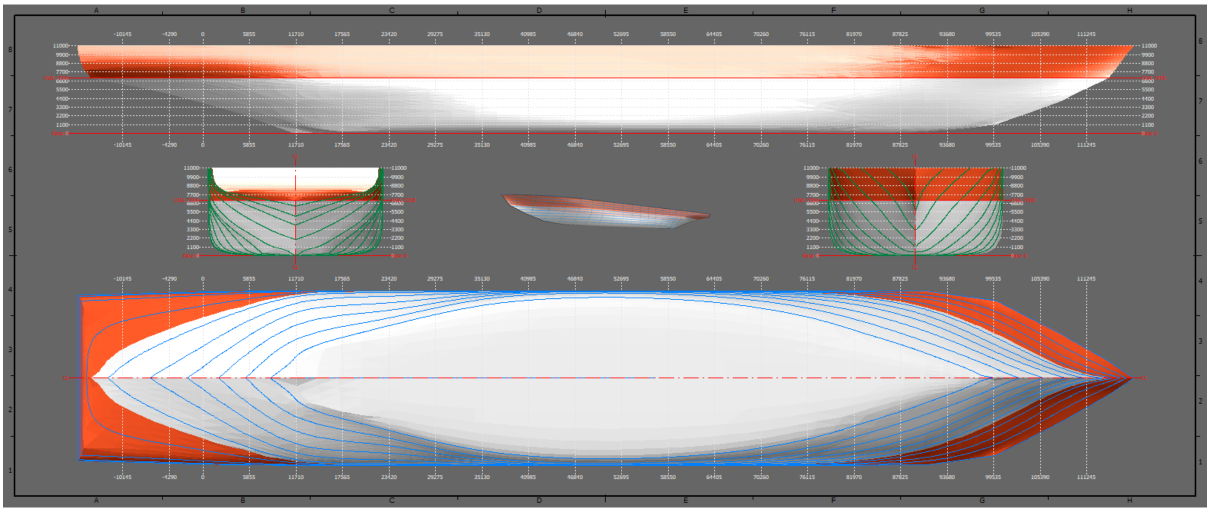

Figure 11 The Lines Plan

Main Dimensions and Coefficients of Fineness

The design's hydrostatic report in the Appendix 1: Design hydrostatics report. The main dimensions can be seen to match the dimensions determined earlier: length of perpendiculars being 117.10 m, beam of 22 m and draft being 7 m.

The total displacement volume of this design according to Delftship is 11605.5 m^3. The Hull Lines -excel that was used to create this model, approximated the displacement volume to be 12 163 m^3. Thus, there is a slight change. The block coefficient according to Delftship was 0.6436 and according to excel 0.678. The difference checks out as the main dimensions are the same, but the displacement volume is not.

These and the other coefficients of fineness from the different sources compared below in table 1:

Table 6 Coefficients of fineness

	Delftship	**Hull Lines -excel**
Block coefficient	0.6436	0.678
Prismatic coefficient	0.6803	0.691
Waterplane coefficient	0.8701	0.903
Midship area coefficient	0.9460	0.981

The Longitudinal Centre of Buoyancy

The longitudinal centre of buoyancy (LCB) according to the Delftship model was -6.094 %. This turned out to be worse than what the excel had predicted. Compared to empirical equations, the LCB should be in the range of -2 % to 0.15 % for block coefficient around 0.68 and in the range of -2.8 to -0.8 for block coefficient around 0.64. The excel predicted based on the draft sketch LCB to be -3.3 % which in this case would have been better.

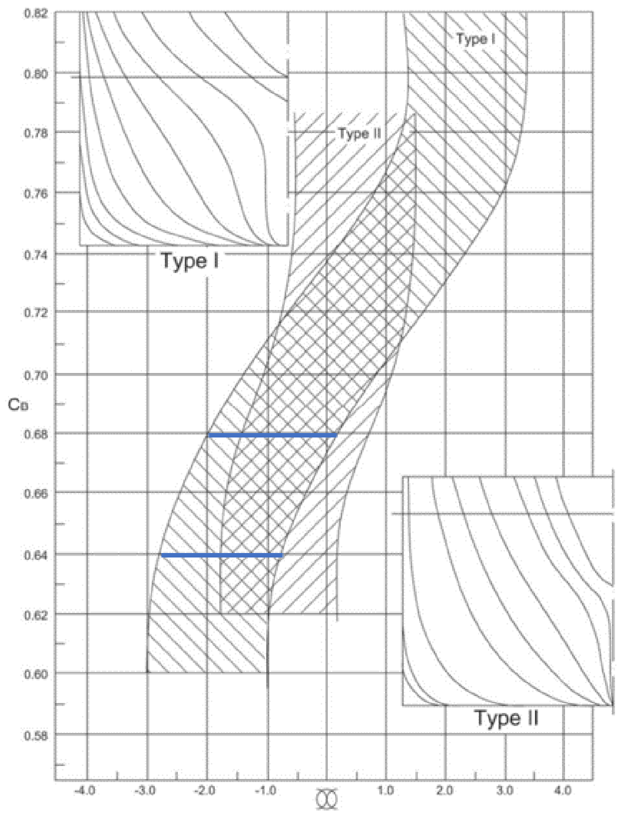

Figure 12 Empirical data about LCB

3.4 Hull Volume Estimation

Calculating Whole and Part Cubes

We estimated hull volume of our project ship by calculating whole and part cubes inside the hull using excel (Figure 13). Part cubes counted as half of the cubes. We determined volume of cubes using formula below.

$$x \times y \times z = frame\ length \times 1\,m \times distance\ between\ WPAs$$

Hull lines are from our hull line calculations and our estimation excel (Figure 13) checks if coordinates of corners of cube are inside the hull lines or not and then one can see if the whole cube is inside the hull lines or not.

		% step 2: 0.5 if corner touch whole square										
Y [m]	11	0	0	0	0.5	0.5	0.5	0.5	0	0	0	0
	10	0	0	0.5	0.5	0	0	0.5	0	0	0	0
	9	0	0	0.5	0	0	0	0.5	0.5	0	0	0
	8	0	0	0.5	0	0	0	0	0.5	0	0	0
	7	0	0.5	0.5	0	0	0	0	0.5	0.5	0	0
	6	0	0.5	0	0	0	0	0	0	0.5	0	0
	5	0	0.5	0	0	0	0	0	0	0.5	0	0
	4	0	0.5	0	0	0	0	0	0	0.5	0	0
	3	0.5	0.5	0	0	0	0	0	0	0.5	0.5	0
	2	0.5	0	0	0	0	0	0	0	0	0.5	0
	1	0.5	0	0	0	0	0	0	0	0	0.5	0
	0	0.5	0	0	0	0	0	0	0	0	0.5	0
		0	1	2	3	4	5	6	7	8	9	10
												X [frames]

		% step 1: 1 if square corner inside of awp curve										
Y [m]	11	0	0	0	0	0	0	0	0	0	0	0
	10	0	0	0	0	1	1	0	0	0	0	0
	9	0	0	0	1	1	1	0	0	0	0	0
	8	0	0	0	1	1	1	1	0	0	0	0
	7	0	0	0	1	1	1	1	0	0	0	0
	6	0	0	1	1	1	1	1	1	0	0	0
	5	0	0	1	1	1	1	1	1	0	0	0
	4	0	0	1	1	1	1	1	1	0	0	0
	3	0	0	1	1	1	1	1	1	0	0	0
	2	0	1	1	1	1	1	1	1	1	0	0
	1	0	1	1	1	1	1	1	1	1	0	0
	0	0	1	1	1	1	1	1	1	1	0	0
		0	1	2	3	4	5	6	7	8	9	10
	cubes:	43	% step 3: calculating whole squares from coordinates manually									X [frames]
	1/2 cubes	16										

Figure 13 Screenshot of excel sheet used to estimate hull volume. Blue cells mark coordinates of whole squares and green cells mark part squares on one specific WPA.

Estimation of the volume of the hull to upper deck is 21933.5 m^3, which is relatively good value when compared to 22700 m^3, value obtained from excel for our hull lines. Calculated volume is the volume of the hull between frames 0 and 10, meaning stern and the bow are not calculated.

Simpson's First Rule

The given excel for Simpson's first rule integration was used to calculate volume both with cross section areas on different frames and with waterplane areas on different waterlines. For the volume based on cross-sectional areas the ships length was divided into frames and calculated cross-sectional areas for each off them. The result for that became 21300 m^3 and for the second integration 21467.4 m^3. In the second integration the waterplane areas were calculated. There is a bit difference between the volumes which can be due to that the values used was approximated from the values we have used to calculate hull lines. Also, when integrating the volume with waterplane areas the area for stern and bow was not considered.

Table 7 Result from Simpson integration using cross-sectional areas.

Volume	21300	m3
Density of water	1,025	t/m3
Displacement	21832,87149	t
LCB from fr0	59,646	m

Table 8 Result from Simpson integration using waterplane areas.

Volume	21467,4	m3
KB	6,194	m

Volumes obtained from the given excel for Simpson's first rule are both smaller than the estimated volume, this might be due to actual volume of the hull inside the half cubes being smaller than volume of the half cubes.

The hull volume to upper deck has been calculated in the excel for our hull lines as 22700 m^3. And the difference between those can also be due to approximations and difference in waterlines and frames. From calculating the cross-sectional areas, a SAC curve was made and the LCB calculated. The SAC curve consists of the sectional areas for the whole length of the vessel. The LCB calculated is at 59,646 m which is more aft from our previous SAC lines. The SAC lines have difference in the x-coordinates. From the second integration we also get a value for the vertical location for centre of buoyancy from the keel (KB) as 6.194 m. Which seems a bit high since our draft is 7 m (maximum draft 11 m used in the integration) but the hull shape has a quite flat bottom and are not so deep at the aft, which could make the KB a bit high.

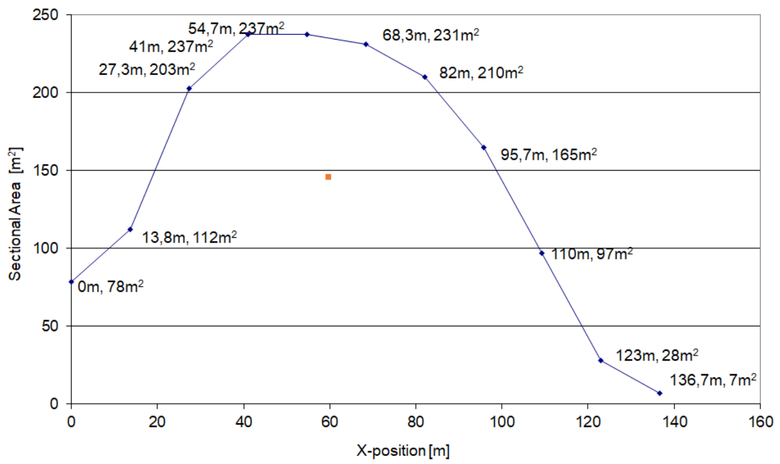

Figure 14 Sectional Area curve

The volume of the hull to upper deck was estimated to be 21933.5 m^3 by using whole and part cubes estimation. The Hull Lines -excel calculated that value to be around 22700 m^3. The Delftship does not calculate this value automatically, but by setting draft to almost at the height of the upper deck, the hydrostatic report can be used to obtain an estimated value. By doing this, we obtained that according to the Delftship, the whole hull volume would be around 21700 m^3. This estimate is valid when compared to the other two estimates.

3.5 NAPA iteration

During this Ship Design Portfolio course the hull is modelled in NAPA which made it possible to do an hydrostatics iteration based on the new model. Our Delftship model presented earlier had an design draft at 7 m with a displacement at 11605.5 m^3. From the hydrostatics calculations done in NAPA and inspecting our displacement a draft at 6.3 m would be enough. With a safety margin of about 3 % the new design draft is 6.4 m.

T m	DISP t	LCB m	VCB m	KMT m	CB	WLA m2	MCT tm/cm	TPC t/cm
5.600	9949.3	56.428	3.064	10.949	0.6169	2292.3	185.7	23.5
6.000	10909.4	56.081	3.305	10.889	0.6314	2367.7	201.2	24.3
6.400	11900.7	55.719	3.546	10.833	0.6458	2453.3	221.1	25.1
6.800	12920.5	55.344	3.788	10.795	0.6599	2529.3	239.6	25.9
7.200	13976.2	54.929	4.030	10.788	0.6743	2608.5	261.0	26.7
7.600	15055.0	54.535	4.272	10.747	0.6882	2651.3	271.5	27.2

Figure 15. Hydrostatic values from NAPA model

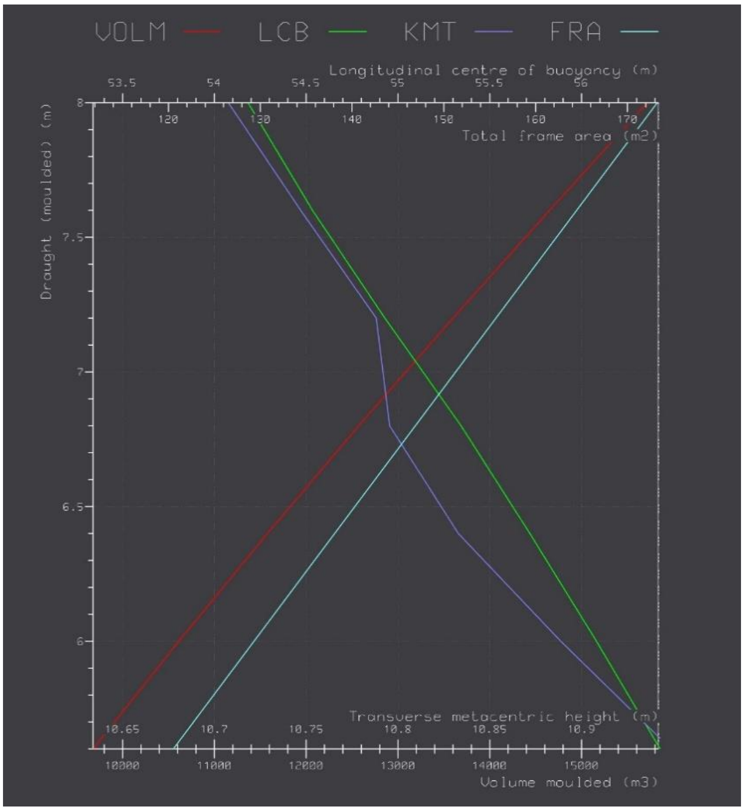

Figure 16. Hydrostatic graph for NAPA model

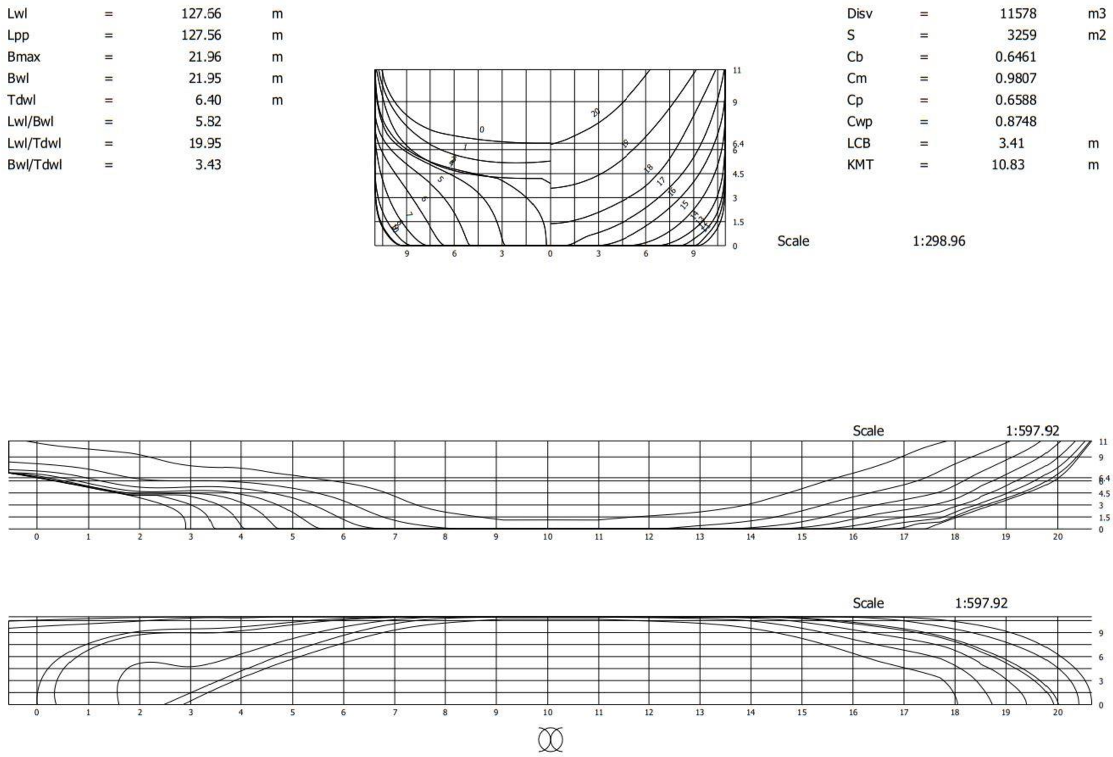

Figure 17. Lines drawing for NAPA model

Comparing the values for the Delftship model and Excel values done in PNA the dimensions and values have small changes but remains in the same range.

4 General Arrangement (GA)

4.1 GA Requirements

Safety

Safety on the vessel is taken into account on many aspects. The personnel safety is considered by making the GA so that it is not labyrinthine. This way, in case of emergency, it is easier for the personnel to get to safety.

On decks 5 and 6, there is enough lifeboats for 110 % of the maximum personnel capacity of the vessel. The lifeboats are stationed so that they are easy to access from all places on board, i.e., close to the stairways and also so that they are accessible also from the outside areas of the decks.

There are also survival kits (both personal and group) suitable for the polar environment on the lifeboats as well as survival equipment on board in case the vessel gets stuck. The capacity of these equipment and kits is 110 % of the maximum number of persons on board.

The helicopter safety is taken care by arranging helicopter deck and hangar big enough so that the helicopter operations can be carried through safely.

Also, lot of other arrangements are also vital for the survivability of the crew. For example, the GA includes a hospital area so that minor infections can be handled accordingly and also needed help in case of personal injuries are available on board.

For safety according fire, detection and extinguishments systems are placed on the vessel. Since the vessel is operating in cold climate the fire-fighting systems need to be anti-freezing. It's also necessary that equipment with high fire risk is placed safely for example engine rooms and storage of liquids, and to use fire safe materials. The stairways are placed so that personnel and crew can escape to the lifeboats from each deck. For safety and escape reasons the breadth and length of stairs and corridors in different locations shall also meet the SOLAS regulations.

Water: Fresh, Grey and Black

The ship water treatment systems will make sure that the ship can continuously re-use freshwater, so that fresh water does not have to be replenished. This is because the ship should be able to stay on the sea for 32 days without going back to shore for water. Still some amount of fresh water will be lost in amongst the dry waste. However, if all water treatment systems fail in such a way that no new drinking water can be produced, there should be enough fresh water in the tank left for 14 days for normal service in order to fix the problem.

To calculate this required amount of fresh water, one must know how much freshwater people use. The amount of fresh water used by people in different countries varies, for example it is about 0.142 m^3 in the UK (Energy Saving Trust, 2013) and 0.150 m^3 in Finland (Helsinki Times, 2012) but sadly, there seems to be little clear data on how much water is used on ships. The number 0.3 m^3 per person per day seems to be pop up as well a lot, though often in the contexts of cruise ships. Given that the only two real sources use numbers at around 0.145 m^3 per person per day, a 0.16 m^3 of freshwater per person per day will be assumed, providing 15 extra litres of water per person per day as a safety factor. This leads to a total required amount 274 m^3 of water that needs to be added into the freshwater tank. 10 percent will be added to this to compensate for the potential water loss during treatment. Leading to a final total requires freshwater capacity of 300 m^3.

Now to determine how much of this 300 m^3 will go into the grey water tank and how much will go into the black water tank should the water treatment system fail. The average person in Germany uses 25-50 L of water every day just by flushing the toilet. Taking the highest number of 50 L, 110 L of the 160 L used per person per day would be left for showering, drinking, cooking and washing clothes. That means 69 % of the total is grey water and the remining 31 % is black water. That in turn means that the grey water and black water tanks need a capacity of 206 m^3 and 94 m^3 respectively.

Fuel: Hydrogen and Diesel

As power for other uses than propulsion will partly be powered by hydrogen. As presented further in chapter 6 the vessel will be equipped with fuel cells of 1000 kW capacity. The vessel will carry hydrogen for operating time of 64 days which makes the total amount of hydrogen required to be 651 m^3, calculated using the energy required and specific energy and energy density for hydrogen. The tanks volume can be divided into two tanks and fitted at the aft.

On 32-day voyages fuel consumption for whole voyage is 454 tons of fuel oil. Given a marine diesel density of 0.9 ton/ m^3 (Danish Environmental Protection Agency, 1998), the total required tank volume for diesel is 505 m^3.

Table 9 Fuel oil consumption per voyage

	Open water	Light ice cond.	Heavy ice cond.	Sum.
Per voyage	70%	25%	5%	100%
Hours / voyage, h	537.6	192	38.4	768
Prop. Power, kW	1400	4203	29000	
Fuel cons., kg/h	238	715	4933	
Fuel/voyage, ton	128	137	189	454

Lubrication oil and bilge water

In chapter 8 of this report, it is calculated that during 32-days voyage of our vessel 1.35 m^3 of lubrication oil is required. DNV GL has a requirement for bilge water tanks. For vessels with engine power above 20 000 kW required bilge water tank capacity can be calculated with following equation: $40 + \frac{Main\ engine\ rating\ (kW)}{500}$. In our case the overall output power of the engines is 31720 kW, thus required minimum bilge tank capacity is 104 m^3.

Ballast Water

At full capacity the ship should have a draft of 6.4 m, with a displaced volume of 11900 m^3 according to NAPA, which is equivalent to 12198 tonnes. Of this, the deadweight is 2967.2 tonnes. Even without the dead weight the ship should preferable still be close to design draft. This is achieved with ballast water. Our ship has ballast water capacity of 1400 m^3 ,which equals in 1435 tonnes of salt water.

Frame Spacing

The frame spacing used in the GA is 800 mm as it in practise is between 500-900 mm. Since our vessel has an overall length over 120 m it will also have longitudinal frames with wider spacing (mixed framing system).

Height of the Bow

ILLC and Classification societies have regulations and guidelines for the minimum bow height for ships. Sufficiently high bow provides enough area for anchoring and mooring equipment and prevents water from splashing on the deck in rough sea conditions.

According to ILLC regulations (Mochammad, 2014), minimum bow height for ships below 250 m in length can be derived by following equation:

$$H_b = 56 \times L \times \left(1 - \frac{L}{500}\right) \times \frac{1.36}{C_b + 0.68} = 5587.18\, mm$$

, where $C_b = 0.688$ and L=139 m.

DNV GL (DNV GL, 2016) has it own requirements for the minimum bow hight derived with following equation:

$$H_b = \left[6075 \times \left(\frac{L_f}{100}\right) - 1875 \times \left(\frac{L_f}{100}\right)^2 + 200 \times \left(\frac{L_f}{100}\right)^3\right] \times \left[2.08 + 0.609 \times C_b - 1.603 \times C_{wf} - 0.0129 \times \left(\frac{L_f}{T}\right)\right] = 4637.96\, mm$$

, where freeboard length $L_f = 130.3\, m$, T=7 m, $C_b = 0.688$ and water plane area coefficient forward of $\frac{L_f}{2}$ $C_{wf} = 0.85$.

Bow height of our ship is 7 m, which fulfils both requirements.

Double Bottom

Classification society requires double bottom from collision bulkhead to aft peak bulkhead. DNV GL has defined minimum height for the double bottom by following equation:

$$H_{db} = 1000 \times \frac{B}{20} = 1100\, mm\, , minimum\ allowed\ H_{db} = 760 mm$$

, where B=22 m. (DNV GL, 2016)

Height of double bottom of our vessel is 1200 mm which exceeds the value required by DNV GL.

Bulkheads

For ships with length between 125 m and 145 m DNV GL requires 6 transverse bulkheads if engine is aft, and 7 if engine is anywhere else. In our case engine is not at aft, thus required number of bulkheads is 7. All ships are required to have at least one collision bulkhead, one aft peak bulkhead and one bulkhead at each end of the engine room.

For ships with an electrical propulsion plant, like our ship, both the generator room and the engine room must be enclosed by watertight bulkheads. Space of four frames must be left between the engine and bulkheads to have space for maintaining and service. Aft peak bulkhead should form watertight compartment enclosing the stern tube and the rudder trunk.

Collision bulkhead is designed as a barrier for water in case of collisions. Location of collision bulkhead is defined by SOLAS and classification societies. For ships without bulbus bow and with length less than 200 m position of collision bulkhead should be

$$0.05 \times L_f \geq X_c \geq 0.08 \times L_f$$

, where X_c is the distance from forward perpendicular in meters.

In result, our collision bulkhead should be from 6.5 m to 10.4 m from the forward perpendicular. Our value is 8 m, which fits the required values.

Fire Zones

The fire zones will be divided by bulkheads, decks and doors. In practise the bulkheads for fire zones are constructed from steel and insulated to prevent spreading of smoke, heat and flame. According to SOLAS the main vertical zones have a maximum length of 48 m to meet the subdivision of watertight bulkheads (IMO, 2002), which means that at least three of our bulkheads (at frames 32, 85 and 105 in the GA) will be constructed to meet the regulations of fire zones and watertight. The maximum area of a main vertical zone is 1600 m^2 (IMO, 2002).

4.2 GA Definition

General arrangement is developed further from last design cycle. In this design more attention is paid to safety, material and people flows and tank volume capacities.

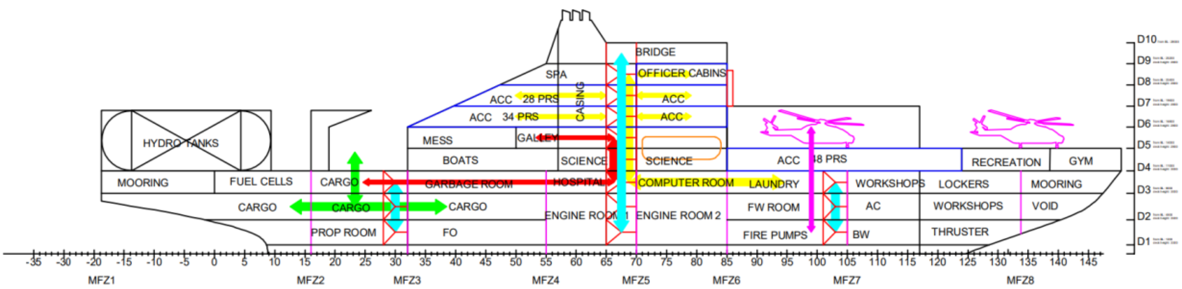

Figure 18 Sideview of Khione showing cargo flows (green), people flows (blue) through stairs, provision and garbage flow (red) from provision storages to galley to garbage room and laundry flows (yellow).

As can be seen the flow of people between decks is possible at three locations on the ship marked in blue. At these locations there are stairs and elevators, making it possible to move heavier items using carts or trolleys. This will make it easy to transport provisions to the galley for preparation and waste back to the garbage room. The flow of cargo will go through a single large hatch using the onboard crane down into the cargo hold. As can be seen the cargo hold is much larger than the hatch, but the cargo hold can despite this be filled to maximum capacity by using pallets and pallet jacks. Another type of flow through the ship will be the flow of fuels and in this case, there are three types. Starting from the aft the flow of hydrogen to the fuel cells is very simple since the tanks are on top of the fuel cells. At the midship the main engines will take their fuel from tanks in the double bottom below. The most complicated flow is that of fuel for the helicopters, since it will pass through several decks. We however think that this is the best solution compared to storing the fuel closer and therefore higher in the ship. This significant amount of fuel would be too detrimental to the stability of the ship especially when the free surface effect in a half-filled tank is considered.

Dead-ends are not permitted in crew accommodation areas (IMO, 2012) and thus on deck 8 (Figure 19) there is escape stairs at and of a corridor at frame #85. The bridge is on deck 9 and deck 8 has in addition to officer cabins and office, recreation and spa area.

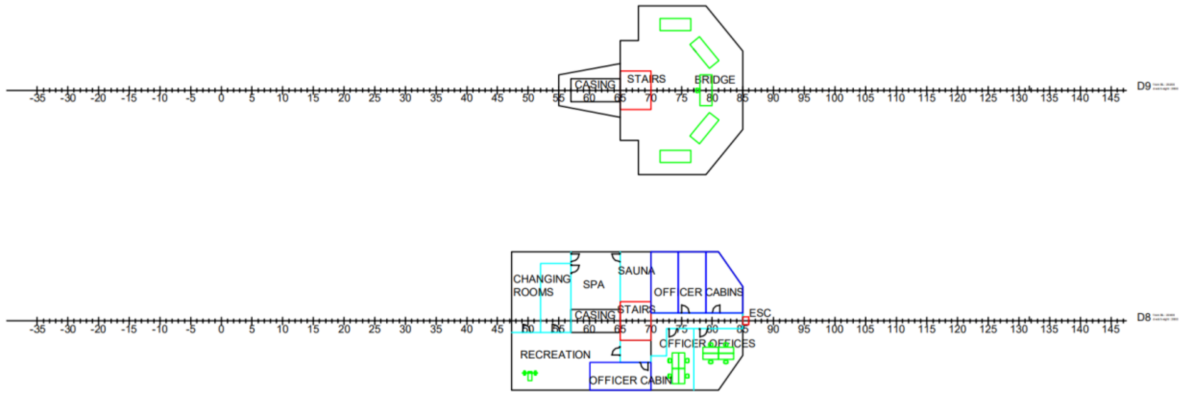

Figure 19 Decks 9 and 8.

Decks 6 and 7 have together 31 crew cabins. Standard cabins are 21.5 m^2 and for two persons. In deck 7 there are offices and in deck 6 there are LSA locker, AC room and linen lockers. Lifeboats and helicopter hangar reach from deck 5 to deck 6.

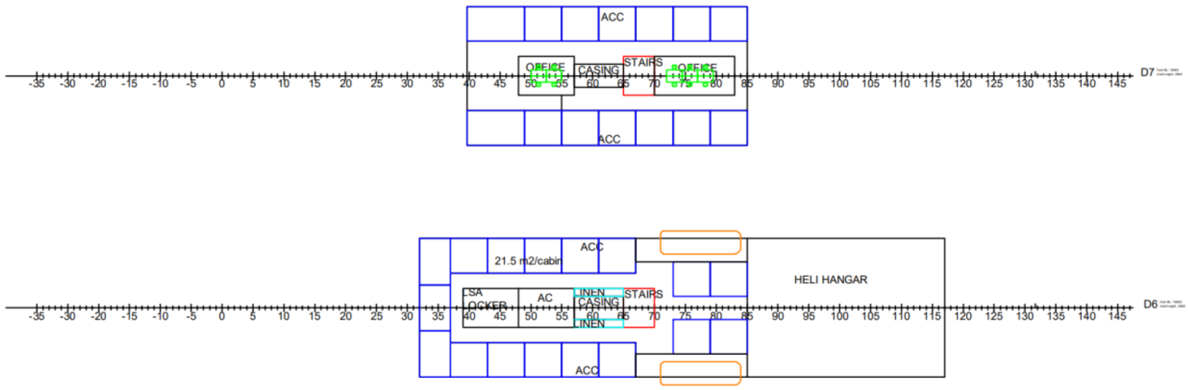

Figure 20 Decks 7 and 6.

Galley, mess, first gym and helipad are located in deck 5. Hydrogen tanks, cargo hatch, boat room, science laboratories, switchboard, AC room, accommodation area, auditorium, recreation area and second gym are on deck 4. In accommodation area cabins are in three rows to prevent dead-ends in corridors. Cargo cranes are used to load and unload cargo but also lift boats, that are used in field research.

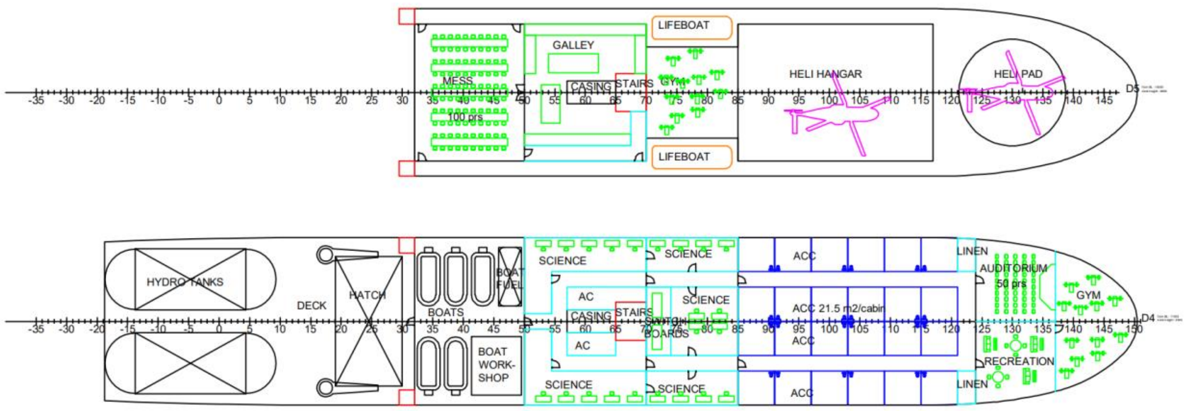

Figure 21 Decks 5 and 4.

Deck 3 has aft and fore moorings, fuel cells, first cargo space, garbage room, hospital, computer room, laundry, workshops and lockers. Side ballast tanks reach from deck 1 to deck 3. On deck 3 in service areas there is dead-end corridors, this is allowed as it is separated from accommodation areas (IMO, 2012). Computers are used in research and big data handling and allow semi-autonomous operating of the ship. There is switchboard in computer room. Deck 2 has more cargo spaces, fresh water room, workshops and void forward from collision bulkhead. Main engine rooms reach from deck 1 to deck 2.

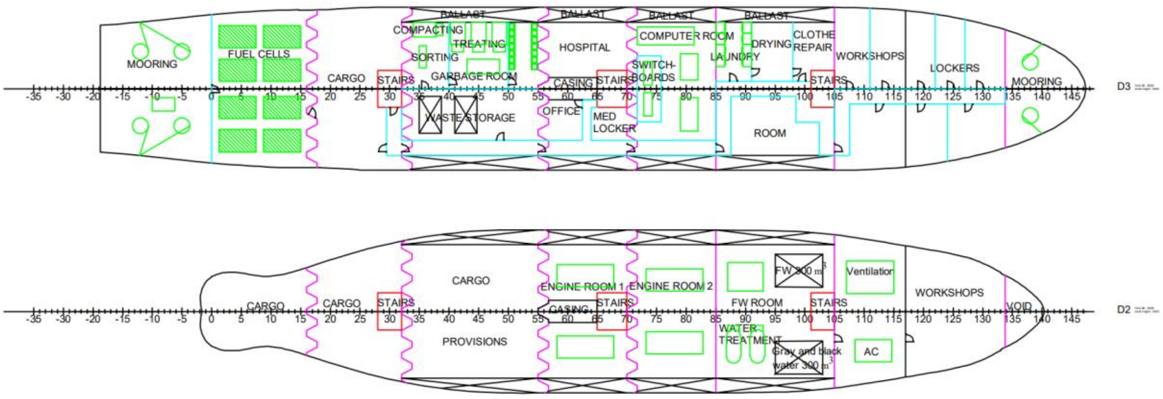

Figure 22 Decks 3 and 2.

On deck 1 there are propulsion room, tanks, switchboard, engine rooms, boiler room, fire pump room for sprinkler system, aviation fuel and bow thuster room. Double bottom has tanks in it.

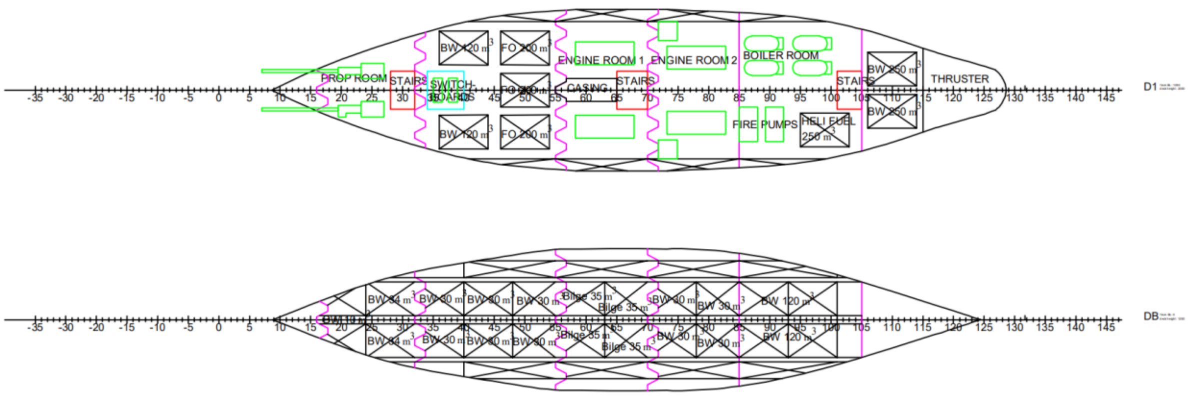

Figure 23 Double bottom and deck 1.

5 Structure

This part will be covered with the report from Ship Structures and Construction report which is submitted afterwards as a separate report.

After modelling the hull in NAPA the modelled surfaces was used for steel model. The hull consists of the shell, bulkheads and deck all stiffened with suitable stiffeners. For the superstructure the decks and shells have been modelled mostly for weight estimations and visualisation. In figures below the steel model is shown. The model includes both transverse and corrugated bulkheads.

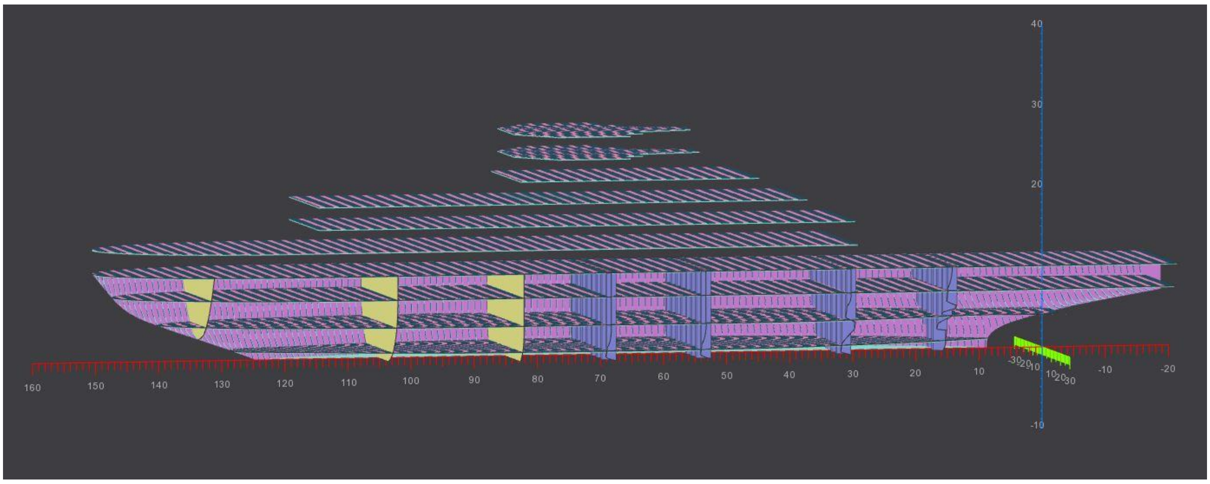

Figure 24. NAPA steel model

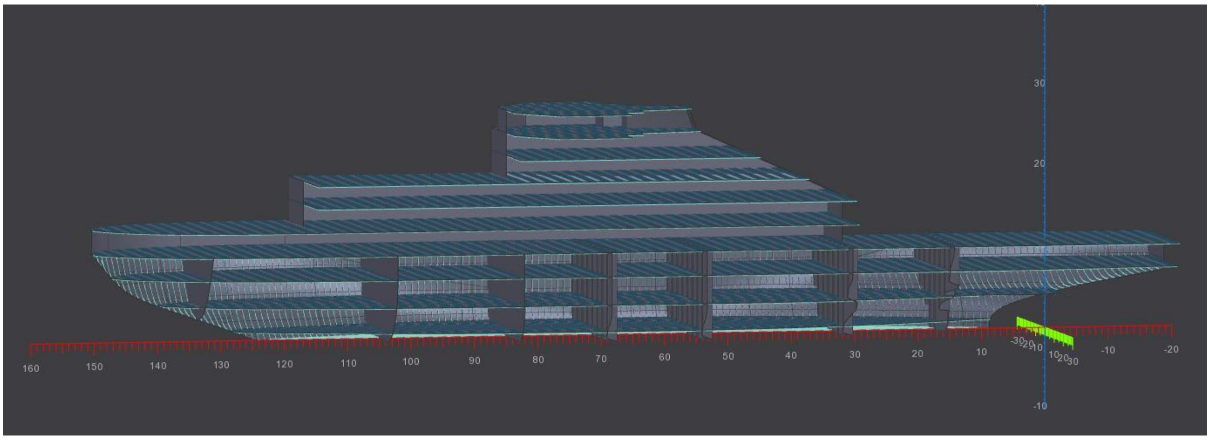

Figure 25. NAPA Steel model

The steel weight has been estimated in Chapter 8.1, including all structural elements to be 3178,8 tons. Weight estimation in NAPA is possible and the estimation for main elements is 2764,5 tons. Only for plates including the decks and shell the estimation is 2296,6 tons and for only stiffeners 468,9 tons. The plate thicknesses in ice belt region are used as plate thickness value evaluated in Ship Structures and Construction course others is used with NAPA default values. As the manual estimation includes more structural elements the difference is understandable.

6 Power and machinery

6.1 Operating profile

Our vessel will not have a specific route that she will always follow. Her operating profile changes as the route she takes varies. Also, the profile will also vary regarding the re-supply and research operations during the voyage. A possible voyage, shown in Figure 29 The example voyage drawn on a map, was defined to give an example of the types of routes we envisage for our vessel. This voyage

Figure 26 The example voyage drawn on a map

starts from Tuktoyaktuk, Canada to Reykjavik, Island via Aasiaat, Greenland. The voyage also includes a stop at the Canadian Arctic Archipelago close to Resolute Bay, where a re-supply operation is conducted with helicopters. The depth of the port of Resolute Bay is 4.9 – 6.1 m (Ports.com, n.d.), which is too low for our vessel.

The first 200 nautical miles include the departure from Tuktoyaktuk and open sea voyage until the sea ice is met as the vessel approaches the Canadian Arctic Archipelago. When meeting the ice our vessel will have to slow down substantially, but it is impossible to give an exact constant speed. The speed is not constant as the ice thickness and condition may vary in the archipelago. As the vessel reaches Resolute, it will stop for the duration of the re-supply operation. Simultaneously, research operations can be conducted, if possible and necessary. The range of a voyage between Tuktoyaktuk and Resolute is approximately 860 nautical miles. The vessel will continue its journey through the ice until it reaches the Baffin Bay, where the sea is no longer ice covered. After approximately 900 nautical miles, the vessel reaches Aasiaat. From there, the voyage will continue to Reykjavik, which it will reach after 1500 nautical miles.

As our vessel will travel a substantial distance in ice, which is always changing, our operating profile does not show a specific route. Instead, it shows our vessels behaviour in different conditions. The profile can be divided into two general areas, one where the ship is operating in open water at its cruising speed of 13 knots and the other one where it is operating in ice.

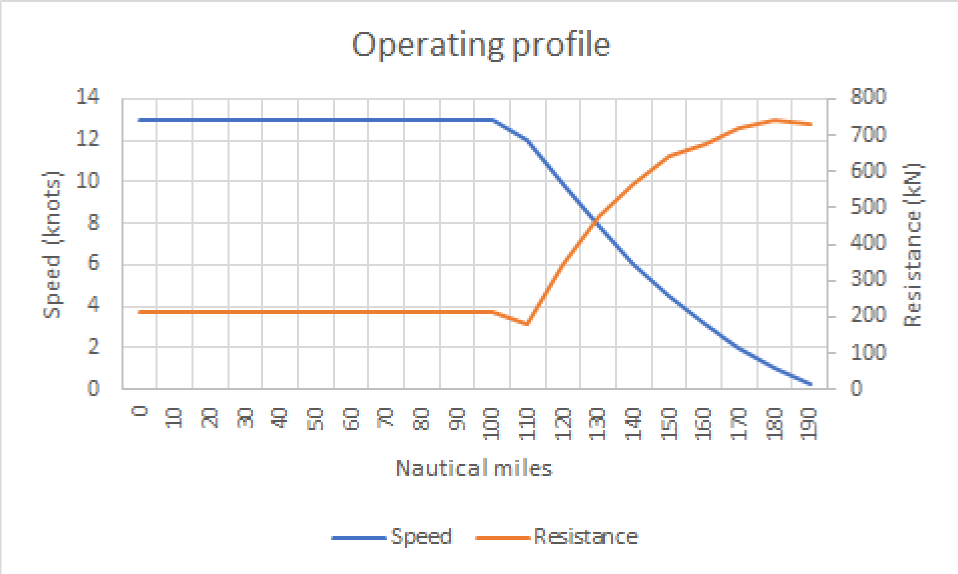

Figure 27 Example operating profile

This Operating profile is of a representation of an idealised voyage for our vessel. On this voyage the first 100 nautical miles are open sea. This means that our ship is doing its cruising speed of 13 knots and encountering the associated resistance. After the first 100 nautical miles the vessel encounters ice. The ice gets thicker and thicker at a rate of 10 cm per 10 nautical miles, meaning that at 110 nautical miles its 10 cm thick, at 120 nautical miles is is 20 cm thick and so on.

6.2 Resistance and Propulsion Power

Air Resistance

The air resistance of a ship is usually quite low compared to the hydrodynamic and if applicable ice resistance on the ship. Usually being about 2 % of the total resistance in open water. Presence of a box-like superstructure increases this resistance and during a headwind the resistance can increase even more, even up to 10%. Khione has a frontal area of 712 m^2 at draft of 6.4m. Frontal area can be divided into the superstructure with area of 230 m^2 and to the bow of the hull with area of 482 m^2. An estimation of the drag can be given by the following equation:

$$R_{AA} = \frac{1}{2}\rho(V + V_{wind})^2 \cdot (A_{Frontal,s} \cdot C_{d.s} + A_{Frontal,b} \cdot C_{d.b})$$

In which $A_{Frontal,s}$ is the frontal area of the superstructure, $A_{Frontal,b}$ is the frontal area of the bow, $C_{d,s} = 1.05$ is the aerodynamic resistance coefficient for cube, as our superstructure is, $C_{d,b} = 0.75$ is the aerodynamic resistance coefficient for the bow of our ship, and R_{AA} is the aerodynamic resistance. Furthermore, ρ is the density of the air at sea level, being 1.225 kg/m^3, V is the speed of the ship and V_{wind} is the expected head wind speed. The latter two are 6.69 m/s and 12 m/s respectively, the wind speed being the average wind speed found on the Atlantic side of the artic ocean. Resulting in average air resistance in headwinds R_{AA} = 129 kN and air resistance without any wind R_{AA}= 16.5 kN. However, the highest wind speed in the artic is about 50 m/s (Przybylak, 2003), which would in headwind situation lead to air resistance of more than R_{AA}= 1000 kN.

Results of the previous calculation compares well with our NAPA resistance results as they predicted air resistance of 10 kN at cruising speed of 13 knots. Superstructure was not taken into consideration in NAPA calculations.

Ice Resistance

Khione is Arctic vessel and thus IACS polar class rules are applied in design. However, Finnish-Swedish Ice Rules have method for calculating ice resistance in brash ice channel and this method is used to estimate easy ice conditions. We calculate FSICR ice resistance for 1A Super class, as it is nearest to Khione's ice class. The FSICR method for Ice resistance in brash ice channel and then minimum power is shown below.

Table 10. Input values for ice resistance.

Input values		
L	127.56	m
L_{bow}	41.4	m
L_{par}	36.2	m
B	21.95	m
T	6.4	m
A_{wf}	500	m^2
α	31	deg.
φ_1	45	deg.
φ_2	44	deg.
K_e	1.44	
D_P	5	m

Table 10 shows input values. L is length of water level and L_{BOW} is length of bow and L_{PAR} is length of parallel mid part on water plane. A_{wf} is area of fore part of the ship on water plane. α is waterline angle, φ_1 is stem angle and φ_2 is rake of the bow at B/4. Minimum engine output is calculated with formula:

$$P = K_e \frac{\left(R_{CH} / 1000\right)^{3/2}}{D_P} \left[kW\right]$$

Where K_e is propulsion system coefficient and R_{CH} is resistance of the ship in ice:

$$R_{CH} = C_1 + C_2 + C_3 C_\mu \left(H_F + H_M\right)^2 \left(B + C_\psi H_F\right) + C_4 L_{PAR} H_F{}^2 + C_5 \left(\frac{LT}{B^2}\right)^3 \frac{A_{wf}}{L}.$$

C_1 and C_2 are coefficients calculated for only for 1A Super class as they take a consolidated layer on top of brash ice in account. Formulas for C_1 and C_2 are:

$$C_1 = f_1 \frac{BL_{PAR}}{2\frac{T}{B}+1} + (1+0.021\varphi_1)(f_2 B + f_3 L_{BOW} + f_4 BL_{BOW}),$$

$$C_2 = (1+0.063\varphi_1)(g_1 + g_2 B) + g_3 \left(1 + 1.2\frac{T}{B}\right)\frac{B^2}{\sqrt{L}}.$$

H_F is the thickness of the brash ice layer displaced by the bow, and H_M is the thickness of the brash ice in middle of channel. C_3, C_4, C_5 and f_1, f_2, f_3, f_4 and g_1, g_2, g_3 are coefficients. Formulas for C_μ and C_ψ are:

$$C_\mu = 0.15\cos\varphi_2 + \sin\alpha\ ,$$

$$C_\psi = 0.047\psi - 2.115\ ,$$

Where $\tan\psi = \tan\varphi_2 / \tan\alpha$. The term $\left(LT / B^2\right)^3$ must equal or greater than 5, for our project ship the term is 4.85 so value 5 is used instead. Ice resistance in channel is R_{CH} = 597.2 kN and minimum power is P = 4203 kW.

For ice resistance in level ice Lindqvist method (Lindqvist, 1989) is applied, which is shown below.

Crushing component R_c for ice resistance is:

$$R_c = 0.5\sigma_B H_{ice}{}^2 \frac{\left(\tan\varphi + \dfrac{\mu_H \cos\varphi}{\cos\psi}\right)}{\left(1 - \dfrac{\mu_H \sin\varphi}{\cos\psi}\right)}$$

, where σ_b is flexural strength of ice and H_{ice} is thickness of ice. μ_H is friction coefficient between ship and ice. φ is the stem angle and α is the waterline entrance angle. ψ is then obtained $\psi = \tan^{-1}\left(\tan\phi / \sin\alpha\right)$.

Bending component R_b for ice resistance is:

$$R_b = 0.003\sigma_B B\left(H_{ice}\right)^{3/2}\left(\tan\psi + \frac{\mu_H \cos\varphi}{\sin\alpha\cos\psi}\right)\left(1 + \frac{1}{\cos\psi}\right)$$

, where B is beam of the ship.

Submersion component R_s for ice resistance is:

$$R_s = \left(\rho_w - \rho_i\right) g H_{ice} B\left[\frac{T\left(B+T\right)}{B+2T} + \mu_H\left(0.7L - \frac{T}{\tan\varphi} - \frac{B}{4\tan\alpha} + T\cos\varphi\cos\psi\sqrt{\frac{1}{\sin^2\varphi} + \frac{1}{\tan^2\alpha}}\right)\right]$$

, where ρ_w and ρ_i are water and ice densities. L and T are length and draft of the ship.

Finally, ice resistance R_{ice} is obtained from formula:

$$R_{ice} = \left(R_c + R_b\right)\left(1 + 1.4\frac{v}{\sqrt{gH_{ice}}}\right) + R_s\left(1 + 9.4\frac{v}{\sqrt{gL}}\right).$$

Ice resistance calculated for different ice thicknesses is shown in Figure 28. The figure also shows net thrust, which is calculated with formula:

$$T_{net} = (P_D D_P)^{2/3} \cdot \left(1 - \frac{V}{3V_{ow}} - \frac{2}{3}\left(\frac{V}{V_{ow}}\right)^2\right),$$

where V is speed of the ship and V_{ow} is open water speed. For delivered power, P_D, main engine output 31720 kW is used.

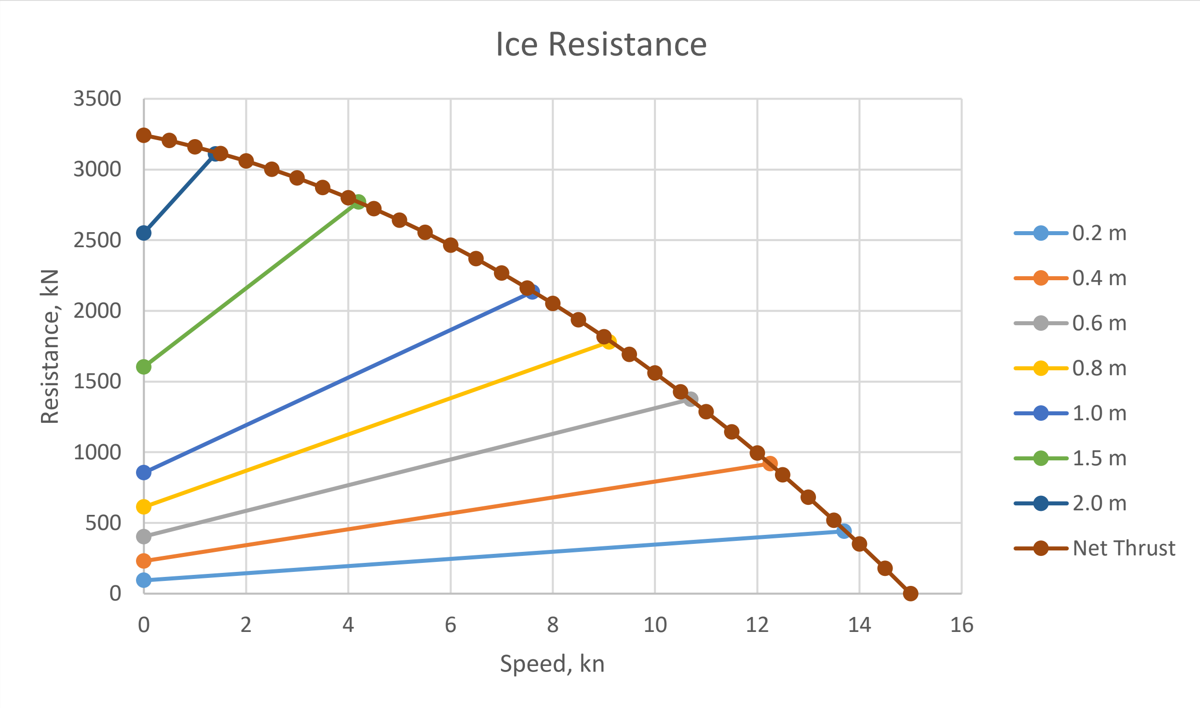

Figure 28 Ice resistance calculated with Lindqvist method and net thrust.

Our project ship is required to obtain speed of 3 knots in 1.65 m thick level ice. Ice resistance in 1.65 m thick level ice when sailing 3 knots is R_{ice} = 2807 kN, which is just below net thrust, meaning requirement is met.

To evaluate our ice resistance calculations, we used different methods to calculate ice resistance. Below is shown Lewis and Edwards method:

$$R_i = C_B \sigma_f Bh + C_S \rho_i gBh^2 + C_V \rho_i gBh^2 \frac{v}{\sqrt{gh}},$$

and Vance method:

$$R_i = C_B \sigma_f Bh + C_S \rho_\Delta gBh^2 + C_V \rho_i LB^{0.35} h^{0.65} v^2,$$

where C_B, C_S, C_V are coefficients, σ_f is flexural strength of the ice, ρ_i is density of ice and ρ_Δ is difference between densities of water and ice. For Lewis and Edwards method Mackinaws coefficients are C_B = 0.019, C_S = 3.455 and C_V = 4.68. For Vance method Mackinaw's coefficients are C_B = 0.034, C_S = 16.91 and C_V = 0.165 and FinnCarrier's coefficients are C_B = 0.05, C_S = 53.7 and C_V

= 0.183. As shown in Figure 29, other methods, especially with Mackinaws coefficients, give similar results to result obtained from Lindqvist method. Ice resistance was calculated for 1 m thick ice.

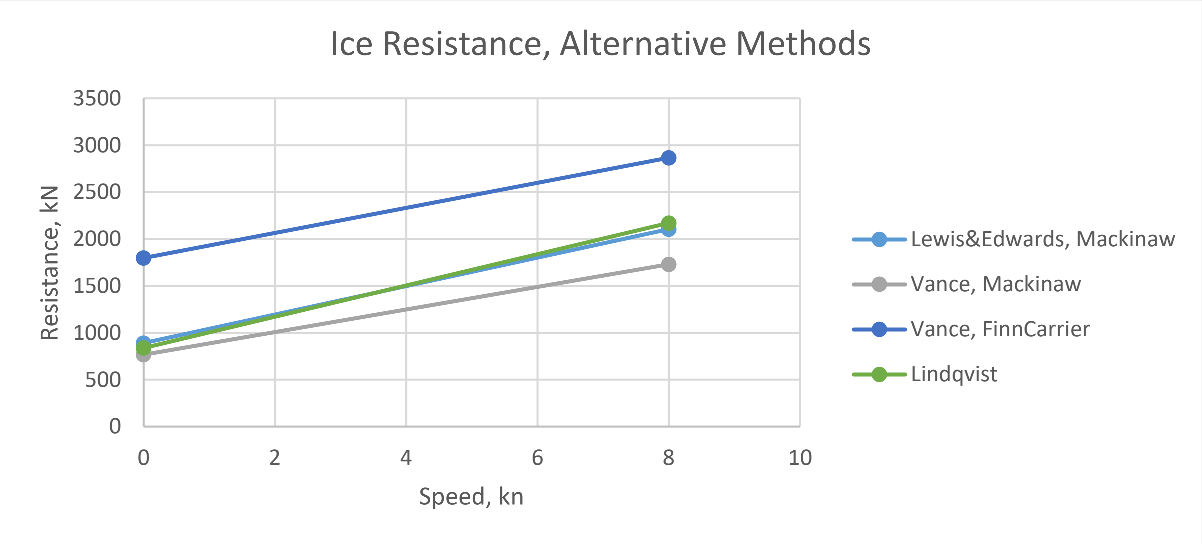

Figure 29 Ice resistance in 1 m thick ice calculated with alternative methods.

Open Water

The open water resistance was during PNA calculated with an given excel sheet. Results at that point were 260 kN for total resistance and effective power 2.6 MW at the cruising speed of 13 knots. Updating and checking the values to new values from the NAPA model the result came out 195.7 kN for total resistance and 1.3 MW. The excel calculation and NAPA iterations presented in the next section both uses two fixed pitch propellers with diameter of 5 m.

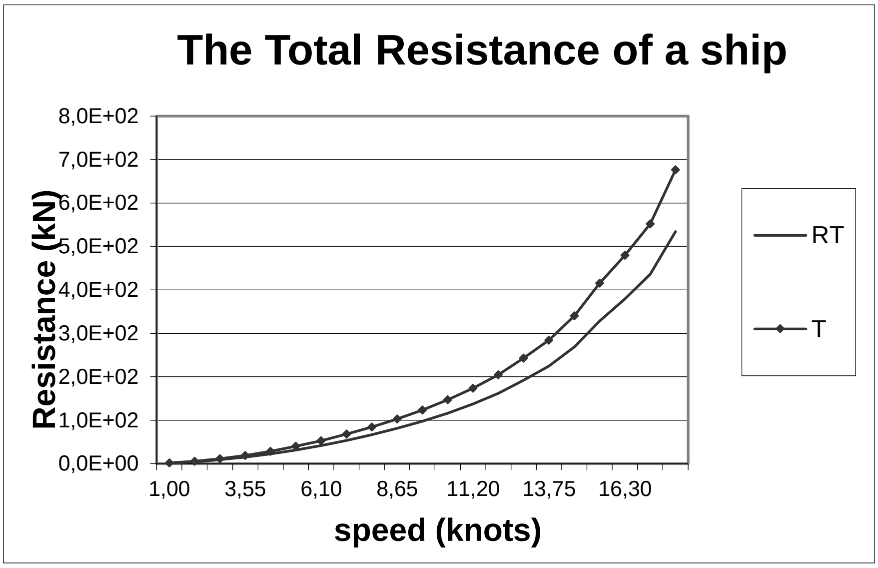

Figure 30 RT is the total resistance and T is the propeller thrust.

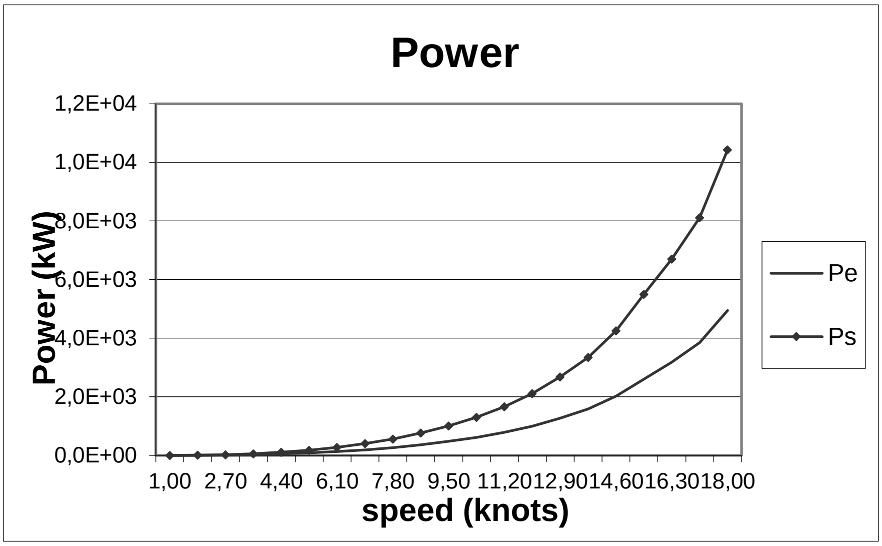

Figure 31 Pe is effective power and Ps is shaft power.

NAPA iteration

To check the manual resistance calculations, we did resistance calculations using NAPA with the Holtrop 84 and Holtrop 82 methods. The iteration was done without appendages.

Fn	Vs	Rt	Pe	RHull	Rwind	RVisc	Rwave
	knots	kN	MW	kN	kN	kN	kN
0.000	0.000	0	0.000	0	0	0	0
0.015	1.000	1	0.001	1	0	1	0
0.029	2.000	5	0.006	5	0	4	0
0.044	3.000	12	0.018	11	1	9	0
0.058	4.000	20	0.041	19	1	16	0
0.073	5.000	31	0.079	29	2	24	0
0.087	6.000	43	0.133	41	2	34	0
0.102	7.000	58	0.208	55	3	45	0
0.116	8.000	75	0.307	71	4	57	0
0.131	9.000	94	0.434	89	5	72	1
0.145	10.000	116	0.596	110	6	87	2
0.160	11.000	142	0.803	134	7	104	5
0.174	12.000	173	1.071	165	9	123	12
0.189	13.000	213	1.424	203	10	142	25
0.204	14.000	263	1.895	251	12	164	47
0.218	15.000	328	2.529	314	14	186	81
0.233	16.000	410	3.373	394	16	210	131
0.247	17.000	508	4.446	491	18	236	195
0.262	18.000	637	5.901	618	20	262	288
0.276	19.000	819	8.001	797	22	290	432
0.291	20.000	1042	10.717	1017	24	320	615
0.305	21.000	1257	13.581	1230	27	350	789
0.320	22.000	1436	16.251	1406	29	382	924

Figure 32. Holtrop 84 Resistance calculations from NAPA.

From the Holtrop 84 method the total resistance resulted to 213 kN and 1.4 MW at Khiones cruising speed 13 knots.

Fn	Vs	Rt	Pe	RHull	Rwind	RVisc	Rwave
	knots	kN	MW	kN	kN	kN	kN
0.000	0.000	0	0.000	0	0	0	0
0.015	1.000	1	0.001	1	0	1	0
0.029	2.000	5	0.005	5	0	4	0
0.044	3.000	11	0.018	11	1	9	0
0.058	4.000	20	0.040	19	1	15	0
0.073	5.000	30	0.077	28	2	23	0
0.087	6.000	42	0.131	40	2	33	0
0.102	7.000	57	0.204	54	3	44	0
0.116	8.000	73	0.301	69	4	56	0
0.131	9.000	92	0.426	87	5	70	1
0.145	10.000	114	0.584	108	6	85	2
0.160	11.000	139	0.787	132	7	102	5
0.174	12.000	170	1.049	161	9	119	12
0.189	13.000	210	1.407	200	10	139	26
0.204	14.000	256	1.843	244	12	159	44
0.218	15.000	329	2.536	315	14	182	87
0.233	16.000	407	3.354	392	16	205	134
0.247	17.000	480	4.202	463	18	230	174
0.262	18.000	618	5.722	598	20	256	276
0.276	19.000	855	8.354	833	22	283	475
0.291	20.000	1117	11.495	1093	24	311	699
0.305	21.000	1296	13.997	1269	27	341	836
0.320	22.000	1399	15.835	1370	29	372	897

Figure 33. Holtrop 82 Resistance calculations from NAPA.

Respective for the results from Holtrop 82 method at cruising speed is 210 kN and 1.4 MW. The results from these methods are close to each other which makes the results reliable. The results have more difference in the higher speeds, 0.42 MW at 22 knots, although the difference will probably not affect our later choices which depends on the power demand. Comparing these results to the previous calculation using excel the values are in the same range. The changes can be due to NAPA using the exact hull and the values in excel can be approximated and some appendages included.

The resistance and engine output was also calculated for the ship in a channel with brash ice. Since Khione has the PC4 the highest 1AS class was used to get an estimation. Using the input values with to fixed-pitch propellers with diameter 5 m the results was at the load draft 6.4 m, $R_{CH} = 597.97\ kN$ and an engine output at 4.7 MW. With NAPA's estimated ballast draft 5.76 m the values resulted to $R_{CH} = 453.8\ kN$ with engine output at 3.1 MW.

Since Khione are operating in heavy ice conditions and will need the icebreaking power ice resistance using the Lindqvist method will be followed.

6.3 Total Power Demand

Total power demand of main engines is set by propulsion power as the hotel load of the ship is powered with auxiliary fuel, hydrogen. As calculated in the previous chapter the resistance in 1.65 m ice is 2807 kW which results to a power requirement of 29150 kW (engine output using Finnish-Swedish ice class rules). In the assignment description we started with a requirement to have the ability of producing propulsion power of 26600 kW which now will be fulfilled, with our power requirement and engine choices presented in later chapters. The power in open water is stated in the resistance calculations as 1400 kW and in lighter ice conditions to 4200 kW.

As auxiliary energy source Khione will use hydrogen. During PNA the estimated power powered with hydrogen solution was calculated to 617 kW including the hotel load, research and helicopter hangar. Since auxiliary power also includes lighting, HVAC, emergency system, navigation, galleys etc. By estimating power needed for different systems and based on installed machinery on reference vessels the total auxiliary power demand is estimated to roughly 3000 kW. As the main engines will have the capability to produce a high amount of power and the full capacity will only be needed when breaking thick ice, the diesel-electric system can also be used to power auxiliary consumers. Thus, hydrogen is installed for to power 1000 kW to limit the size for tanks etc. needed onboard. As mentioned, the use of hydrogen is used as an auxiliary power as an innovation for less emissions and for possibilities for research of use of hydrogen on vessels and especially in arctic environment.

Table 11 Power demand with estimated power need for auxiliary systems

Column1	Open water	Light ice cond.	Heavy ice cond.
Hours	538	192	38
Propulsion	1400	4203	29150
Hotel	600	600	600
HVAC	800	800	800
Research	500	500	500
Navigation	80	80	80
Deck equipment	500	500	500
Safety equipment	400	400	400
Water and waste systems	20	20	20
Sum	4300	7103	32050

6.4 Propulsor(s)

Propulsion systems that are usually used are controllable pitch, fixed pitch propellers or podded/azimuthing propulsors. Khione is design to have two shafts which means the alternatives left is to use propellers on shaft, thus it is chosen to use fixed-pitch propellers.

The fixed-pitch system does not have any mechanical or hydraulic connection which makes it simple and reliable although the manoeuvrability would be better with a controlled pitch propeller. Fixed-pitch propellers are also cost-effective since the manufacturing, installation and operational costs are lower than for other types. Since the components of the propulsion system can be located nearly anywhere the generators for the shafts can be fitted at the aft of the ship making the shafts shorter. Manufacturers for fixed pitch propellers is for example Wärtsilä and Kongsberg. Kongsberg offers propellers with 5 blades and other set features whereas propellers from Wärtsilä can more tailor-made and optimised to the needs and requirements. Wärtsilä also has a long experience of manufacturing propellers and has also specialization for more advanced applications. Propellers with diameter of 5m has been used in the resistance calculations.

Another alternative for shafts is controllable pitch propellers. These alternatives have benefits for better manoeuvrability but would not necessarily be that cost-effective. Advantages with controllable pitch propellers is the ability to change propellers pitch for changing direction or speed of ship rather than changing the speed for main engines (Marine Insight, 2019).

6.5 Energy Sources

Ice-going vessels usually uses diesel-electric propulsion system, medium speed diesel and gearbox or low-speed diesel with direct shaft. The diesel-electric system is common in icebreakers since it’s efficient at slow speeds and has excellent manoeuvring characteristics (Traficom, 2019). Generally, diesel-electric systems consist of diesel engines that drives electric generators to then produce electric power for the propellers and other use. Another alternative for propulsion system is to use diesel generators to produce energy directly for the propulsors, but with using a diesel-electric system the use of space is more efficient and flexible since the diesel generators, switchgear and propulsion motors can be located nearly anywhere (Aichele, 2007). Our vessel has two engine rooms with two engines each.

The vessel will be equipped with diesel-electric machinery which will use marine diesel oil as an energy source. Since the vessel will operate in arctic the fuel used must be reliable and easily available for the long operating times. Also, the environmental impact of the fuel is important. Marine diesel oil (MDO) is a blend of distillates (marine gasoil) and heavy fuel oil (HFO). Since the content of heavy fuel oil is low, the marine diesel oil has a maximum sulphur content of 3,5%. It comes also in a low-sulphur variant (1%) which can be used if the vessel crosses areas with stricter emission limits. In the Arctic polar code area ships mostly uses distillate marine fuels (marine gas oil and marine diesel oil) and 10% uses HFO. MDO fuels are more expensive than HFO and therefore a big part of shipping still uses HFO. (Marquard & Bahls, 2015)

Figure 34 Example of Diesel-electric propulsion system (Ocean Time Marine, 2020)

Accommodation and Auxiliary Power

As an auxiliary power source for other power than propulsion it's decided to use hydrogen. The use of hydrogen is an innovation for our vessel and has been chosen since renewable energy sources like sun and wind would not have been effective and reliable enough. It would be possible to use the main energy source for accommodation, but the choice to use hydrogen has also a research aspect and will enable testing the use of hydrogen in both vessels and the function in cold areas. The tanks for liquid hydrogen are fitted at the aft on the vessel and will also need fuel cells and other systems for the power transmission.

According to the example operating profile, the ship will spend 16 days at sea with one stop in between at Asiaat. It will start in Tuktoyaktuk and end its journey in Reykjavik. A normal ship could probably refuel in both Tuktoyaktuk and Asiaat, however this ship uses hydrogen as its non-propulsive power source. One can hardly expect small settlements like Tuktoyaktuk and Asiaat to have the infrastructure to refuel hydrogen. So realistically, the ship can only potentially refuel its hydrogen once every two trips, which is when it is in Reykjavik. Thus, the ship is required to carry enough hydrogen for 32 days, a little over one month. However, since this is just an example operating profile, it is possible that the ship would also be taking longer trips. Therefore, it was decided that the ship will carry enough energy for 64 days at sea without refuelling hydrogen.

The hydrogen tanks are fitted at the aft due to safety. A solution for fuel cells is from the company Ballard and their FCwave fuel cell solution designed for powering vessels. The system is scalable from 200kW to MWs and can be used for example to power ferries and hotel load on cruise ships (Ballard, 2020). One fuel cell has rated power of 200 kW, dimensions are 1,22m x 0,738m x 2,2m and weight 875 kg which makes it the integration modular and flexible (Ballard, 2020). The system is also DNV GL compliant and has hydrogen safe enclosure. The engine room will need to have temperature between 0 – 45 °C. Using this solution 1000 kW could be implemented with 5 systems. As fuel cells produce waste heat and water the heat it can be used for purposes needing heat e.g. water heating and laundry. Regarding rules and requirements of the use of hydrogen and fuel cells the tanks shall be pressure tanks for maritime use and safe handling of hydrogen ensured, the ventilation for the fuel cell system should be specified for hydrogen and safety assessments for the piping (EMSA, 2017). As mentioned the vessel will carry hydrogen for 64 days, thus the required volume for tanks are calculated to 651 m^3 using the energy required and specific energy and energy density for hydrogen. The tanks volume can be divided into two tanks and fitted at the aft as planned.

6.6 Machinery Configuration

The main components of diesel-electric machinery are diesel generators, electric switchboards, electric propulsion motors and control room.

The diesel generators used will be 4-stroke engines by Wärtsilä, 2 x 12V31 and 2 x 14V31. Wärtsilä 31 is suitable for main propulsion, in hybrid installations, as auxiliary engine and as our case in diesel-electric configurations. These engines are chosen since Wärtsilä is a well-known company and the Wärtsilä 31 engines are designed to have good efficiency and emission performance, and to meet our needed power demand. The diesel version of the engine is separately optimised for heavy or light distillate fuels. (Wärtsilä, 2019)

The rated power for the 12V31 is 7320 kW and for 14V31 8540 kW which all together will provide 31720 kW.

Table 12 Engine dimensions (mm) and weights (tonnes).

Engine type	A*	A	B	C	F	Weight
12V31	7900	7840	3137	3500	1496	77.1
14V31	8540	8480	3137	3500	1496	84.6

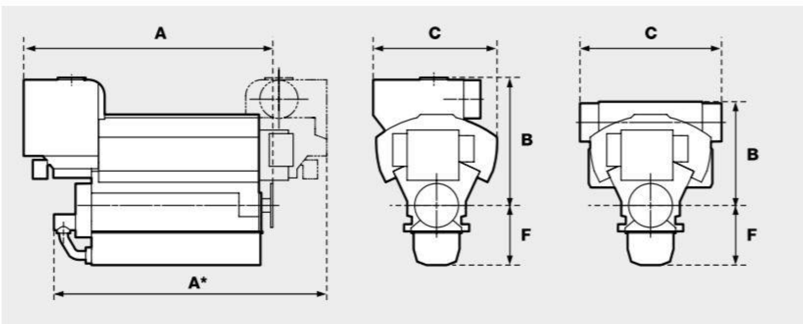

Figure 35 and 36 Wärtsilä 31 and definition of dimensions (Wärtsilä, 2019).

Table 13 Engine alternatives

Engine	Volume (m3)	Weight (tonnes)	Power (kW)
Wärtsilä 12V31	128.1	77.1	7320
Wärtsilä 14V31	138.5	84.6	8540
Wärtsilä 12V32	74.9	57	6960
Wärtsilä 16V32	101.7	71	9280
MAN 12V32	176.2	117	7200
MAN 14V32	186.0	131	8120

For selecting suitable diesel engines, the aspects like volume, weight and power have been considered.

Another option would be Wärtsilä 32 motors, 2 x 12V32 and 2 x 16V32 which together would provide 32480kW. The 32 range diesel engines provide the best power-to-weight and power-to-space ratio. Similar to the Wärtsilä 32 is same kind of engines from MAN, the difference compared to the Wärtsilä engines in size and weight is relatively large. Wärtsilä 12V31 and 14V31 engines are also chosen for their stated high efficiency and environmental aspects with low fuel consumption and high cylinder power 610kW/cylinder.

Since this ship is diesel-electric, the two propellers are not directly driven by the engine but by electric motors. The electricity generated by the engines is used to power these electric motors. An electric motor is a device that turns the generated electric energy into angular kinetic energy i.e., the rotation of the propeller shaft. The big advantage of electric motors over diesel motors is that motors can provide maximum torque at any speed, which is great for an icebreaker that must push its way through ice at a low speed (Wärtsilä, 2016). Each motor needs to be able to produce about 15 MW of power. One option is to apply one ABB AMZ 1250 synchronous motor per shaft. These motors have an output up to 30 MW at 750 rpm (ABB, 2018).

The switchboard of the system distributes the power from the generators. The main switchboard is usually located in the main engine room or machinery control room. In event of fire there's required installations which will shut down all ventilation and fuel oil systems (ETO, 2020).

Table 14 and 15 Engine details and fuel consumption

Engine details		
Engine output 14V31	8540	kW
Engine output 12V31	7320	kW
Engine output total	31720	kW
Fuel consumption	170.1	g/kWh
Lube oil consumption	0.45	g/kWh

	Open water	Light ice cond.	Heavy ice cond.	Sum.
Per voyage	70%	25%	5%	100%
Hours / voyage, h	537.6	192	38.4	768
Prop. Power, kW	1400	4203	29000	
Fuel cons., kg/h	238	715	4933	
Fuel/voyage, ton	128	137	189	454

Space requirements from the Polar Code according the machinery is to take the environmental conditions into account. Machinery shall be installed so that it's protected for example from ice accretion, snow accumulation and freezing and increased viscosity of liquids. DNV GL also sets requirements for the environmental conditions in the machinery space and general requirements for the construction. All spaces with machinery need ventilation under all conditions.

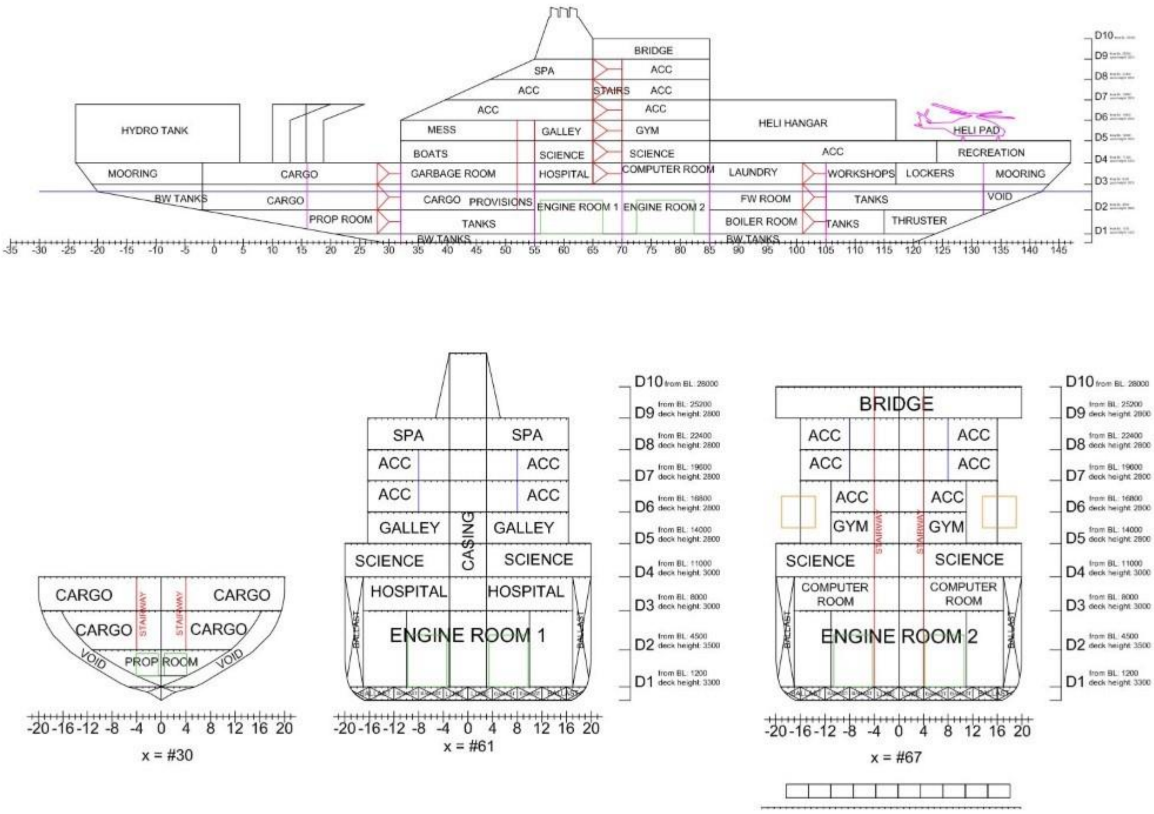

Figure 37 General Arrangement with fitted engines.

6.7 Machinery optimization

For comparing different fuel types CO_2 emissions the amount of emissions can be calculated with emission factors and the amount of fuel. For the comparison LNG and ethanol are chosen, as LNG are becoming more common in ships and ethanol could also be a future energy source in ships.

Table 16 Fuel comparison

	Open water	Light ice cond.	Heavy ice cond.	Sum.
Marine diesel oil 170.1 g/kWh **Fuel cons., kg/h** **Fuel/voyage, ton**	 238 128	 715 137	 4933 189	 454
LNG 140.5 g/kWh **Fuel cons. Kg/h** **Fuel/voyage, ton**	 197 106	 591 113	 4075 156	 375

Ethanol 251.5g/kWh **Fuel cons. Kg/h** **Fuel/voyage, ton**	352 189	1057 203	7296 280	672

Table 17 Emission calculation for different fuel types per voyage

Fuel Type	Emission factor (t CO_2 / t fuel)	Fuel Consumption (t)	CO2 emissions (t CO_2)
Diesel/Gas Oil	3.206	454	1455.5
Liquefied Natural Gas	2.750	375	1031
Ethanol	1.913	672	1286

The approximations shows that LNG has the lowest fuel consumption (140.5 g/kWh) and a emission factor at 2.75 t CO2/t fuel which results in less emissions. Otherwise, ethanol has the lowest emission factor from the compared fuels but highest fuel consumption (251.5 g/kWh). From the compared fuels diesel/gas oil has the highest amount of emission. Due to the operating profile in arctic it's not likely that fuels like LNG and ethanol can be refueled. Choosing a dual fuel engine would enable the use of LNG when operating outside arctic areas.

An alternative to lower costs from the machinery implementation would be to not use hydrogen as energy source for the auxiliary power. For easier implementation, the vessel could have engines with higher power output to cover the hotel load as well. As the power from hydrogen is around 1000 kW it would not necessarily cause big changes in the machinery. The diesel-electric solutions should also be suitable and efficient for that. But the choice to use hydrogen was chosen as an innovation and for the possibility to do further research on the use of hydrogen as a fuel.

For better efficiency, the vessel could have been designed to use azipod or other azimuthing thrusters. They have been used on several icebreakers and could have benefits for maneuvering in harsh conditions. Since the vessel was firstly designed for shaft propulsion and the cost of azipods are higher the azimuthing thrusters left without further consideration. Controllable pitch propellers were considered for Khione in the PNA-course but from received feedback on their pros for our purpose and cost-effectiveness they don't seem to be the best option.

6.8 Risk-based design and environmental aspects

Evaluating the risks in the design and solutions is important. Following the given rules and legislation in designing the vessel ensures some level of safety but that doesn't neglect risks.

The risks coming from the operating area as harsh weather conditions can be minimized by following the rules regarding ice strengthening. The machinery installations will also be implemented as described in the chapter rules and legislation. Our example operating profile shows that the places in the arctic doesn't have that much population or services which means that the vessel needs to have enough storage for all necessary for the crew for the whole operating time.

There are also risks regarding the machinery solutions. The diesel-electric machinery is commonly used which will make it a safe solution but of course accidents can happen. Use of hydrogen is not that widely used, and the use of hydrogen needs some solutions regarding the safety, as example the tanks are fitted at the outside on the aft in cause of hydrogen spills.

In this case the environmental aspects are important. As the vessel operates in the arctic areas the emissions should be controlled not to be too high. In the machinery optimization chapter, it's shown that other fuels like LNG and ethanol would enable less emissions than marine diesel oil, but they are not mainly suitable for our main operating purpose. As using hydrogen as auxiliary fuel the emissions can be lowered and enable further research on the possibilities with using hydrogen as a fuel.

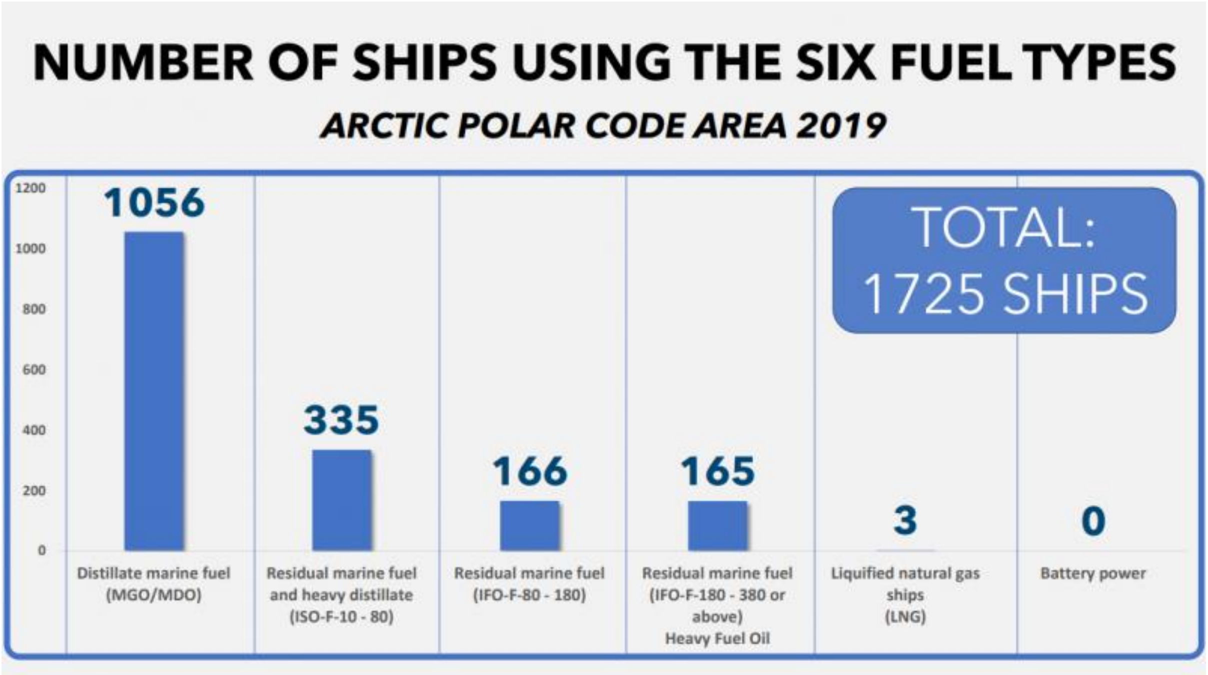

Figure 38 Number of ships using different fuels (Arctic Council, 2020)

As seen from the figure most of the vessels in the arctic polar code area uses marine gas oil and marine diesel oil as fuel. Compared to heavy fuel oil they have less emissions and better abilities in fuel spills into water or ice. (Arctic Council, 2020)

As an environmental aspect as well the power of the machinery needs to be enough for icebreaking which is solved with calculating the power requirements with ice class rules.

6.9 Rules and legislation

The DNV GL IMO Polar code will ensure that the machinery systems maintains functions necessary for safety when operating in the polar area. The requirements are divided into groups: all ships, operation in cold air temperature and ships with ice strengthening. The requirements for all ships is that the machinery installations shall be protected from ice and snow accumulation, snow and ice ingestion at seawater and ventilation intakes, seawater intake temperature, and freezing and increased viscosity of liquids, gases and other essential substances. For the ice-strengthened ships the loads from the interaction with ice shall be considered in machinery installations. (DNV GL, 2017)

Following the DNV GL gives rules for ice-going vessels and contains rules for polar classes, defining the propeller blade design, propulsion line components, steering system, prime movers, and auxiliary systems etc., the machinery and propulsors haven't been inspected on a level where these have been needed. (DNV GL, 2017)

Finnish-Swedish ice class rules are followed in the calculation for ice resistance and power requirements cause the rules for polar classes doesn't have similar formulas. As regulations regarding fuels used in arctic areas the use of heavy fuel is restricted and will also be banned in 2024 and 2029 for countries with coastline bordering arctic waters (Reuters, 2020).

7 Outfitting

7.1 Main Equipment

For effective and safe operation, and for fulfilling our mission, ship must be properly equipped. Most crucial equipment for our ship is listed and categorized below.

Table 18 Required equipment.

Life saving	Lifeboats (enclosed type) Fast rescue boat Adequate thermal protection clothing for the crew Insulated immersion suits Searchlights Emergency signal equipment
Navigation equipment	Compasses Radar Autopilot GPS
Communication equipment	Lifeboat communication system and transmitters Ship communication system
Fire safety equipment	Fire extinguishers Fire pump Sprinkler system Firefighting outfits Fire hoses
Cargo handling	Cranes
Mooring and anchoring equipment	Mooring winches Anchor windlasses Chain stoppers Fairleads Anchors Chains Ropes
Research	Chainsaws Sensors Drills

7.2 Properties of main equipment

Anchoring and mooring

Required properties of anchoring and mooring system can be defined by equipment number provided by DNV GL. Formula for equipment number is presented in DNV GL Rules of Classification of Ships part 3 chapter 3 as below.

$$EN = \Delta^{2/3} + 2 \times B \times H + 0.1A$$

, where:

H = height from the summer load waterline to the top of the uppermost deckhouse, in m

Δ = Displacement in tonnes

B = Greatest breadth

A = area of profile view of the hull, in m^2

With our values of $A = 1585\ m^2$, $B = 22$ m, $H = 18.2$ m, $\Delta = 12\ 900$ t, equipment number EN results in 1509. Based on the equipment number DNV GL requires our vessel to have:

- 2x 4590 kg anchors
- 550 m chain
    - Diameter of the chain 52 mm – 68 mm (depending on material)
- 5 mooring lines
    - 190 m each
    - Minimum breaking strength 324 kN

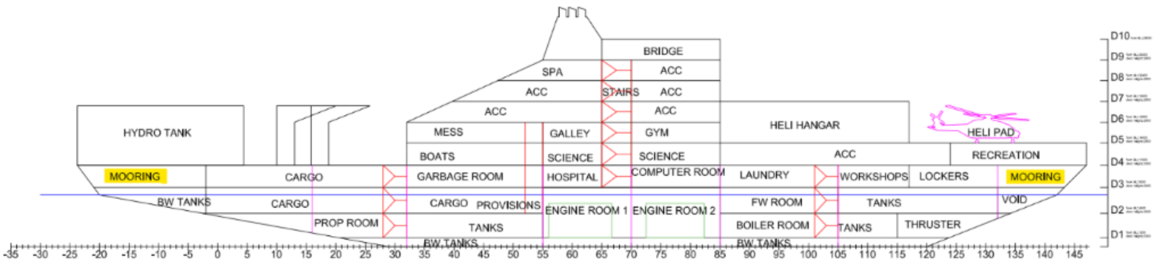

Figure 39 Location of mooring

Life-saving

Our ship is equipped with two lifeboats and one fast rescue boat, which is also used in research operations. SOLAS demands that lifeboats are at least 7.3 m long and have capacity of accommodating at least 125 % of the crew. In our case crew consists of 100 members, thus capacity must be 125 pax. Capacity of each lifeboat on board is 63, thus requirement is fulfilled. Lifeboats should have equipment like food, first aid, compass, signaling mirror and communication systems.

The type of totally enclosed lifeboats used on our ship are JYN-80.

Cargo Handling

Our cargo handling capability consists of two hydraulic TTS Cargo cranes on aft deck. Main crane is type CCL -crane and is capable of lifting 30 tonnes at 18 m. Auxiliary crane has smaller lifting capability but longer reach. Auxiliary crane is type GPC -crane with capability of 10 tonnes at 24 m. Main crane weights 47 tonnes and auxiliary crane 20 tonnes.

7.3 SFI Classification

This extensive section contains the SFI classification of the ship. It is mostly taken from (Wärtsilä, 2011), with changes where that is appropriate.

Main Group	Group	Sub groups	Code
Group 1 Ship General	11 Trials and tests	Trials general	111
		Machinery testing	112
		Inclining experiment	113

		Fuel- oil, lub. oil and hydraulic oil	114
		Dock testing and Trial trip	115
		Post Seatrial Inspection	116
	12 Guarantee	Guarantee	121
Group 2 Hull	20 Hull materials	Hull general	200
		Hull materials	201
		Sandblasting, Priming and Painting	202
		Testing of Tanks, Bulkheads etc	203
		All decks, flats, shell, bulkheads, etc. shall be hose tested as required by classification rules.	204
		X-ray and Ultrasonic testing of Hull parts	205
	21 Aft body	General from stern to bulkhead 3	210
		Shell plates	211
		Steering gear room	212
	22 Engine area	General, bulkhead 4 to 6	220
		Shell Plating	221
		Bottom, keel	222
		Inner bottom	223
		Deck platforms	224
	23 Midship/Cargo area	General, bulkhead 3-4 and 6-7	230
		Deck	231
		Bulkheads	232
	24 Forebody	General, bulkhead 7 to bow	240
		Shell plates	241
		Bow and stem section	242
	25 Superstructure and deckhouse	Superstructure and Deck house	250
		Superstructure	251
		Wheelhouse	252
	26 Hull outfitting	Hull marking	261
	27 Material protection external	Painting - General	270

		Superstructure, deckhouse	271
	28 Material protection internal	Ballast tanks, oily water tank, chain lockers, cofferdams, void spaces, roll reduction tank	281
		Fresh water tanks	282
		Fuel oil, lube oil, and hydraulic oil tanks	283
		Water ballast tanks	284
	29 Miscellaneous hull work	Miscellaneous internal areas, vent and air trunks, all other surfaces	290
Group 3 Equipment for cargo	31 Equipment for cargo	Cargo fittings on 2nd deck	311
Group 4 Ship Equipment	40 Manoeuvring machinery and equipment	Maneuvring control	401
		Rudder	402
		Steering gear	403
		Bow thrusters	404
	41 Navigation equipment	Radar system	411
		Sonar system	412
		GPS	413
		Gyro plants, Auto pilot, Compasses	414
		Echo sounder, Speed log	415
	42 Communication equipment	Radio plant	421
		Local area network (LAN)	422
		Calling, command and telephone systems	423
		Light and Signalling equipment	424
	43 Anchoring and mooring equipment	Anchor with chain and equipment	431
		Fixed mooring equipment	432

	44 Repair and cleaning equipment	Repair and maintenance equipment	441
		Washing system	442
		Incinerator	443
		Outfitting in store rooms	444
		Piping	445
Group 5 Equipment for crew and passengers	50 Lifesaving Equipment	General lifesaving equipment	500
		0 MOB boats	501
		503 Emergency marking	502
		Medicine and First Aid Equipment	503
		Loose firefighting equipment	504
	51 Insulation, panels, bulkhead, doors, side scuttles and windows	General	510
		Insulation, bulkheads and panelling	511
		Doors with coamings in accommodation	512
		External doors with coamings	513
		Side scuttles and windows	514
	52 Internal deck covering, ladders, steps, railing	Deck base covering, internal	521
		Deck top covering, internal	522
		Stairs, handrails in accommodation	523
		Floor plates, ladders and pl.forms in engine room	524
		Ladders, Platforms, Railings etc in tanks	525
	53 External decks	Deck covering	531
		Hand rails, Railings and Gates	532
		Ladders and Steps	533

	54 Furniture and Inventory	Crew Furniture	541
		Researcher furniture	542
		Communinal furniture	543
		Hospital supplies	544
	55 Galley, pantry, Provisions and laundry equipment	Galley and pantry equipment	541
		freezing and refrigeration system	542
		Laundry	543
		Garbage	544
	56 Transport equipment for crew	Gangway	561
		Ladder	562
	57 Ventilation, aircondition and heating systems	Ventilation and Aircond. systems for Accomodation	571
		Ventilation for the remaining parts of the vessel	572
		Ventilation for engine control room/ SW-board room	573
		Ventilation Engine room	574
		Ventilation of cargo area	575
	58 Sanitary system and equipment	Sanitary supply system	581
		Sanitary discharge system	582
		Sanitary equipment	583
		Drinking water system	584
Group 6 Machinery main components	61 Generator and motor	1 Generator	611
		2 Motor	612
		3 Switchboard	613
	62 Propellers, Transmission and foils	Propeller	621
		Propshaft	622

Group 7 Systems for machinery components	70 Fuel oil system	General fuel oil system	700
		Fuel oil transfer and drain system	701
		Fuel purification plant	702
		Fuel oil service system	703
	71 Lub oil system	Lub oil transfer and drain system	711
		Lube Oil Purification System	712
		Lub oil system for propulsion machinery	713
	72 Cooling system	General	720
		Sea water cooling system	721
		Fresh water cooling system	722
	74 Exhaust system	Exhaust gas system	741
		Exaust heat distribution system	742
	76 Distilled & make up water systems	Freshwater generators	761
		Fuel cell water recapture system	762
	79 Automation system for machinery and cargo systems	General	790
		Engine control room	791
		Common automatic equipment, engine alarm etc.	792
		Automation equipment for propulsion machinery and transmission, engine telegraph etc.	793
		Fuel cell control system	794
Group 8 Ship systems	80 Ballast, bilgde and drain systems, gutter pipes outside accomodation	General	800
		Ballast system	801

		Bilge system	803
		Scupper pipes outside accomodation	804
	81 Fire and lifeboat alarm systems, fire fighting systems	Fire fighting general	810
		Fire detection, fire and general alarm system	811
		Fire and washdown system	812
		Fire fighting system with gas	813
	82 Air and sounding system	Air and sounding systems in tanks	821
	83 Special common hydraulics oil systems	Special hydraulic oil systems	831
	85 Electrical systems, general part	Electrical system general	850
		Administrative net work	851
	86 Electrical supply system	General electrical supply system	860
		Shore Connection box	861
	87 Electrical common distribution	Main Ship service and Emergency switchboards	870
		Main Switchboard	871
		Emergency switchboard	872
		Distribution boards and panels	873
	88 Electrical cables and installation	Cableways general	880
		Cableways in accommodation	881
		Cableways on external decks	882
	89 Electrical distribution system	Electrical lighting systems for engine room etc	891

		Electrical lighting for superstructure/accommodation	892
		Electrical lighting system for weather decks	893
		Electrical motors, general	894

7.4 Machinery systems

The SFI-classification is a classification used in the maritime and offshore industry. The SFI system is built up as a 3-digit decimal classification system, which contains of 7 main groups (1 digit) consisting of 10 groups (2 digit) which are divided into 10 sub-groups (3 digit).

The main groups according to machinery systems are 5-8. Main group 4 consists of ship equipment, such as navigational, anchoring, communication equipment's and manoeuvring machinery. Main group 5 consists of equipment which serves crew and passengers such as equipment's for lifesaving, furniture, catering and sanitary. Machinery main components like main and auxiliary engines, propeller plant etc. are in main group 6. Group 7 is for systems serving the machinery main components like fuel and lubrication oil systems, exhaust systems etc.. Central systems for ballast and bilge, firefighting, and electrical distribution are in main group 8.

Group 4 Ship Equipment	40 Manoeuvring machinery and equipment	Wärtsilä
	41 Navigation equipment	
	42 Communication equipment	
	43 Anchoring and mooring equipment	2 x Anchors 4590kg each 550m chain ca. 36t
	44 Repair and cleaning equipment	
	48 Special equipment	Main crane 47t Auxiliary crane 20t
Group 5 Equipment for crew and passengers	50 Lifesaving Equipment	2 x Lifeboats 3802.5kg each 2 x LHD-110 davits 6000kg each MOB-boat FRB-700 2546kg
	51 Insulation, panels, bulkhead, doors, side scuttles and windows	
	52 Internal deck covering, ladders, steps, railing	
	53 External decks	
	54 Furniture and Inventory	
	55 Galley, pantry, provisions and laundry equipment	

	56 Transport equipment for crew	
	57 Ventilation, airconditioning and heating systems	Heinen & Hopman
	58 Sanitary system and equipment	
Group 6 Machinery main components	61 Generator and motor	Engines Wärtsilä 2 x 12V31 77.1t each and 2 x 14V31 84.6t each Ballard fuel cells 5 x 875kg
	62 Propellers, Transmission and foils	Wärtsilä
Group 7 Systems for machinery components	70 Fuel oil system	
	71 Lub oil system	
	72 Cooling system	
	74 Exhaust system	
	76 Distilled & make up water systems	
	79 Automation system for machinery and cargo systems	Wärtsilä
Group 8 Ship systems	80 Ballast, bilgde and drain systems, gutter pipes outside accomodation	
	81 Fire and lifeboat alarm systems, fire fighting systems	
	82 Air and sounding system	
	83 Special common hydraulics oil systems	
	85 Electrical systems, general part	
	86 Electrical supply system	
	87 Electrical common distribution	
	88 Electrical cables and installation	
	89 Electrical distribution system	

7.5 HVAC

The electricity consumption of the HVAC system can be up to 30 % which makes the definition of HVAC system important. The needed heating and cooling power required can be estimated with air exchange factors given in DNV GL rules for ventilation.

According to ISO 7547:2002 with requirements for accommodation the design conditions according to temperatures shall be in summer; outdoor +35°C and indoor +27°C and in winter temperatures; outdoor -20°C and indoor +22°C. As the vessel will mostly operate in the arctic the equipment for winter temperatures should especially be applied. Heat gain comes also from persons lightning which in these calculations are neglected as it is a rough estimate.

For hospitals supply air with a non-return flap should be installed, exhaust air devices should be installed over areas with high heat emissions and high humidity in for example laundries and drying- and ironing-rooms. The air distributing systems should not exceed 55dB measured 1m from the air terminal device. Temperature control should also be fitted in all accommodation spaces, by installing systems for controlling airflow, different valves, solenoids etc..

Heinen & Hopman is one company providing HVAC solutions for marine industry and research vessels. According to them the different types of laboratories, accommodation and machinery spaces need climate controlling and special air conditioning and ventilation systems. The HVAC systems for accommodation and laboratories should be separate, as the laboratory spaces have more conditions to consider. (Heinen & Hopman, 2021)

The heating on a research vessel can be done by installing different types of units and heaters. As the temperatures in polar regions are low the installed heating system must have power for keeping the inside area warm and ensure enough temperatures in the engine room, technical spaces and emergency generator. The ventilation in engine rooms allows the engines to cool down and ensure better performance. The noise level on ventilations is restricted but the research equipment onboard should also be taken into account. Ventilation is mainly done by fans and dampers of different types for type of rooms. Induction ventilation for galley will remove the heat (puts the most of heat into a vessel) and prevents fumes and contamination to other areas of the vessel. Air conditioning contains of chillers, colling plant, and several units which can be specialized for cabins. using refrigeration, the vessel can extend voyages without provisioning trips. For example, trips to remote area will require larger capacity for waste cooling. (Heinen & Hopman, 2021)

7.6 Automation and control systems

An automation and control system on a vessel covers aspects for ship operation like propulsion plant operation, power management operation on auxiliary engines, auxiliary machinery operation, loading off cargo, navigation and administration of maintenance. (shippipedia, n.d.)

The propulsion and power are monitored and controlled to keep efficiency and safety. The systems for these considers for example fuel consumption, temperatures for engines and combustion, control of diesel electric propulsion etc.. Monitoring and control of auxiliary machinery contains several systems for auxiliary machinery like pumps, pressure, tank level and other parameters for the water cooling systems, water control, boiler/steam systems etc. (shippipedia, n.d.)

To keep the cargo loading safe the process can be monitored by level gauging, valve control, ballast and ballast pump control etc. (shippipedia, n.d.)

Solutions for these are provided by several companies for example siemens and Wärtsilä. The Wärtsilä NACOS Platinum navigation automation control system covers ship automation, propulsion, navigation, power management, dynamic positioning and general alarm and control. It is used globally on hundreds of vessels which makes it a reliable solution. The system complies with all relevant international rules and regulation from the classification society DNV GL, which is used for Khione. (Wärtsilä, 2016)

Some features the NACOS system contains and which is needed on Khione is radars, ECDIS, WECDIS, autopilot and track control, VDR, joystick system and dynamic positioning, alarm, monitoring and control system, power management system, propulsion control system, main engine safety system, HVAC and cargo monitoring and control. (Wärtsilä, 2016)

7.7 Special equipment and systems

As Khione is a polar research and re-supply vessel the special equipment is equipment needed mainly for the research, cargo and helicopters. The special equipment belongs to SFI group 48. Groups fulfilled from the SFI classification is diving equipment (483), laboratory equipment (484), aircraft/helicopters (485) and de-icing equipment (487). Due to the operating area equipment for de-icing will be needed.

The research will be done with equipment for biological, chemical, or physical oceanography as well as paleoceanography research. Different sensor arrangements will be needed for hydrographic and oceanographic research. There will be a CTD system to measure the conductivity, temperature, and pressure of the seawater. The vessel also has supporting technology for remotely operated underwater vehicles (ROV's) and autonomous underwater vehicles (AUV's). Two cranes will be fitted which can be used for both cargo handling and research.

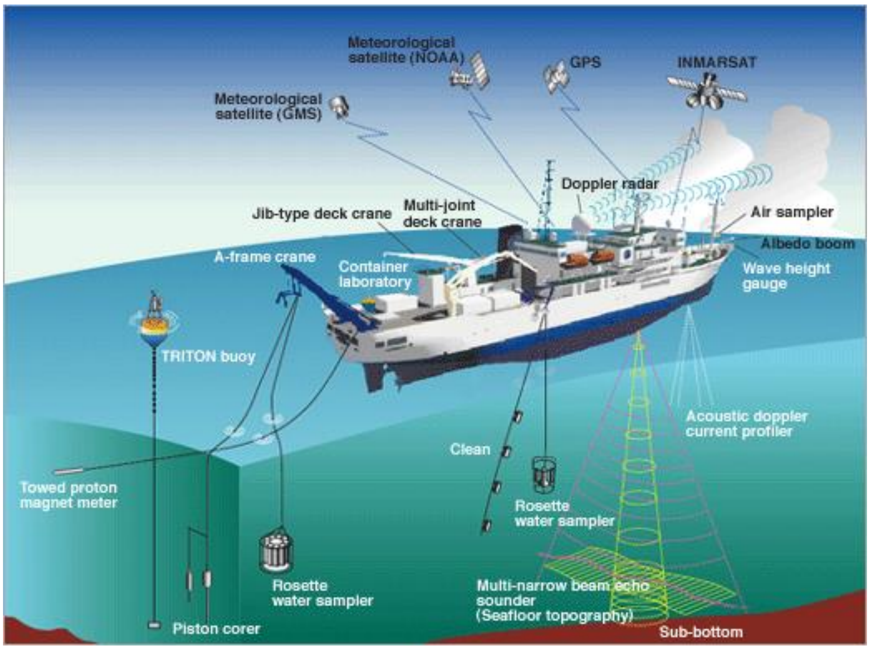

Figure 40 Example of special equipment for research purpose (Jamstec, n.d.)

8 Weight and Stability

8.1 Weight

Deadweight

Deadweight, DWT, is calculated with following formula:

DWT = DWTc + DWTFO + DWTFW + DWTC&E + DWTPR.

In which: DWTc is the cargo deadweight; DWTFO is the fuel oil weight; DWTFW is the lube oil weight; DWTH is the hydrogen weight; DWTC&E represents the weight of crew and their effects and finally DWTPR is the weight of provisions.

Propulsion is powered with diesel engines and rest of the energy is produced with hydrogen. Fuel and lubrication oil capacities are calculated with engine loads due propulsion. FO and LO capacities are calculated for 32-day (768 h) voyages.

Table 19. Main engine consumption details

Engine details		
Engine output 14V31	8540	kW
Engine output 12V31	7320	kW
Engine output total	31720	kW
Fuel consumption	170.1	g/kWh
Lube oil consumption	0.45	g/kWh

Table 20. FO and LO consumptions per voyage.

Sea condition				
	open water	light ice	heavy ice	sum
per voyage	70%	25%	5%	100%
hours per voyage, h	537.6	192	38.4	768
prop. power, kW	1400	4203	29000	
fuel cons., kg / h	238	715	4933	
lube cons., kg / h	1.17	1.97	9	
fuel cons. per voyage, ton	238	137	189	**454**
lube cons. per voyage, ton	0.63	0.38	0.35	**1.35**

Minimum amount of fuel and lubrication oil required for 32-day voyages are rounded up to obtain DWTFO and DWTFW. DWTFO is 454 tonnes and DWTFW is 1.35 tonnes.

Hydrogen tank capacity is 651 m^3 and can contain 4.6 tons of liquid hydrogen. DWTH is 4.6 tons.

The vessel has 20 crew members and 80 scientists onboard (100 persons total). Weight of provisions, DWTPR, is calculated with formula:

$$DWTPR = 0.01t \times persons \times days$$

Weight of crew and their effects, DWTC&E, is calculated with formula:

$$DWTC \,\&\, E = 0.17t \cdot persons$$

For the vessel formula for DWTR gives 32 tonnes and formula for DWTC&E gives 17 tonnes.

The vessel has 960 m^2 of cargo space in two floors equaling 3074 m^3. The cargo of the vessel is manly supply for arctic research centers. Supply is usually food and spare parts. Cargo space can't be filled 100 % full, but more like 80 % full. Average density of supplies is assumed to be 1000 kg/m^3. Cargo deadweight, DWTc, of the vessel is then 2460 tonnes.

Table 21. Deadweight calculation

	DWTFO	DWTFW	DWTH	DWTC&E	DWTPR	DWTc	**DWT**
tons	454	1.35	4.6	17	32	2460	**2968.95**

Lightship Weight

The lightship weight estimation was conducted with the given excel sheet. The initial values including the main dimensions and coefficients of fineness were obtained from the latest iteration of our hull model design from Delftship. The vessel's main characteristics and the final lightship weight estimation and the vessel's estimated vertical center of gravity can be seen in Figure 1 below:

Table 22 Ship's main characteristics

Ship's main characteristics	
L(m)	132.92
B(m)	22
T(m)	7
D(m)	11
CB	0.6436
LCB(m) @AP (m)	50.751
Lightship weight	
9450.87	
KG_{Light}	
6.394	

The structural weight of our vessel was estimated based on the modified Lloyd's equipment number E. E is defined as following in the lecture notes.

$$E = L(B+T) + 0.85\, L(D-T) + 0.85 \sum_i h_i + 0.75 \sum_j h_j$$

The h_i is a profile area of a superstructure element and the h_j is a profile area of a deckhouse element. The size of the vessel's superstructure and deckhouse was estimated using the general arrangement we have drawn on AutoCAD. This gave us E of the rough value of 5990. Finally, the structural weight was estimated with the following formula from the lecture notes:

$$W_S = K\, E^{1.36}[1 + 0.5\,(C_B - 0.7)]$$

Here, the factor K varies with the ship type. The data provided in the lecture notes stated that a research vessel's typically have K of value 0.045. However, this data implemented that the value of E in these cases would vary between 1350 and 1500. This did not correspond our value of E at all. Thus, we decided to use the value of 0.038 for K, typical for passenger ships, as it corresponds to our value of E better. The data claimed that typical E for this value of K was between 5000 and 15000.

These values lead to our structural weight estimation to be 5068.44 tonnes.

The machinery weight includes propulsion machinery (prime mover, reduction gear, shafting and propeller). Since we are using diesel-electric machinery we calculated the estimated weight for the total machinery with the equation from lecture notes.

$$W_M = 0{,}72 * (MCR)^{0{,}78}$$

As MCR we used the total produced power from the diesel engines, 31720 kW. Resulting to a machinery weight of 2335.46 tonnes. The Wärtsilä diesel engines has given weights on 77.1 tonnes (12V31) and 84.6 tonnes (14V31) which means exact weight of the diesel engines are 323.4 tonnes. Since the estimated machinery weight calculates the total machinery, the remaining 2012 tonnes is for the other machinery needed. The other machinery includes four electric generators, two shafts and two propellers.

The outfitting weight is estimated with the following formula from the lecture notes:

$$W_O = C_O \, L \, B$$

where the factor C_O is the outfitting weight coefficient. In order to retrieve the value for this factor, there is Figure 44. Our vessel's length between perpendiculars is 117.1 m. Since there is no ship type represented in the figure, that corresponds ours, we decided to calculate our vessels value L/B and use one of the coefficients for passenger vessels. In our case, $L/B = 132.92/22 \approx 6.04$. Thus, as seen in Figure 44, we used the value of 0.7 as our C_O.

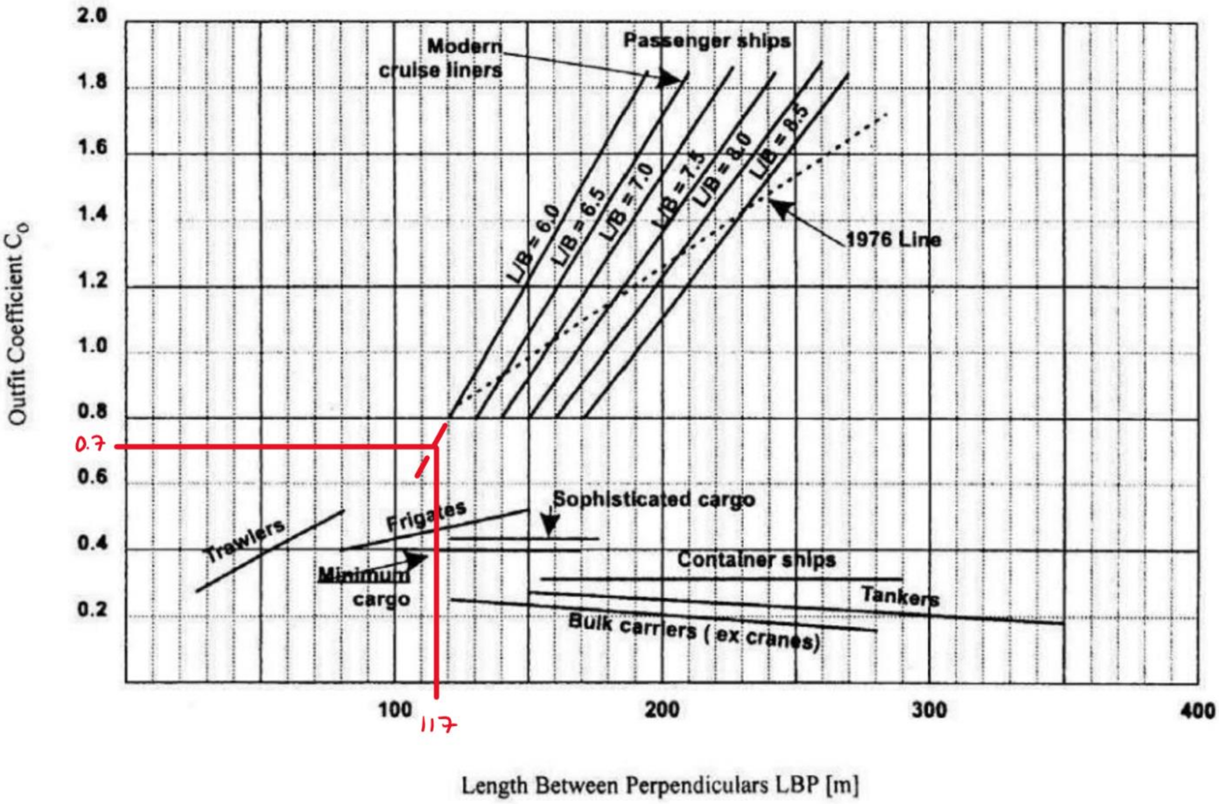

Figure 41 Outfit coefficients.

Thus, our outfitting weight ended up being 2046.968 tonnes.

Further, our final lightship weight estimation was made by summing the structural, machinery and outfitting weight together:

$$W_{LS} = W_S + W_M + W_O$$

As earlier seen in Table 42, this value ended up being 9450.87 tonnes. Deadweight is difference between displacement and lightship weight, resulting that displacement is sum of deadweight and lightship weight. Displacement of the vessel is 12517.72 tonnes according to the calculations. The displacement of our hull form according to Delftship is 11895.7 tonnes. Between the calculated value and the Delftship calculation, there is a difference of 622.02 tonnes. This is clearly a problem as the calculations go over the weight that this hull form can carry. Thus, more iterations and more precise calculations are needed as adding the weight reserve increases this difference further.

Lightweight

Direct method is used to calculate steel weight of the project ship. This is done by obtaining areas of steel structures from AutoCAD and NAPA and then summing up these areas and multiplying them with steel thicknesses in Excel. Density of 8 g/cm^3 is used for the steel. Table 23 shows weight of the different elements. Total steel weight is 3178.9 tons. This is radically less than the value, 5068.44 tons, obtained in PNA course using formula described in subchapter above.

Table 23. Steel weight calculations.

thick, [mm]	6	12	12	12	18	12	9	9
element	decks	trans. BH	trans. Walls	long. Walls	super str.	stairs	girders	floors
area sum, [m2]	15107.5	2095.6	367.3	2111.2	3922.4	777.1	649.8	262.7
tons	725.2	201.2	35.3	202.7	564.8	74.6	46.8	18.9

thick, [mm]	26	10	10	10	10	30	
element	shell	stiffs	stingers	trans frame	web frame	pillars	
area sum, [m2]	2394.8	5493.7	810.4	1289.2	709.1	613.2	**sum**
tons	498.1	439.5	64.8	103.1	56.7	147.2	**3178.9**

Using SFI classification we get sum of different weights of different ship systems, 462.1 tons. This is low as the weight data is difficult to obtain, as there isn't much information freely available, thus we keep using earlier weight estimations for outfitting weight done in subchapter above. Even though the SFI classification isn't complete it can be used in estimation of weight distribution of the ship.

Weight Reserve

Ship of our project is highly unconventional and specialized. In concept design phase, weight assessments are rough, and done by statistical methods, and by direct calculations. In case of highly specialized ship, like a ship in question, statistical methods can be problematic due to lack of applicable data. As known, using statistical method with lack of data can cause uncertainty in weight calculations. Highest uncertainty of statistical method is related to the structural weight due to the ice strengthening. Ice strengthening narrows down the possible data used for statistical method, leading to increased uncertainty.

Uncertainty in weight calculations is dealt by adding reserve to weight and to vertical centre of gravity of ship. Weight reserve is expressed as percentage of ships total lightweight and reserve in vertical centre of gravity is expressed in meters. At the time of delivery of a ship, target values are 0 % and 0.1

m, respectively. Values are considered again at every new iteration of the design process. In the conceptual stage, values are at their extreme. Values depends on deadweight/displacement -ratio of the ship. At current conceptual design stage of our project, with deadweight/displacement -ratio of roughly 0.3, weight reserve of 15 % and vertical centre of gravity reserve of 1 m should be applied.

8.2 Loading conditions

Four different loading conditions were defined for the stability calculations with NAPA. Our ship performs resupply operations to villages and research centers in Arctic and Antarctic region, which are very much scattered and relatively small. Due to scattered locations and small sizes, all the cargo onboard is not expected to be delivered to one location. Thus, loading conditions of the vessel will vary during the voyages all the way from fully loaded to the ballast condition. Defined loading conditions are fully loaded (FULL) and ballast (EMPTY) conditions, additionally overloaded (OVERFULL) and completely empty (OVEREMPTY) conditions were defined to analyze the sensitivity of intact stability of the ship. Overloaded and completely empty loading conditions are not designed to be used in practice, their only purpose is to show the sensitivity of the intact stability to possible misuse or technical errors. Defined full loading condition illustrates situation when ship is loaded to maximum capacity and ballast condition illustrates light ballast loading condition.

In designed fully loaded condition, deadweight of the ship is not completely in line with previously made calculations. Previously, in chapter 8.1, it was calculated that total deadweight capacity of the ship would be 3000 tons. Now, by help of NAPA, it can be noticed that fuel and liquid cargo tanks of our vessel are oversized related to previous calculations. Resulting in growth of deadweight up to 5500 tons in fully loaded condition. With this increase in deadweight our draft changes from 6.4m to 7.5m and displacement from 12517 tons to 14924 tons. In designed ballast condition draft of our vessel is 6.3m and displacement 11624 tons. Draft change between the designed loading conditions is thus 1.2m. In defined theoretical overloaded condition ship is loaded with 2000 tons over the design deadweight and in theoretical completely empty condition ship is completely empty, including the ballas tanks. Illustrative figures of designed load conditions in Figure 41. More detailed report on loading conditions can be seen in Appendix 2.

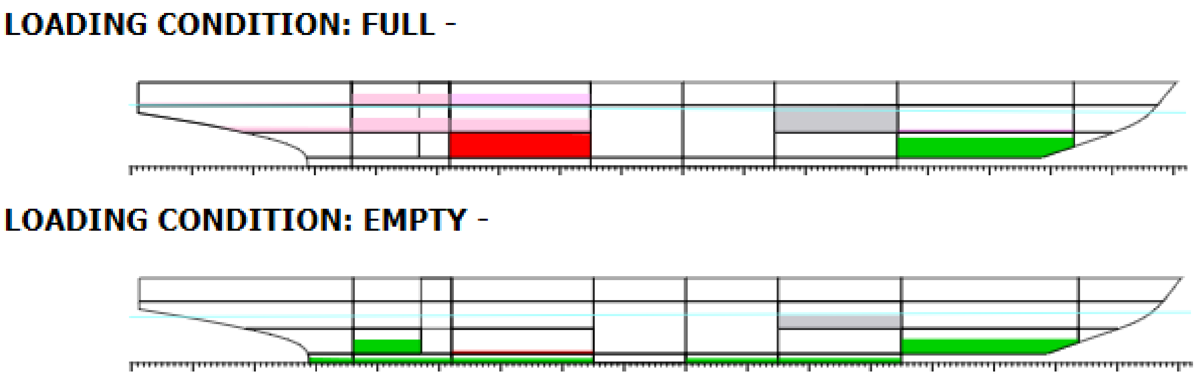

Figure 42 Designed loading conditions

8.3 Intact stability

Intact stability analysis was performed by use of NAPA. Criteria for intact stability performance were regulations set by IMO. Used regulations were IMO's basic intact stability criteria and IMO's weather stability criteria. Openings like tank ventilation and mooring holes were not included in assessment. At sensible heeling angles previously mentioned opening shouldn't cause any notable effect. Analysis was performed to both designed loading conditions and to the theoretical overloaded and completely empty conditions to see the sensitivity of the stability calculations. Results of the intact stability criteria calculations are presented in following Tables 34, 35, 36 and 37.

Table 24 Intact stability results of fully loaded design condition.

Loading condition: FULL

RCR	TEXT	REQ	ATTV	UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.491	mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	0.793	mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.302	mrad	OK
GZ0.2	Max GZ > 0.2	0.200	1.749	m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	37.833	deg	OK
GM0.15	GM > 0.15 m	0.150	3.535	m	OK

RCR	TEXT	REQ	ATTV	UNIT	STAT
LR.IMOWEATHER	IMO weather criterion	1.000	3.406		OK
LR.IMOWINDHEEL 1	HEEL < 16 deg	16.000	0.932	deg	OK
LR.IMOWINDHEEL 2	HEEL < 80% of FRB immersion	13.491	0.932	deg	OK

Table 25 Intact stability results of ballast loaded design condition.

Loading condition: EMPTY

RCR	TEXT	REQ	ATTV	UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.567	mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	1.002	mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.435	mrad	OK
GZ0.2	Max GZ > 0.2	0.200	2.595	m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	40.808	deg	OK
GM0.15	GM > 0.15 m	0.150	3.546	m	OK

RCR	TEXT	REQ	ATTV UNIT	STAT
LR.IMOWEATHER	IMO weather criterion	1.000	5.365	OK
LR.IMOWINDHEEL 1	HEEL < 16 deg	16.000	1.271 deg	OK
LR.IMOWINDHEEL 2	HEEL < 80% of FRB immersion	18.193	1.271 deg	OK

Table 26 Intact stability results of theoretical overload condition.

Loading condition: OVERFULL

RCR	TEXT	REQ	ATTV UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.421 mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	0.659 mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.238 mrad	OK
GZ0.2	Max GZ > 0.2	0.200	1.376 m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	37.185 deg	OK
GM0.15	GM > 0.15 m	0.150	3.764 m	OK

RCR	TEXT	REQ	ATTV UNIT	STAT
LR.IMOWEATHER	IMO weather criterion	1.000	2.690	OK
LR.IMOWINDHEEL 1	HEEL < 16 deg	16.000	1.767 deg	OK
LR.IMOWINDHEEL 2	HEEL < 80% of FRB immersion	8.494	1.767 deg	OK

Table 27 Intact stability results of theoretical completely empty condition.

Loading condition: OVEREMPTY

RCR	TEXT	REQ	ATTV UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.645 mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	1.125 mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.480 mrad	OK
GZ0.2	Max GZ > 0.2	0.200	2.871 m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	40.270 deg	OK
GM0.15	GM > 0.15 m	0.150	4.611 m	OK

RCR	TEXT	REQ	ATTV UNIT	STAT
LR.IMOWEATHER	IMO weather criterion	1.000	3.768	OK
LR.IMOWINDHEEL 1	HEEL < 16 deg	16.000	1.311 deg	OK
LR.IMOWINDHEEL 2	HEEL < 80% of FRB immersion	23.399	1.311 deg	OK

As can be clearly seen from the status of each intact stability result, all the design and theoretical loading conditions fulfilled the criteria's set by IMO. Even the theoretical exaggerated loading conditions fulfilled the criteria's, meaning that intact stability of our design is not sensible to possible uncertainties. More detailed results of intact stability analysis shown in Appendix 3.

8.4 Damage stability

Damage stability calculations are done in NAPA for different scenarios. The dryship without loadings is used for the calculations to limit the amount of calculations.

Since the vessel is operating mostly in ice-covered waters damage by ice must be taken into account. The vessel can be damage the hull, propellers, rudder or other appendages can get damaged by hitting ice. When hitting something heavier than the surrounding e.g., multi-year flow or edge of ice in open water the hull gets larger impacts. The damage can also be affected by wave-induced impacts, or hit from large floes or bergs. The sides of the vessel can be damaged if the vessel gets trapped in ice while the pressure can increase the loads on the sides (Gard, 2012). Despite damage from ice other scenarios could be flooding to engine room caused by different incidents and failures in machinery.

Three calculations where done in NAPA using the damage stability functions. The first case is damage in the bow on deck 1-3 compartments, which could be possible by bow collision or hitting higher ice. Second case is damage in the fore lower bow part and partly the bottom, this scenario could be possible if the ice becomes thicker than the vessel is designed for. Last third case is flooding to both engine rooms. The figures present the floating positions after the damage calculations and shows that the chosen damages wouldn't in the shown stage cause sinkage, should still be remembered that the ship is without loading.

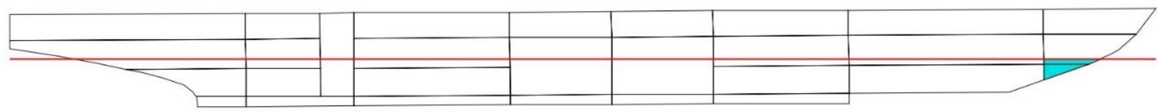

Figure 43. Floating position after damage in bow

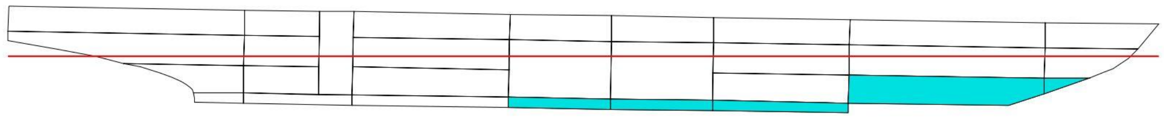

Figure 44. Floating position after damage in fore part of bottom and deck 1in bow

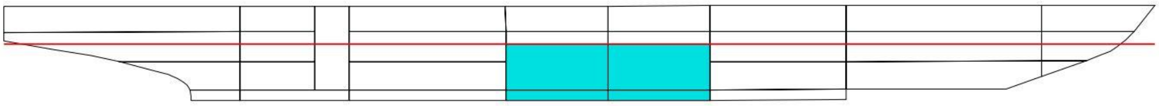

Figure 45. Floating position after damage in engine rooms

The above damage scenario results is checked to meet damage stability criterias. The IMO Res. A.534 (13) code of safety for special purpose ships is used as criterias. As seen in the table below all the criterias are met in the final stage of damage.

Table 28. Damage stability criteria checks

CASE	STAGE	PHASE	RCR	REQ	ATTV	UNIT	STAT	MINGM
INI1/DAM_BOW1		EQ	V.PROGR..	0.000	999.900	m	OK	-999.900
INI1/DAM_BOW1		EQ	V.MAXHEE.	7.000	0.000	deg	OK	0.006
INI1/DAM_BOW1		EQ	V.MINGM..	0.050	4.587	m	OK	-0.335
INI1/DAM_BOW1		EQ	V.MINGZ..	0.100	2.796	m	OK	-0.146
INI1/DAM_BOW1		EQ	V.RANGE..	20.000	50.000	deg	OK	-0.084
INI1/DAM_EN.1		EQ	V.PROGR..	0.000	999.900	m	OK	-999.900
INI1/DAM_EN.1		EQ	V.MAXHEE.	7.000	0.000	deg	OK	-0.116
INI1/DAM_EN.1		EQ	V.MINGM..	0.050	4.724	m	OK	-0.499
INI1/DAM_EN.1		EQ	V.MINGZ..	0.100	2.740	m	OK	-0.269
INI1/DAM_EN.1		EQ	V.RANGE..	20.000	50.000	deg	OK	-0.271
INI1/DAM_BOT1		EQ	V.PROGR..	0.000	999.900	m	OK	-999.900
INI1/DAM_BOT1		EQ	V.MAXHEE.	7.000	0.018	deg	OK	-1.249
INI1/DAM_BOT1		EQ	V.MINGM..	0.050	5.837	m	OK	-1.498
INI1/DAM_BOT1		EQ	V.MINGZ..	0.100	3.221	m	OK	-1.259
INI1/DAM_BOT1		EQ	V.RANGE..	20.000	49.982	deg	OK	-1.306

9 Building costs

As the data available regarding the cost coefficients of different ship systems and spaces was limited, we first estimated the total building costs based on the estimated hull structure cost and comparing that to a possible cost distribution. As the building cost of our reference ship RRS David Attenborough was also available, we also calculated another estimate of total building cost based on it.

9.1 Hull structure costs

The cost of the hull structure can be estimated with the following equation:

$$C_{ST} = K_{STEEL} \times W_S + K_{MAN} \times MHS$$

where K_{STEEL} is the cost of steel per ton, W_S is the structural weight, K_{MAN} is the man hour cost and MHS is the required number of man hours (Papanikolaou, 2014). The cost of steel per ton is at the moment roughly 575 € per tonnes (MEPS, 2020). Our structural weight has been estimated to be roughly 5000 tonnes. Cost of man hour is estimated to be 45 €/h. The required working hours can be estimated with the following formula:

$$MHS = a \times (W_S)^b$$

where a in our case can be estimated to be 243 and b to be 0.85 (Papanikolaou, 2014, p. 445). Thus, the required working hours in our case is roughly 339 000 hours. Further, the hull structure cost can be estimated to be approximately 18.3 million €.

9.2 Distribution of Cost

As the data regarding costs of other systems of the vessel was limited, we used the obtained estimate for hull structure cost to approximate the total building cost based on a possible cost distribution. In Table 42, the cost distribution of a platform supply vessel (PSV) is represented (Shetelig, 2013).

Table 29 Distribution of costs for PSVs.

Technological group	Portion of total cost
Hull	20 – 30 %
Machinery and Propulsion	25 %
Cargo containment and handling	20 – 25 %
Ship common systems / Ship assembly and systems integration (for outfitting yard)	20 %
Hotel and accommodation	5 %
+ Financial costs	+ Financial costs

As the cargo containment and handling is a relevant part of our vessel's mission, not to forget that we also have to build a fully functional helicopter hangar onboard, this part of the total cost was approximated to be 25 %. Also, the hotel and accommodation cost are estimated to take a bigger slice of the total cost, 10 %, since they must be more lavish compared to a usual PSV. Financial costs were approximated to take a 5 % part of the total costs. Thus, 15 % of the total cost is the share of hull structure cost. Therefore, based on this method, our total building cost is roughly 122 million €.

9.3 Comparison with Reference Ship and Final Total Building Cost Estimation

The RRS Sir David Attenborough took £200 million to build (Anon., 2019). Based on current market conditions, that equals to roughly 225 million euros. Given this, our previous estimate was not realistic.

The total building cost can be approximated with the following formulas from lecture notes:

$$P = C_1(DWT)^B$$

where B is typically 0.7 - 0.8, or

$$P = C_2(W_E)^{0.87}$$

The value of coefficients C_i can be approximated as the deadweight and lightship weigh, W_E, are known. In RRS Sir David Attenborough's case, the deadweight is 4475 tonnes, and the displacement of the vessel is 12790 tonnes. This leads to the lightship weight of 8315 tonnes. Thus, if the exponent B is varied from 0.7 to 0.8, the first formula gives that the coefficient C_1 is from approximately 270 000 to 626 000, and the second gives the C_2 of rough value 87500, as the total building cost P is 225 million €.

As stated earlier in this report our vessel's deadweight is 3066.85 tonnes, and the lightship weight is roughly 8538.65 based on displacement. Thus, the first formula, varying C_1 from 270 000 to 626 000, gives us the total building cost of 166.2 to 172.7 million €. The second formula gives the total building cost of 230.3 million €. As Khione is close in size to RRS Sir David Attenborough also the cost should be close 225 million €. To conclude the total building cost analysis, the mean value of upper first P and second P value are used. This gives cost estimation of 202 million €.

The final cost distribution of our vessel based on the SFI system, can be seen in **Error! Reference source not found.**. The distribution was based on the approximated roughly 202 million € of total building costs and the earlier calculated roughly 18.3 million € of hull structure costs. The hull can be seen to take 8.9 % slice of the total costs. The mission of our vessel is quite versatile, which explains why different equipment for both the crew and for the ship herself along with different ship systems, take bigger slice than what was shown in Table 42. The hydrogen implementation also increases the costs regarding the main components of machinery and the systems for it.

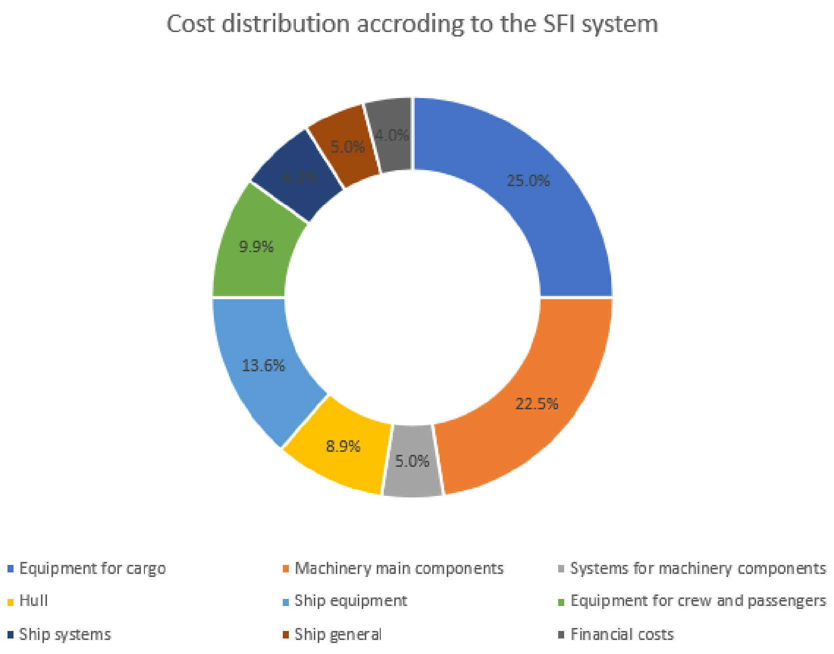

Figure 46 Cost distribution of Khione.

In Table 30, the cost estimate for each group is presented. System costs are specified in subchapter 9.4 Hydrogen System Costs.

Table 30 Cost distribution.

	M €	%
Equipment for cargo	50.5	25.0%
Machinery main components	45.5	22.5%
Systems for machinery components	10	5.0%
Hull	18	8.9%
Ship equipment	27.5	13.6%
Equipment for crew and passengers	20	9.9%
Ship systems	12.5	6.2%
Ship general	10	5.0%
Financial costs	8	4.0%
Total building cost	202	100%

9.4 Hydrogen System Costs

Cost estimations of main systems are defined in earlier subchapter. Finding specific costs of systems is difficult as system providers don't usually publish their prices. As hydrogen is in important role in our concept design, cost estimation for hydrogen is calculated. Cost of hydrogen system is part of machinery costs.

Costs of hydrogen system consists of tank for liquid hydrogen, fuel cells, piping, controlling systems and installation. Fuel cells cost about 100 $/kW and liquid hydrogen tanks 160$/kgLH$_2$ (Rivard, et al., 2019). Ship has 1000 kW of fuel cells and 651 m^3 tanks for liquid hydrogen, which leads to cost of 6.3 million euros. Total cost of hydrogen system can be estimated then be 8 million euros.

9.5 Contract Price

The contract price should cover building costs, other expenses of the shipyard and also profit margin. Shipbuilding is highly competitive field and in addition the corona virus has had negative effect on the markets. This can lead even lower profit margins. The contract should be priced so that it isn't higher than the market price, which can cause losing the contract, and not lower than the costs, which leads to losing money (Shetelig, 2013). Figure 47 shows average profit margins of European shipbuilding companies. It should be noted that the Figure 47 shows profit margins of shipbuilding companies and not singular contracts, such as a shipbuilding contracts, but the information can be used to estimate profit margin for our contract price. The shipyard building Khione can be stated as medium sized company and so profit margin of 3-5 % is used when determining contract price. Final contract price is calculated with profit margin of 4 % leading to contract price of 210 million euros.

The contract price is close to RRS Sir David Attenborough's cost. Price information about our other reference vessel, Xue Long 2, isn't available. Russian Project 00903, which is smaller (L = 84 m) than Khione and but is also Arctic research vessel, has been reported to cost 85 million euros being two times cheaper than Khione. Norwegian Arctic research vessel, also smaller than Khione (L = 100 m), cost 156 million euros. Belated Canadian icebreaker CCGS John G. Diefenbaker, which is larger (L = 150 m) than Khione, has budget of 1.1 billion euros. Khione can seem expensive to smaller ships with same purpose, but cheaper than more similar sized ships. The contract price of Khione is most likely below market prices and still has good profit margin, that is optimal situation.

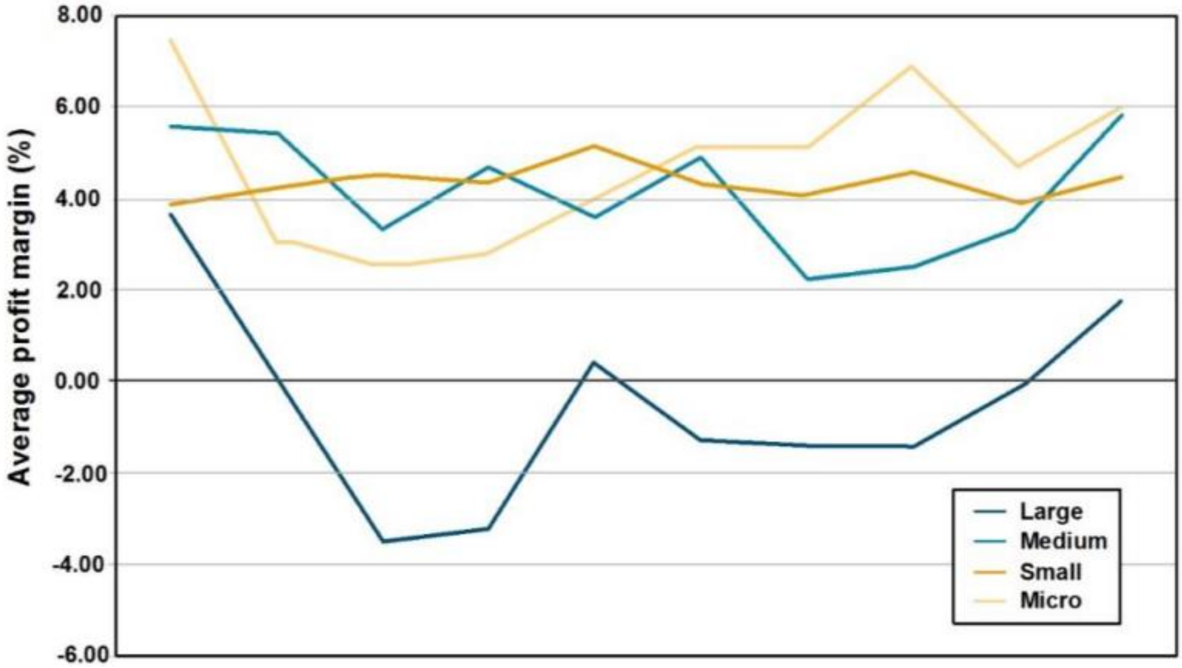

Figure 47 Average profit margin of European shipbuilding companies (Legorburu, et al., 2016)

10 Evaluation

10.1 KPIs

Suitable key performance indicators KPIs should be selected to illustrate the performance of the ship. Main tasks of our ship are to deliver cargo to remote research centres and communities, and to conduct research in arctic area. Because of our partly non-commercial mission, incoming annual cashflow is limited. Operation of the vessel is widely supported by the government. However, resupply operations are conducted during 10 months of each year bringing cashflow for the vessel. Since big part of the resupply is done to small and isolated communities and research centres, rate of transporter cargo each year is not great, but very important for the people receiving the cargo.

Even that operation of our vessel is financially supported by government, it is important to know eventual costs caused by the operations. To monitor these costs, net present value (NPV) analysis was selected as first KPI. Performance of research and resupply operations are difficult to measure directly. Measuring the amount of collected data or delivered cargo may not tell the real quality of performance. However, ship is going to operate in remote areas without external help. For operations in such places, it is crucial that ship is reliable. Failures in ship could potentially cause life threat not only to the crew of the ship, but also to the people of remote communities and research centres, which relies on cargo that is delivered by our vessel. Thus, another suitable key performance indicator for our vessel is average time between failures.

NPV

Cargo capacity of our vessel is 2460 tonnes. Since big part of the resupply is done to small and isolated communities and research centres, rate of transporter cargo each year is not great. It is estimated that on average it takes a month for the vessel to deliver the full capacity of cargo. Resulting that yearly 2460 tonnes x 10 = 24600 tonnes of cargo is transported. Freight rates for the transported cargo is significantly higher than typical because of the challenging transport routes and locations. It is estimated that on average freight rate would be 1500 € per tonne.

Net present value (NPV) -analysis has been performed to the ship. Lifetime of the ship is designed to be 30 years and initial investment 190 million €. Annual costs consist of maintenance, salaries, and fuel. Estimations for yearly maintenance and fuel costs are based on Swedish Viking ice breakers, which are normally used in offshore operations of Northern Sea. For severe winters fuel cost of those ice breakers has been 4.3 million € and maintenance costs 3.95 million € (Lindborg & Andersson, 2020). Annual cost for our vessel is roughly estimated to be 15 million € based on previously presented values. High number of crew members (100 pax) adds significant annual cost in form of salaries. Interest rate is 10%. Performed net present value -analysis is presented in Table 44 below.

As mentioned in beginning, vessel is designed for research and resupply operations which are not only driven by commercial success. NPV -analysis shows that governmental support is indeed required for ship to operate. During its lifetime, our vessel is not able to turn NPV to positive. After 30 years of operation, net present value of the ship is –28.6 million €.

Net present value at the end of the lifetime of the ship could be improved by several ways. In production lowering the building costs by increasing the efficiency could be done. Efficiency can be improved by lowering required work hours for example by using prefabricated modules, and by precise design and planning work. Now income has assumed to come just from delivered cargo to the research centres and communities, but some cargo, like recyclable trash could be transported also away, bringing new income. As mentioned, great piece of annual costs comes from salaries of our crew. Reduction of the people on deck would lead in significant savings. However, interest rate has a crucial role on development of NPV of the ship. Now interest rate is assumed to be 10 %. Even small reductions in interest rate would have significant effect on the NPV of the vessel. For example, with reduction of 1.5 %, from 10 % to 8.5 %, our NPV value at the end of the lifetime of the ship would change from –28.6 million € to -2 million €, which could be already considered as a breaking even. Further reduction in interest rate could alone make our ship profitable with current costs and incomes.

Table 31 Net present value analysis

			Year	**NPV, annual cash flow**	**NPV**
Initial investment	190,000,000	EUR	0	-190,000,000 €	-190,000,000 €
Transport capacity	24,600	ton/year	1	19,909,091 €	-170,090,909 €
Freight rate	1,500.00	EUR/ton	2	18,099,174 €	-151,991,736 €
Annual revenues	36,900,000	EUR	6	12,361,979 €	-139,629,756 €
Annual costs	15,000,000	EUR	7	11,238,163 €	-128,391,594 €
Annual cash flow	21,900,000	EUR	8	10,216,512 €	-118,175,082 €
Interest rate	10.0 %		9	9,287,738 €	-108,887,344 €
			10	8,443,398 €	-100,443,946 €
			11	7,675,816 €	-92,768,130 €
			12	6,978,015 €	-85,790,115 €
			13	6,343,650 €	-79,446,465 €
			14	5,766,954 €	-73,679,510 €
			15	5,242,686 €	-68,436,825 €
			16	4,766,078 €	-63,670,747 €
			17	4,332,798 €	-59,337,948 €
			18	3,938,907 €	-55,399,041 €
			19	3,580,825 €	-51,818,216 €
			20	3,255,295 €	-48,562,920 €
			21	2,959,360 €	-45,603,561 €
			22	2,690,327 €	-42,913,234 €
			23	2,445,752 €	-40,467,482 €
			24	2,223,411 €	-38,244,072 €
			25	2,021,282 €	-36,222,789 €
			26	1,837,529 €	-34,385,260 €
			27	1,670,481 €	-32,714,779 €
			28	1,518,619 €	-31,196,159 €
			29	1,380,563 €	-29,815,596 €
			30	1,255,057 €	-28,560,539 €

Average time between failures

This KPI measures average time between failures i.e., need for unscheduled maintenance of the vessel. It is very clear way of measuring reliability of the ship. Only failures that compromise fulfilling the mission and cannot be repaired immediately onboard of the ship are considered. Failures that place the ship out of service and into state which requires repair to continue its mission are considered in the KPI. Failures repaired during normal scheduled maintenance are also out of consideration.

This KPI cannot be calculated in advance but has to be measured during operation of the ship. The average length between failures must be as great as possible. Low value can indicate problems in maintenance or design of the vessel and is clear indicator that changes must be done. Average time between failures is especially important key performance indicator in ships like ours, because of remote operation areas and novel technological solutions. Average time between failures can be calculated with following equation:

$$ATBF = \frac{\Sigma(Time\ between\ failures)}{Number\ of\ failures}$$

10.2 SWOT

Our mission was to design a safe, reliable, and efficient research and re-supply vessel for use in Arctic and Antarctic areas. The operation of the vessel will focus on research and to provide supply to small communities and research centres. Since the vessel operates in extreme conditions and for long periods and the arriving time to next port can be uncertain the vessel needs to have enough capacity for storage, tanks etc. for long periods.

The opportunities of Khione are to bring new services to the arctic and Antarctic areas where the amount of service is limited. And of course, contribute the state with new research and broaden information of the operating areas. Because of the operating area the vessel needs to be designed with ice as a big aspect, the design and size of the vessel is still flexible for other operational areas as well.

Khione will use new technologies including hydrogen as an auxiliary energy source. The use of hydrogen will bring both positive and negative aspects. Since it's a new energy source in shipping it will bring weaknesses and threats because it is not so researched and the safety and security with the use needs to be considered. Another technology is the vessel being semi-autonomous which will bring the amount of crew down but it's also a new technology and needs a lot of implementing. The use of these technologies will bring new experience and visibility in terms of opportunities for their future use. However, Khione is also equipped with traditional technology like reliable shaft propulsion system, which makes it cheap to build and maintain but at the same time the solution will have effect on manoeuvrability.

Since it's a research and resupply vessel it will mostly be funded by the government. The amount of supply is small and will not bring that much cashflow, thus a large amount of economic support and investments are needed for fulfilling the operation and research work. There are several requirements and regulations according the design and operation of the vessel which must be met and new regulations are probably coming during the vessels life-time, and we also want that the vessel is safe and reliable for the arctic areas.

Table 32 SWOT analysis.

<table>
<tr><td>Strengths
New technologies
• Hydrogen
• Semi-autonomous
➢ Improved environmental friendliness and efficiency

Conventional propulsion
➢ Reliable, cheap, easy to maintain

Operational flexibility</td><td>Weaknesses
New technologies
• Hydrogen
• Semi-autonomous
➢ Possible problems with costs, reliability, and safety

Conventional propulsion
➢ Effects on manoeuvrability

Probably economic governmental support needed</td></tr>
<tr><td>Opportunities
Services to arctic and Antarctic

New research</td><td>Threats
Safety and security regarding new technology used

New and changing regulations

Long operation times</td></tr>
</table>

10.3 Prototype Problems

Prototype problems concerning our project ship are about introduction of new technologies and expenses of new design. The new technologies applied in Khione are hydrogen fuel and semi-autonomous operation. As Khione doesn't have earlier sister ships she is prototype herself. This leads to common problems of prototypes such as great cost of design and risks of creating something new.

Implementing the hydrogen auxiliary fuel system in Khione requires a great deal of design work as hydrogen as maritime fuel is new. Hydrogen is highly flammable and system failures can lead in worst case scenario to explosion. Thus, careful following of regulations and using professional contractors is important. The knowhow gained form successful utilisation of hydrogen as fuel can be asset in future, that for example be sold as consulting services.

Semi-autonomous operation is new technology that is part of Khione project. Different levels of autonomy are used in different systems of the ship in way that safe and efficient operating of the ship is assured. Definitions for levels of autonomy by Lloyd's Register are shown in (Lloyd's Register, 2017). Systems considered having some level of autonomy are the systems that enable navigation and moving the ship and also coupled to basic functions inside the ship. Navigation system has human user deciding the destination on operations, but the system optimizes and proposes routes. Fuel and engine systems are in dialogue with navigation system, also steering system couples with these systems. High autonomy level (AL) is suggested for systems mentioned, AL 4-6. Basic functions such as HVAC, water treatment, waste and sewage systems. Garbage and recycling require some physical labour, but otherwise it can also be highly automated like the other systems mentioned. These systems could have AL 5-6.

AL 0) Manual: No autonomous function. All action and decision-making performed manually (n.b. systems may have level of autonomy, with Human in/ on the loop.), i.e. human controls all actions.

AL 1) On-board Decision Support: All actions taken by human Operator, but decision support tool can present options or otherwise influence the actions chosen. Data is provided by systems on board.

AL 2) On &Off-board Decision Support: All actions taken by human Operator, but decision support tool can present options or otherwise influence the actions chosen. Data may be provided by systems on or off-board.

AL 3) 'Active' Human in the loop: Decisions and actions are performed with human supervision. Data may be provided by systems on or off-board.

AL 4) Human on the loop, Operator/ Supervisory: Decisions and actions are performed autonomously with human supervision. High impact decisions are implemented in a way to give human Operators the opportunity to intercede and over-ride.

AL 5) Fully autonomous: Rarely supervised operation where decisions are entirely made and actioned by the system.

AL 6) Fully autonomous: Unsupervised operation where decisions are entirely made and actioned by the system during the mission.

Figure 48 Levels of autonomy (Lloyd's Register, 2017).

Design of Khione compromises use of reference ships and possibility of creating something new. Despite of references, great deal of work has to be put in design, which is can be costly. In addition to reference ships and experience of designers also rules and regulations guide the design process. Khione has a lot of advanced technology and systems that regular icebreakers don't have, which increases furthermore needed design work. Some challenges linked to building icebreaking vessels can be overcome by choosing shipyard, which has experience in icebreaking ships. Use of digital simulation and modelling tools is also important part of modern shipbuilding, that can ease and make shipbuilding process more efficient.

10.4 Regulatory challenges

Our ship has been designed to meet all regulations set by the various relevant regulatory organizations, such as the IMO and the classification society DNV. Therefore, we do not expect to have any challenges under unchanged regulations. If we do we see those challenges coming from being able to produce our design according to plan and we would therefore classify them not as regulatory challenges but instead challenges in production. The risks of such challenges are however in our opinion small.

Possible challenges would come from changes in regulations such as the Polar Code, in our opinion. The areas we want to operate in are sensitive from an ecological standpoint and are even currently protected to some extent from excessive pollution. If regulatory institutions were to decide this protection must be strengthened our design could become problematic, depending on the regulations. Our ship is to a large extent powered by fossil fuels and therefore changes limiting or forbidding such fuels would make it necessary to retrofit our ship with an acceptable system. Such a retrofit would be expensive as well as complicated and could make the tight economic margins our ship is supposed to operate on non-viable.

11 Visualisation

The progress on visualisation for Khione is at a stage where the NAPA steel model has been exported to a cad software, where details can be added and parts colored to get a more realistic view of the vessel. Figure 50 shows a rendered picture of the model so far. For the gala presentation the model will be further outfitted with outfitting on decks as hydrogen tanks and cranes showing the purpose and characteristics of the ships.

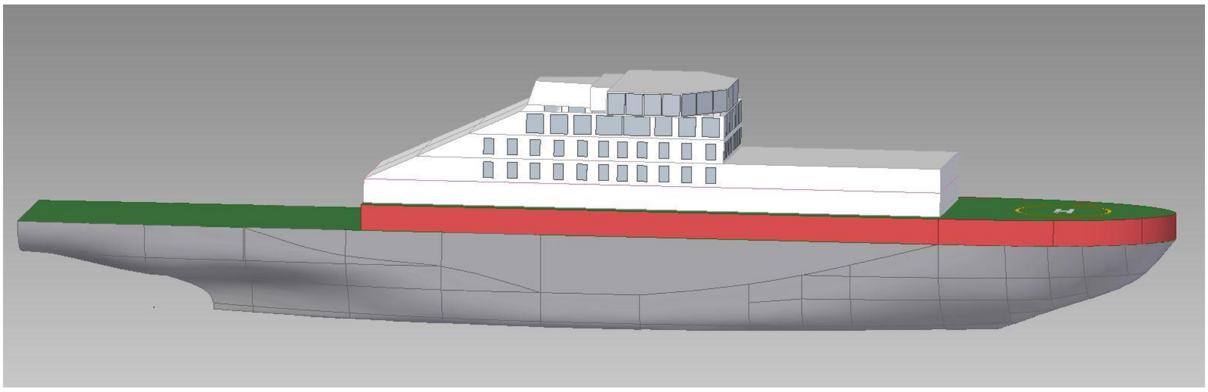

Figure 49. CAD model of Khione.

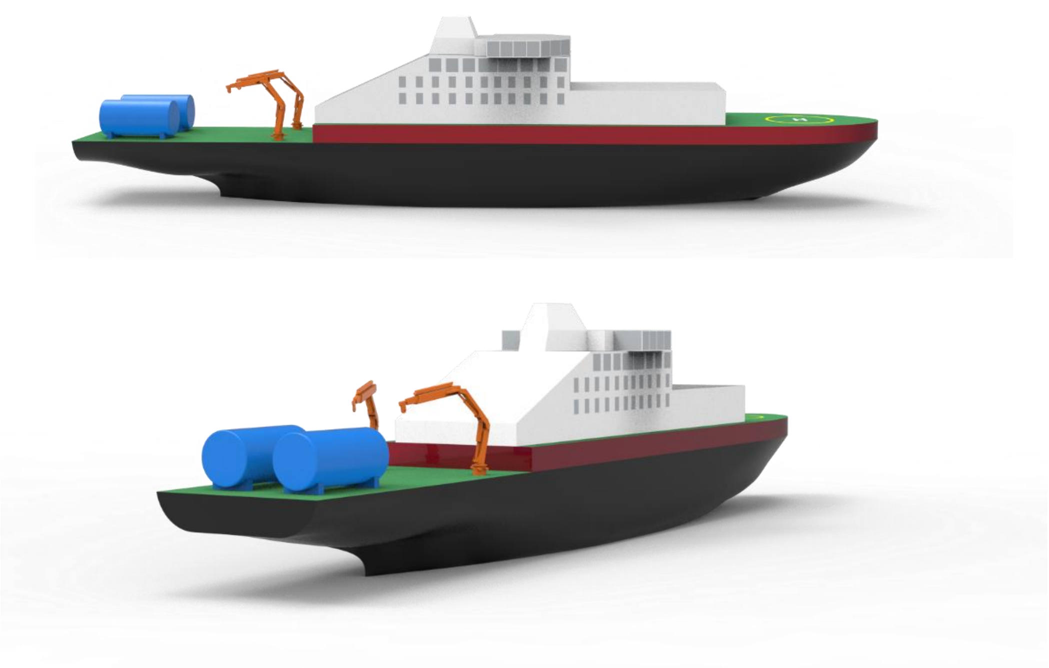

Figure 50. Final rendered pictures.

12 Discussion and Conclusion

12.1 Ship project

Designing a ship from the very beginning is always demanding process. This applied also to our project. Special purpose of the ship and new innovations caused some further challenges to already demanding project but they also made it more rewarding in terms of learning. Design process began with the initial requirements related to ice breaking and research capabilities and with keeping the semiautonomous operations in mind. Process pushed forward using the ship design circle. In the Principles of Naval Architecture course, the whole conceptual design phase was executed following the structure of the design circle. Design process got continued during the Ship Design Portfolio course, when the first round of preliminary design stage was added on top of conceptual design.

Aim of the design was to create modern, ecologic, and effective ice breaking research and resupply vessel for the most demanding seas of the world. New innovations were performed in the fields of autonomous operations and energy, which indeed caused some challenges during the design. Initially, hydrogen was even considered to be the energy provider for the propulsion system but with current technology, we had to reduce the idea. Hydrogen is currently designed to be used as energy source for auxiliary purposes. Autonomous operations were given big attention in design of the ship, which can be seen in great amount of computing space and sensors on board.

All the innovations and decisions in the design of the ship are made keeping the mission of the vessel in mind. Autonomous operations and top-level research facilities and scientific equipment will guarantee the best possible environment for the researchers to do their job. Resupply capabilities are taken care by large cargo handling and storing capacity.

A lot of effort was put on improving the efficiency and operational capabilities of the vessel. However, the safety and seakeeping issues were comprehended to be at least as even important aspect. Khione is designed to operate in remote polar seas at North and South poles, where external help is often not available. Thus, the vessel has to perform well in terms of fire safety, flooding, functional safety, and reliability. Seas in the high polar regions may not have the most severe sea states, but surrounding areas like the notorious North Atlantic have been measured to be one of the most difficult areas in terms of rough seas. Due to rough seas surrounding the planned operational area of the vessel, special attention was put on the seakeeping and stability characteristics.

As mentioned, conceptual phase and the first round of preliminary design has been executed. Great amount of work has been done and the design is slowly but surely evolving from just a concept into more detailed plan of an icebreaking research and resupply ship. Even so, the design is still far from complete and more iterations of ship design spiral would be needed. Knowledge gained from the design process until now is absolutely crucial for possible next design stages of the design, and for future designs.

12.2 Group work

Discussing about the group work during this project is impossible without mentioning the effects of the global pandemic. Every one of us had to learn new style of group working. In the beginning of the project i.e., in the beginning of Principles of Naval Architecture course, this new online group working caused some minor issues and it took some time for the members of the group to get used to the new system. However, after the slow start this new way of working started to feel more natural and did not cause any significant problems for the project.

Our team has remained largely intact since the Principles of Naval Architecture and it has been functioning well since the beginning. Work has been done very democratically and no leaders or experts for any field has been named in advance. Overall, the working atmosphere in the group has been very much casual. Nevertheless, all the submissions have been done well and delivered on time. This approach of group working has been possible because of the motivated group members and clear dividing of work.

Dynamics within the group functioned very well, possibly due to mutual interest to the subject and similar attitude towards working. These same factors influenced the workload distribution within the group, which was without some exceptions, evenly distributed.

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13 Appendices

13.1 Appendix 1: Design hydrostatics report

Design hydrostatics report

Designer	
Created by	
Comment	
Filename	Straightfromexcel.fbm

Design length	117.10 (m)	Midship location	58.550 (m)
Length over all	132.92 (m)	Relative water density	1.0250
Design beam	22.000 (m)	Mean shell thickness	0.0000 (m)
Maximum beam	22.000 (m)	Appendage coefficient	1.0000
Design draft	7.000 (m)		

Volume properties		Waterplane properties	
Moulded volume	11605.5 (m^3)	Length on waterline	127.98 (m)
Total displaced volume	11605.5 (m^3)	Beam on waterline	21.962 (m)
Displacement	11895.7 (tonnes)	Entrance angle	8.030 (Degr.)
Block coefficient	0.6436	Waterplane area	2241.7 (m^2)
Prismatic coefficient	0.6803	Waterplane coefficient	0.8701
Vert. prismatic coefficient	0.7396	Waterplane center of floatation	47.350 (m)
Wetted surface area	3076.1 (m^2)	Transverse moment of inertia	76093 (m^4)
Longitudinal center of buoyancy	50.751 (m)	Longitudinal moment of inertia	2121929 (m^4)
Longitudinal center of buoyancy	-6.094 %		
Vertical center of buoyancy	3.942 (m)		

Midship properties		Initial stability	
Midship section area	145.7 (m^2)	Transverse metacentric height	10.499 (m)
Midship coefficient	0.9460	Longitudinal metacentric height	186.78 (m)

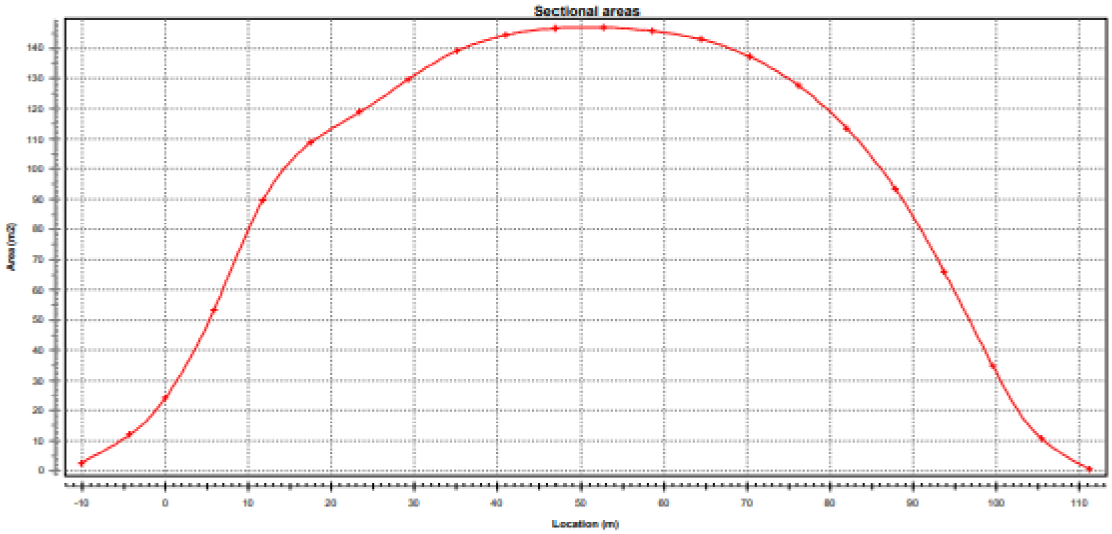
Lateral plane	
Lateral area	734.8 (m^2)
Longitudinal center of effort	52.732 (m)
Vertical center of effort	3.754 (m)

The following layer properties are calculated for both sides of the ship

Location	Area (m^2)	Thickness (m)	Weight (tonnes)	LCG (m)	TCG (m)	VCG (m)
Layer 2	4417.6	0.000	0.0	47.729	0.000 (CL)	4.429

Sectional areas

Location (m)	Area (m^2)	Location (m)	Area (m^2)	Location (m)	Area (m^2)	Location (m)	Area (m^2)	Location (m)	Area (m^2)
-10.145	2.6	17.565	108.8	46.840	146.6	76.115	127.7	105.390	10.8
-4.290	12.0	23.420	118.8	52.695	146.8	81.970	113.5	111.245	0.6
0.000	24.2	29.275	129.7	58.550	145.7	87.825	93.5		
5.855	53.1	35.130	139.1	64.405	142.9	93.680	66.1		
11.710	89.6	40.985	144.4	70.260	137.2	99.535	34.9		

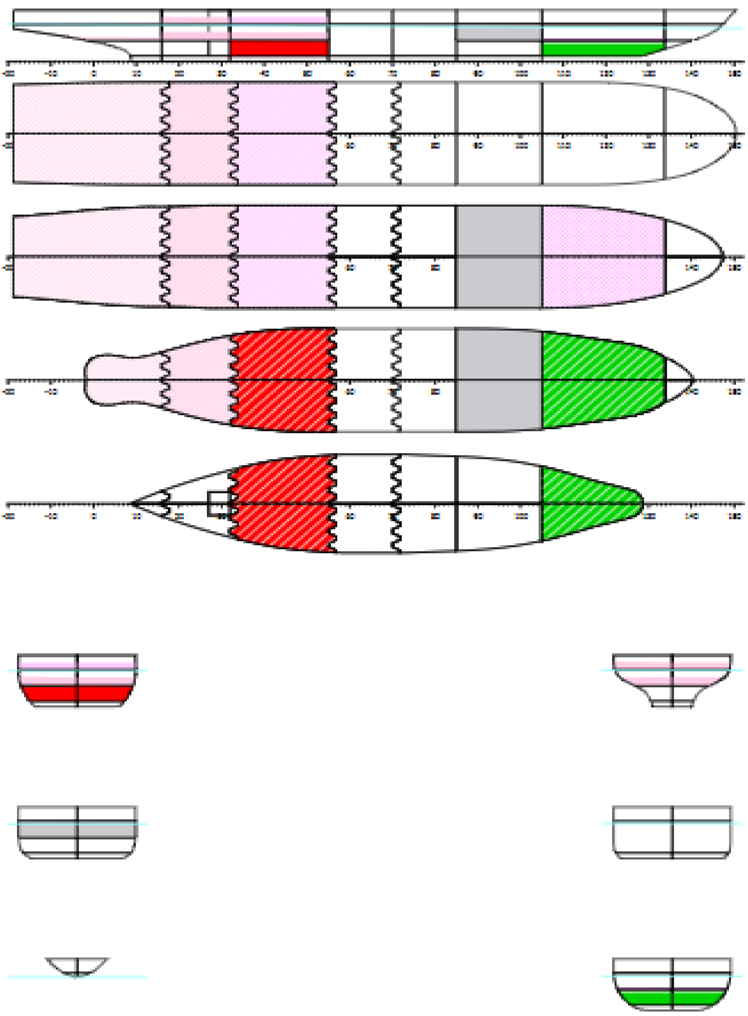


NOTE 1: Draft (and all other vertical heights) is measured from base Z=0.000
NOTE 2: All calculated coefficients based on project length, draft and beam.

13.2 Appendix 2: Loading conditions

Napa Oy | LOADING CONDITIONS | DATE 2021-03-24
NAPA/D/LD/201214 | | TIME 13.17
KHIONE-DESKTOP-CEJJNDH-8/A | FULL | USER ADMI
KHIONE-DESKTOP-CEJJNDH-8 | | 1

LOADING CONDITION: FULL -



LOAD	MASS t	XM m	YM m	ZM m	FRSM tm
Ballast Water	743.2	93.257	0.000	2.651	2196.56
Cargo	1841.6	22.890	0.000	6.385	14800.04
Liquid cargo	1175.7	75.979	0.000	6.219	0.00
Chain Locker	0.0	0.000	0.000	0.000	0.00
Cold Room Store	605.8	35.545	0.000	8.750	0.00
Diesel Oil	961.4	35.899	0.000	2.775	1456.76
General spaces	146.8	94.449	0.000	4.693	0.00
Hospital	0.0	0.000	0.000	0.000	0.00
Lift	0.0	0.000	0.000	0.000	0.00
Linen Store	0.0	0.000	0.000	0.000	0.00
Apparat Space	0.0	0.000	0.000	0.000	0.00
Machinery Sp.	0.0	0.000	0.000	0.000	0.00
Repair Shop	0.0	0.000	0.000	0.000	0.00
Sludge	0.0	0.000	0.000	0.000	0.00
Void	0.0	0.000	0.000	0.000	0.00
Deadweight	5474.4	49.448	0.000	5.425	18453.36

Lightweight	9450.0	55.000	0.000	6.400
Deadweight	5474.4	49.448	0.000	5.425
Total weight	14924.4	52.963	0.000	6.042

Floating Position

Draught moulded	7.528 m	KM	10.81 m
Trim	-0.899 m	KG	6.04 m
Heel, PS=+	0.0 deg		
TA	7.978 m	GM0	4.77 m
TF	7.079 m	GMCORR	-1.24 m
Trimming moment	-24069 tonm	GM	3.53 m

Longitudinal Strength

			X	Frame
Shear force (min)	-1099.8 t	(-)	91.103 m	113.879
Shear force (max)	1296.5 t	(-)	44.000 m	55.000
Sagging moment	-			
Hogging moment	50677.0 tm	(-)	58.084 m	72.604

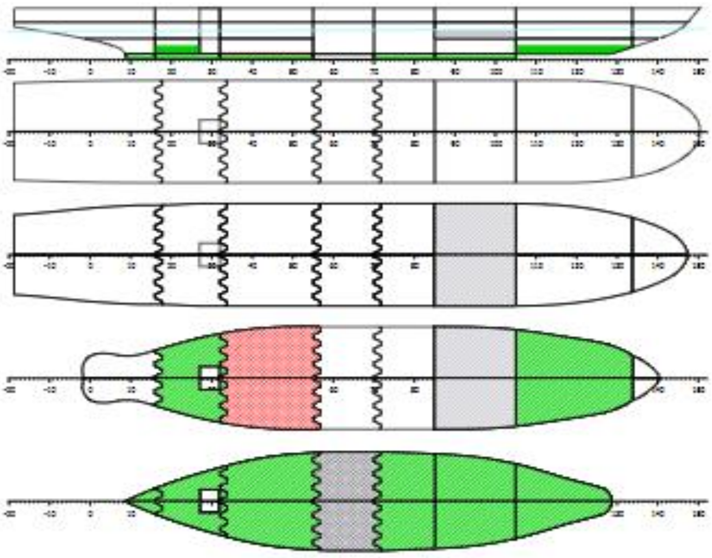
Napa Oy
NAPA/D/LD/201214
KHIONE-DESKTOP-CEJJNDH-8/A
KHIONE-DESKTOP-CEJJNDH-8

LOADING CONDITIONS

EMPTY

DATE 2021-03-24
TIME 13.19
USER ADMI
1

LOADING CONDITION: EMPTY -



Napa Oy	LOADING CONDITIONS	DATE 2021-03-24
NAPA/D/LD/201214		TIME 13.19
KHIONE-DESKTOP-CEJJNDH-8/A	EMPTY	USER ADMI
KHIONE-DESKTOP-CEJJNDH-8		2

LOAD	MASS t	XM m	YM m	ZM m	FRSM tm
Ballast Water	1402.3	62.753	0.000	1.312	9094.27
Cargo	0.0	0.000	0.000	0.000	0.00
Liquid cargo	599.8	75.963	0.000	5.380	3431.42
Chain Locker	0.0	0.000	0.000	0.000	0.00
Cold Room Store	0.0	0.000	0.000	0.000	0.00
Diesel Oil	103.3	36.006	0.000	1.378	1456.76
General spaces	0.0	0.000	0.000	0.000	0.00
Hospital	0.0	0.000	0.000	0.000	0.00
Lift	0.0	0.000	0.000	0.000	0.00
Linen Store	0.0	0.000	0.000	0.000	0.00
Apparat Space	0.0	0.000	0.000	0.000	0.00
Machinery Sp.	0.0	0.000	0.000	0.000	0.00
Repair Shop	0.0	0.000	0.000	0.000	0.00
Sludge	69.2	50.677	0.000	0.064	4265.99
Void	0.0	0.000	0.000	0.000	0.00
Deadweight	2174.7	64.742	0.000	2.398	18248.44

Lightweight	9450.0	55.000	0.000	6.400
Deadweight	2174.7	64.742	0.000	2.398
Total weight	11624.7	56.823	0.000	5.651

Floating Position

Draught moulded	6.291 m	KM	10.77 m
Trim	0.549 m	KG	5.65 m
Heel, PS=+	0.0 deg		
TA	6.016 m	GM0	5.12 m
TF	6.566 m	GMCORR	-1.57 m
Trimming moment	11335 tonm	GM	3.55 m

Longitudinal Strength

			X	Frame
Shear force (min)	-1162.6 t	(-)	88.713 m	110.892
Shear force (max)	1311.6 t	(-)	21.600 m	27.000
Sagging moment	-			
Hogging moment	55894.4 tm	(-)	52.032 m	65.041

13.3 Appendix 3: Intact stability

Napa Oy
NAPA/D/LD/201214
KHIONE-DESKTOP-CEJJNDH-8/A
KHIONE-DESKTOP-CEJJNDH-8

LOADING CONDITIONS

DATE 2021-03-24
TIME 10.25
USER ADMI
1

Loading condition: FULL

RCR	TEXT	REQ	ATTV UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.491 mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	0.793 mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.302 mrad	OK
GZ0.2	Max GZ > 0.2	0.200	1.749 m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	37.833 deg	OK
GM0.15	GM > 0.15 m	0.150	3.535 m	OK

Loading condition: EMPTY

RCR	TEXT	REQ	ATTV UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.567 mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	1.002 mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.435 mrad	OK
GZ0.2	Max GZ > 0.2	0.200	2.595 m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	40.808 deg	OK
GM0.15	GM > 0.15 m	0.150	3.546 m	OK

LIMIT CURVE

T m	TR m	MINGM m	MAXKG m	DCRI
3.840	0.000	0.465	11.452	AREA3040
4.264	0.000	0.150	11.329	
4.480	0.000	0.150	11.106	GM0.15
5.120	0.000	0.150	10.887	GM0.15
5.482	0.000	0.150	10.823	
5.524	0.000	0.157	10.809	
5.760	0.000	0.207	10.717	AREA40
5.957	0.000	0.259	10.637	
6.400	0.000	0.458	10.375	AREA3040
7.029	0.000	0.918	9.883	
7.040	0.000	0.937	9.863	MAXGZ25

Loading condition: OVEREMPTY

RCR	TEXT	REQ	ATTV UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.645 mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	1.125 mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.480 mrad	OK
GZ0.2	Max GZ > 0.2	0.200	2.871 m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	40.270 deg	OK
GM0.15	GM > 0.15 m	0.150	4.611 m	OK

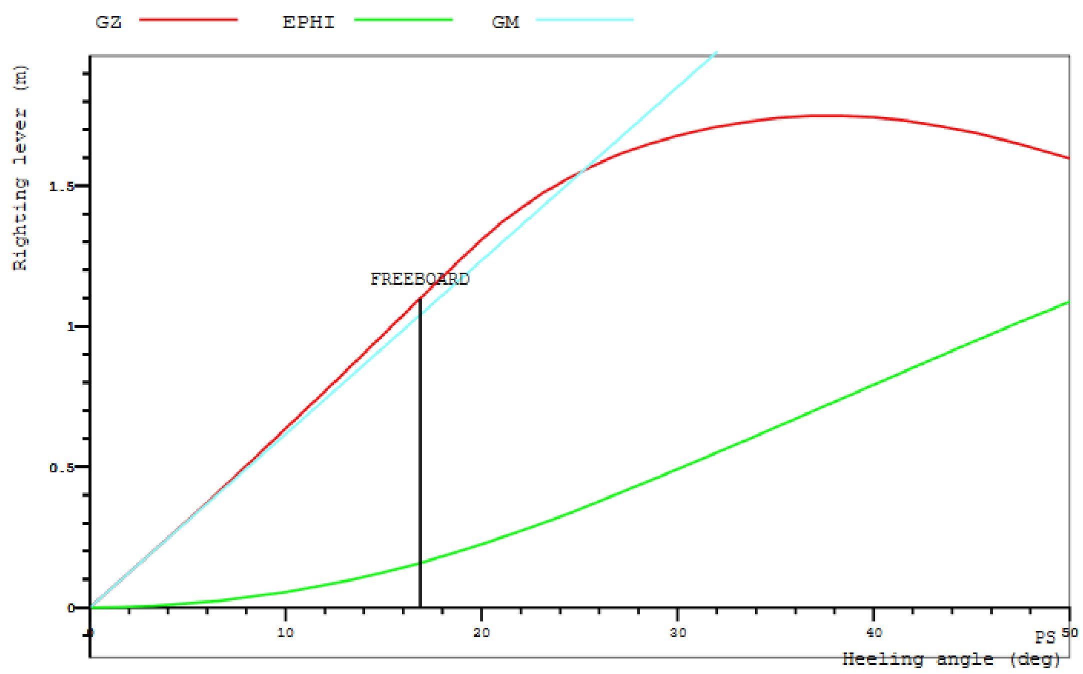
Loading condition: OVERFULL

RCR	TEXT	REQ	ATTV UNIT	STAT
AREA30	Area under GZ curve up to 30 deg.	0.055	0.421 mrad	OK
AREA40	Area under GZ curve up to 40 deg.	0.090	0.659 mrad	OK
AREA3040	Area under GZ curve btw. 30-40 deg.	0.030	0.238 mrad	OK
GZ0.2	Max GZ > 0.2	0.200	1.376 m	OK
MAXGZ25	Max. GZ at an angle > 25 deg.	25.000	37.185 deg	OK
GM0.15	GM > 0.15 m	0.150	3.764 m	OK

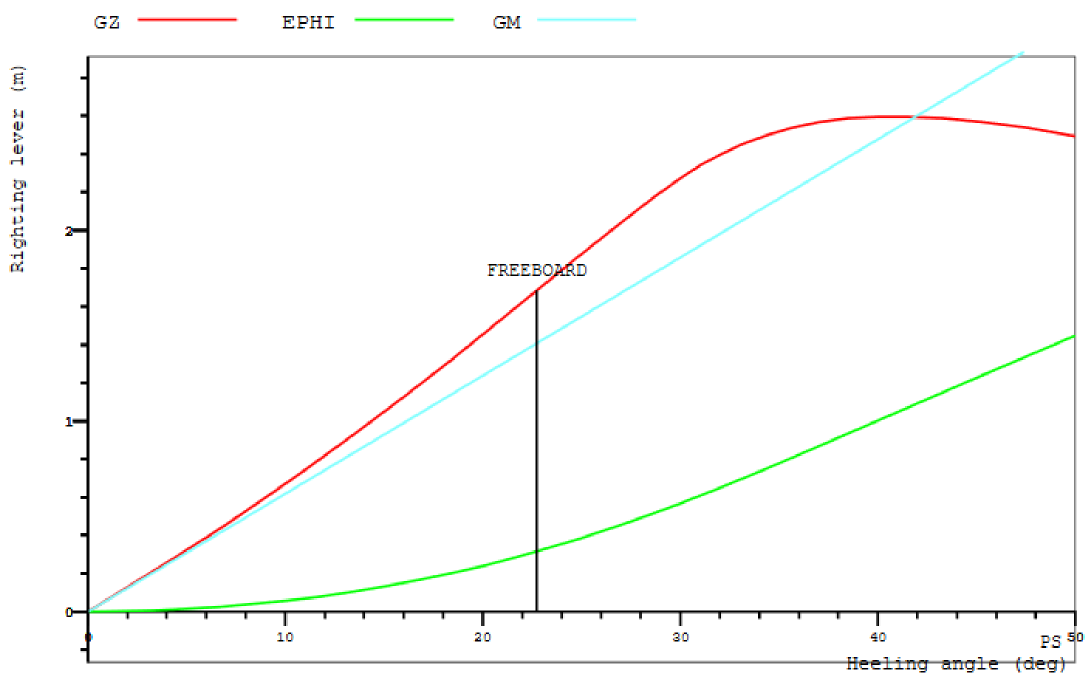
LIMIT CURVE

T	TR	MINGM	MAXKG	DCRI
m	m	m	m	
3.840	0.000	0.465	11.452	AREA3040
4.264	0.000	0.150	11.329	
4.480	0.000	0.150	11.106	GM0.15
5.120	0.000	0.150	10.887	GM0.15
5.482	0.000	0.150	10.823	
5.524	0.000	0.157	10.809	
5.760	0.000	0.207	10.717	AREA40
5.957	0.000	0.259	10.637	
6.400	0.000	0.458	10.375	AREA3040
7.029	0.000	0.918	9.883	
7.040	0.000	0.937	9.863	MAXGZ25

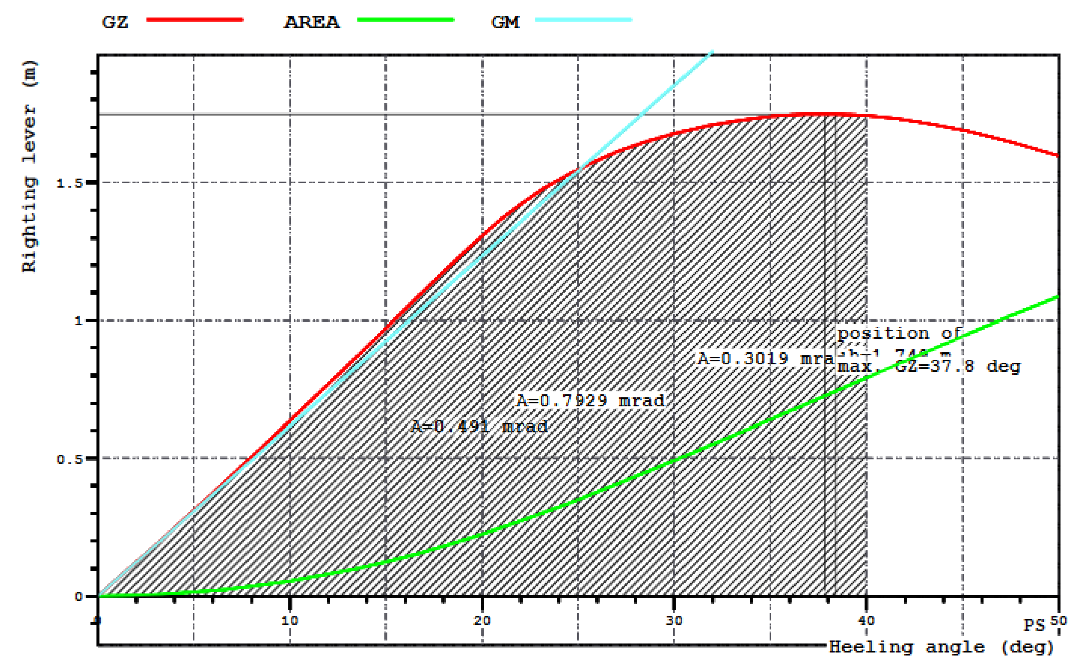
FULL loading condition rightening lever



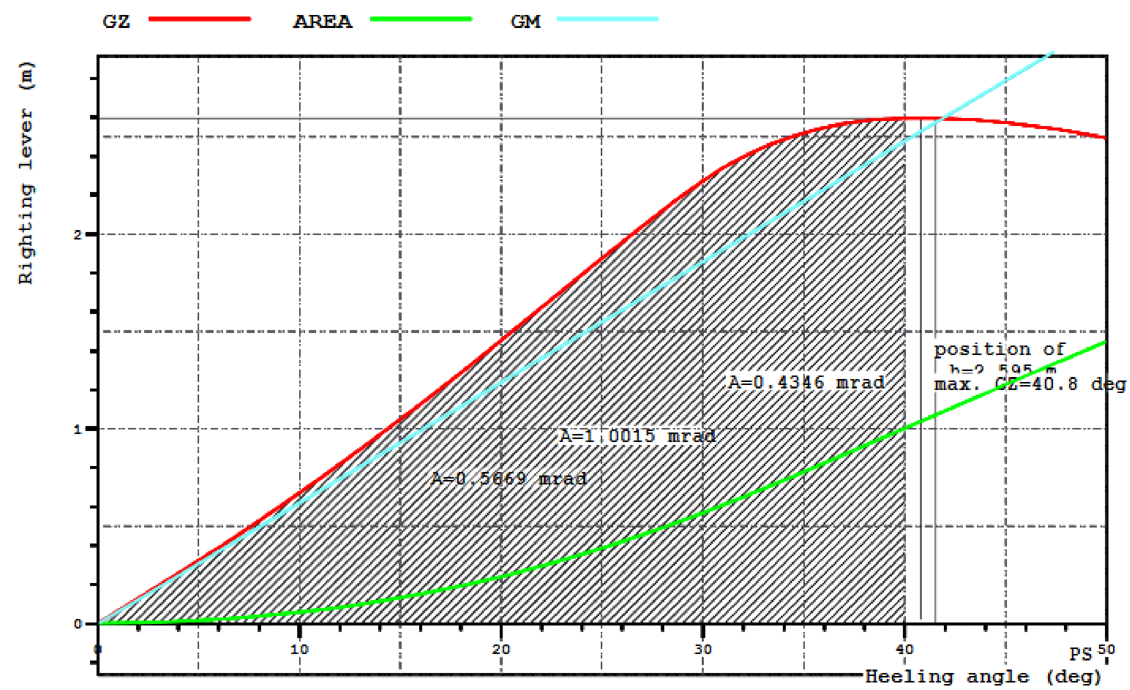
EMPTY loading condition rightening lever



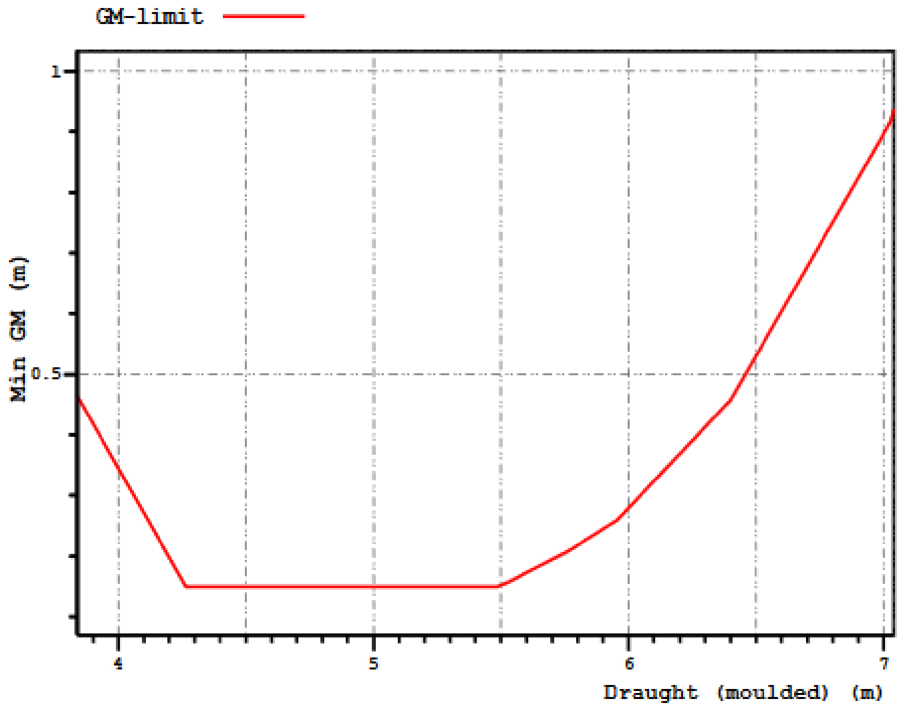
FULL loading condition intact stability criteria check



EMPTY loading condition intact stability criteria check



GM limit plot



13.4 Appendix 4: Technical Specification

IB Khione

Technical Specification

General description

Designed vessel is research and re-supply vessel for remote Arctic and Antarctic areas. Khione is designed to have worldwide operation area and will ability to independently navigate in ice with thickness up to 1.65 meters. Ship will be capable of conducting long voyages without need for external support. Latest technology is used for research and re-supply purposes. Environmental friendliness is the key factor in whole design of the ship.

Warranted data

Speed

Design speed of the vessel in open water is 13 knots and design speed in 1.65 m thick ice is 3 knots. These speeds are warranted to be achieved in specific environmental conditions. Conditions listed below.

- Calm wind and wave conditions i.e., Beaufort and Douglas scale 0
- No sea currents
- Deep water
- Clean hull and propellers
- All the engine power is usable
- Water density 1025 kg/m^3
- Even keel trim
- Draft at 6.4 m

Deadweight

Total deadweight of the ship is 2970 tons. Warranted cargo deadweight is minimum 2460 tons.

Main parameters

- Length over all 132.9 m
- Length between perpendiculars 135.34 m
- Beam 22 m
- Design draft 6.4 m
- Depth 11 m
- Freeboard 4.6 m
- Block coefficient 0.68

Net tank capacities

- Diesel oil 1200.7 m^3
- Liquid cargo 1199.7 m^3
- Ballast water 2736.2 m^3
- Other 290.8 m^3

Ice class

- Polar Class 4

Designed crew/passengers

- 20 ship crew members
- 80 research scientists

Accommodation

- 53 x 10 m^2 rooms with two berths
- 2 x 15 m^2 rooms with single berths

Locations and distribution of accommodations are according to General Arrangement of the ship.

Utility spaces

Science and laboratory spaces occupy in total area of 500 m^2. Additionally, to research and scientific spaces ship has various utility spaces, which are listed below.

- Laundry
- Workshops
- Locker rooms
- Recreation space
- Cargo hatches
- Hospital
- Helicopter hangar
- Mess
- Gym
- Spa
- Bridge

Size and location of utility spaces are according to General Arrangement of the ship.

Cargo

- Cargo deadweight capacity is 2460 tons
- General cargo hatches have total net volume of 6018 m^3
- Liquid cargo tank net volume 1199.7 m^3
- Cold room storage with net volume 1211.6 m^3
- Cargo handling done with two hydraulic TTS Cargo cranes
  - Main crane capability 30 tons at 18 m
  - Auxiliary crane capability 10 tons at 24 m

Hull

Hull structure of the ship is designed in accordance with rules of classification society and keeping the icebreaking in mind. Ship has mixed framing system to withstand longitudinal bending and ice loads. Double bottom and decks are framed longitudinally with frame spacing of 550 mm. Transversal side structure framing has spacing of 800 mm. Side girders are placed on every fourth longitudinal frame (2200 mm) and web frames every fifth transversal frame (4000 mm). Ice belt is used to form ice strengthening of the hull. High tensile steel is used for the structural components and ship will be constructed by welding.

Machinery and main systems

Propulsion

Propulsion provided by two shafts with fixed pitch propellers with a diameter of 5m.

Engines and generators

The ship is diesel-electric and will have four main diesel engines driving generators connected to the two electric motors giving power to the driveshafts.

Anchoring and mooring

The anchoring equipment on board will consist of mooring winches, anchor windlasses, chain stoppers, fairleads, anchors, chains and ropes. To comply with DNV rules there will be two anchors, 550m of chain and five mooring lines of 190m each on board.

Life-saving equipment

The ship will have two enclosed lifeboats on davits and one fast rescue boat that can also be used in research operations. The fast rescue boat can be put to sea using the ships on board cranes. In addition, there will be personal lifesaving equipment for the crew such as thermal clothing and immersion suits. There will also be searchlights and emergency signalling equipment on the ship.

Boilers and steam generators

Waste heat from the engines and electricity are the intended heating sources for the ship. If future estimations and measurements show this to be insufficient steam boilers can be added to the engine rooms.

Emergency diesel generators

The ship is not intended to have separate emergency diesel generators. This is because the main engines in conjunction with the hydrogen fuel-cells are estimated to provide enough redundancy and capacity for emergency power.

Machinery main components

Primary components in the engine room, for example main engines, propellers, plant, boilers, pressurized air system for starting and generators.

Fuel and lubrication systems

The ship's fuel system includes storage and service tanks and a purification system. The lubricating oils are stored in separate tanks in the technical spaces.

Hydrogen systems

The ship will have hydrogen tanks on the aft deck and five fuel cells inside the hull.

Exhaust gas system

The exhaust line is to be equipped with SCR-system for cleaning of exhaust gases. Systems for waste heat utilization are also provided in the engine casings.

Common systems

Fresh water system

The ship will have a combination of a water treatment plant and tanks to provide the ship with fresh water. The tanks are dimensioned so that even in the case of technical problems with the purification system there is still enough to either make repairs or go into port.

Shore connection

The ship is to have electrical shore connection on both sides. The stations have sockets to connect the ship to the land based electrical network.

Voltage

Voltage onboard follows the land standardin Finland, 230V/ 50Hz alternating current.

Lighting

Energy efficient technologies, such as LED to be utilized throughout the ship. Motion detectors for lights on spaces not manned constantly. The lighting design to be made according to architectural drawings and visions in the accommodation, and according to the regulations to provide enough lighting in the working areas.

Ballast system

The ballast system is designed to ensure correct floating position in all possible loading conditions.

Fire systems

The ship has on board fire protection and detections systems in accordance with governing regulations.

Sewage system

The ship has an on board treatment plant with backup tanks that ensure no sewage is released into the environment.

Classification and rules

Flag

The flag state of our ship is Finland.

Classification society

The ship is designed according to the rules set out by DNV.

Class notation

✠ 1A Multi-purpose dry cargo ship FC(POWER) E0 RP(2) Crane HELDK SPS PC(4) Shore power

International rules

- Polar code
- SOLAS

Makers list

Main engines and generators	Wärtsilä
Electrical systems and motors	ABB
Bow thruster	Wärtsilä
Fuel cells	Ballard
HVAC	Heinen & Hopman
Evacuation system	Viking
Software	NAPA
Cabin modules	Piikkio Works Oy
Fire protection and detection	Marioff
Water management system	EVAC

13.5 Appendix 5: Ship Dynamics

Aalto University School of Engineering
Department of Mechanical Engineering
Marine Technology
MEC-E2004 Ship Dynamics

Khione

Ship Dynamics

Submitted 28.05.2021

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1 Ship Dynamics

1.1 Operational profile

Our vessel will not have a specific route that she will always follow. Her operating profile changes as the route she takes varies. Also, the profile will also vary regarding the re-supply and research operations during the voyage. A possible voyage, shown in Figure 1, was defined to give an example of the types of routes we envisage for our vessel. This voyage starts from Tuktoyaktuk, Canada to Reykjavik, Island via Aasiaat, Greenland. The voyage also includes a stop at the Canadian Arctic Archipelago close to Resolute Bay, where a re-supply operation is conducted with helicopters. The depth of the port of Resolute Bay is 4.9 – 6.1 m (Ports.com, n.d.), which is too low for our vessel.

Figure 51. Voyage for operating profile

The first 200 nautical miles include the departure from Tuktoyaktuk and open sea voyage until the sea ice is met as the vessel approaches the Canadian Arctic Archipelago. When meeting the ice our vessel will have to slow down substantially, but it is impossible to give an exact constant speed. The speed is not constant as the ice thickness and condition may vary in the archipelago. As the vessel reaches Resolute, it will stop for the duration of the re-supply operation. Simultaneously, research operations can be conducted, if possible and necessary. The range of a voyage between Tuktoyaktuk and Resolute is approximately 860 nautical miles. The vessel will continue its journey through the ice until it reaches the Baffin Bay, where the sea is no longer ice covered. After approximately 900 nautical miles, the vessel reaches Aasiaat. From there, the voyage will continue to Reykjavik, which it will reach after 1500 nautical miles.

As our vessel will travel a substantial distance in ice, which is always changing, therefore our operating profile does not show a specific route. Instead, it shows our vessels behaviour in different conditions. The profile can be divided into two general areas, one where the ship is operating in open water at its cruising speed of 13 knots and the other one where it is operating in ice.

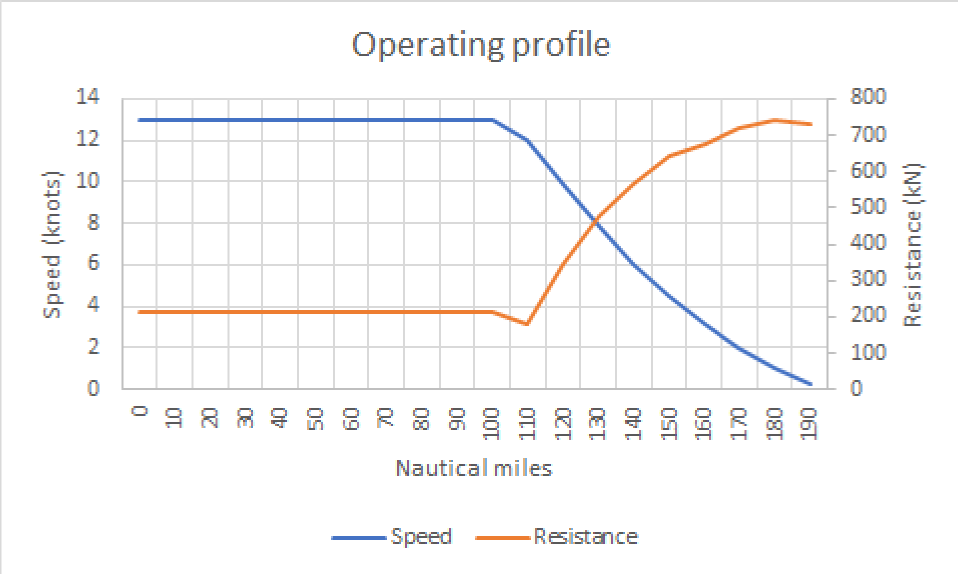


Figure 52. Speed and resistance during operating profile

This Operating profile is of a representation of an idealised voyage for our vessel. On this voyage the first 100 nautical miles are open sea. This means that our ship is doing its cruising speed of 13 knots and encountering the associated resistance. After the first 100 nautical miles the vessel encounters ice. The ice gets thicker and thicker at a rate of 10 cm per 10 nautical miles, meaning that at 110 nautical miles its 10 cm thick, at 120 nautical miles is 20 cm thick and so on.

Ship dynamics requirements

The ship's dynamics requirements are dependent on the operational area of the ship. As was described in the section on the operational profile, our ship does not have a regular route it follows. What can be said about the operational area is that it is very likely to include some of the most challenging waters in the world. This means that our ship should be designed for unrestricted service and given the service area notation R0 by DNV GL and comply with their Rules for Ships Pt.3 Ch.1 Sec.4.

1.2 Maneuvering devices

Khione is fitted with diesel-electric propulsion system with two shafts with one fixed pitch propeller each, and a tunnel thruster in the bow. The bow thruster will enable better manoeuvrability at low speeds and increase safety in berthing in bad weather (Marine Insight, 2021). The diesel-electric makes it possible to place the components of the propulsion system more flexible allowing the generators for the shafts to be fitted at the aft of the ship making the shafts themselves shorter.

A twin rudder arrangement will be used, the rudders are located behind the propellers to produce transverse force and steering moment by generating the water flow. The rudders could be spade or semi-balanced skeg rudders as they are usually fitted on twin-screw vessels. As the vessel will operate in ice the rudders should be fitted with ice knifes to prevent the rudders from head-on impact of the ice floes when going astern, as the ice knifes will push ice floes downwards. The strength and shape of the ice knife should be designed to meet its function and should be below the waterline when operating in ice (Traficom, 2019).

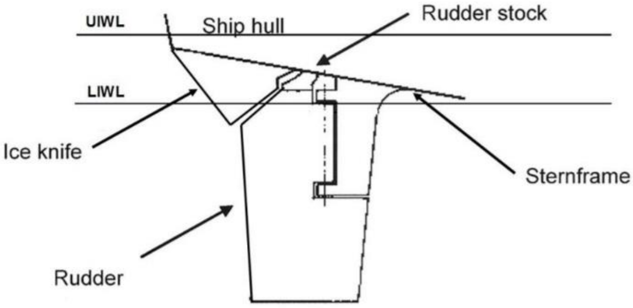
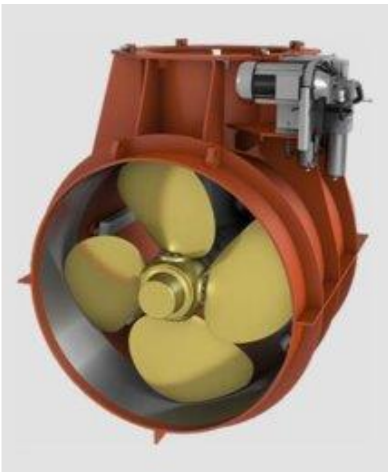


Figure 53. Twin rudder arrangement and 54. Ice knife implementation (Traficom, 2019)

A bow thruster will support manoeuvring, mooring operations, station keeping and dynamic positioning. According to Wärtsilä the bow thruster should be located as far in the bow as possible, and the suitable tunnel length is 2-3 times the propeller diameter (Wärtsilä, 2017). The propeller would be controllable pitch to enable change in direction of thrust. Sizewise suitable alternatives would be Wärtsiläs WTT models with propeller diameter around 2 m providing maximum power in manoeuvring between 1450-1850 kW (Wärtsilä, 2017). The components for the bow thruster are shown in figure 6. The bow thruster room should be easily accessible by the crew and have space for the motor and other necessary equipment needed (Marine Insight, 2021).

Bow thrusters are usually not used in ice since it can damage the thruster blades but to prevent ice from entering the thruster tunnel a grid can be installed. Even though the grids can have a negative impact on the thruster's open water performance. (Traficom, 2019)



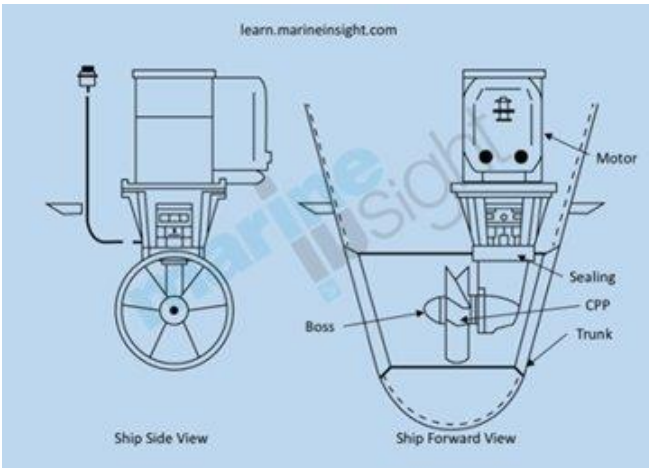


Figure 55. Wärtsilä WTT transverse thruster (Wärtsilä, 2017) and 56. Bow thruster components (Marine Insight, 2021)

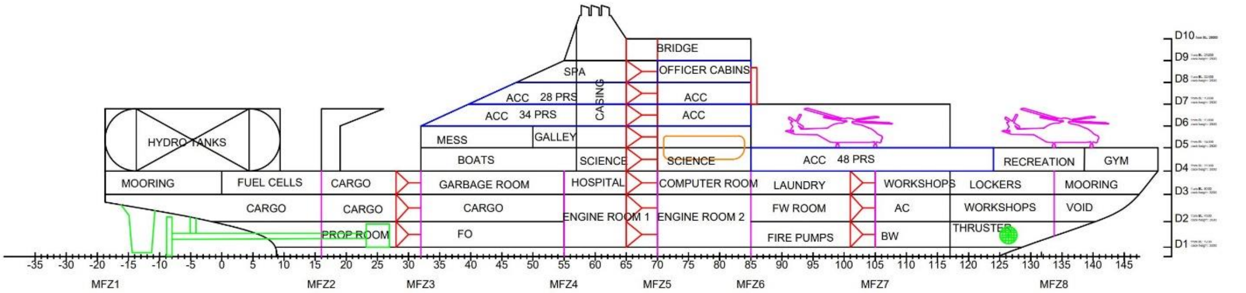


Figure 57. Profile view showing bow thruster and rudder

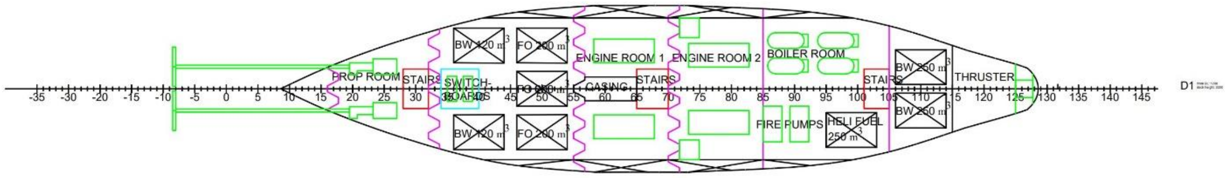


Figure 58. Deck 1.

The propulsion system as well as bow thruster is located on deck 1, shown in figure 8, main engine rooms reaching from deck 1 to 2. The thruster will have the thruster room besides it which will have space for all the needed equipment for the thruster. The propellers for propulsion have diameter of 5 m and the shafts will be supported to the hull.

1.3 Effects of the Hull Form on the Ship Dynamics

The hull form of our project ship is typical icebreaking hull design, which affects to open water characteristics of the ship. Especially shapes of the stern and the bow are important factors is ships open water characteristics and so dynamics of the ship.

Our project ship has spoon shaped bow to minimize crushing ice and then ice resistance. This solution for shape of the bow can cause heavy slamming in open water conditions. Stem angle at water level is 37° and below water level about 20°, which is typical for icebreaking vessels. Due icebreaking bow the vessel has higher resistance in open water. Large bottom area of spoon bow hits water surface when ship emerges and submerges in waves. Interaction between water surface and hull can cause extreme forces in slamming, that can cause deformation of the hull and vibrations along the hull beam.

Stern should have large enough clearances between tip of propeller blades and stern frames and bottom of the level ice sheet. Clearances must be large enough to avoid loads that can occur when ice floes are forced between the propeller and the stern frame and when propeller can hit large ice floes (Traficom, 2019). To have large clearance propellers must be deep enough. Number of propellers affects greatly to the stern design. Stern design must such that it protects rudders and propellers (Canadian Coast Guard, 2012). We have employed flat transom design in our hull design as it is typical for icebreaking vessels (Figure 59). Flat transom stern can have negative effect on ship stability in open water. In their paper Silva and Guedes Soares present different features of the ship that make ship vulnerable to extreme rolling (Silva, 2010). Flat transom stern is one of the identified features.

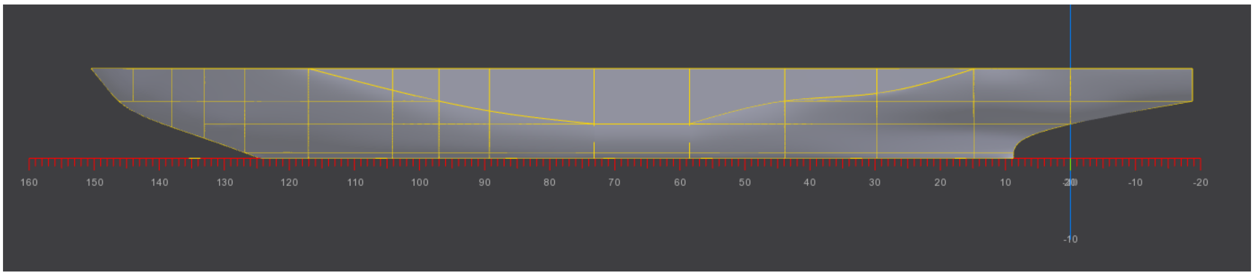


Figure 59. Snapshot from NAPA of Khione's hull design.

2 Ocean Waves

2.1 Water Depths of Operational Area

As our project ship operates in arctic waters, water depths of arctic shipping routes are used as water depths of operational area. Shipping routes considered are: Northwest Passage (NWP), Northern Sea Route (NSR), and Arctic Bridge (AB) (Figure 60).

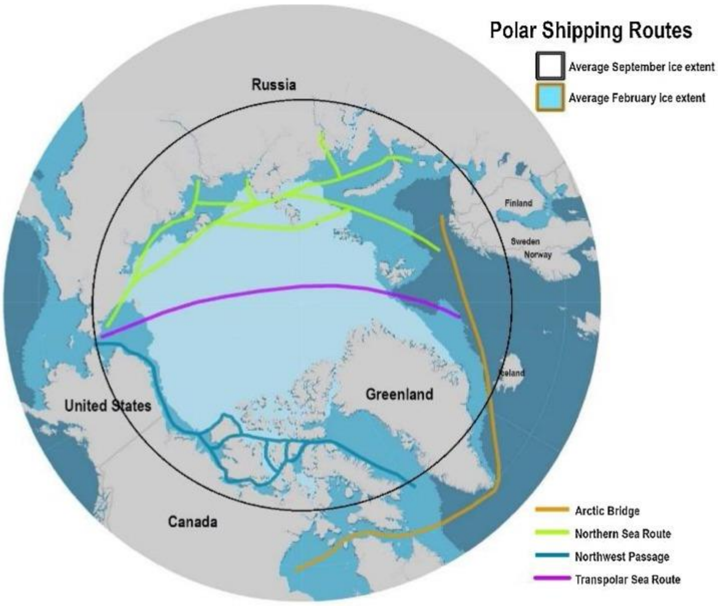


Figure 60. Arctic shipping routes (Rodriguez, 2010).

Northwest Passage includes seven different routes through the Canadian Arctic Archipelago (Headland, 2020). Two of the routes are considered deep water routes. Rest of the routes have shallows, restricting the draft to less than 10 m. (Arctic Council, 2009). Parts of the Northwest Passage which require draft of less than 6.4 m can be avoided while navigating through the Northwest Passage, thus constrain for draft set by the (NWP) can be considered as 14 m. (Headland, 2020).

Northern Sea Route has multiple paths. Including straits with depth less than 10 m overall with depths varying between 8-250 m. However, shallowest areas can be avoided by taking different route. Shallowest part of NSR is Dmitry Laptev Strait restricting draft of the ship to 6.7 m, which is larger than draft of our project ship (6.4 m). (Arctic Council, 2009).

Arctic Bridge lays in relatively deep waters. Shallowest part of the Arctic bridge is the Hudson's Bay (NOAA, 2020). Average depth in the bay is 125 m (W. Burt, 2016). Arctic Bridge route doesn't set any realistic constrains for draft of our vessel. Bathymetric chart of Arctic area (Figure 61) shows that arctic waters outside the shipping routes are relatively deep, allowing our project ship operate quite freely in limits of ice conditions.

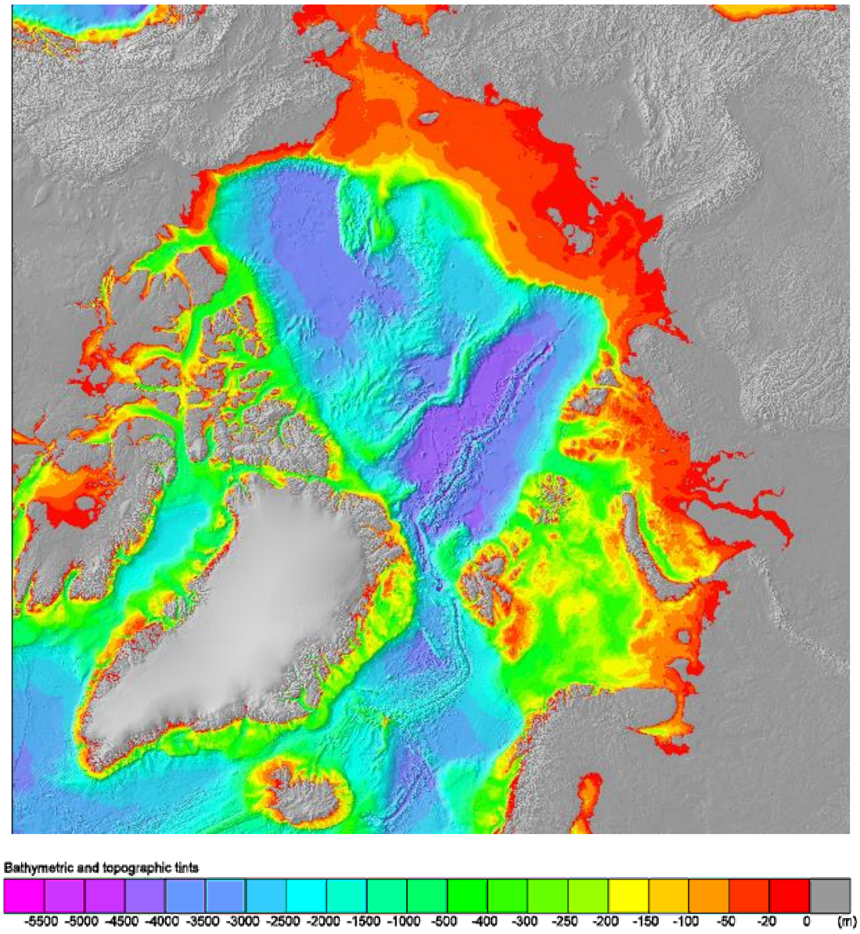


Figure 61 Bathymetric chart of Arctic area (NOAA, 2020)

2.2 Seasonal Variations of Wave Conditions

In arctic areas, that is our planned operational area, waves and swells are affected by mainly by ice coverage, wind strength and direction, but also interannual climate oscillations (Stopa, et al., 2016). In summer when the arctic ice coverage is in its minimum the waves are at their largest as the there isn't ice to damp waves (Rainville & Woodgate, 2009). Thomson and Rogers (2014) suggest that with decreasing ice coverage waves size of waves increase and swells get more common. In the future larger and more energetic waves will then break ice and accelerate decrease of ice coverage. (Thomson & Rogers, 2014). Second maximum of wave heights is in early winter, when wind speed is highest (Dosser & Rainville, 2016). Figure 62 shows seasonal variation of wave height and how waves are highest at summer and second maximum is reached in early winter.

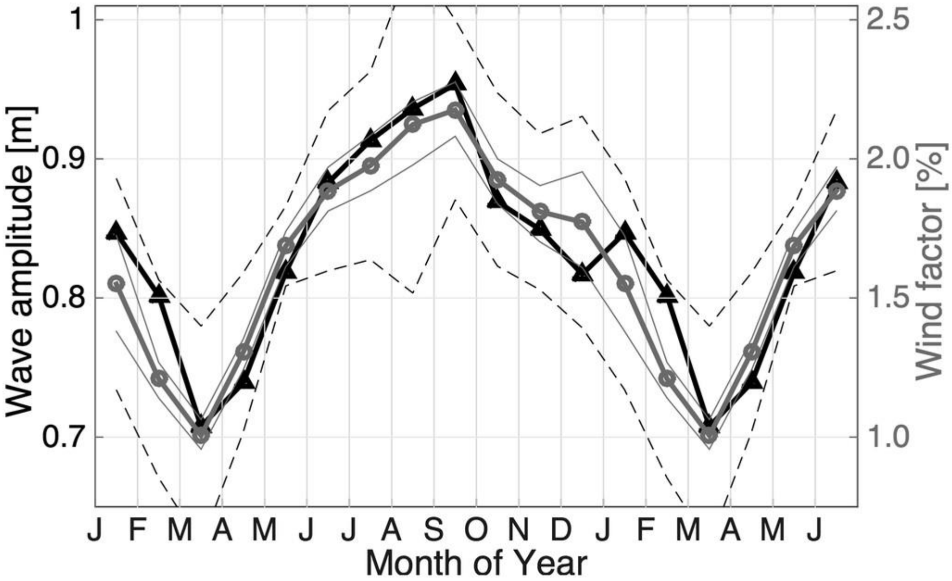


Figure 62. Wave height varies seasonally. Black line is Near-inertial internal wave amplitude data, thick grey line is wind factor, dashed lines are standard deviation and thin grey lines are bootstrapped uncertainty estimates.

2.3 Wave types

A typical research journey as presented in our operating profile will be in the Canadian Arctic Archipelago and around the arctic ocean and Labrador sea. The wave characteristics are dependent on the depth of the water, strength of wind, duration of the wind and the length of the water surface over which it acts, which in our case also are affected by the ice-free ocean.

To study the wave characteristics in the arctic seas the following pictures illustrates the mean and extreme wave heights on an average base don years between 2002 and 2012. As examples necessary for our operating profile the Labrador Sea has on average a mean wave height at 4 m in March and 3 m in September, while the extreme wave height is between 5-7m depending on the season. In Baffin Bay the wave heights is on average around 2m and maximum 4m, when it's not covered in ice. In the arctic the wind speed is approximately 8 m/s, in Baffin Bay about 7 m/s and highest in the Norwegian sea 9,5 m/s.

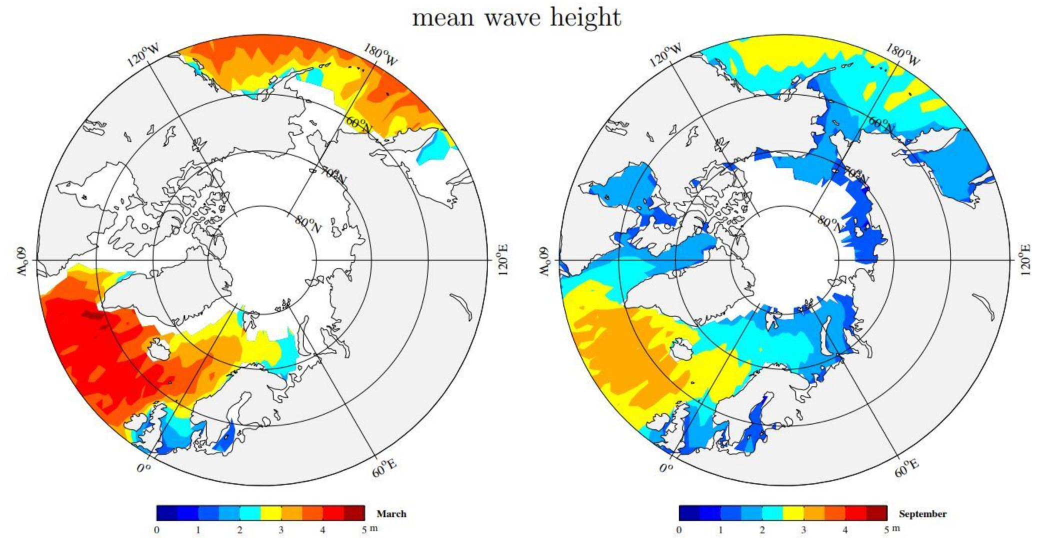


Figure 63. Mean wave height in the Arctic Seas, average over 2002-2012. Left March, right September. (Babanin, et al., 2014)

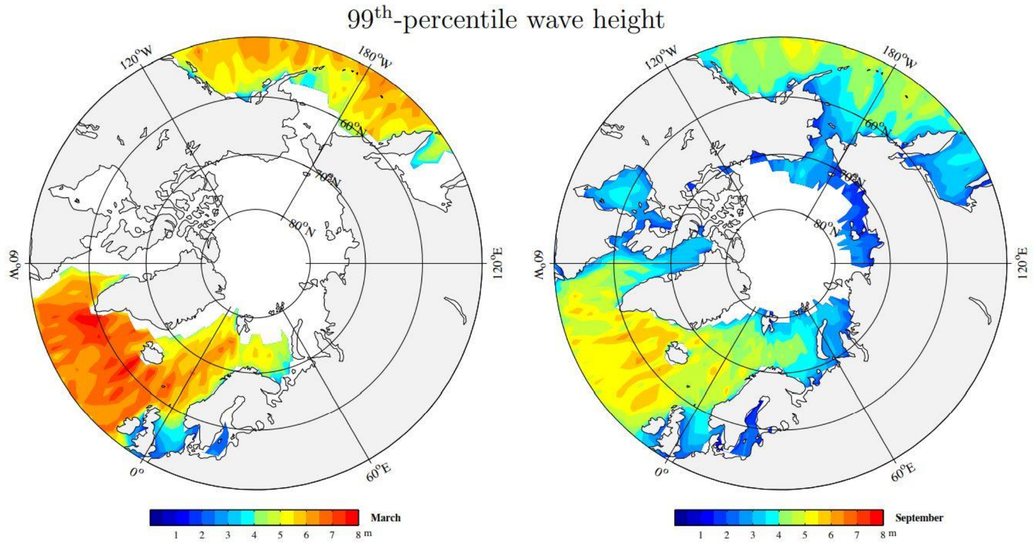


Figure 64. Extreme wave heights in the Arctic, over 2002-2012. Left March, right September. (Babanin, et al., 2014)

During March it’s expected that the area is covered in ice despite North Atlantic and since September is the end of Northern summer the time should have the maximal area of ice-free ocean (Babanin, et al., 2014). As the area of ice-free ocean increases it affects the building of ocean surface waves.

The different types of ocean waves are presented in the picture below with period band and energy scale. The shortest-period waves are on the right side and is the phase when wind starts blowing. Gravity waves is occurred when wave length is between 1.5-900 m or period 1-25 s. In storm conditions the wave period is between 10-12 s and wave length 150-220m. Swells occur when the waves propagate over a larger depth than the wavelength and have typically wavelength greater than 260 m or period 24 s. (Toffoli & Bitner-Gregersen, 2017)

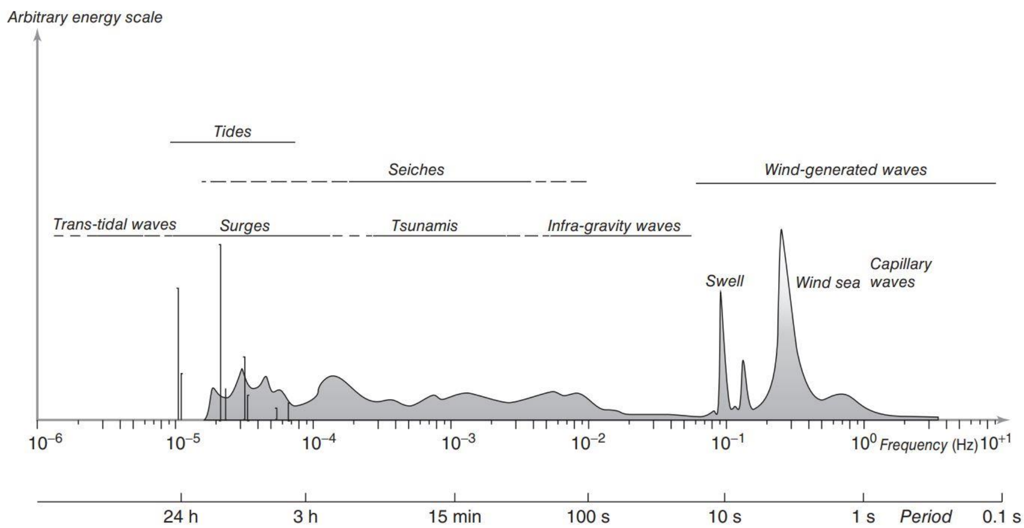


Figure 65. Types of ocean waves (Toffoli & Bitner-Gregersen, 2017)

To sketch waves that can be encountered on a journey the wave height data and different periods is used. The wave height 2 m is chosen as an average wave height in Baffin Bay and 7 m as an extreme wave height. As periods 2 s is chosen to illustrate general waves and 6s for light storm conditions.

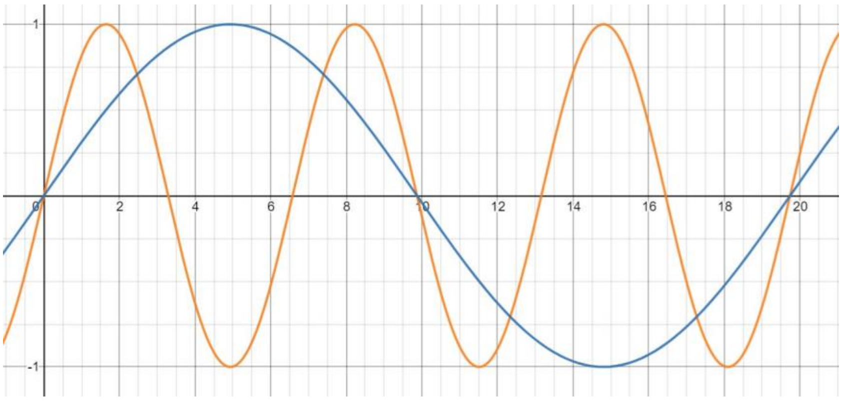


Figure 66. Sketched waves with wave height 2 m and period 2 s (blue) and 6s (orange).

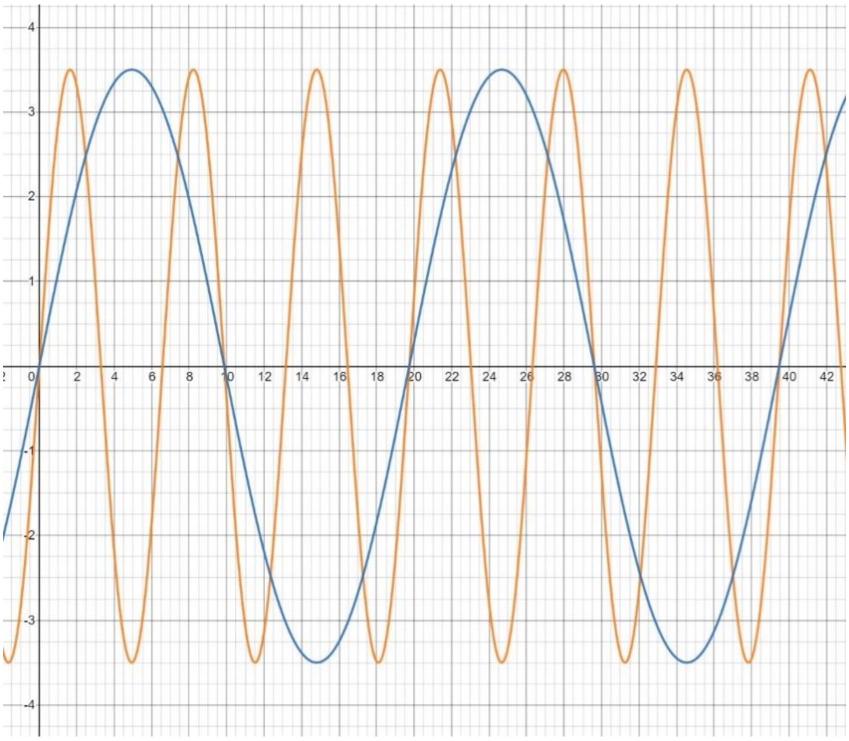


Figure 67. Sketched waves with wave height 7 m and period 2 s (blue) and 6s (orange).

2.4 Extreme events

Extreme events like extreme ocean surface waves can become larger and increase to occur in the arctic due to the climate change. Extreme wave events which have used to occur every 20 years can according to studies increase to occur every couple of years and the waves could get up to 2 m higher than the current wave heights. (ScienceDaily, 2020)

Since the area of ice-free ocean increases the occurrences of extreme waves increases aswell. This allows also increase in arctic cyclones affecting extreme waves, most notable in September when it's ice-free. Since the locations of open water in the arctic varies by years and days the likelihood of extreme waves generated by cyclones is a stochastic process. In years 1979-2016 events related to low-pressure systems, mostly cyclones have occurred 23 times during August, 25 during September and 21 during October. Despite the higher amount in September the statistics show increased amount during the latter years. (Waseda, et al., 2020)

2.5 Our chosen spectra

We are going to use the ISSC spectra for our project. We believe that this is a good choice based on the availability of data and the formulation of the spectrum. The spectrum is defined by two parameters, significant wave height and period. This formulation makes the spectrum easy to combine with wave data from different areas also available from the ISSC.

From what we have read the accuracy of the spectrum appears to be satisfactory, although Michel (W., 1968) mentions that there are some issues with the spectral density of the Bretschneider spectrum, which is very similar to the ISSC spectrum. Despite this Michel, in the same paper, describes a great benefit of using a spectrum defined partially by the period. That is the ability to manipulate the period to study the maximum effect on a vessel. This seems very useful to us and is partly why we have chosen this spectrum.

3 Equations of Motion

3.1 Equations of motion

When describing ship rigid-body motions there are six degrees of freedon to consider, three translations and three rotations. Between these degrees of freedom there are differences in how they should be described stemming from the fact that some have associated restoring forces and others do not. Those with restoring forces are often considered the most important since they can oscillate harmonically. The individual equations for rigid-body motions have their origin in the equation of movement for a single degree of motions system consisting of a spring, a mass and a damper. This simple equation however must be modified to apply to the motions of a ship. In its modified form the equation of motion for a ship becomes.

$$[-\omega_e^2(M+A) + i\omega_e N + S]\,\widehat{\overrightarrow{u}} = \widehat{\overrightarrow{F}_e}$$

In this equation M, A, N and S are six-by-six matrices. As in the simpler equation we have three terms, one for the mass, one for the damping and one for the restoring force. However, we now have A which is the added mass for the system. The added mass deals with hydrodynamic forces stemming from the water's movement around the hull. If we assume symmetry for the mass distribution the mass matrix becomes.

$$M = \begin{bmatrix} m & 0 & 0 & 0 & mz_g & 0 \\ 0 & m & 0 & -mz_g & 0 & mx_g \\ 0 & 0 & m & 0 & -mx_g & 0 \\ 0 & -mz_g & 0 & \theta_{xx} & 0 & -\theta_{xz} \\ mz_g & 0 & -mx_g & 0 & \theta_{yy} & 0 \\ 0 & mx_g & 0 & -\theta_{xz} & 0 & \theta_{zz} \end{bmatrix}$$

We can see that we are dealing with both mass and mass moments of inertia. In our case the mass distribution is going to be relatively consistent. This is because although we have cargo and consumables on board their weight is proportionally small. This means that in our seakeeping analysis the differences between loading conditions are hopefully quite small and can be made smaller with appropriate ballasting.

$$S = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \rho g A_w & 0 & -\rho g A_w x_w & 0 \\ 0 & 0 & 0 & gm\overline{GM} & 0 & 0 \\ 0 & 0 & -\rho g A_w x_w & 0 & gm\overline{GM}_L & 0 \\ 0 & 0 & 0 & 0 & 0 & \theta_{zz}\omega_g^2 \end{bmatrix}$$

Above the matrix for the restoring forces can be seen. This matrix can prove challenging for us because of our flat transom. This means that the assumption in many sea keeping analysis methods of small motions and constant waterplane area may not be valid and therefore the restoring forces can be a problematic area for us that requires special attention.

For the damping and added mass there are no general matrices to present. Both depend on hydrodynamics and will have to be evaluated with seakeeping software. Most likely this software will use some implementation of strip theory or a panel method. Strip theory is described later in the section.

3.2 Effects on Equations of Motions

Effects of general arrangement, hull form and operational on equations of motions are observed in this chapter. The equations of motions and their components are identified in chapter before. The effects on EOM are considered relative to the safety terms. The equation of movement is as follows:

$$\left[-\omega_e^2\left(M+A\right)+i\omega_e N+S\right]\hat{\vec{u}}=\hat{\vec{F}}_e$$

General Arrangement

General arrangement related features effecting on EOM are mass distribution, cargo movement, tanks (sloshing), that effect on structural mass M , stabilisation systems, appendages, that effect on damping N .

M means total mass of the ship and is matrix consisting of parameters; mass moments of inertia θ, mass m and its coordinates x_g and z_g . The ship is assumed to be symmetric longitudinally, meaning that $y=0$. The mass acts with square of encounter frequency ω_e^2 . (Bertram, 2012). The greater structural mass M is, the greater absolute of exciting force $\hat{\vec{F}}$ is. By minimizing parameters of M (θ, m , x_g and z_g) the M itself is minimized. This can be done by even mass distribution and weight optimization, also careful attachment of cargo is important. Sloshing of liquids in tanks can be reduced by reducing tank sizes.

Damping N is loss of energy and acts against the mass term and so the greater damping the smaller absolute of exciting force. The N matrix consist of effects of the ship induced waves. Damping is increased by appendages, such as parts of propulsion system, that are outside of the hull. (Cheirdaris, 2021) In Khione's case these are propellers, rudders and bow thrusters.

Hull Form

Hull form effects on added mass A , which is the liquid having same acceleration and phase as the ship. A matrix is calculated with sectional added mass coefficients a , that depends on sectional underwater geometry, ship speed U , encounter frequency ω_e and damping component b (Cheirdaris, 2021) (eq. 7-18). Damping matrix N is calculated similarly (Cheirdaris, 2021) (eq. 7-19).

Area, first moment and moment of inertia of the waterplane are parameters of restoring matrix S . Like damping also restoring force acts against mass term (structural mass and added mass). As restoring force is analogous to spring stiffness, so increasing to waterplane area to ship comes "stiffer". Stern transom, which Khione has, complicates the calculation of restoring force. (Cheirdaris, 2021).

Hull form also effects on excitation force $\hat{\vec{F}}$, which consists of Froude-Krilov force and diffraction force. Froude-Krilov force, also incident wave, is the pressure over the wetted surface of the ship, when the hull has no effect on waves. Diffraction force is the pressure caused by the presence of the ship disturbing the water. (Bertram, 2012).

Operational Profile

Operational profile tells how and where the ship operates. Parameters linked to operational profile are wave conditions and ship speed and velocity. Wave conditions are linked to excitation force $\hat{\vec{F}}$, via wave forces.

The heading of the ship effects on encounter frequency ω_e , which then is linked to mass and damping terms. Encounter frequency is highest with head seas and lowest with following seas. As mentioned in subchapter above, the ship speed U is used to calculate added mass and damping. Ship speed also effects on ship's vulnerability on parametric roll. (Bertram, 2012). Parametric roll is discussed more in paper reviews. With proper seakeeping, controlling the speed and course, dangerous conditions can be avoided.

3.3 Motions and loads in design software

For the coming motion and load calculations we will use NAPA as software since we use it in Ship Design Portfolio course. The motion calculations in NAPA the Seakeeping Manager tool can be used. The seakeeping manager consist of folders, which can be seen in Figure 68 for the different calculations where the needed input values are set. For the calculations either panel or strip theory can be used. The difference in these is that the strip theory does not include surge, whereas panel method includes all six degrees of motion. After defining the method, the steps for calculations are basically the same for either of the methods.

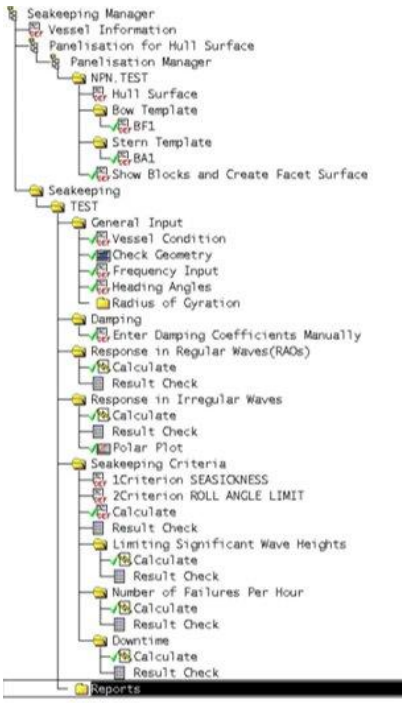


Figure 68. NAPA Seakeeping Manager steps and inputs.

For Khione the panelisation worked out by choosing similar templates for stern and bow, and choosing suitable amount of panels (Figure 69). In the General input-step the modelled hull is chosen and values for operating conditions set, for more exact values an earlier defined loading condition can be used, the loading conditions is used for the radius of gyration. The heading angle can be chosen with 30 and 45 degree step.

Response in regular waves calculates motion amplitudes. For irregular waves input values as water depth, sea spectrum etc. which has been defined in the previous chapter is needed. In the seakeeping criteria folder criteria for checking operational limits or to estimate downtime can be defined. At this point the seasickness and roll angle limit criteria has been set for our case but there are several alternatives that can be added.

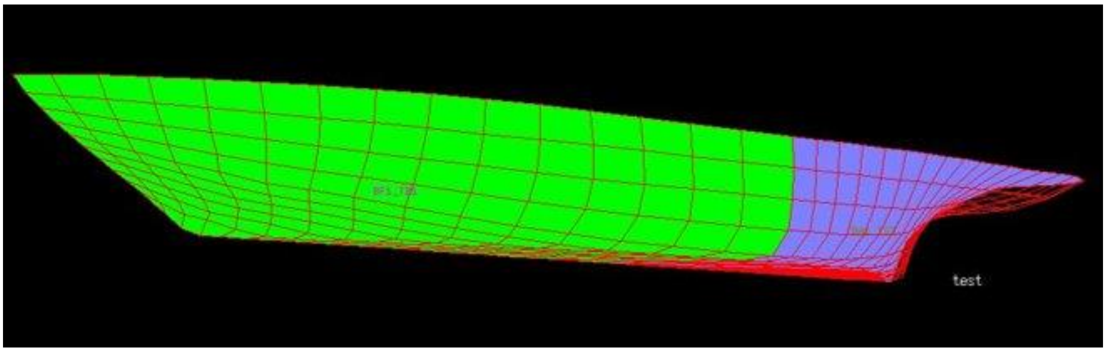


Figure 69. Panelisation of hull.

The results from the seakeeping manager can be checked during the go through of the manager and in the report afterwards consisting of all calculations, both as tables and graphs.

Calculation of loads, still water bending moment and vertical bending moment can be done by commands and with defined tasks in NAPA, by using input values as movement calculations and loading conditions which are defined in NAPA earlier. Loads in irregular sea state can be done by using macro.

4 Motions and loads

4.1 Seakeeping analysis

As written in previous chapter the software used for seakeeping analysis in our group is NAPA. The calculation for responses in regular and irregular waves for all 6 motions is done by using the built-in seakeeping manager in NAPA and loads are calculated with macros provided in NAPA workshops. Unfortunately, NAPA isn’t able to compute the motions and loads directly with required 3 hour maximums.

The seakeeping manager uses the hull model done In NAPA and references for geometry like length (Lpp=127,6m), breadth (B=21,96m), and draught (T=6,4m) directly from the hull model. For more accurate values an earlier defined loading condition for ballast condition is also used. To calculate the six degrees of motion the panel method is used for the wetted surface. To do the panelisation templates for bow and stern is chosen dependent on the design of the hull, bulb type etc.. The number of mesh is set to be 10 at left/right and 15 at top/bottom in bow part, and stern part to have 20 at bottom/top, which results to 350 panels in total, seen also in Figure 69.

The heading angles is calculated with step of 30 degrees (0, 30, 60, 90, 120, 150, 180) and speeds used 0, 8, 13 knots. As the design speed in open water is 13 knots for Khione. Determination of the coefficients for equation of motions is generated with the panels, where hydrodynamic pressures due to ship motions are calculated. These are added masses, damping and exciting forces. Equations of motions in regular waves are solved by matrix operations with the hydrodynamic quantities and matrices for ship real mass and restoring force. Despite ship motions, relative motions between two points can be needed. The procedure explained is then transformed to get results for irregular waves.

On top of this evaluation for seakeeping criteria can be done where several criteria can be set, for example seasickness and roll angle limit. These uses also set wave heights and given wave statistics.

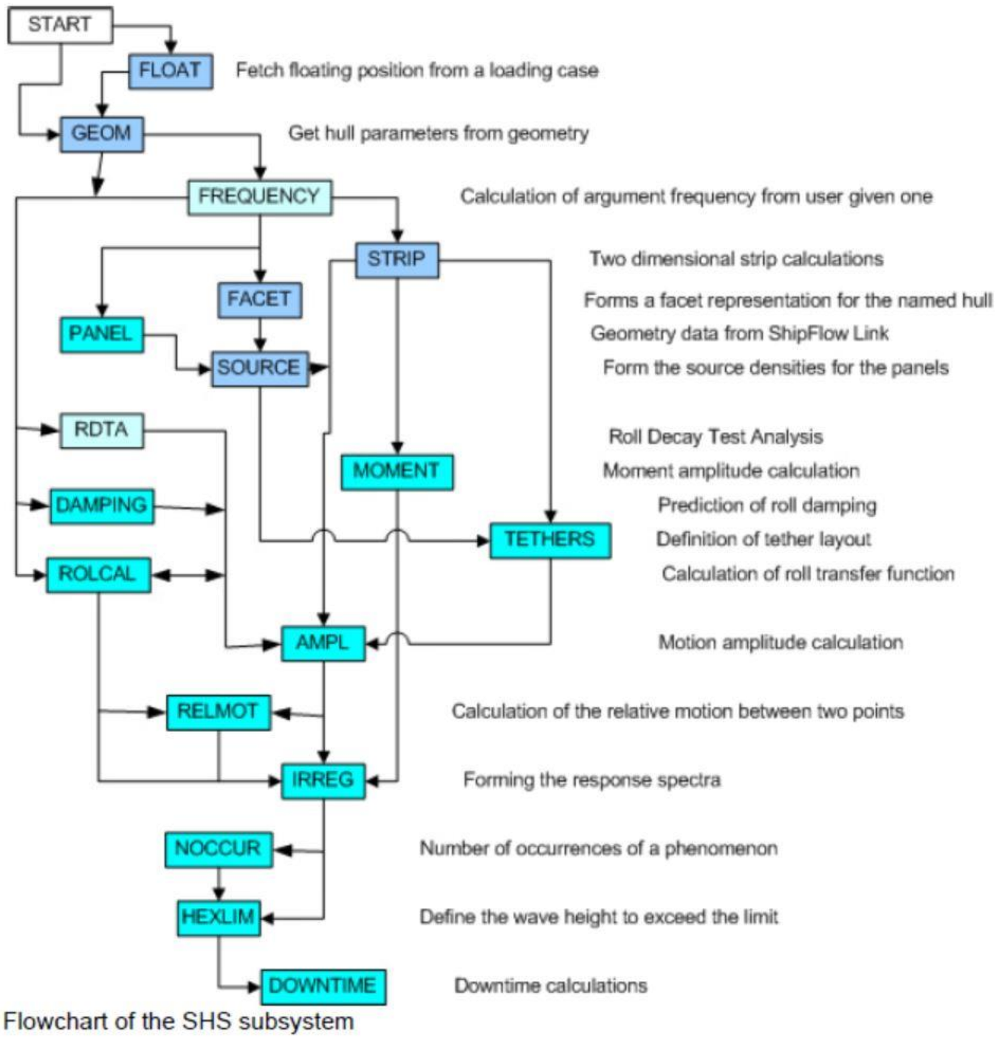


Figure 70. Flowchart of Seakeeping subsystem in NAPA (NAPA, 2020).

4.2 Downtime

Downtime is the time the ship is inoperational. Seasickness and roll angle limit curves are limit curves used in downtime calculations. Seasickness and limit roll angle curves are plotted in significant wave height to exceed the limit cause against zero crossing period on different speeds and headings. Then these curves are plotted on scatter diagram of seastates. The number of seastates above the limit curves is then the downtime. Figure 71 shows example of downtime plot on ship speed of 13 knots and heading of 90 degree. All cases are similar to the example in a way that the seasickness curve defines the total downtime. Table 33 shows the total downtime per cents for all speed and headings. The seasickness curve tends to be low after 3 seconds on zero speed leading to extremely high downtimes. The ship has the better downtime the higher ship speed is and the lower heading degree is. Heading seas cause extremely high downtimes on every calculated speed. This could be cause large motions such as slamming.

Table 33 Total downtime per cents for all speed and headings

Total Downtime			
deg	**0 knots**	**8 knots**	**13 knots**
0	97.5%	7.9%	0.3%
30	98.4%	19.4%	1.2%
60	99.6%	94.4%	31.2%
90	99.6%	86.0%	30.7%
120	99.9%	98.6%	96.8%
150	99.1%	97.6%	93.5%
180	99.2%	97.2%	91.8%

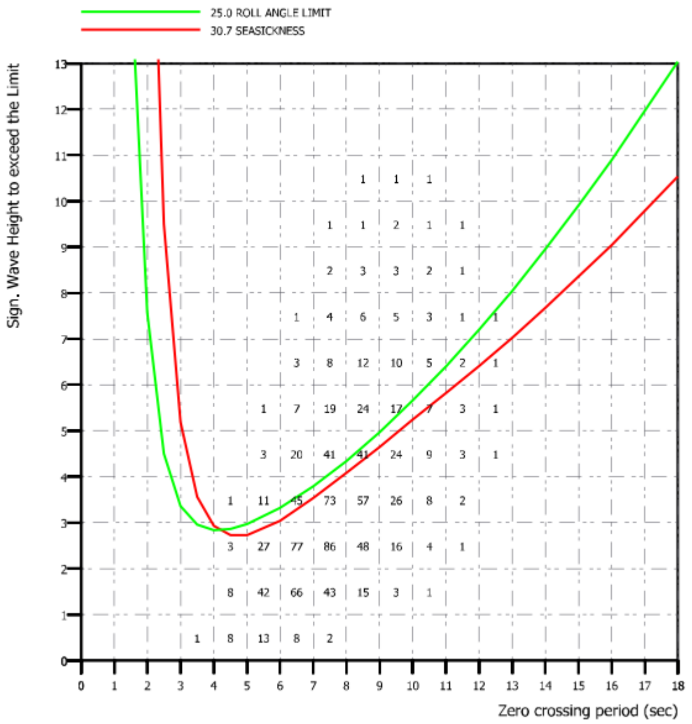


Figure 71 Downtime plot on ship speed of 13 knots and heading of 90 degree

4.3 Response amplitude operators

Response in regular waves

RAO graphs calculated for all six degrees of freedom are shown in Figure 72. In graphs Y-axis is shows response function in m/m or deg/m for transfers and rotations, respectively, and x-axis is non-dimensional wave length ($\sqrt{L_{pp}/\lambda}$). Input for regular wave analysis is presented in Table 34.

From graphs in Figure 72 it can be seen that in most cases largest response occurs in 90- or 120-degree headings and maximum responses range between 0.7 and 2.3, with exception of sway response which has its maximums, when non-dimensional wave length is low, meaning relatively long wave lengths. Short wave lengths don't cause response motions. Also, headings of 60 and 30 degree causes high responses on pitch, surge and yaw motion. Surge motion has especially high peak in 13 knot speed and following seas (0 degree heading), this could be caused by surf riding. Head seas (0 degree heading) cause only heave, pitch and surge motions, that may be caused by slamming. Beam seas (90 degree heading) cause highest rolling motions, which seems reasonable.

Coupling can be seen especially well between heave and pitch motion; motion peaks occur on same non-dimensional wave length and heading. Coupling is influence of one motion to other motions. Also, first peaks of roll motion on ship speed 13 and headings 30 and 120 degree occur on same non-dimensional wave length than peaks of pitch motion.

With higher speeds motions develop second peaks, that can be lower or higher than the first peak depending on the heading and motion in question. Beam seas is exception and does have only one peak for each motion.

Table 34 Inputs for regular wave analysis. Columns are non-dimensional wave length $\sqrt{L_{pp}/\lambda}$, wave length, L_{pp}/λ, λ/L_{pp}, wave period and frequency in rad/s and 1/s.

SqrLPLa	WLen	LPLA	LaLp	WPer	Omega	Hz
	m			sec	rad/s	1/s
0.1500	5669.5	0.023	44.444	60.260	0.1043	0.0166
0.2000	3189.1	0.040	25.000	45.195	0.1390	0.0221
0.2500	2041.0	0.063	16.000	36.156	0.1738	0.0277
0.3000	1417.4	0.090	11.111	30.130	0.2085	0.0332
0.4000	797.3	0.160	6.250	22.597	0.2780	0.0443
0.4500	629.9	0.203	4.938	20.087	0.3128	0.0498
0.5000	510.3	0.250	4.000	18.078	0.3476	0.0553
0.5500	421.7	0.303	3.306	16.435	0.3823	0.0608
0.6000	354.3	0.360	2.778	15.065	0.4171	0.0664
0.6500	301.9	0.423	2.367	13.906	0.4518	0.0719
0.7000	260.3	0.490	2.041	12.913	0.4866	0.0774
0.8000	199.3	0.640	1.562	11.299	0.5561	0.0885
0.9000	157.5	0.810	1.235	10.043	0.6256	0.0996
1.0000	127.6	1.000	1.000	9.039	0.6951	0.1106
1.1000	105.4	1.210	0.826	8.217	0.7646	0.1217
1.2000	88.6	1.440	0.694	7.532	0.8341	0.1328
1.3000	75.5	1.690	0.592	6.953	0.9037	0.1438
1.5000	56.7	2.250	0.444	6.026	1.0427	0.1659
1.8000	39.4	3.240	0.309	5.022	1.2512	0.1991
2.2000	26.4	4.840	0.207	4.109	1.5293	0.2434
2.7000	17.5	7.290	0.137	3.348	1.8768	0.2987
5.5000	4.2	30.250	0.033	1.643	3.8232	0.6085

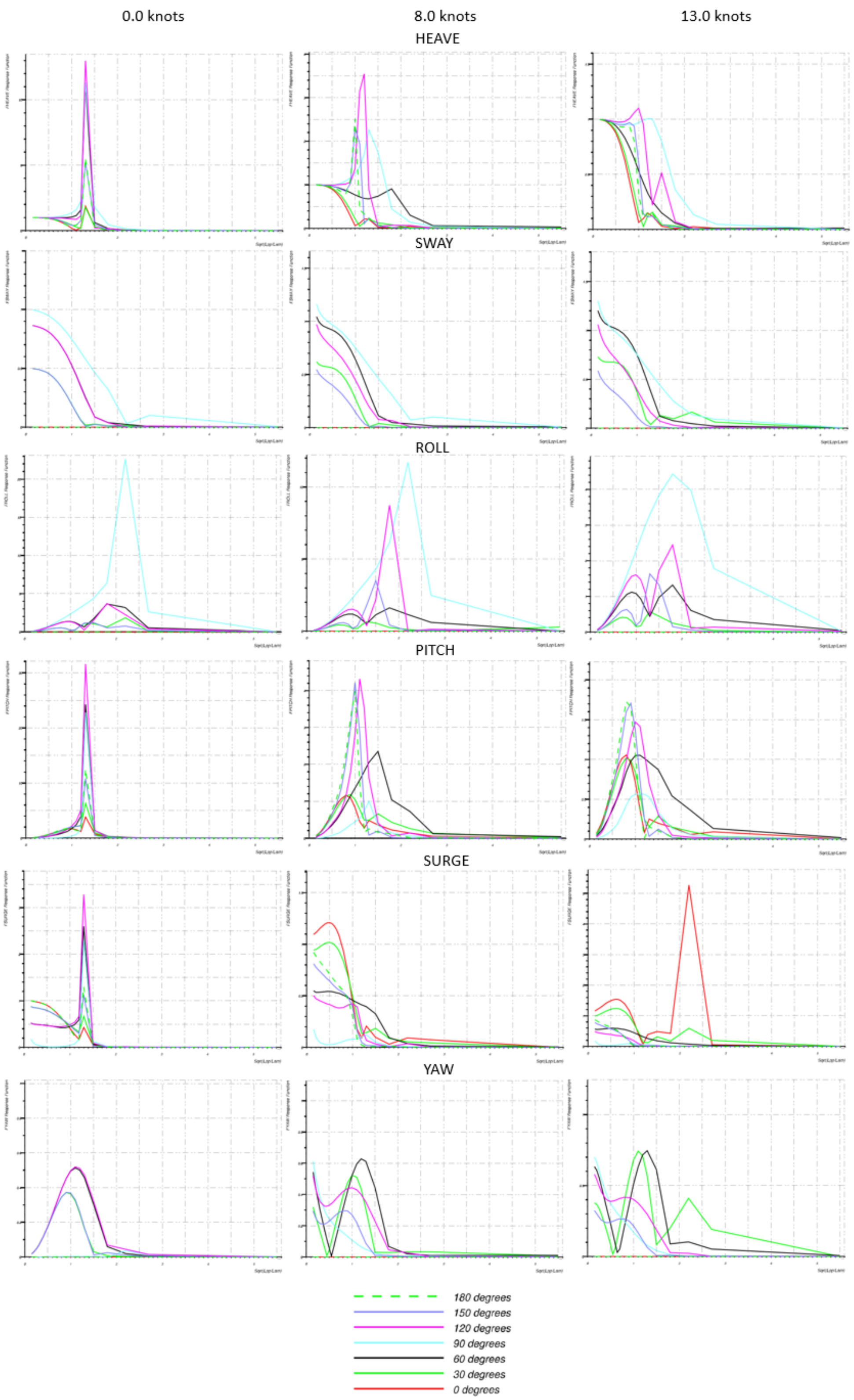


Figure 72 Response functions for regular waves for ship speeds 0, 8 and 13 knots

Response in irregular waves

In calculation of responses in irregular waves input values for significant wave height, zero-crossing period TZ, wave spectrum and spreading function is set. The significant wave height used in this calculation is 7m, as it's stated as extreme wave height in our earlier chapter, and as wave spectrum the modified Pierson-Moskowitz is used. Response of irregular ways is shown in Figure 73. The response function is in m or deg for transfers and rotations, respectively, and x-axis is zero crossing period. Response functions approach non zero values with large zero crossing periods, except head and following seas are zero in sway, roll and yaw motions.

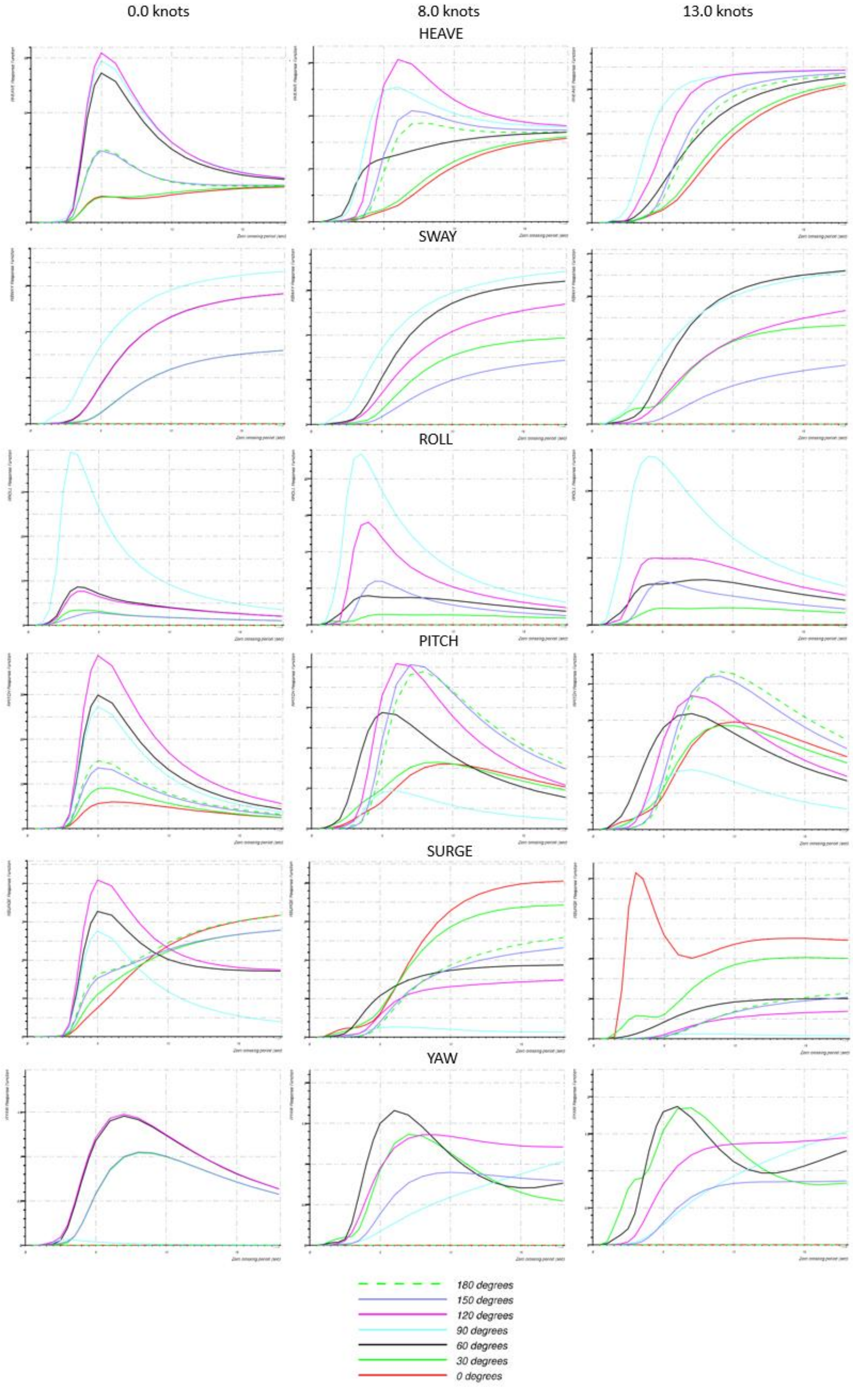


Figure 73 Response functions for irregular waves for ship speeds 0, 8 and 13 knots calculated with panel method

From Figure 73 maximum values of ship motions for ship speed 0, 8 and 13 knots are compiled to Table 35, which also shows which speed results largest motion with bolding. Similar table is done also for strip method in Table 36. Calculations were carried out using NAPA.

Table 35 Maximum responses of each motion calculated with panel method

Motion	0 knots	8 knots	13 knots
Heave	**3.5 m (90°)**	3.5 m (90°)	3.5 m (90°)
Sway	**3.4 m (90°)**	3.4 m (90°)	3.3 m (90°)
Roll	**38.8 ° (90°)**	24.8 ° (90°)	16.2 ° (90°)
Pitch	4.3 (150°)	4.7 ° (180°)	**4.7 ° (180°)**
Surge	0	0	0
Yaw	1.8 ° (120°)	2.3 ° (60°)	**2.9 ° (60°)**

Table 36 Maximum responses of each motion calculated with strip method

Motion	0 knots	8 knots	13 knots
Heave	**15.5 m (120°)**	6.1 m (120°)	3.4 m (120°)
Sway	3.3 m (90°)	3.4 m (90°)	**3.6 m (60°)**
Roll	**38.9 ° (90°)**	23.4 ° (90°)	12.6 ° (90°)
Pitch	**38.7 ° (120°)**	8.2 ° (120°)	4.3 ° (180°)
Surge	4.1 m (120°)	4.1 m (0°)	**8.3 m (0°)**
Yaw	1.5 ° (120°)	1.7 ° (60°)	**1.9 ° (60°)**

4.4 Global loads

The global loads are calculated with macros in NAPA for chosen speeds and headings, using predefined loading condition. The first one calculates vertical shear force and vertical bending moment in regular 1 meter waves. From the headings calculated for the speeds used in response amplitude operator analysis the largest bending moment and shear force was evaluated with the speed of 13 knots and heading 180 degrees, head seas. The results for this are presented in table and figures below, where the FVMOM is the vertical bending moment, FVSHE vertical shear force and SQRLPLA is Lpp divided by wavelength.

Table 37. Vertical bending moment and vertical shear force, 13 knots and heading 180.

SQRLPLA	FVMOM	FVSHE
	Nm	N
0,15	358	10898
0,2	597	16617
0,25	846	20688
0,3	1059	22399
0,4	1154	47415
0,45	922	91197
0,5	441	161309
0,55	456	262137
0,6	1605	398537
0,65	3227	575873
0,7	5563	801038

0,8	14824	1430856
0,9	32938	2265505
1	46677	2653512
1,1	44405	2348098
1,2	33357	1880255
1,3	17482	1147388
1,5	21147	603564
1,8	9825	244790
2,2	9256	148859
2,7	8458	357590
5,5	12841	592064

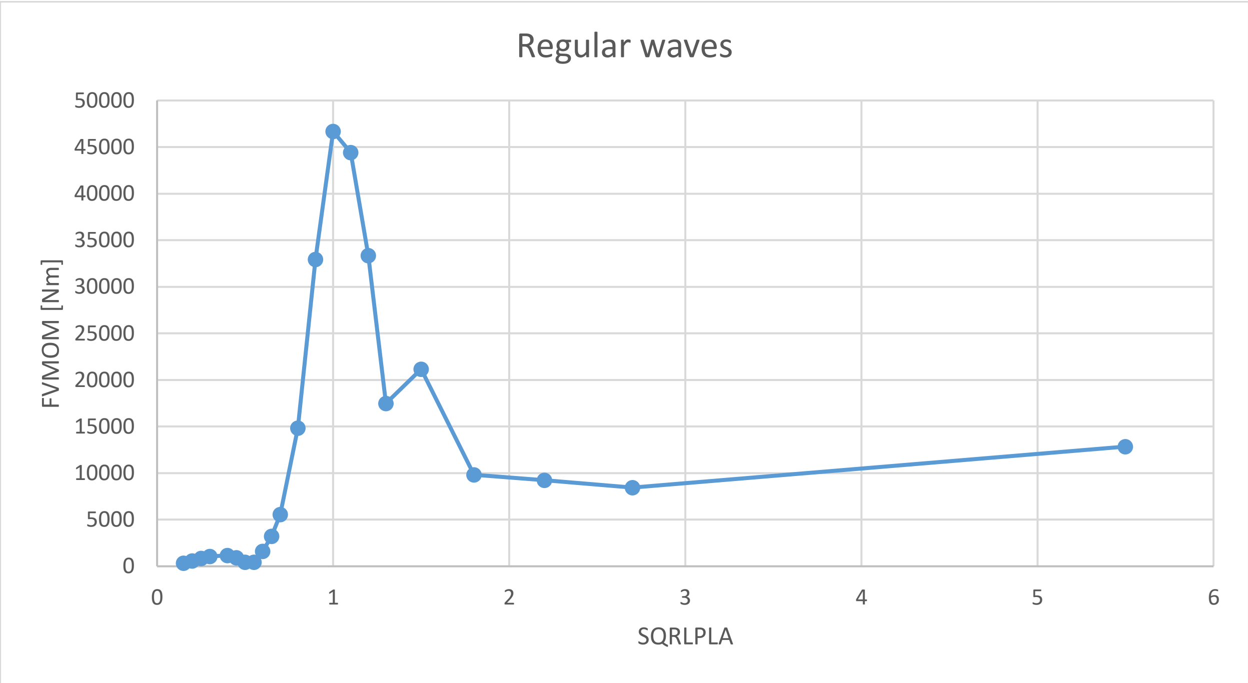


Figure 74. Vertical bending moment at speed 13 knots and heading 180.

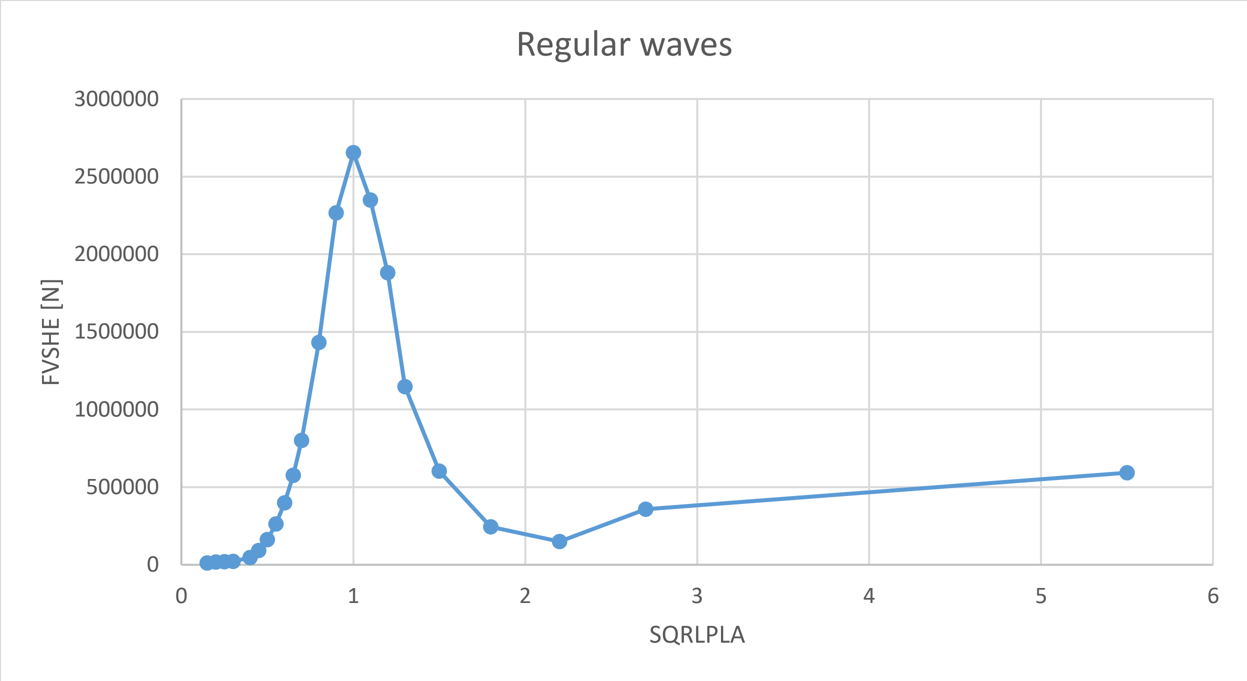


Figure 75. Vertical shear force at speed 13 knots and heading 180.

As seen in the figures the maximums of the vertical bending moment and vertical shear force is where the ship length and wavelength are equally large, which sound reasonable.

Vertical bending moment curve (Figure 74) has similar shape to vertical bending moment curve from article by Temarel et al., which also states that computational methods can successfully compute wave-induced loads. Yet there are still some challenges. (Temarel, et al., 2016). Both plots have first higher peak and second lower peak. It should be noted that x-axis in our plot and plot from literature are different but as shown in Table 34 we also have frequencies corresponding non-dimensional wave lengths. The article of Temarel et al. handles different loads connected to ship motions and their calculation methods. Accuracy of current calculation methods is in good state and challenges linked to computation methods are example linked to headings and numerical simulations. Loads handled are for example green water, sloshing, slamming and fatigue. (Temarel, et al., 2016).

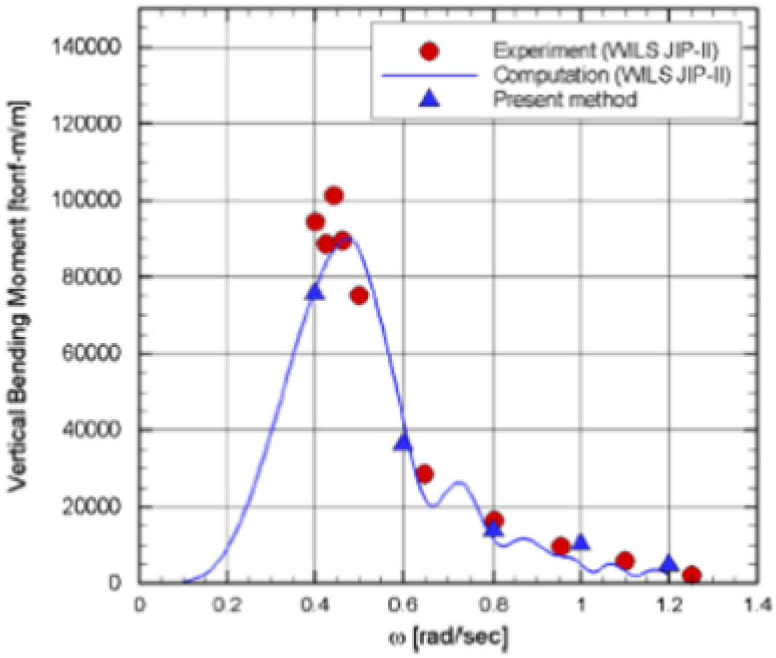


Fig. 2. Comparison of amidships VBM predictions and measurements for a 10,000 TEU containership traveling at 5 knots in regular head waves (Kim et al. 2013a).

Figure 76 plot from (Temarel, et al., 2016)

Kukkanen and Matusiak present non-linear panel method in their paper (Kukkanen & Matusiak, 2014). The method is used to calculate loads in head seas. The method is validated with model tests showing good accuracy. Figure 77 shows how maximum bending moment of the model of Kukkanen and Matusiak is reached at about 0.6 rad/s. Figure 74 shows our calculation of vertical bending moment, which reaches its maximum at 1 SQRPLA, which corresponds 0.70 rad/s (Table 34) so our results seem reasonable.

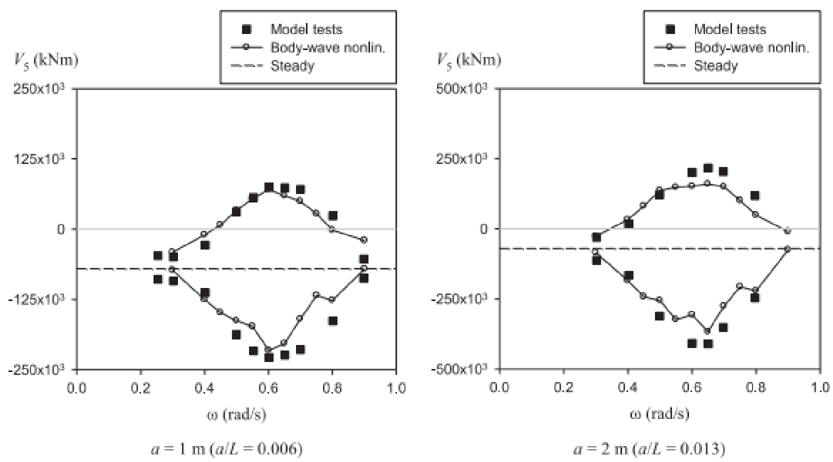


Figure 77 maximum and minimum bending moment peaks in regular waves (Kukkanen & Matusiak, 2014)

The second macro is able to calculate the significant wave bending moment in irregular waves. The macro is set to 1m waves as the bending moment is linear the results can be multiplied with the wave height wanted. In our case 7 m. As the heading of 180 degrees with speed of 13 knots was the most critical in bending and shear forces it was used to this aswell. The results are presented in the table below for the zero-crossing period TZ of 4,5-18 seconds, RVMOM as significant wave bending moment first for 1m and secondly for 7 m significant wave height.

Table 38. Significant wave bending moment at wave height 1 m and 7 m, speed 13 knots and heading 180.

TZ	RVMOM 1m	RVMOM 7m
sec	MNm	MNm
4,5	9,6	67,5
5	11,9	83,6
6	14,6	102,3
7	14,6	102,0
8	13,2	92,2
9	11,5	80,2
10	9,8	68,9
11	8,5	59,2
12	7,3	51,2
13	6,4	44,5
14	5,6	39,0
15	4,9	34,5
16	4,4	30,7
18	3,5	24,8

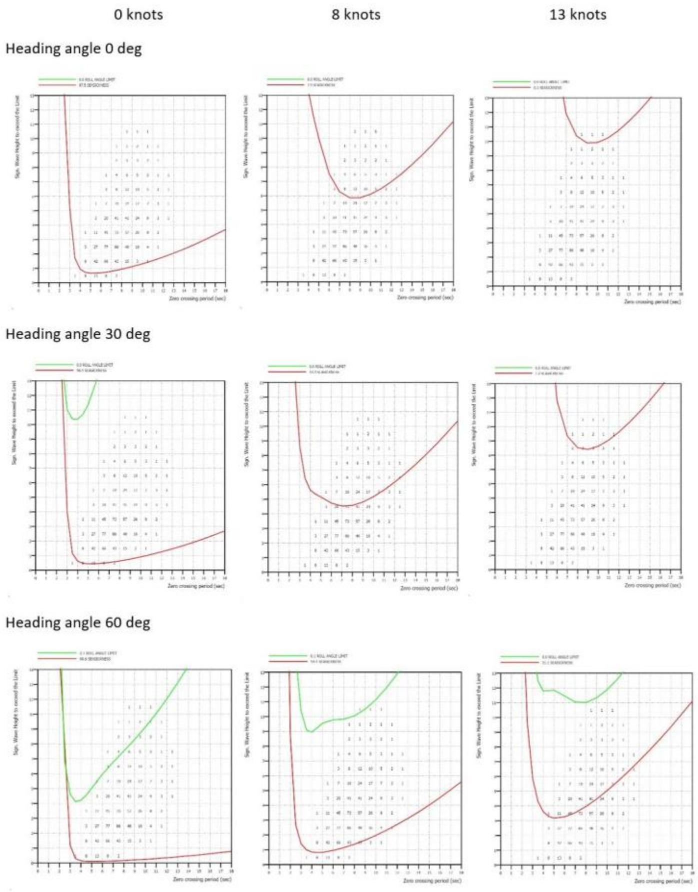
The highest wave bending moment for the given zero-crossing period at 6s with result of 102,3 MNm at 7 m wave height.

5 Seakeeping, Added Resistance and Manoeuvring

5.1 Seakeeping Criteria

Seakeeping criteria is assessed with the Seakeeping manager in NAPA used and discussed in chapter Seakeeping analysis. Seakeeping criteria is used to understand behavior of the ship in waves. Common criteria are slamming, deck wetness, seasickness, exceedance of transfer function values (for example acceleration) and flare slamming. These can be questioned by when the criterion is exceeded and how often exceedance is occurred. (NAPA, 2020)

Seakeeping criteria computed for Khione is seasickness and roll angle limit. For seasickness an index, MSI, describing the percentage of vomiting experienced by persons onboard is defined. It’s defined by a function including vertical acceleration, excitation frequency and exposure duration. Single amplitudes for motions can be used as limits, for roll angle a limit of 8 degrees is used as recommendation for accustomed naval personnel. (NAPA, 2020) The results for these are resented below, plotted with the adjusted sea state values from global wave statistics for our operation area.



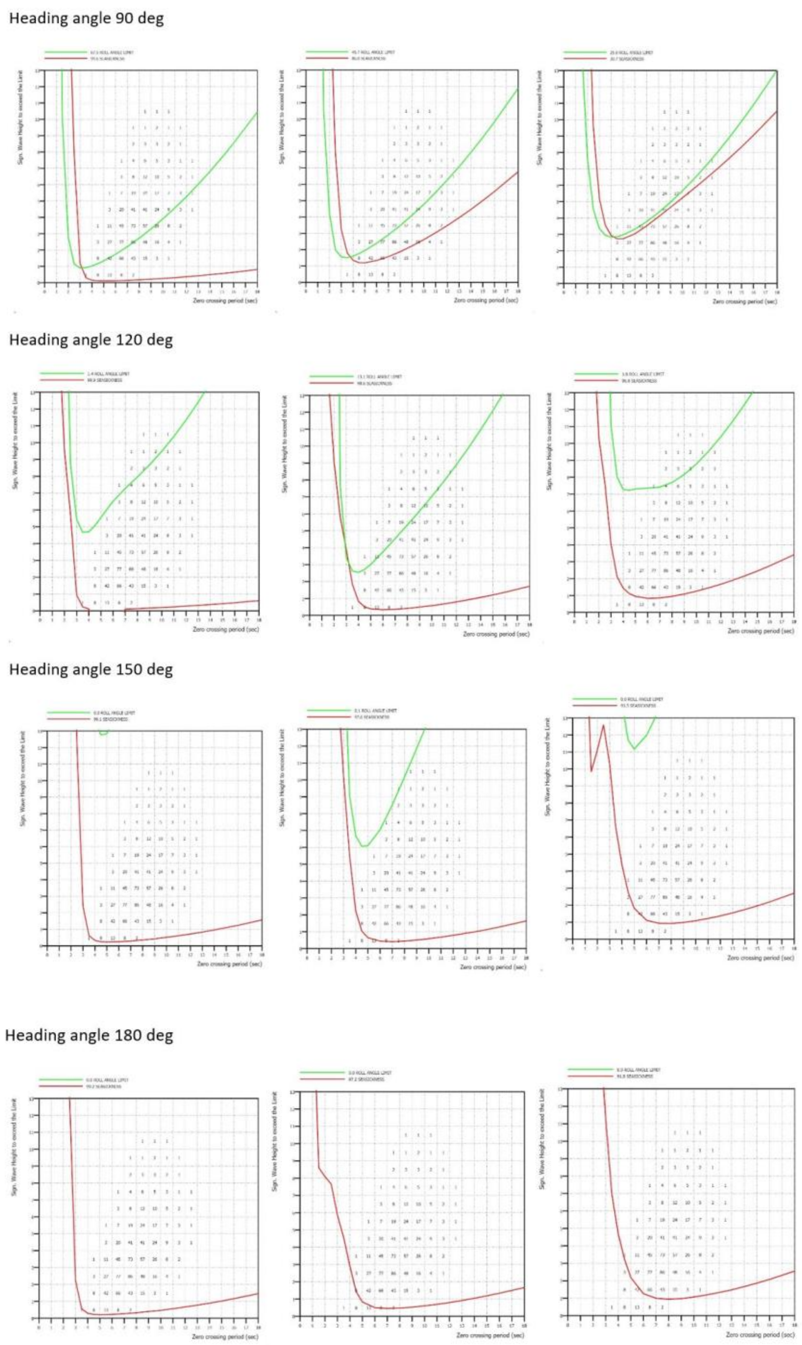


Figure 78. Seasickness (red) and roll angle limit (green) curves

The downtime due to seasickness and downtime is presented and discussed in chapter Downtime.

The accuracy of these results are dependent on the Seakeeping Manager in NAPA. How the Seakeeping Manager works and how we have implemented it has been described previously and therefore will not be gone over here. Based on previous material in this project such as (Mansour & Liu, 2008) and (Matusiak, 2011), we feel that the panel method used in the Seakeeping Manager provides sufficient accuracy for the design stage the project is in. This however does not mean that our results are perfect and without need for further iteration. As with most of our results further investigation is needed if the design is going to be taken further. Because of the investment that would be needed in these further investigations, they would nor be practical at this stage and the results we have obtained are within a reasonable margin of error.

5.2 Added resistance

The added resistance is assessed with the Seakeeping manager in NAPA, as used to the for the motions. The calculations for added resistance in NAPA uses the strip theory and a reflection coefficient method which applies corrections to short wavelengths. Calculating added resistance with strip-theory assumes that it is related, integrated over the ship's length, to the product of the sectional damping and the vertical velocity squared. (NAPA, 2020) The input values are the same as used for the computation of motions and presented in chapter Seakeeping analysis.

NAPA states that the results yields to 15 – 25% of model test values in the motion regime. For discouraging results for short waves a reflection of a cylindrical wall is added so that the results will apply for the whole wavelength. The added resistance presented is for irregular seas with the significant wave height of 7m and for more concrete comparison to open water resistance also with wave height of 1m. These are computed for head seas, 180 deg, for the same speeds used in the seakeeping manager for motions 0 knots, 8 knots and 13 knots (cruising speed). For added resistance calculations in irregular seas the error is approximated to be between 15-30% (NAPA, 2020).

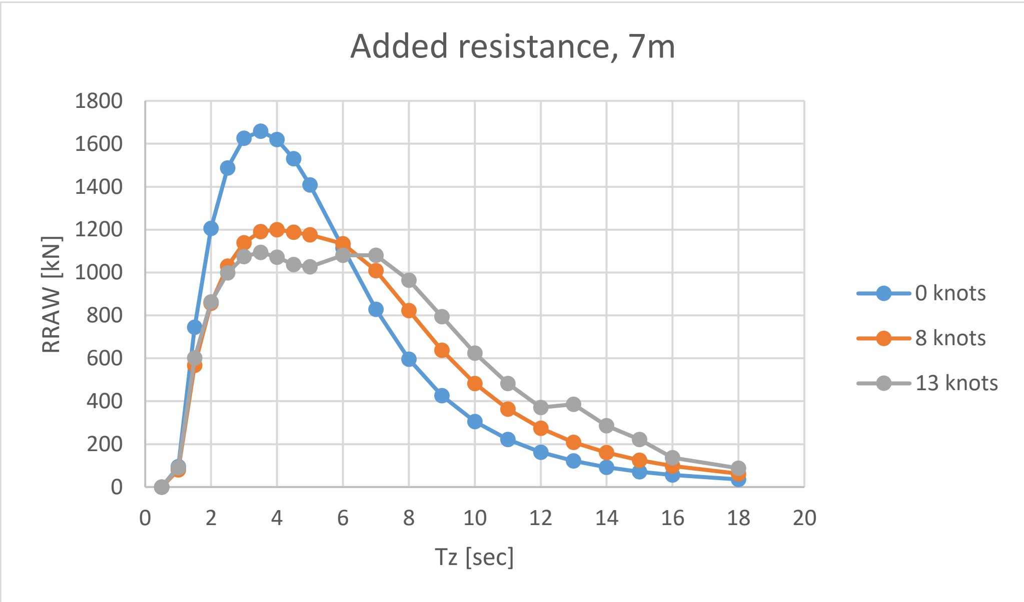


Figure 79. Added resistance for significant wave height 7m

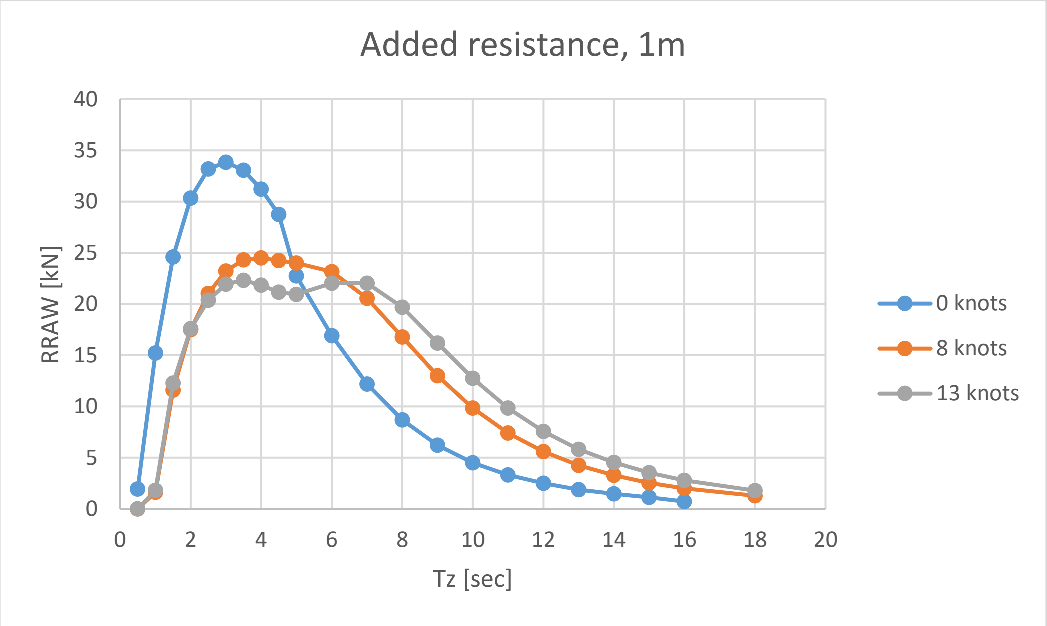


Figure 80. Added resistance for wave height 1m.

The significant wave height used is 7m, as in earlier computations and 1m. The figures above presents the added resistance for the three speeds. The maximum values are presented in Table 39.

Table 39. Maximum values for added resistance calculations.

Wave height	0 knots	8 knots	13 knots
1m	34 kN	24 kN	22 kN
7m	1660 kN	1200 kN	1090 kN

It can be seen that the resistance is highest at the speed of 0 knots and decreases with the speed. the maximum values are obtained approximately at the zero-crossing period of 3-4 s. The stillwater resistance for Khione at cruising speed is 210 kN and ice resistance in 1,65m thick ice with speed 3 knots is 2800kN (obtained in Ship design portfolio course). Comparing the values of added resistance from waves the results seem reasonable. With the wave height of 1m the total resistance at cruising speed would be 232kN, when the added resistance due to waves is 9%. For the extreme conditions with wave height 7m, the total resistance results to 1300kN, where the added resistance takes up significantly larger part.

As previously mentioned, NAPA gives an estimate of the accuracy of the results of 15-30% in irregular waves and we have no reason to doubt this estimation. Using a panel method and potential flow theory has advantages especially in how fast results can be obtained. With these advantages there are also disadvantages in the accuracy of the results. As is pointed out in (Liu & Papanikolau, 2016) potential flow theory suffers from irregular frequency issue, neglect of viscous effects and neglect of the hull form above the calm water line. There are also problems with using the method with short waves, but as was mentioned earlier some corrections have been made in NAPA to address this. How well the corrections work we cannot determine, since we only have an estimate for the total error.

With these possible sources of error pointed out we feel that our obtained values are fit for purpose. This is a first estimate of added resistance sufficient for this stage of the design process. Further iteration would require more detailed investigations using other methods, for instance model tests or CFD modelling. This however would mean a significant investment in time and resources that at least in our opinion lay outside of the scope of this course. We can see no obvious errors in our results when we compare the shape of the graphs we have obtained and those presented in (Liu & Papanikolau, 2016).

5.3 Maneuvering tests

Manoeuvring tests are performed with the Manoeuvring manager in NAPA. The manoeuvring manager is built similarly as the manager for seakeeping with folders for input values and computations.

Input values for the manoeuvring calculations are; hull data (directly from the hull model), data for roll, quantity, dimensions and locations of rudders, propellers, and tunnel thrusters. Wind data, current forces and wave drift forces are as input values if they are needed for the simulations. (NAPA, 2020)

The coordinate system for the manoeuvring part in NAPA follows JTTC (Japan Towing Tank Conference) standard where positive turning direction is resulted with positive rudder angle. The results for velocities and locations are fixed to the centre of gravity of the ship. (NAPA, 2020)

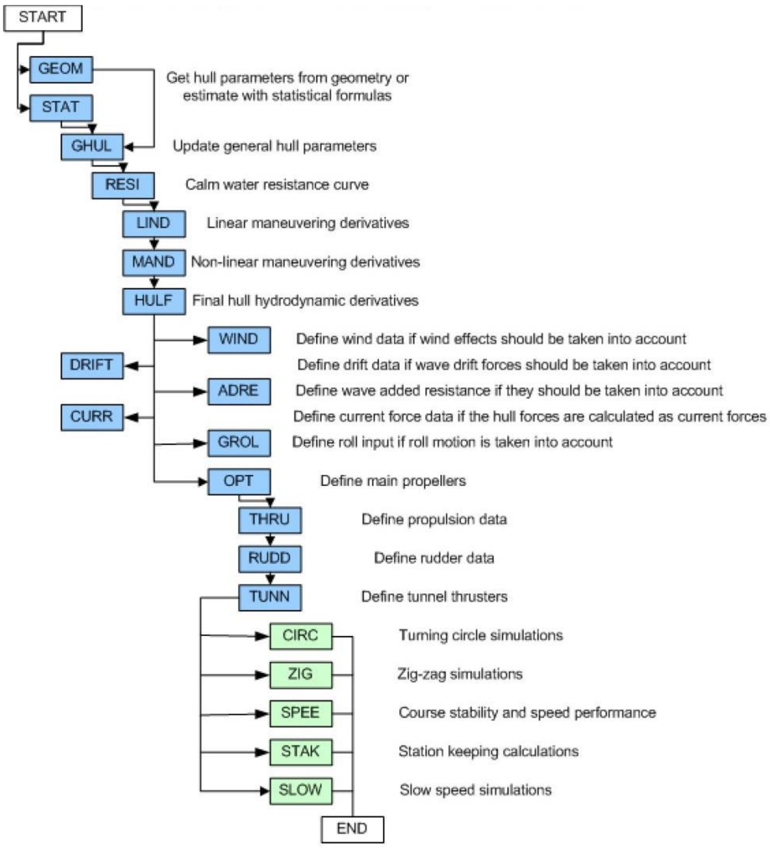


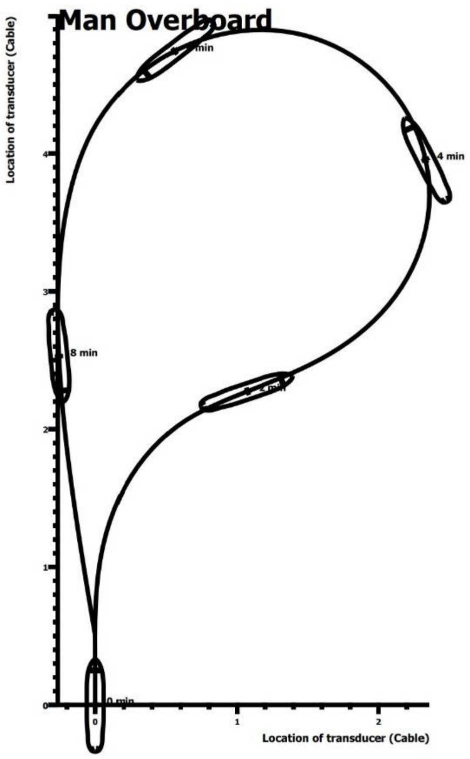
Figure 81. Flowchart for Manoeuvring calculations (NAPA, 2020).

The hull data is in our case directly from the hull model. Method used in NAPA for linear manoeuvring coefficients is Clarke. Main propellers, 2 fixed pitch propellers, have a diameter of 5 m and 4 blades. Used maximum shaft power is 1,4 MW as it's the power demand for cruising speed in open water, NAPA has an restriction of 2 MW for shaft power which denies the use of our total installed power and the manoeuvring simulations is still done in open water. Input values for rudders are defined with Ogawa switch, rudder area 16,33 m^2 and rudder height 4,48 m. The input values for tunnel thruster are diameter 2 m and power 1,45 MW, as defined in chapter Maneuvering devices. Location coordinates for propellers, rudders and thruster are obtained as drawn in Figure 57.

Manoeuvring tests performed is man overboard, turning circle and emergency stop characteristics. These are recommended by several organizations and are included in IMO A601 test (ITTC, 2017). They are also chosen since they are included in the manoeuvring manager used.

Man overboard

The man overboard tests are done for providing information of the time it takes to manoeuvre back to point where a person or object has gone overboard. There are several turns wherefrom elliptical turning manoeuvre and Williamson turning manoeuvre is common. These have differences in how the manoeuvre is done considering rudder angles. Parameters obtained from this test is plot of the track, time taken to reach the starting point and lateral deviation. (ITTC, 2017)



	TIME [sec]	PPOR [%]	RDOR [deg]
1	10.00	80 80	0 0
2	80.00	80 80	60 60
3	220.00	80 80	−35 −35
4	90.00	80 80	−25 −25
5	160.00	80 80	1 1

Figure 82. Man overboard manoeuvre and simulation inputs

Simulation inputs for the man overboard manoeuvre are ships initial speed (13 knots), time interval for saving (4 sec) and time interval for ship figure plot (120 sec). The simulation inputs for the manoeuvre are shown in Figure 82. Man overboard manoeuvre and simulation inputs Figure 82, where each rudder angle (RDOR) is set for a suitable time to result the manoeuvre. The total time for the simulated man overboard manoeuvre, to return to the starting point, is 9 min 20 sec.

Turning circle

In full scale manoeuvring trials, the turning circle tests are performed with approach speed with maximum rudder angle both to starboard and port sides. The turning circle should be at least 540 degrees to determine necessary main parameters. (ITTC, 2017)

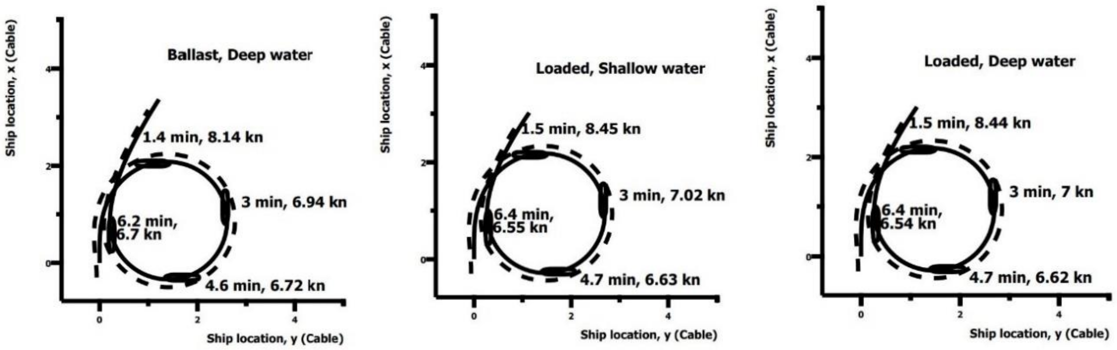


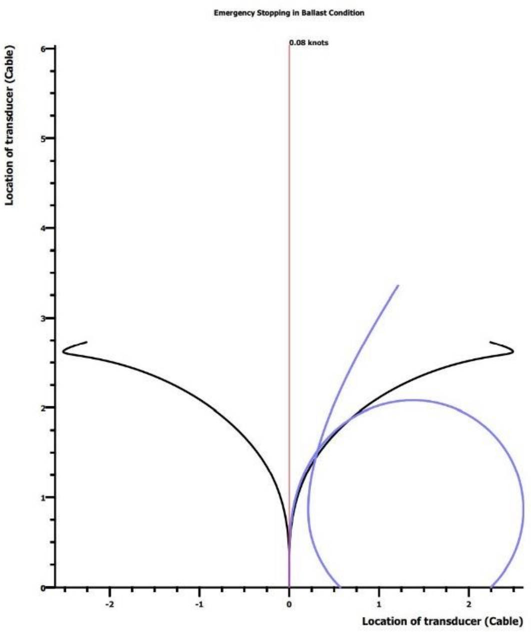
Figure 83. Turning circles in ballast and loaded conditions

The turning circle is assessed for ballast condition in deep water and for loaded condition in deep and shallow water. The input values for deep water turning circles are the earlier defined loading conditions and water depth of 5000 m (used as our depth in motion calculations). For the shallow water the water depth is 1000 m. As seen in Figure 83 the radius for the three turning circle simulations is of similar size. The times for the circles; ballast condition in deep water is 6,2 min (last speed 6,7 knots), loaded condition in shallow water 6,4 min (last speed 6,55 knots) and loaded condition in deep water 6,4 min (last speed 6,54 knots). The tactical diameter in all cases are near 2,5 cable lengths.

Emergency stopping

Stopping test ar in full scale trials usually done by starting with full ahead speed by applying full astern power. The test is completed when the propelling unit has reached full rpm for astern and the speed is down to 0 knots. Parameters necessary after the test is head reach (distance travelled in direction of the initial course), length of the track and lateral deviation (distance normal to the initial course). (ITTC, 2017)

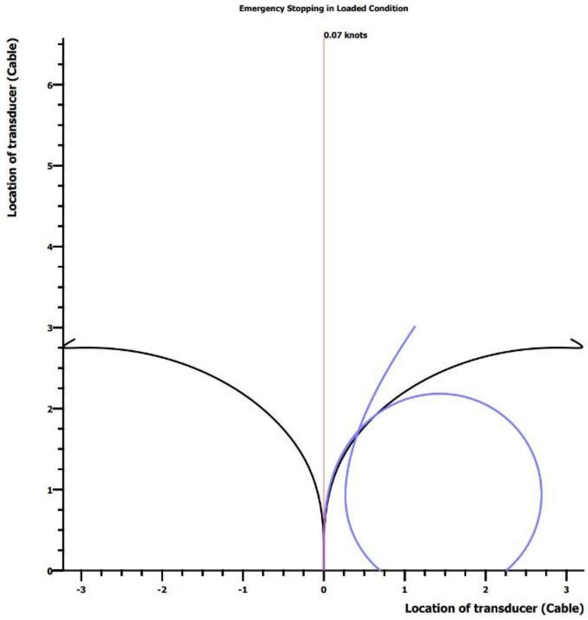
The emergency stopping manoeuvre simulates stopping and turning manoeuvre for zero rudder angle to port and starboard side. The simulation is done by adjusting the last defined time until the speed reaches 0 knots. The manoeuvre for stopping is plotted in the figures below for ballast and loaded condition, with the corresponding turning circles aswell.



	TIME [sec]	PPOR [%]
1	20.00	40 40
2	20.00	0 0
3	30.00	-30 -30
4	30.00	-60 -60
5	210.00	-70 -70

Figure 84. Emergency stopping in ballast condition

To reach the speed of 0 knots in ballas condition the time results to 5 min 10 sec. The head reach results to approximately 2,6 cable lengths and lateral deviation 2,5 cable lengths.



	TIME [sec]	PPOR [%]
1	20.00	40 40
2	20.00	0 0
3	30.00	-30 -30
4	30.00	-60 -60
5	255.00	-70 -70

Figure 85. Emergency stopping in loaded condition

To reach the speed of 0 knots in ballas condition the time results to 5 min 55 sec. Both the head reach and lateral deviation in loading condition is larger than compared to the ballast condition. The results is 2,75 cable lengths for head reach and 3,25 cable lengths to lateral deviation.

5.4 Accuracy of results

The accuracy of our results depends on our inputs and the modelling method in NAPA's Manoeuvring Manager. The inputs we have used are the best available to us. The hull form is taken directly from NAPA, the roll characteristics are the best available to us and the manoeuvring devises are inputted as planned. The linear manoeuvring coefficients are not specific to our design but reputable because of their inclusion in (Anon., 2012) . Based on a holistic overview of the methods for testing manoeuvring available to us, we feel that the results are the best we can achieve and sufficient for the design iteration of our ship. Further investigation is needed in following iterations, but the current results are satisfactory for now.

5.5 Following Improvements

In addition to project courses there are many useful courses that help studying of this course. Some of those our group members have already taken. These courses are ship systems, ship hydrodynamics, Random loads and processes, that helped in deciding steering systems, understanding themes of this course and understanding stochastic nature of sea waves, respectively. These and other courses, such as winter navigation, also helped in other design aspects of our project ship.

Three major themes of improvements to Khione's design that are linked to themes of this course are study of stabilizers, hull form iteration and detailed propeller study. Courses about fluid dynamics, such as Computational Fluid Dynamics, Computational Fluid Modelling, Computational Marine Hydrodynamics, Fluid Dynamics, further understanding of improvement themes. We have talked about stabilizers and next step would be analyze their effect and choose best option for Khione. For iteration of the hull form, simulations should be carried out changing different parameters of the hull. Also, study and optimization of propeller requires simulations.

In this course we have calculated loads that ship structure need to withstand so structures are indirectly linked to this course. There are many structure courses for improving structures of our project ship, such as Dynamics of Structures and Thin-walled Structures.

6 Book and paper reflections

This chapter concludes the book and paper reflections done in the Ship Dynamics assignments.

Paper - On the Parametric Rolling of Container Vessels. Silva, S., Guedes Soares, C. 2010

Paper describes features of the ship that can cause parametric roll and focuses studying parametric roll of container ships. Parametric roll is stability failure mode, that causes roll motion of the ship and is effect of righting arm stability variation (Figure 86). Our project ship isn't container ship, but the results are still useful, when studying vulnerability of our project ship to parametric roll. Hull form features that lead to change of wetted area of the hull, when waves travel along the ship, make the ship vulnerable to parametric roll. Flat transom stern is such feature and our project ship has that kind of stern shape. Ship and wave angle lengths are also important factor in occurrence of parametric roll. Wave length needs to be larger than 0.8 * L_{BP} and smaller than 2 * L_{BP} to parametric roll to occur. Parametric roll can occur in head and following seas and also small oblique heading angles. As the ships speed and wave frequency affects occurrence of the parametric roll, several conditions with different ship speeds and wave frequencies can lead to parametric roll. Classifications societies have their own guidance's on avoiding parametric roll. (Silva, 2010)

Parametric roll is mainly problem for large vessels, such as large container vessels. Parametric roll can still occur on smaller ships, such as our project ship which is 136 m long, when conditions (wave length, ship speed and heading) are right. Parametric roll can lead to high accelerations, which can cause of damage of personnel, cargo and laboratory and ship equipment. (Silva, 2010)

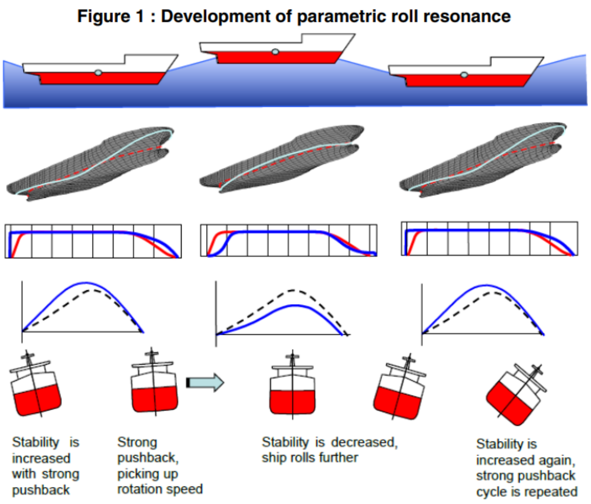


Figure 86. Parametric resonance rolling (Bureau Veritas, 2019)

Paper - Wave loads and flexible fluid-structure interactions: current developments and future directions. S.E. Hirdaris et al. 2010.

The paper describes the work done and being done by Lloyd's Register to improve and validate the methods used for determining design loads on hulls. The improvements described are highly linked to the increased computing power available to current designers. This increase has led to it being possible to model and predict loads caused by more complex phenomena than previously. The paper focuses particularly on whipping, springing and sloshing. Whipping and springing are both resonance phenomena. The difference between them is that springing is a steady state phenomena and whipping is transient. Sloshing is the movement of liquid cargo.

The paper goes into great detail describing the work being done to obtain reliable hydroelastic models. These models move away from treating the ship as a rigid structure moving through a liquid and instead models the ship bending and twisting as it would in the real world. These models are very simply put the combination of a CFD program with FEA. The hydroelastic models are tested for accuracy with model tests and full-scale measurements and the results are very good, although there is still work to be done.

The paper also goes into great detail about sloshing in tanks and the work that is being done to understand the loads that arise from sloshing. Although this is interesting it has less relevance for the project we are currently working on. The shiptype examined in the paper was a LNG-carrier with huge tanks and our ship is a research and supply vessel with comparatively tiny tanks.

The relevance of this paper for our project is the hydroelastic models that have been used and if they could be used to determine loads in our ship. At the moment it is unclear if we are going to use anything like the models described in the course and project, but whether or not we will we are now more aware of the option.

Paper - An Overview of Roll Stabilizers and Systems for Their Control. K.S. Kula. 2015.

The paper presents various solutions and systems for roll motion. The type of roll stabilization system which is needed depends on the ship type. For example, on cruise ships the excessive motions interfere with the activities offered onboard and passengers' comfort. When on Ro-Ro ships the stabilization is needed for cargo areas e.g., to keep containers stable.

The mathematical model to choose systems is done by evaluating the motion of the ship with coordinates. The coefficients are first determined analytically and then corrected by model tests. Later on, the results is verified in sea trials. The sea trial can include for example turning circle, zigzag trials, and surge-sway-yaw-roll models.

One solution for stabilization is anti-roll tanks which has been introduced already 1874 by William Froude. There are passive, controlled-passive and active tanks, where passive tanks include free-surface tanks and U-tubes. The U-shaped U-tube tank is fitted on both sides of the ship and is monitored by pressure sensors calculating the best stabilizing moment. Passive tanks don't increase hull resistance and the efficiency doesn't depend on ship's speed. But requires space onboard. Active tanks are generated has an actuator pressing water between the sides of the ship. This type needs a large amount of energy but are capable of larger stabilizing moments than the passive tanks.

Moving outside the ship fin stabilizers are fitted on the sides of the hull. As rudder blades the fin generates lift and drag when water flows around it. In regular waves fin stabilizers have ability to 90% roll damping. Heel can also be controlled with rudder roll stabilization.

As conclusion the choice and need of stabilizers depends on the type of ship and operating purpose and if it is useful. To control the performance of ship motion a set of controllers for different speeds, environmental and sea conditions should be designed. Using only one controller for all conditions would not lead to efficient results.

For our project, the most relevant solution would be rudder roll stabilization, where the existing rudders and steering gear is used. As the vessel is mostly operating in arctic areas fins would not necessarily be safe. The market has retractable fins which can be retracted into a fin box while not needed. The anti-roll tanks could be used for stabilisation since they are not affecting outside the hull and are therefore not dependent on the ice. Negative side about them is that they take up space from spaces designed for other purposes.

Paper - Wave Statistics. Michel K. Ochi. 1978

The paper is divided into two distinct parts and sets out a question for the first part and presents a method for the second. The first part is about short-term responses and the spectra used to determine them. The paper's main question in this part is whether the spectra can be relied upon since many factors affect the actual sea spectra in a certain location. The second part is about long-term responses and how to design with them in mind.

The first part starts with presenting and describing two families of spectra to be used in the paper. One family is defined by two parameters and the other is described by multiple. These families of spectra will be compared to data measured at stations presented in the paper. The measured data was also used to create probability distributions for both wave heights and period. According to the paper both the wave height and period are log-normally distributed. This distribution can be used to define the families of spectra. The result is multiple spectra per sea state, which then can be evaluated to determine their likelihood of occurring. This likelihood can be translated into weights for the different spectra. Responses for the different spectra are also evaluated and the difference in responses gives an upper and lower bound for them. This range of responses has then been compared with the measured data from all around the world. The comparison has yielded good results and the computed range has been shown to be in good agreement with the measured values. The difference between the two families of spectra was that the range was wider for the two-parameter family of spectra. In evaluating extreme values for design the spectra used in the paper have to be combined with other factors such as service life etc. The paper claims that the spectra give good values for mild and moderate seas, but that the two-parameter spectra give excessive values in severe seas.

In the second part the series of spectra used is determined by two methods. The first method uses the same two families of spectra combined with the weights mentioned earlier and the second method uses the joint probability function of significant wave height and period, derived from the log-normal distributions, directly. The severest long-term situation was determined using a concept called asymptotic distribution of extreme values, but the accuracy of the results is highly dependent on the amount of data available. The long-term response prediction is performed using factors such as sea severity, spectral shapes, the speed of a ship and likely headings. When the extreme values of the short- and long-term responses were compared they were very similar. The difference between them was the complexity of obtaining results. In that respect the short-term situation was much easier to evaluate.

The paper has given us good insight into how our spectra should be chosen and the methods used for deriving design values from it. The calculations can sometimes seem difficult, but the paper gives a good guide in how to perform them.

Paper - Marine Environments and Its Impact on the Design of Ships and Marine Structures. Michel K. Ochi. 1993.

The paper presented in a comprehensive way how the environmental conditions e.g. different type of waves are relevant to the design and safe operations of ships and marine structures. It goes through the information about winds, waves and currents that are necessary for stochastic prediction of the systems responses in a seaway. The paper is highly practical and describes e.g. what kind of waves should be chosen for spectra etc.

Some aspects that is useful for this report and our design is information about different types of waves, choosing wave spectra and extreme events. The paper introduced different wave spectras and what input parameters is used. For developing wave spectra measured data is used for wave heights and frequency etc. The Pierson-Moskowitz, two-parameter, six-parameter and JONSWAP spectrums where introduced. For more extreme events like hurricanes the JONSWAP is most suitable.

The paper also considered extreme and special wave events. Since the sea is random the ship will probably encounter unusual wave conditions which can cause damage and be unsafe. In cases like hurricanes and tropical cyclones the severest wind speed and associated sea severity should be considered. The evaluation of loadings caused by wind and waves should also be done. For extreme wave heights the design wave height for the vessel should be chosen including a risk parameter to cover the probability for extreme wave height to exceed the thought height. By using long-term wave data with gamma probability distribution, the most severe sea state the vessel will encounter can be estimated.

Book chapter - Principles of Naval Architecture, Vol. III – Motions in Waves and Controllability. Lewis, E., V. 1989. Section 2 Ocean Waves by Cummins, W., E., with paragraphs by Dalzell, J., F.

Ocean waves are extremely irregular but also often statistically stationary over time periods, longer than a half-hour. Sea waves are stochastic and wave elevations at some time intervals roughly follow Gaussian probability density function (PDF). This behavior of sea waves enables use of mathematical models in seakeeping. Estimating size and motion of the waves is important for safe operating of the ship.

Storm generates waves with friction between air and water surface and local pressure fields caused by wind blowing over sea surface. Simultaneously happening events between air and sea surface add disturbances on the wave system, which then develops statistically stationary over time. This fully developed condition is reached with long enough observation distance and time interval when wind speed is steady. Wave-breaking limits wave growth. Wave systems are formed from set of events and they can be resolved into set of wave components, that have lengths and directions, using Fourier transform. Wave components have sinusoidal form and non-normal distribution.

Waves travel along water and water-particles have cycle-like motion, that leads into deep water. Motion of water particles and pressure changes in water affect the whole body of the fluid. Pressure changes affect wave speed and so does water depth. In shallow waters wave speed is function of water depth and in deep water function of wave length. Motion of the water particles and potential energy of the waves form energy of wave trains.

Many models have been developed to describe waves. For example, Stokes waves, Trochoidal waves and Froude’s approach have their cons and pros. Statistical approach is used when describing occurrence of waves. Fast Fourier Transform (FFT) is the best method for analysing waves. Directional spectrum is a great way to describe sea state as it gives more complete picture of the sea. DFT of the wave amplitudes can be Gaussian or Rayleigh, depending on band broadness of the spectrum. Narrow band causing Rayleigh distribution and broad band causing Gaussian distribution. Wave spectrum can then be used to calculate ship response in waves. (Cummins & Dalzell, 1989)

Book chapter - Rawson, K.J., Tupper, E.C. (2001). Basic Ship Theory (5th Edition). Ship motions (p. 459-479)

First in the chapter all the motions in the six degrees of freedom are defined (surge, sway, heave, roll, pitch and yaw). Disturbing yaw, surge and sway modes of the ship doesn’t set the ship in simple harmonic motion, like disturbing heave, roll and pitch modes does. Equations of motions are presented for roll, pitch and heave in undamped and damped still water scenarios.

In undamped still water equations of motions for ship in roll and pitch modes are very similar as they both are rotational motions. The equations are based on Newton's law; moment equals to product of moment of inertia and angular acceleration. Period of motion is solved with equations of motions. The length of period of the ship is compared to spring stiffness and the shorter the period the stiffer the ship is. Periods of roll and pitch motions are dependant of transversal and longitudinal metacentric height, respectively, and period of heave motion is dependant of waterplane area. Equations for a ship in a damped motion in still water are more complex than in undamped motion. The simplest case of damped motion accounts damping with linear damping constant and it can be solved with differential equation. When damping is non-linear, equations of motions can't be solved using differential equations.

Period of roll changes likely depending on ship design and there are many methods to approximate the period of roll. The ship motions are observed in regular ways. Modifying rolling equation in still water the equation for rolling in beam seas is obtained. The equation has many simplifications and wave is described with its frequency and maximum slope. The amplitude of the forced oscillation, that tells how the ship rolls in the wave, depends on frequencies of ship rolling in still water and the wave. Pitching and heaving motions in waves are considered in head seas and can be viewed as mass/spring system, where ship has a mass and sea acts as a spring. Ship's motion in irregular seas is presented with RAO and irregular seas are presented with wave spectrum and finally ship motion is obtained from RAO and wave spectrum. The ship motion is calculated for different ship speeds. Surge, sway and yaw are discussed. Forward motion of the ship is an example of surge, sideward drifting of the ship is swaying and changing course of the ship is yawing. Based on linear theory the ship rolls in large amplitudes when wave encounter frequency is close to ship's natural frequency of roll. Metacentric height of the ship changes with underwater geometry of the ship, which changes in longitudinal waves. Oscillating change of metacentric height causes then large amplitude rolling.

Paper - Quick Strip Theory Calculations in Ship Design, J.MJ. Journée. 1992.

The paper explains the theory behind strip theory calculations and introduces to a method with shorter computation time than the general strip theory calculations used. To analyze seakeeping (local motions, accelerations, added resistance etc.) programs using numerical methods that includes linear theories can be used. These programs are often complicated and the calculations time-consuming. To fasten up the process a strip theory based computational method has been developed.

For application of ship motions in early design stage of the ship the strip theory is the most known for calculation of wave induced motions. The strip theory is based on potential flow theory where viscous effects are neglected and means that effects for viscous roll damping effects should be accounted by empirical formulas. The ships motions are supposed to be small and relative to the cross-sectional dimensions for the ship, the hull area below the water level is only accounted. The strip theory provides a good knowledge for early-stage design and additional model experiments can be done in detailed design stage for evaluating added resistance or extreme events.

The paper goes through the strip theory method and coefficients used for the calculations. For getting results in shorter time two-parameter Lewis transformation can be used. This mapping method is depending on only two parameters, half breadth to draught ratio and area coefficient of the cross section, which makes the method simpler. Total hydrodynamic coefficients for sway, roll and yaw motions can be evaluated by integrating cross sectional values for the ship's length. Pitch and yaw coefficients follow heave and sway moments around the center of gravity. Separate approach must be used for surge motions. Additional damping coefficient for surge and roll can be estimated with Ikeda method. Wave loads in strip theory calculations is also calculated by integrating two-dimensional loads on the cross section by the ship's length, which consists of Froude-Krilov and diffraction parts. In addition, the effects of added resistance, wave spectra, shipping water and bow slamming is described.

The quick strip theory calculations will deliver information with short computation time. The calculation time is mostly consumed for calculation of potential coefficient and the solutions of the equation of motion. By using the Lewis conformal mapping method risks in input errors can me minimized. The faster calculations have been used for calculation ship types like container ships and crude oil tankers and the results shows that the method is safe to use for conventional mono-hull ships in preliminary design stages.

As conclusion the paper had good overview on the strip theory calculations in general and what is included in the calculations. The introduction of quick calculations was interesting and will for sure find use in ship design when calculations have to be done in short notice. For our project NAPA will be used as software where the alternatives for calculations are panel method and strip theory, so despite we don't use the software described in the paper, it gave insight to the background of the strip theory alternative.

Paper - On the non-linearities of ship's restoring and the Froude-Krylov wave load part, J. Matusiak, 2011

In the paper a method called Laidyn is presented to determine ship motions in waves. The method tries to extend a previous method so it can handle long-crested irregular waves. In the method the ship is regarded as a rigid intact body and a linear surface wave theory of Airy is used to model surface waves. The good thing for us is that the method evaluates both the wetted surface of the ship's hull and pressures up to the actual position of the free surface. This can be very beneficial to us since there are some concerns regarding our flat transom. The method does this to take the non-linearities of the Froude-Krylov loads into account.

$$
\begin{aligned}
X_g - mg\sin\theta &= m(\dot{u} + Qw - Rv) \\
Y_g + mg\cos\theta\sin\phi &= m(\dot{v} + Ru - Pw) \\
Z_g - mg\cos\theta\cos\phi &= m(\dot{w} + Pv - Qu) \\
K_g &= I_x \dot{P} - I_{xy}\dot{Q} - I_{xz}\dot{R} + (I_z R - I_{zx} P - I_{zy} Q)Q \\
&\quad -(I_y Q - I_{yz} R - I_{yx} P)R \\
M_g &= -I_{yx}\dot{P} - I_y\dot{Q} - I_{yz}\dot{R} + (I_x P - I_{xy} Q - I_{xy} R)R \\
&\quad -(I_z R - I_{zx} P - I_{zy} Q)P \\
N_g &= -I_{zx}\dot{P} - I_{zy}\dot{Q} - I_z\dot{R} + (I_y Q - I_{yz} R - I_{yx} P)P \\
&\quad -(I_x P - I_{xy} Q - I_{xz} R)Q,
\end{aligned}
$$

The method formulates the equations of motion as can be seen above. The method uses both an inertial coordinate system and a body-fixed one. The equations of motion are solved numerically using 4^{th} order Runge-Kutta integration scheme. The method operates in the time domain so the forces acting on the system must be evaluated at each time step. The extension to the method is how these forces are evaluated.

A panel method is used and three different models for evaluating the pressure at each panel's center point are tested. The irregular waves are created by the superpositioning of 19 component waves. When evaluating the results between the three models of determining pressure and a linear model, the motions are greater in the non-linear results. This is due to resonant behavior.

The method is very interesting to us since it effectively deals with our stern design and non-linearities. Unfortunately, the conclusion states that further investigations need to be conducted to evaluate and validate the accuracy of the method.

Book chapter - The Principles of Naval Architecture Series, Strength of Ships and Ocean Structures, A. Mansour, D. Liu, 2008, Section 2

The book chapter we have selected gives a thorough overview of the types of loads a ship will experience and the moves on to presenting the possible ways of determining them. The chapter classifies the loads in to four different categories, static, low-frequency dynamic, high-frequency dynamic and impact. In this report we are mostly interested in the static and low-frequency dynamic. This is because we are not taking resonant behavior into account nor are we taking phenomena like slamming into account.

The static loads a ship experiences is a function of the mass and buoyancy distribution of the ship. In still water the forces due to gravity and buoyancy are equal in magnitude but not in distribution over the ship's length. This difference leads to a force distribution that when integrated gives the distribution of shear force and moment. As the chapter points out this is largely an exercise in accounting and therefore computer programs like NAPA are very useful. All that the user must do is to make sure that all the mass is present in its right place and the program can adjust the hydrostatic forces accordingly.

The dynamic loads are more difficult to determine and can be done using several different methods ranging from approximations to direct calculations and stochastic methods. The simplest approximate method is to place the ship on a wave with a length equal to the ship's LBP and a height of L/20. Two situations, hogging and sagging, need to be considered. This is done by placing the crest of the wave at midship for hogging and at the bow and stern for sagging. This method is simple but gives overestimations for the loads. The chapter also presents some empirical formulas that can be useful for quick calculations.

When using a direct computation of wave-induced fluid loads hydrodynamic theories are applied to calculating the pressure forces on the hull and the response of the hull to the forces. The chapter mentions several levels of approximation that can be found in the programs used for this.

- Frequency linear strip theory method based on two-dimensional potential flow theory
- Frequency linear three-dimensional theory based on potential flow boundary element method
- Frequency quadratic strip theory method, which consists of a perturbation method of potential flow theory expanded up to the second-order terms for the wave theory, the nonlinearity of restoring forces due to non-vertical ship sides, and the hydrodynamic forces
- Time domain strip theory method, where the hydrodynamic problem is handled according to linear theory but the hydrostatic and Froude-Krylov wave forces are included up to the incident wave surface
- Time domain three-dimensional potential flow boundary element method, where the hydrodynamic problem is handled according to linear theory but the hydrostatic and Froude-Krylov wave forces are accounted for either up to the mean water line (i.e., three-dimensional time-domain linear) or up to the incident wave surface (i.e., three-dimensional time-domain moderately nonlinear)
- Time domain three-dimensional nonlinear theory approach, which satisfies the body boundary condition exactly on the portion of the instantaneous body surface below the incident wave. It is assumed that both the radiation and diffraction are small compared to the incident wave so that the free surface boundary conditions can be linearized with respect to the incident wave surface, whereas the hydrostatic and Froude-Krylov wave forces are included up to the incident wave surface. This approach solves a three-dimensional time-domain potential flow termed "body-nonlinear" problem.

Of these approaches there can't really be said that one is better than the other. It all depends on the application purpose. What mainly separates the approaches and are the factors when choosing which to use is the speed of the process and the accuracy.

Everything that has been discussed previously in the chapter has been about the motions and loads that a ship experiences when subjected to a certain environment, but what is equally important to obtaining good results is that the environment used in calculations is representative of real-life conditions. That is what the chapter brings up next in its subsection called Probabilistic Analysis of Wave-Induced Loads in Random Seas. The content of the subchapter has been dealt with in previous chapters in this report and therefore won't be gone over here.

Book chapter - Rawson, K.J., Tupper, E.C. (2001). Basic Ship Theory (5th Edition). Seakeeping (p. 457-487)

Seakeeping is a quality of a ship describing its seaworthiness in different conditions and it is affected by many features of the ship. Seakeeping criteria are used to estimate seakeeping capability of the ship, which needs to perform in acceptance levels. The limits concern safety, efficiency and comfort of the ship operating. The most common seakeeping criteria are speed and power in waves, slamming, wetness, propeller emergence and impairment of human performance. Also, ship motions and effects of waves on the ship reflects seakeeping performance of the ship and are in major part of seakeeping analysis. For example, zero speed causes most severe rolling motion, this observation can be also seen in Figure 73.

More power is needed to maintain a certain speed in more severe wave conditions. Increase of waves causes resistance in hull and appendages and change of propeller conditions decreases propulsion efficiency. Severe wave conditions can also lead to voluntary speed reduction via captain's decision, especially in slamming conditions, which our project ship will most probably encounter as it has spoon bow. Ship features such as low longitudinal moment of inertia, low longitudinal radius of gyration, low displacement-length ratio, fine form and bulb helps the ship maintain higher speed in severe wave conditions. Our project ship lacks these features as it has hull design common for ice breaking vessels.

Slamming is a notable problem for our project ship, which has to be acknowledged by the captain when operating in conditions susceptible to slamming. Slamming can be decreased by reducing speed and changing the heading. Light loading condition increases possibility of slamming. Khione is resupply vessel and its loading condition changes during its operations. Hull response to the slamming depends on the area the slamming load acts. High pressure on limited area can cause plate deformation and if the area is larger, the pressure can set the whole hull girder vibrating.

Wetness, meaning green water over the ship, cannot be calculated accurately but rougher assessments can be obtained from relative vertical movement of the bow and water surface and from model tests. Figure 73 shows the heave and pitch motion, from which it can be assumed that bow of the ship is relatively large, causing wetness. High forecastle of Khione can prevent wetness. In addition to wetness, large motions can also cause propeller emergence, which is regarded when quarter of propeller diameter is above water surface.

In severe wave conditions motions of the ship can affect the ability and motivation of humans to work. Similar to estimation of wetness there is not accurate method to estimate effect of motions to degradation of human performance. Yet accelerations and its frequency are used assess degradation of human performance. Also, statistical method motion sickness incidence (MSI) can be used.

Book chapter - Lewis, E. (1989). Principles of Naval Architecture (Second Revision), Volume III - Motions in Waves and Controllability. Derived Responses (p. 109-125)

Added resistance is one the ship responses on rough seas effecting on seakeeping performance evaluation. Added resistance in rough consists from added resistance caused by wind, waves, motions due waves and rudder action. Several methods have been developed to obtain added resistance. Some notes from methods are that still water resistance does not affect added resistance, oscillating of the ship generates damping waves that causes added resistance and amount of resistance is coupled with ships natural motion periods. Common to all different methods is that added resistance is proportional to the square of wave height in regular waves. Added resistance is calculated for our project ship with 7 m significant wave height resulting relatively high resistance (Figure 79). Based on the principle of superposition the mean of added resistance can be calculated for irregular seas. Internal stabilizers can decrease added resistance as the rolling motion likely increases resistance. Minor resistance can be caused by yawing and swaying motion coupled with rudder action in oblique seas. Obtaining added resistance in short waves with wave lengths is challenging as small waves do not excite large enough ship motions.

Book chapter - Rawson, K.J., Tupper, E.C. (2001). Basic Ship Theory (5th Edition). Manoeuvrability (p. 523-573)

The maneuverability of the ship can be measured with following criterion: ability to maintain course, ship's response to movement of rudders, response on other control devices and ability to turn around in finite space. Directionally stable ship keeps the new straight course after being disturbed away from earlier straight course. Long and slender hull form increases directional stability of the ship. Our project ship is not particularly long and fine formed but neither short and tubby. Also, large skeg would increase directional stability, but our design is not yet so far in details. Model tests and full-scale trials are important and common when assessing maneuverability of the ship due the lack of analytical methods for maneuverability.

In ship turning rudder force sets and hold the ship in angle of attack in water flow causing sufficient radial force to the ship to turn. Holding the ship in angle of attack causes drag leading loss of speed during the turn. Turning also causes moment, which causes the ship heel outwards of turning path.

Turning circle tells ship's ability to turn around but does not indicate initial response. The zig-zag maneuver can be used to indicate initial response of the ship. Indicators for directional stability of the ship are the spiral maneuver and the pull-out maneuver. Features of the ship that increase maneuverability usually decrease directional stability and vice versa, but not in all cases. Our project ship has more fine than short and tubby hull form, which increases directional stability without decreasing turning ability. Khione needs to have great maneuverability to operate safely in polar waters, where she needs to avoid collisions with ice floes.

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