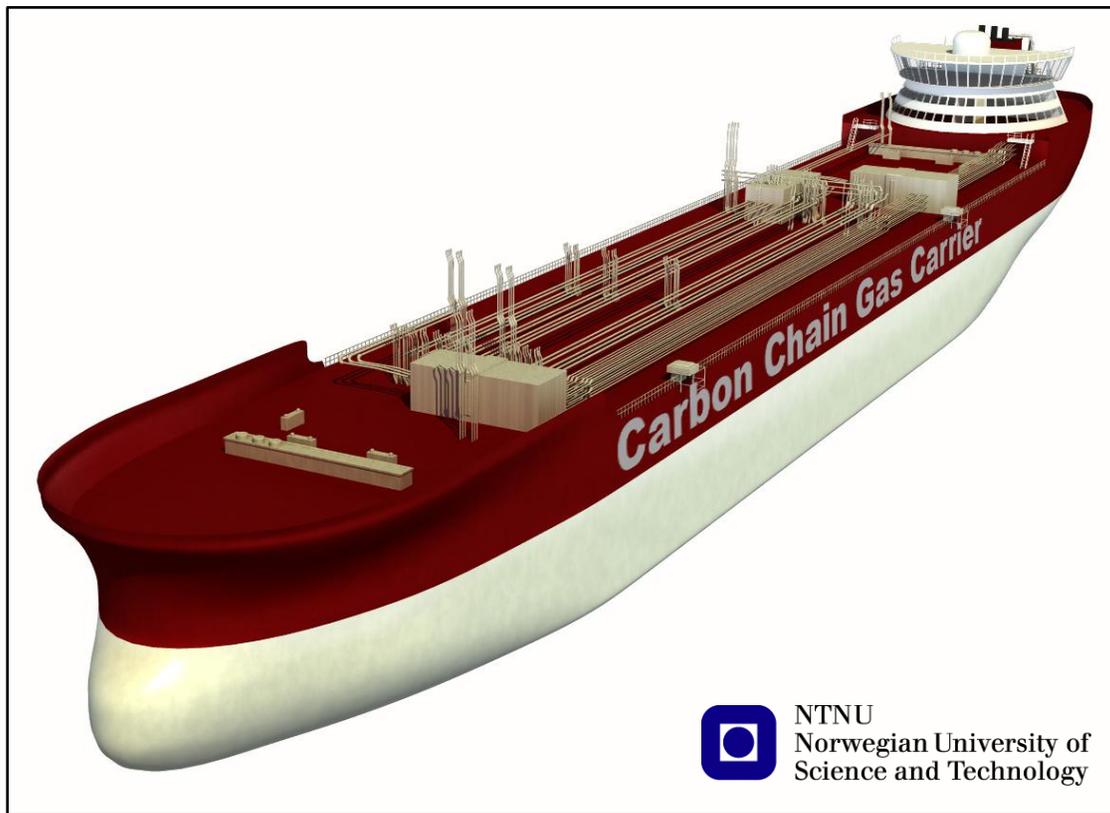


Carbon Chain Gas Carrier



2008-2009 Dr. James A. Lisnyk
student ship design competition

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Preface

This report is a contribution to the 2008-2009 Dr. James A Lisnyk Student Ship Design Competition organized by the Society of Naval Architects and Marine Engineers (SNAME) and the American Society of Naval Engineers (ASNE).

The members of the group are all students in the third year of a five year Master's degree program in Marine technology at the Norwegian University of Science and Technology (NTNU) in Trondheim. Participating in the competition has been both challenging and rewarding for all group members. It proved to be a necessity to expand our knowledge beyond that of the lectures, as well as refreshing knowledge from earlier courses. The time scale and problems concerning access to essential information have been the main restrictive elements.

First, we would like to thank our advisor, Professor Emeritus at NTNU, Stian Erichsen. Special thanks are given to Teekay by Project Manager for the Technology department Bjørn-Olav Gilje, Solvang Shipping by Bertram Engelsen and Environgas by General Manager Ola Ravndal. They have given us motivation as well as much of the essential information.

We have also greatly benefitted of the assistance from the Department of Marine Technology at NTNU by professors Stein Ove Erikstad, Sverre Steen, Maurice F. White, Harald Valland, Bernt J. Leira, Jørgen Amdahl, Anders Endal, Bjørn O. Sillerud and first amanuensis Håvard Holm, as well as SINTEF by Audun Aspelund, Hamworthy by Stein Thoresen, I.M. Skaugen by Kjetil S. Strand and MARINTEK by Erik Lehn, Lars Øien and Erik Henney. Last year's winners of the competition have helped with both motivation and software. This applies specially to Ane Christophersen.

To all the above individuals and to several others who have assisted us in one way or another, we feel very much indebted.

Last, but not least, we would like to thank family and friends for all support, love, patience and forbearance whilst we spent all our time working on this report.

Trondheim 06.05.2009

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Executive Summary

The Owner's Requirements demands a ship able to annually transport three million tonnes of liquefied carbon dioxide for storage as part of large-scale Carbon Capture and Storage. In addition, Liquefied Petroleum Gases is to be taken as return cargo.

As no vessels for this purpose exist today, the main challenge was to find a viable, environmental friendly, economical and efficient way to develop a vessel for a large-scale transportation.

The result is the Carbon Chain Gas Carrier; a weight-critical ship transporting both cargoes in liquid condition. They are stored in the same independent type "C" tanks. Using the same tanks requires purging on cargo change, which is done efficiently with use of the Rapid Purge Technology developed by Envirogas AS.

This report shows that the Carbon Chain Gas Carrier presents a viable solution to the Owner's Requirements, performing the tasks in a successful manner.

System Based Ship Design has been used for guiding the work process, and the end result is presented in the table below:

Length PP	[m]	220
Length OA	[m]	228
Breadth	[m]	31
Depth	[m]	19
Draught	[m]	12.1
GM, LCO₂ / LPG	[m]	2.9 / 2.8
Deadweight, LCO₂ / LPG	[tonnes]	49 152 / 24 743
Displacement, LCO₂ / LPG	[tonnes]	70 000 / 45 000
Annual Delivery, LCO₂ / LPG	Million [tonnes]	3 / 1.5
Number of tanks		6
Cargo capacity	[m ³]	40 000
Service speed	[knots]	16.7
Propulsion		Medium Speed Lean Burn Gas Engine
Fuel		LNG
Installed power	[kW]	2× 9000 + 3×1440
MCR	[%]	86
On/offloading system		Deep Well LCO ₂ Cargo Pumps
On/offloading technology by Environ Gas		Rapid Purge Technology
Building costs	[million USD]	91
Life Cycle Cost	[million USD]	316
Required Freight Rate, LCO₂	[USD/ton]	10
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Table 1: Table of Principal Characteristics

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Symbols

α	Angle	Cwl	Waterline coefficient
β	Angle	Cm	Mid ship coefficient
ε	Strain	d	Diameter
ζ	Thermal conductivity	DNV	Det Norske Veritas
η_1	Surge	DWT	Dead weight tonnage
η_2	Sway	f	Safety factor
η_3	Heave	Fn	Froude's number
η_4	Roll	Fo/Fo	Float on/ Float off
η_5	Pitch	FP	Fore perpendicular
η_6	Yaw	g	Gravity
η_o	Open water efficiency	GM	Metacentre height
η_D	Propulsive efficiency	GM _T	Transverse metacentre height
η_H	Hull efficiency	GT	Gross tonnage
η_R	Relative rotative efficiency	GZ	Restoring lever
\emptyset	Heel angle	h	Height
λ	Air ratio	H _s	Significant wave height
ρ	Density	I	Moment of inertia
σ	Stress	IMO	International maritime organisation
t_G	Correction factor for side ratio	IPCC	Intergovernmental panel on climate change
χ	Heat transfer coefficient	ITTC	International towing tank conference
ω	Frequency	k	Form factor
Δ	Displacement	KB, VCB	Centre of buoyancy
a	Acceleration	KG, VCM	Centre of gravity
Awl	Water plane area	KM	Metacentre over keel
AP	Aft perpendicular	LOA	Length over all
Asp	Aspect ratio of foil	LPP	Length between the perpendiculars
B	Breadth	LWL	Length water line
BM	Centre of buoyancy to metacentre	LCB	Longitudinal centre of buoyancy
C _{app}	Application resistance coefficient	LCC	Life cycle cost
Cb	Block coefficient	LCF	Longitudinal centre of flotation
CCS	Carbon Capture and Storage	LCM	Longitudinal centre of mass
Cf	Frictional resistance coefficient	LCO ₂	Liquefied CO ₂
CL	Centre Line	LNG	Liquefied natural gases
C _m	Mid ship coefficient	LPG	Liquefied petroleum gases
CO ₂	Carbon dioxide	LWT	Lightweight tonnage
C _p	Prismatic coefficient	M	Moment
CP	Critical Point	MCR	Maximum continuous rating
C _T	Total resistance coefficient		
CV	Clipper Victory		

MS	Rest stability
MSI	Motion sickness incidence
M_S	Still water bending moment
M_W	Wave bending moment
NH ₃	Ammonia
NO _x	Nitrogen oxide
p	Pressure
P	Effect
PUR	Polyurethane
Q	Shear Force
RAO	Response amplitude operator
Ro-Ro	Roll on – Roll off
RFR	Required freight rate
RPT	Rapid Purge Technology
R_T	Total resistance
S	Wetted surface
SAC	Section area curve
SOLAS	International convention of safety of life at sea
t	Thrust reduction/thickness
T	Draught
T_g	Glass transition temperature
TP	Triple point
U_1	Outgoing velocity from the propeller
U_a	Induced axial speed plus rotational speed
U_T	Induced tangential velocities
V	Velocity
VCM	Vinyl chloride monomers
w	Wake
WAC	Waterline area curve
Z	Section modulus

1 Owner's Requirements

1.1 Background and Introduction

1.1.1 Carbon Capture and Storage

12th of October 2006 the Norwegian government and Statoil signed an agreement to start the world's first full-scale CO₂ capturing plant at Mongstad, Norway. The plan is to inject the captured CO₂ into geologic formations in the North Sea for indefinite storage.

The increasing energy demand that leads to huge amounts of CO₂ emissions are an environmental problem. This results in an increasing focus on reducing the CO₂ emissions, and a growing interest in CO₂ capture from industrial processes for storage.

The Norwegian offshore geological formations "Johansen" and "Utsira" are porous enough to absorb the CO₂ and have the appropriate cap rocks to ensure that no gas will escape. The Johansen formation is situated about 100 km off the coast, and Utsira somewhat further out. Statoil has planned to start with 0.1 million tonnes annually and increase this over time to 1.1 million tonnes annually through a secondary gas power plant in Kårstø. Simulations shows that the injection wells probably can take somewhere in the range of 3.5 million tonnes in each well in the Johansen region.

In 2004, Germany emitted about 886 million tonnes of CO₂. This makes it the country with the 6th largest annual emissions in the world. From estimates of the IPCC (Intergovernmental Panel of Climate Change), the CO₂ emission fee will have to be at least 80 USD/ton to reduce the CO₂ emissions sufficiently. Plans for CO₂ capture is taking form all over the European Union, making safe storage of CO₂ an upcoming business.

1.1.2 Transport of CO₂

The link between capture and storage is transportation. Pipes may be regarded as preferable to ships, but only for short distances and large amounts of CO₂. Where that is not the case, ships will be favourable. It is clear that the need for transport cannot be covered entirely by pipes as the CO₂ emissions are scattered over vast areas, and not necessarily constant. Therefore, the need for a cost effective and flexible transport system is present.

Today, small ships (1000 – 5000 m³) are used for transportation of CO₂. However, due to the huge amounts of CO₂ emissions, it would be expensive and undesirable to utilize these smaller ships for transportation. As the CO₂ emission fee is expected to increase drastically, there is likely to be a need for vessels that can transport large quantities of CO₂.

A few of the LPG ships in operation today will be capable of CO₂ transportation (after some simple modifications) as the properties of LPG are very similar to that of liquefied CO₂. It should also be noted that 1/3 of the world's LPG fleet is over 20 years of age, and will have to be replaced the following years.

1.1.3 LPG

There are produced large amounts of LPG at the refinery at Mongstad on the south west coast of Norway that can easily be distributed to Germany and other places in Central Europe. Germany consumed 2.8 million tonnes of LPG in 2006.

The total base demand of LPG in Central Europe and Eastern Europe were 13.2 and 9.7 million tonnes respectively in the same year. It is assumed that the amounts will increase in the next years. In Southern Europe the demand is 16.2 million tonnes, but is not expected to increase.

The total annual amount of LPG produced at Mongstad is 1.2 million tonnes. The refinery at Kårstø, Norway is producing 4.4 million tonnes annually, and could be a source for more LPG if so is needed.

This makes Norway one of the largest exporters of LPG in Europe, capable of supplying most of Central Europe with LPG.

1.2 Mission

The main function of the ship is to transport captured CO₂ from Europe to Norwegian ports for geologic storage. The secondary function of the ship is to transport LPG from Norway to Europe.

1.3 Trade Route

The route will be between Mongstad, Norway and Brunsbüttel, Germany. A roundtrip is as follows: Brunsbüttel-Mongstad- Brunsbüttel.

At Mongstad the CO₂ will be transported in pipes for storage at the Johansen or Utsira formation in the Norwegian Sea. The ship will then transport LPG from Mongstad to Brunsbüttel.

The distance between Brunsbüttel and Mongstad is 930 km.

3 million tonnes of CO₂ is to be transported each year. A corresponding large, but feasible amount of LPG should be taken as return cargo.

1.4 Capacities

The ship is to be in accordance with, but not limited by, the following specifications:

- Ability to transport at least 3 million tonnes CO₂ each year
- Ability to transport LPG

1.5 Cargo Access

The CO₂ will be loaded from intermediate storages on land equal to 1-2 ship loads. The same is assumed for LPG.

1.6 Limitations of Principal Particulars

The ship may not exceed the restrictions of the harbours in Brunsbüttel and Mongstad. They are:

- LOA: 350 meter
- Breadth: 55 meter
- Draught: 13.8 meter

1.7 Classification

Det Norske Veritas (DNV)

1.8 Country of Registration

Norway

1.9 Applicable Regulations

- IMO
- SOLAS
- DNV Rules for classification of ships

2 Area of Operation

The area of operation was determined by three major factors.

- Location of possible future full-scale carbon capture (CC)
- Location of possible future full-scale carbon storage (CS)
- Nearby trade route for LPG

In Europe, the location of future CC facilities will primarily be at the coal fuelled power plants. It is then assumed that the CO₂ will be transported to a temporary storage in Brunsbüttel.

Possible future full-scale storage can be done offshore and onshore. Carbon Capture and Storage (CCS) projects have already been initiated in many countries. (1) (2)

The Norwegian offshore geological formations Johansen and Utsira are porous enough to absorb the CO₂ in saline aquifers, and have the appropriate cap rocks to ensure that insignificant amount of gas will escape. The Norwegian offshore industry is very well established, so a great deal of the required technology and competence for CO₂ storage are already present.

As explained in the Owner's Requirement, the trade route will be:

Route	Payload
Brunsbüttel – Mongstad	CO ₂
Mongstad – Brunsbüttel	LPG

Table 2: Trade route



Figure 1: Area of operation (3)



Figure 2: Trade route (4)

The waters of the North Sea are very rough. Storms are common, demanding a great deal of the ships frequenting the area.

Maximum wave height is assumed to be 32 meters (5), and significant wave height, H_s , is 6.3 metres (6). These and other data from sea spectral wave characteristics are measured in the North Sea nearby Karmøy, which is in the area of operations.

The winter in the North Sea is a period with harsh weather, and severe winter storms often occur to a large extent. The vessel will of these reasons experience a lot of rough sea in transit, making the route and ship speed subject to alteration. To make up for any loss of time, the ship must be able to run at higher speeds to satisfy the schedule. The installed motor rating for propulsion must be high enough to allow for this to happen.

3 Mission Development

3.1 Overall Aim and Focus

The aim of the design team is to show that a cost effective and environmental friendly transport between carbon capture and storage is possible.

3.2 Transport Task

3.2.1 Conditions of CO₂

The CO₂ is to be transported in a cost effective manner. Generally, this will imply the largest possible density of the payload as well as a minimum use of time and energy. The CO₂ can be transported in three different conditions: Solid, liquid and super critical. Figure 3 (7) and Figure 3 (8) forms the basis on which the condition was set.

The pressure-density diagram shows the density of CO₂ as a function of temperature and pressure. The straight vertical and horizontal lines marked by red colour shows the border between conditions. The usual condition used in pipelines as well as the conditions for full- and semi pressurized vessels are shown. The points indicating best condition for fully refrigerated- and semi pressurized ship, and pipelines are suggested by A. Aspelund and Dr. de Koeijer (7).

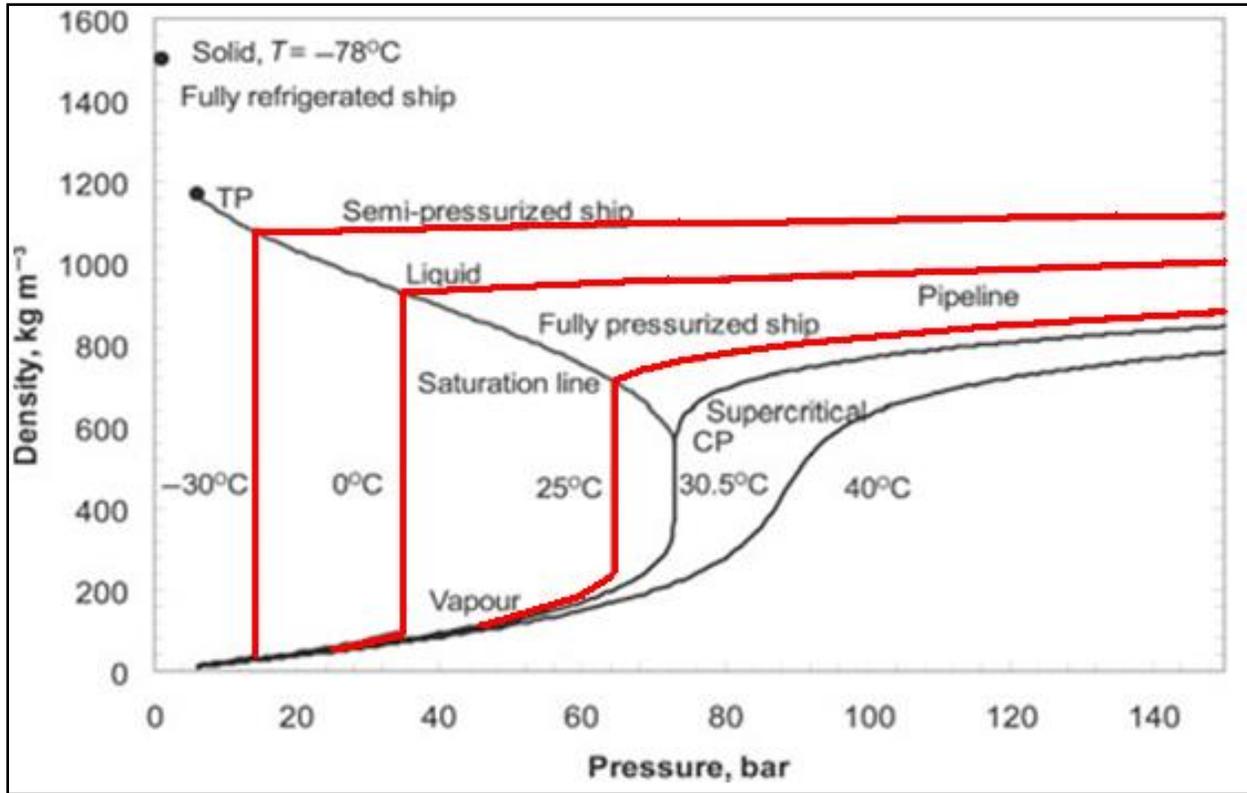


Figure 3: CO₂ pressure and density diagram (7)

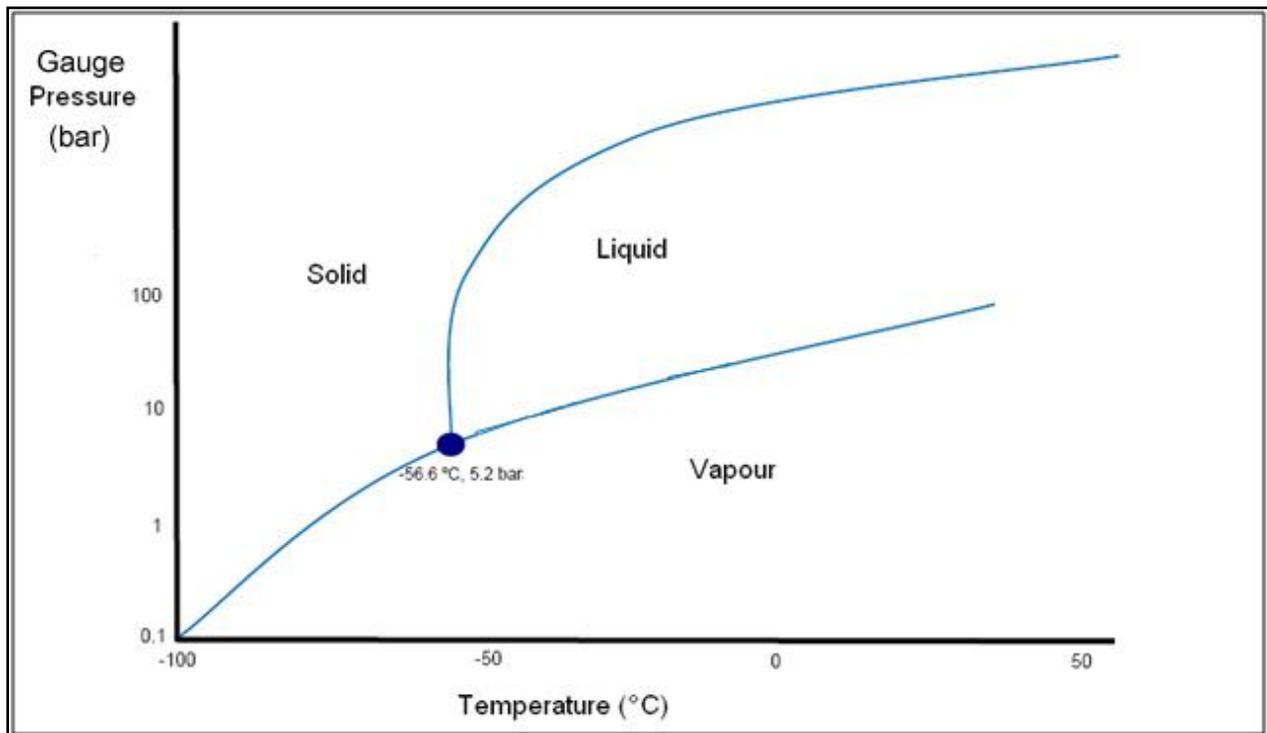


Figure 4: CO₂ phase diagram (9)

As the densities of CO₂ are higher in liquid and solid phases, only they are represented in the selection process.

3.2.1.1 Solid (SCO₂)

Solid phase will ensure the greater density of the two in question, and can be achieved by either cooling down the substance, applying great pressure, or a combination of both. There will, however, be some problems connected to large-scale transfer in solid phase; the on/offloading procedures will be rather complex and require much time and energy (7). The density is, for all practical uses, limited by both temperature and pressure to no more than 1500 kgm⁻³.

Temperature	[°C]	< -78.5
Pressure	[bar]	1
Density	[kgm ⁻³]	< 1 500

Table 3: Properties of CO₂ in solid condition

3.2.1.2 Liquid (LCO₂)

To ascertain the largest possible density of LCO₂ the liquid should be as close to the triple point (TP) as possible. The temperature should be somewhat higher and with a slightly higher pressure. This is to ensure that some distance to the TP is kept; avoiding formation of solid phase which would be very problematic (7).

Liquid CO₂ near the TP has a large density (1150 kgm⁻³), only surpassed by that of solid CO₂. The thermodynamic properties of LCO₂ are not far from those of LPG and ethylene. The technologies for transporting the liquids in question are well established. It is, for instance, possible to use pumps much the same way as on LPG ships. The exact properties are suggested by (10) and (7): Properties of liquefied CO₂

Temperature	[°C]	-52
Pressure	[bar]	6
Density	[kgm ⁻³]	1 150

Table 4: Properties of liquefied CO₂

3.2.1.3 Choice of Condition

The loading procedures are considered more complex and expensive with solid CO₂ (7) (10). Of this reason, liquid phase is chosen for transportation.

Temperature	[°C]	-52
Pressure	[bar]	6
Density	[kgm ⁻³]	1 150

Table 5: Properties of CO₂ during transport

3.2.2 Condition of LPG

LPG is a mixture primarily of propane and butane, and the concentration of these components can vary. The boiling point of propane is -42 °C at atmospheric pressure, which is lower than that of butane.

Butane has a higher density as well (11). Therefore, data for propane is considered the more conservative and appropriate to use for LPG. The TP of propane is found at 1.013 bar and -42.1 degrees Celsius. The density is 582 kgm^{-3} at this point.

For LPG, -42 degrees Celsius is the lowest temperature allowable by the Gas Code, as a corresponding negative gauge pressure at phase equilibrium is not allowed in the tank. (12). As this is the closest allowable temperature to LCO₂, it is chosen. The pressure is then set to be 1.125 bar.

Temperature	[°C]	-42
Pressure	[bar]	1.13
Density	[kgm ⁻³]	582

Table 6: Properties of LPG during transport (11)

The properties of LPG (propane) are found at the National Institute of Standards and Technology. (11)

3.3 Time Calculations

Assumptions concerning time spent in harbours and during manoeuvring have been made. With the annual amount and distance fixed, the time of each roundtrip, speed, capacities and number of ships were the variables needed to be taken into consideration.

Manoeuvrability time is set to 24 hours per roundtrip. Time spent at port preparing for on/offloading is included. This is an estimate based on numbers from similar ships ((13) (14)) Time for on/offloading has been fixed at 24 hours (15) with the assumption that larger cargo volumes are compensated for with greater pump capacities. With 7.5 days of maintenance annually, the total number of operating days is 357.5.

The following table shows the alternatives for roundtrips. Delays have not been taken into consideration in the following tables. Delays are expected to be compensated for with an increase of speed.

Choice		1	2	3	4
Annual amount	10^6 [tonnes]	3	3	3	3
Roundtrip	[days]	7	6	5.5	5
Roundtrip	[h]	168	144	132	120
Distance	[nm]	1 000	1 000	1 000	1 000
Manoeuvring	[h]	24	24	24	24
On/offloading in harbour	[h]	48	48	48	48
Transit	[h]	96	72	60	48
Speed	[kn]	10.42	13.89	16.67	20.83
Density	[tonnes/m ³]	1.15	1.15	1.15	1.15
Cargo capacity	[m ³]	51 439	44 091	40 416	36 742
Cargo capacity of CO ₂	[tonnes]	59 155	50 704	46 479	42 254

Table 7: Alternatives for roundtrips with various parameters

As there will be costs attached to the intermediate storages at the harbours as well as to the cooling of the cargo during transport, the time spent on each roundtrip should be minimized. Based on experience from similar projects (16), it was decided that roundtrip time should not be longer than seven days. Roundtrips being shorter than five days will imply very high velocities which give great resistances, and have not been taken into consideration. A scatter diagram of existing comparable gas ships from 10 000-100 000 m³ is shown below.

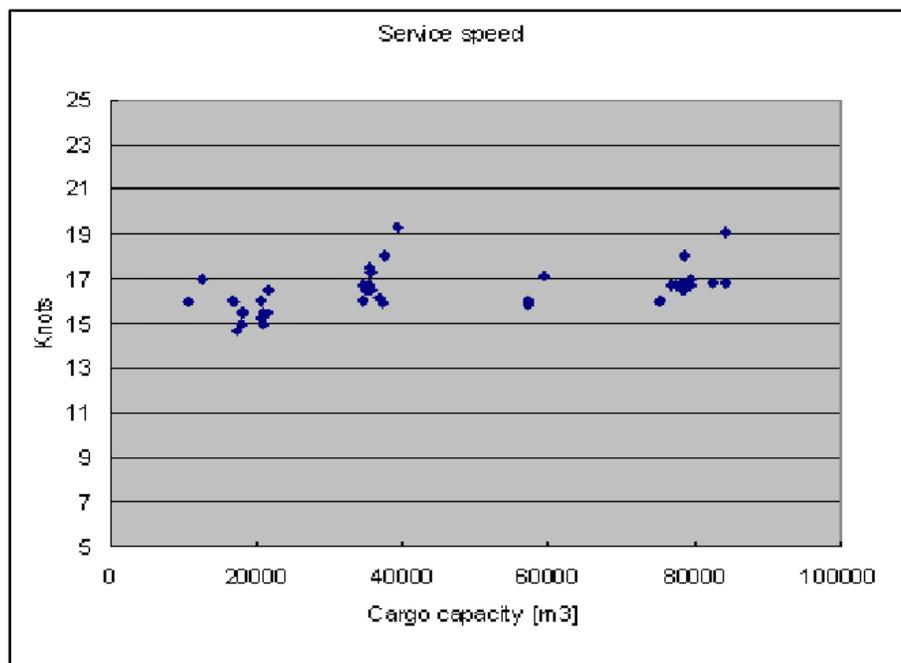


Figure 5: Representation of existing comparable gas ships. (17)

This figure shows that the speed, regardless of cargo capacity, is between 15 and 19 knots, giving an indication of the range of the resulting speed.

Thus, choice number three with a roundtrip of 5.5 days was found as the better choice.

The following table shows the selected roundtrip. The return cargo of LPG should not be considered as fixed, but as the maximum possible amount to be transported if the same tanks are used. Thus, the annual amount of CO₂ is fixed; that of LPG is not.

		Brunsbüttel-Mongstad	Mongstad-Brunsbüttel
Cargo		CO ₂	LPG
Distance	[nautical miles]	500	500
Total time	[h]	66	66
Capacity	[tonnes]	46 000	23 600
Annual amount	10 ⁶ [tonnes]	3	1.5

Table 8: Selected roundtrip

When the established capacity per roundtrip is decided, the number of ships is determined. The options are as follows:

Total capacity	[m ³]	40 000	40 000	40 000	40 000
Number of ships		1	2	3	4
Capacity	[m ³]	40 000	20 000	13 333	10 000

Table 9: Number of ships

Generally, a larger ship has a lower capital cost, is more fuel efficient and thus more environmental friendly than many smaller ships with a total capacity equal to the large one.

Several smaller ships will, however, frequent the harbours more often and render the need for large buffer storages obsolete.

As the aims and goals are making an environmental friendly and cost efficient transport, one ship was selected in favour of two or more.

Number of ships	Volumetric capacity [m³]	LCO₂ capacity [tonnes]	LPG capacity [tonnes]
1	40 000	46 000	23 600

Table 10: Cargo capacity

CO₂ ships exist today, but their capacities are very limited. Yara operates a few, but they do not exceed 1500 m³ (18). These ships are, of this reason, not suitable for full-scale transport of CO₂ (7). As there are, to the knowledge of the design team, no existing combined LCO₂ and LPG ships, a completely new ship design is required.

4 Concept Design

When designing a ship, strategies of how and where the efforts should be placed are needed. *System Based Ship Design* (19) shows a creative, but systematic way of designing a ship. Based on empirical data, it contains coefficients that can be used in the design process. The design spiral is an illustration of the iterative process that is based on repetition of calculations.

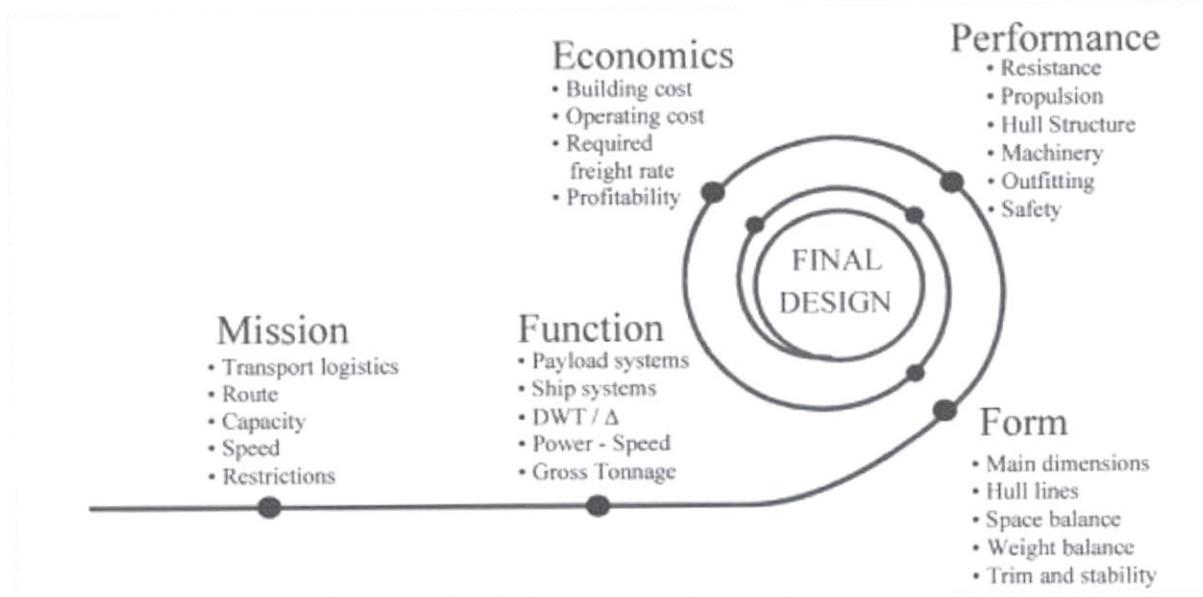


Figure 6: Spiral of design. (19)

This is the method that has been chosen for this project of design.

“By defining each system and the performance requirements for this system we get a framework for the ship design work.” (19)



Figure 7: Order of design stages (19)

The design process is divided into five main stages: Mission, function, form, performance and economics. The design is based on the mission specified for the ship, where aspects such as transport logistics, route, capacity, speed and restrictions are taken into consideration. With these details established, the functions of payload systems and ship systems can be found. Gross volume, power and gross tonnage can also be calculated. Thus, all the required input data for the design are given. It follows

that the form of the vessel can be decided by finding the main dimensions, hull lines, space balance, weight balance and the trim and stability. Performance is the next step in the process where resistance, propulsion, hull structure, machinery, outfitting and safety are established. The last step in the spiral is economics where building costs, operating costs, required freight rate and profitability are calculated and evaluated.

This process will repeat until a satisfactory result has been found.

5 Evaluation of Alternatives

As found in chapter 3, Mission Development, both the CO₂ and the LPG is to be transported as liquids at given conditions.

5.1 Hull Types

The main driving factor in choosing a hull form is that the ship must be able to transport 3 million tonnes CO₂ and a feasible amount of LPG each year as specified. The selection of hull form is based on how the lift force is acquired. This can be obtained by means of buoyancy from displacement, dynamic or powered lift.

5.1.1 Dynamic Lift

These vessels get a change of displacement when in transit speed. The velocity towards the hull or foils makes a lift force that elevates the vessel. This results in less displacement. Dynamic lift vessels are suited for high speed, but at the cost of less efficiency compared to that of displacement vessels. (19). Hydrofoils are of this type. Most of these vessels are navy patrol boats or passenger ships.

5.1.2 Powered Lift

On these vessels, some power is used solely for the purpose of lifting the vessel. Air cushion vehicles (ACV) are of this type. They are suspended slightly above the ground due to high-pressure air flow output. Powered lift vessels generally require smooth surfaces and do not have adequate cargo spaces due to the high power to cargo ratio. (19)

5.1.3 Displacement Vessels

These are vessels acquiring their lift from buoyancy from the displacement. The draught is the same at transit and zero speed. Thus, the displacement does not change by means of speed. All gas carriers operated today are of this type. (19)

5.1.4 Hull Type Selection

As large loads are to be transported each roundtrip, dynamic and powered lift vessels have not been taken into consideration in the selection process. Only displacement vessel hull forms are suitable for this task.

5.2 Displacement Vessels

5.2.1 Ro-Ro Ships

These ships are categorized by the way the cargo is handled. The cargo is rolled on and off by trucks. The cargo can also be automobiles. The alternative for cargo handling could be cranes. This way of doing on/offloading can lower the time at harbour, because it is in some cases less time consuming.



Figure 8: Ro-Ro ship. {Yokohamamotors.com}

With a Ro-Ro ship the pressure tanks would have to be rolled or driven on and off. This idea was based on the wish to decrease the time spent on on/offloading. It would also make the ship more flexible for smaller quanta. Further transportation can easily be done by use of trucks, and storage tanks in the ports may not be needed.



Figure 9: CO₂ tank of 180 m³ transported by truck. {Google.com}

5.2.2 Float on/Float off (Fo/Fo) Vessel

These vessels transport all the cargo on the deck. The ships submerge the deck, and float the cargoes on and off.



Figure 10: Fo/Fo vessel with semi-sub. {Google.com}

Instead of many small tanks, it is possible to transport one large tank. This tank would have to be situated close to the shore, or in the water, ready to be floated on board the ship.

Because of the heavy load, these ships usually have low speeds during transit. The high density of CO₂ can cause major stability problems. Issues are also connected to large tanks that are required as necessity for a new infrastructure.

5.2.3 Standard Gas Carriers

These types of ships can carry a large variety of gases, all depending on the tanks and equipment utilized. Some ships, like Ammonia, LPG and ethylene carriers handles gases in conditions similar to that of liquefied CO₂. The needed speed is 16-17 knots, which is in the range of that of these vessels. Most of the required technology and infrastructure are already in place, and these ships are generally more energy efficient than the other ones mentioned. Time spent on on/offloading will be limited by the pump capacities.

5.2.4 Displacement Vessel Selection

The choice is illustrated in the table below. Points from one to three are given indicating how well the vessels meet the requirements of speed, costs and feasibility. Higher points are favourable.

Vessel Type	Speed	Feasibility	Costs	Sum
Ro-Ro	3	1	2	6
Fo/Fo	1	1	1	3
Standard Gas Carriers	3	3	2	8

Table 11: Choice of displacement vessel

When taking the above-mentioned arguments and the selection table into consideration, it is clear that of the three alternatives considered, the better way of transporting CO₂ at specified condition is by means of standard gas carriers.

6 Liquefied Gas Handling

As specified earlier in the report, the CO₂ is to be transported in liquid phase in the following condition.

Density	[kgm ⁻³]	1 150
Temperature	[°C]	-52
Pressure	[bar]	6

Table 12: Condition of CO₂

The cargo must be stored in a way so that these conditions are ensured. This implies that some sort of closed tanks designed for this purpose will have to be utilized.

6.1 Tank Type Selection

According to the DNV Rules (DNV Pt.5, Ch.5), tanks of type integral-, membrane- or semi-membrane cannot be utilized for a cargo in this condition. This leaves only independent tanks.

Independent tanks are completely self-supporting, and do not form a part of the ship's hull structure. They are to be freely supported with their weight distributed evenly over the entire inner bottom of the cargo space. This implies that they do not contribute to the hull strength, and are not affected by ship deflections. There are three types of independent tanks that are used for transportation of liquefied gas.

6.1.1 Type "A" Independent Tanks

Type "A" tanks are constructed of flat structures and are often prismatic in form. The maximum allowable gauge pressure is 0.7 bar, indicating that the cargoes must be transported in fully refrigerated condition at or near atmospheric pressure. When carrying cargoes below -10 °C a second barrier is required as the tanks are not crack propagation resistant. Fully refrigerated LPG carriers must have a second barrier that covers the entire tanks. This barrier is usually part of the hull structure and should withstand low temperatures. (20)

6.1.2 Type "B" Independent Tanks

Type "B" tanks can be both spherical and prismatic in shape. They are subject to a more detailed stress analysis, which includes fatigue and crack propagation, than type "A". Only partial second barrier is required. The spherical tanks are almost exclusively applied to LNG ships. The prismatic type "B" tanks have a maximum gauge pressure of 0.7 bar like the type "A". (20)

6.1.3 Type "C" Independent Tanks

Type "C" tanks are normally spherical or cylindrical capable of containing cargoes with gauge pressure higher than 2 bar. If appropriate low-temperature steels are used in the tank construction, they can, in addition to semi-pressurized, also be used for fully-refrigerated carriage. For semi pressurized ships the tanks may be designed for gauge pressures up to 8 bar. On these tanks, there is no need for a secondary barrier other than a drip tray. (20)

6.1.4 Tank Type Selection

The desired tanks for the vessel are type "C" independent tanks. This is the only tank type that withstands a gauge pressure of 6 bar or more. The ship will then be classified as semi refrigerated and fully pressurized (21).

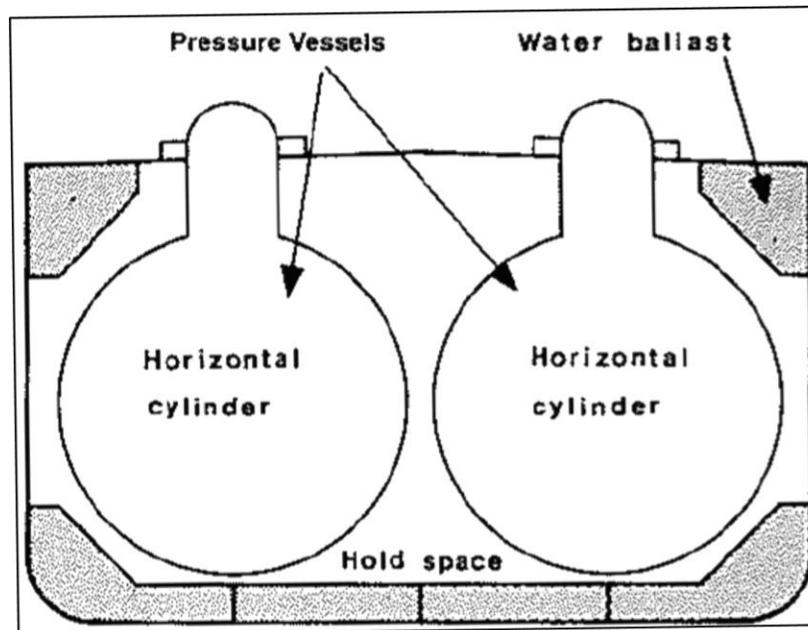


Figure 11: Type "C" tanks (20)

6.2 Tank Configuration

6.2.1 Utilizing Same Tanks

If both cargoes are transported in the same tanks, they have to be purged with cargo change in order to avoid contamination of the other cargo (12).

Both LPG and CO₂ are transported at liquid phase, and in almost similar conditions. The tanks must, however, be cleared on cargo change to avoid contamination of the cargoes. As CO₂ is an inert gas, no additional purging is required.

6.2.1.1 Tank Clearing Procedures of Today

The LPG tank clearing procedures of today are usually divided into the following steps (22):

1. The ship arrives at the harbour with liquid CO₂ and discharges it.

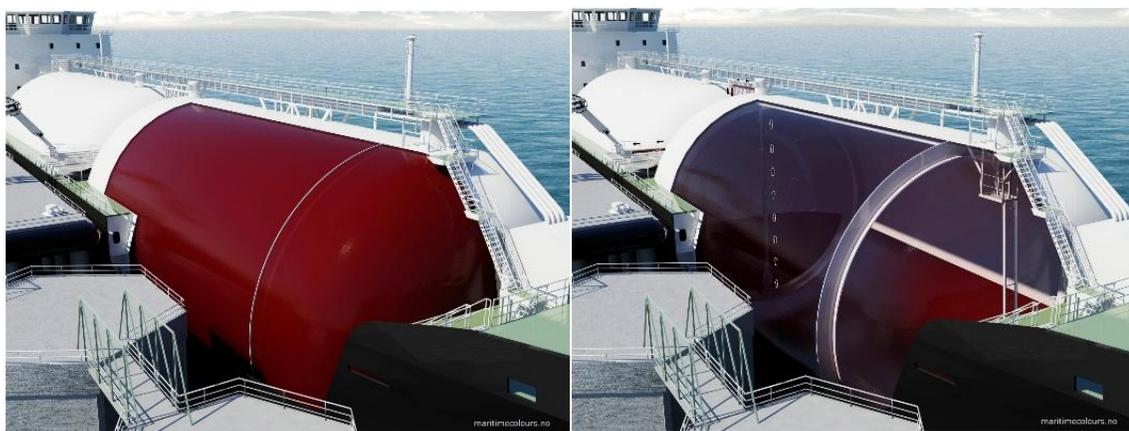


Figure 12: Discharging cargo

2. The gas phase of CO₂ is sent onshore until the ambient pressure is reached.
3. The vessel departs for offshore LPG gas purge. The purged gas mix is released directly into the atmosphere. This is sometimes 300 % of total tank volume.

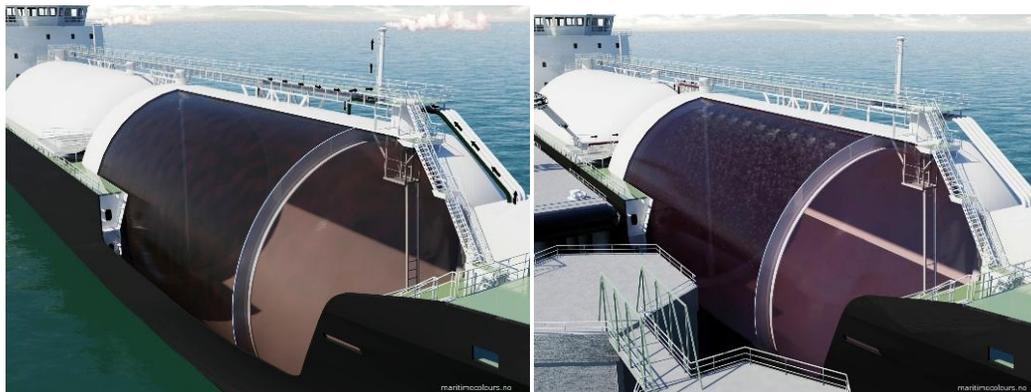


Figure 13: Offshore purge

4. Once sufficiently purged, the vessel returns to harbour for loading of LPG.

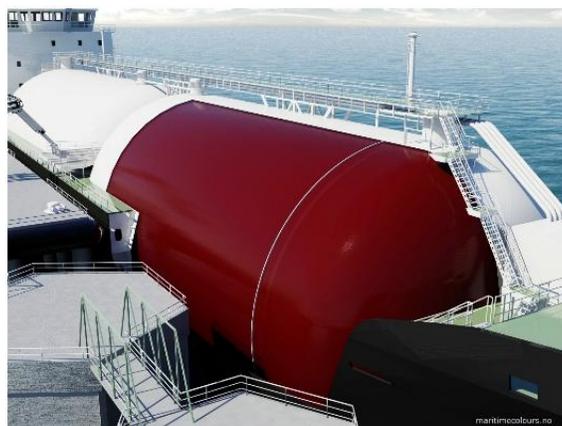


Figure 14: Ready for new cargo

Discharge of LPG is similar to that of LCO₂.

Even though the times vary with different ships and capacities, it is clear that these existing procedures are, in addition to being costly and polluting, very time consuming. The LPG loss on offshore purging is estimated to be one percent of total cargo transported (22), and it is not unusual that this takes an entire week. (22) The comparison ship Clipper Victory, on the other hand, has a total purge time of between eight and eleven days (23). This is, however, also included inert gas purges.

Of these reasons, it was decided to utilize technologies that would permit cargo change without resorting to the procedures explained above. The design team found one that satisfied those demands: The “Rapid Purge Technology”.

6.2.1.2 *Rapid Purge Technology*

Rapid Purge Technology (RPT) is currently being developed by Envirogas AS, and has not yet reached the market. Use of this technology allows cargo change to be done without making use of the procedures mentioned above. Very short summarized, a bellow is inflated inside the tank with cargo discharge to displace the remaining contents. Cargo change from LCO₂ to LPG with RPT is done using the following steps (22):

1. Vessel arrival at harbour. LCO₂ is discharged.

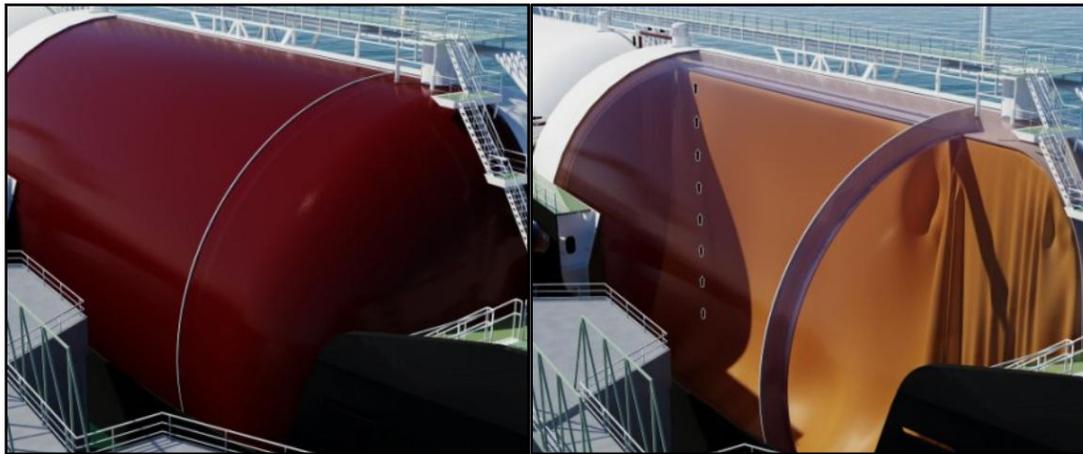


Figure 15: RPT, discharging LCO₂

2. Gas phase is transferred into the bellow.

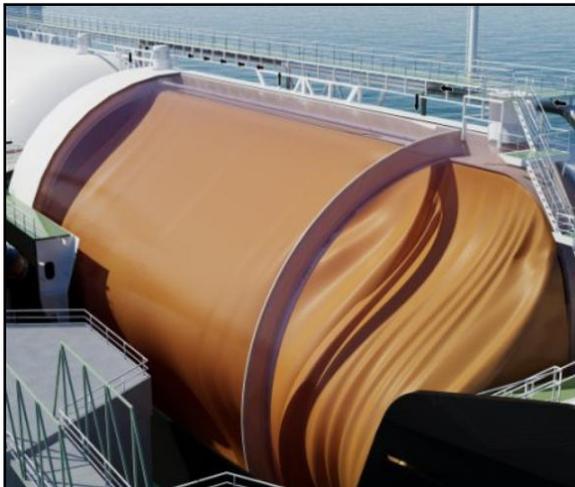


Figure 16: RPT, inflating bellow with CO₂

3. The bellow is fully inflated with CO₂ gas phase. There is virtually no volume in the tank outside of the bellow, thus squeezing any remaining CO₂ gas out.



Figure 17: RPT, bellow inflated

4. LPG gas is introduced in the tank space outside the bellow. At the same time, the CO₂ gas in the bellow is displaced onshore.



Figure 18: RPT, CO₂ gas from bellow and LPG into tank

5. The tank is now ready for LPG loading.

Total time spent is equal to that of liquid on/offloading as the RPT functions simultaneously.

To enable for use of RPT, some additional features must be installed:

- Deck tanks the size of about one per cent of total tank capacity. They are able to contain the gas phases of LPG and CO₂.
- Bellows in the cargo tanks. They are made of light density polyethylene (LDPE). This is a very cheap and common material, the very same used for making plastic bags. The costs attached to the materials will thus be negligible. LDPE has a glass transition temperature of:

$$T_{g, LDPE} = -105 \text{ }^{\circ}\text{C} \text{ (24)}$$

The RPT is not yet on the market, but it is assumed that this will happen in very near future. The use of this technology allows the user to save significant time and money; thus making the vessel more cost efficient and flexible than other gas carriers. Most important, however, is the fact that it is environmental friendly.

It has been established that utilizing the same tanks for both cargoes implies use of the RPT, and the costs are therefore equal to that of this technology.

6.2.2 Utilizing Separate Tanks

Using separate tanks for each cargo will result in larger cargo space and thus higher material and construction costs for both the hull and tanks. No tank purging is required.

6.2.3 Selection of Tank Configuration

The following table shows the alternatives and the method of selection. Points have been awarded from 1 to 3, in which the higher numbers represent the better choice.

	Same tanks		Separate tanks
	RPT	Normal tank clearing	
Costs	3	1	1
Environmental friendly	Yes	No	Yes
Efficient on/offloading procedures	3	1	3
Sum	6	2	4

Table 13: Selection of Tank Configuration

When eliminating the alternative that is not environmental friendly, it is clear that using the same tanks for both cargoes with RPT is the better choice.

6.3 On/Offloading

The on/offloading procedures are to be executed in an efficient and environmental friendly fashion. Furthermore, cargo offloading is done by equipment on board to minimize time spent at harbour as well as to ensure certain flexibility. The time spent on on/offloading in each harbour has already been given in the Mission Development.

On/offloading time in each harbour	[h]	24
------------------------------------	-----	----

Table 14: Time for on/offloading

It has been assumed that both harbours are ready to distribute LPG and CO₂ from intermediate storage tanks on ship arrival. Onloading is done by means of onshore pumps, and offloading is done by means of onboard deep well cargo pumps.

6.3.1 Deep Well Cargo Pumps

Deep wall cargo pumps are designed to enter the tank vertically in the tank sump like shown on the figure.

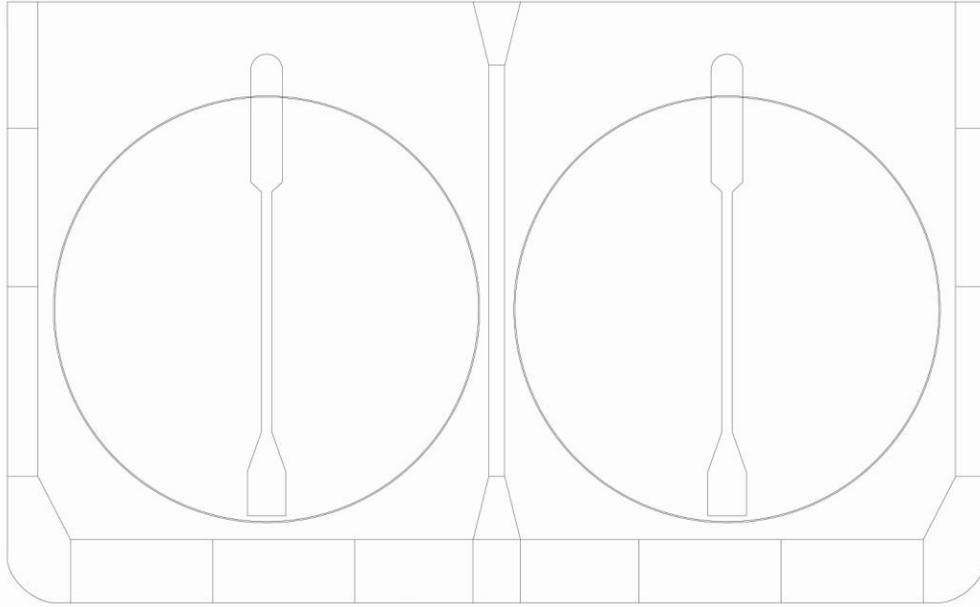


Figure 19: Mid ship section with pumps

Hamworthy Svanehøj AS is one of the biggest developers of these kinds of pumps. This company has designed a new deep well cargo pump specially made for LCO₂ that is suitable for this ship. This is the DW-200. This pump type has the following specifications.

Operation Temperature	[°C]	-52
Capacity	[m ³ /h]	400
Efficiency	[%]	69
Required power at operation point	[kW]	188
Max power	[kW]	196
Length of pump	[m]	15

Table 20: DW-200, specifications (25)

A safety factor of two hours of total on/offloading time are dedicated to unforeseen events as the RPT is not yet a proven technology. This leaves 22 hours to on/offloading. Calculations show that nine pumps are needed with the current pump capacity. To obtain some level of redundancy, ten pumps in total have been selected. They are arranged as shown in the picture below.

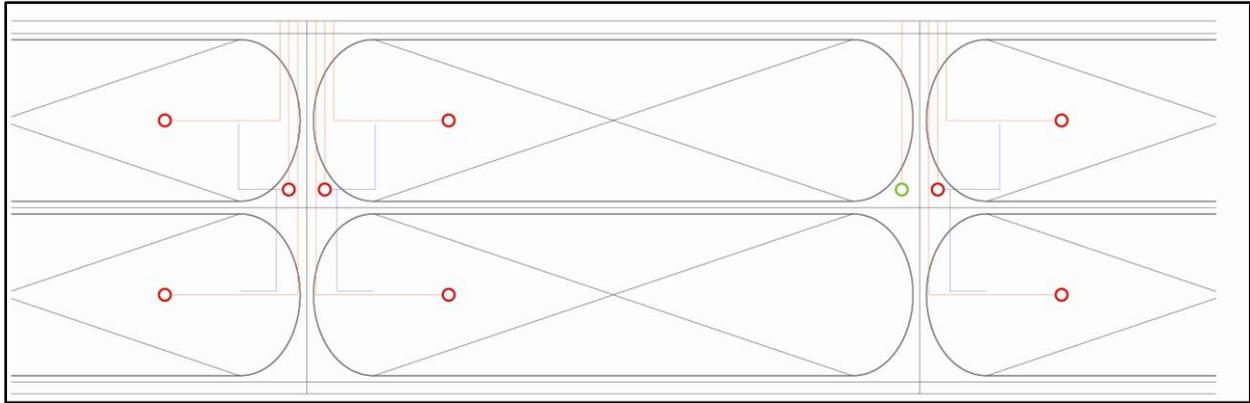


Figure 21: Arrangement of pumps

There is one pump designated to each tank. Every two tanks share one additional pump that is situated in a small sump. The tanks are connected with separate pipelines to the pump. All pumps are connected in parallel to obtain the required total capacity. The tenth pump, which is marked with green, is connected to all tanks, and is used in case of failure in one of the other pumps.

6.4 Heat Gain and Boil-off

Liquefaction and re-gasification will be done at onshore facilities.

An important aspect is how the cargo is kept at the desired phase during transit. The cargo will have a certain heat gain from the surroundings, thus changing the phase of some of the contents into vapour. The increased required volume from this gas will in turn increase the internal pressure. This might compromise the integrity of the tank.

This can be prevented by constantly cooling the cargo and/or recompressing the vapour. This requires special equipment. An alternative is to allow for boil-off to be released into the atmosphere, thus reducing the pressure and maintaining the right temperature. This last option does not require any supplementary equipment

To choose between these two options the liquefied cargo heat gain must be investigated further.

6.4.1 Release of Boil-off

For transit time less than a couple of days coolers and compressors are not needed due to low heat gain. This will, however, require that some cargo is released into the atmosphere to reduce the pressure. This should be done by approximately 0.3 bar per day (26). The scheduled transit time is 30 hours, and pressure loss will be negligible.

6.4.2 Cooling/Compressing Boil-off

A more detailed analysis of the thermal properties of liquid CO₂ has been done in the chapter for Electric Load Analysis. The result shows that 180 kW are needed for cooling of the cargo.

6.4.3 Boil-off Handling Selection

This ship is to be environmentally friendly to ensure the fulfilment of the mission description. Therefore, more CO₂ emissions are to be avoided, and the cargo is to be subject to cooling/compression.

7 Initial Definition and Sizing

The cargo capacity and ship type have now been established. The initial determination of main dimensions and tank sizing will in this chapter be decided. This is more easily done with a suitable comparison ship. There are no existing ships with a capacity near 40 000 cubic metres that is transporting liquefied gas in the same condition. Therefore, a ship with the same estimated deadweight is used. Other dimensions have been derived from this.

The 75 000 m³ class LPG/NH₃-carrier Clipper Victory (CV) from Solvang Shipping has been used in this study. CV transports LPG (propane) with a density of 580 kgm⁻³; thus having a payload of 43 500 tonnes. When transporting LCO₂, the payload of the vessel to be designed is 46 000 tonnes, and is therefore expected to have a similar deadweight and lightweight. This gives a similar weight displacement.

Data of CV given hereunder are with LPG as cargo.

LPP	[m]	217.00
Breadth	[m]	32.25
Depth	[m]	22.20
Draught	[m]	11.65
100% MCR	[kW]	12 270
C_B	[-]	0.86
Displacement	[tonnes]	72 005
Light ship weight	[tonnes]	17 957
Deadweight	[tonnes]	54 048

Table 15: Principal Particulars, CV

The transport condition of LPG is shown in the table below:

Temperature	[°C]	-42
Pressure	[bar]	1.23

Table 16: Transport Condition LPG

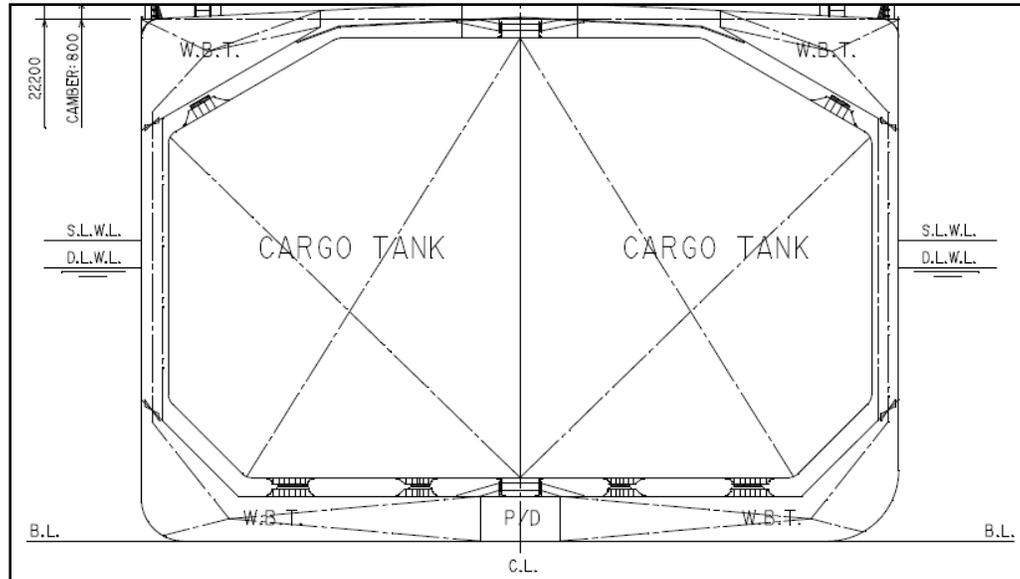


Figure 22: Mid ship section of Clipper Victory (23)

7.1 Criteria of Dimensions

7.1.1 Weight Displacement

The total weight displacement of a ship is divided into lightweight and deadweight. The payload, LCO_2 , is 46 000 tonnes. This is about 2 500 tonnes more than that of CV transporting LPG. The deadweight is therefore assumed to be equally higher. The displacement of the vessel is of this reason set to 75 000 tonnes.

7.1.2 Length Overall

The length of the ship should be no more than what is required to carry the necessary deadweight. The comparison ship suggests that the length should be over 200 meters. It cannot, however, exceed the harbour restrictions, which are 350 metres. (27)

7.1.3 Breadth

The necessary breadth (beam) must be in accordance with the desired length/breadth and breadth/draught-ratios. The beam should be large enough to secure stability. By investigating CV, the beam for the projected ship should be approximately in the range of 25-35 meters with the same length and draught to maintain optimized ship dimension ratios. The harbour restrictions deny the possibility of the breadth exceeding 55 metres. (27)

7.1.4 Draught

The maximum draught at Brunsbüttel is 13.8 meters (27). At Mongstad, there are different draughts for different jetties. The existing LPG-basin is only 8.4 meters deep, but the deepest basin, which is for

crude oil, is 25 meters (28). As there exist no infrastructure today regarding CO₂, it is assumed that new jetties for CO₂ and LPG will be made.

The draught will be less when transporting LPG as the density is roughly half of that of CO₂.

The draught of CV indicates that the harbour restrictions will not pose any problems to the vessel.

7.1.5 Number and Size of Tanks

The numbers of tanks will in this chapter be determined, and there are certain limitations to follow due to the choice of independent type “C” tanks. Short summarized, they are cylindrical in shape with the ability to withstand high inner pressure.

Previous calculations (29) (10) (30) indicate that the tank diameter should not exceed 14 meters because of the required plate thickness. The plate thickness should not, by recommendation by DNV (29), be more than 55 mm. The length of the tanks is limited by the damage stability. Many producers of independent tanks type “C” have not dared to exceed a tank volume of 4 000 – 5 000 cubic metres of this reason. The shipping company IM Skaugen has, on the other hand, a 10 000 cubic metres class LPG/Ethylene/VCM carrier. They have recently implemented tanks of the size 6 000 m³ which are 50 meter long (30). These tanks may also be able to transport LCO₂.

Total tank volume is 40 000 cubic metres. At this point, two aspects must be considered: The costs, and utilization factor of the cargo cross section area. As tanks of this type usually are custom made, there are no prices available. It is assumed that the tank size/cost ratio will decrease with an increase of size. This implies that the costs of one large tank are less than that of two smaller with the same total size. This argument indicates that as few tanks as possible, within the above-mentioned limitations of damage stability, diameter and plate thickness, should be used. For simplicity, all tanks are of the same size.

This gives one favourable result. The tank properties are summarized in the table below.

Number of tanks		6
Length, total	[m]	50
Length of cylinder	[m]	40
Diameter, inner	[m]	13.5
Single tank volume	[m ³]	6 680
Total tank volume	[m ³]	40 080

Table 17: Tank properties

It is implied here that the ends of the tanks are not spherical, but ellipsoidal. This results in more efficient use of cargo space, but somewhat higher loads to the steel. This is explained further later in the report.

7.2 Hull Design

7.2.1 Stepwise Development

The development of the hull is based on a spreadsheet containing several different combinations of ship dimensions that makes it possible to transport 46 000 tonnes of LCO₂ with the current tank dimensions. The better alternative found here is selected for further development.

The length and breadth are determined by tank arrangements. Free spaces around all tanks are included. They are set to one and one and a half for latitudinal and longitudinal directions respectively (DNV Pt.5 Ch.5). With the length of the cargo space established, those of the fore and aft sections of the vessel were established from CV. This is dependent on both breadth and block coefficient as a streamlined form is desired. Rough estimates are done here out of simplicity. They are approximately about 40 percent of total tank length, and LPP is found. The Loa is found by multiplying with a factor of 1.03 (13). The block coefficient and draught is adjusted to ensure the correct displacement of 75 000 tonnes. The block coefficient is manipulated to be somewhere near that of CV, and therefore the draught is set to 13 meters in all except in version IV, where it is set to 20 meters.

The LPP can also be changed, but out of simplicity it was decided that only two parameters should be variables. The table and figure below show the different alternatives. The density of water is set to 1025 kgm⁻³ (32).

The total outer length of each tank is 50 metres, plus insulation and plate thickness. The space between tanks is set to one meter.

		I	II	III	IV	V
Total tank length	[m]	310	104	155	104	155
Total tank breadth	[m]	15,0	46,0	30,0	30,0	30,0
Lpp (L)	[m]	434,0	145,6	217,0	145,6	217,0
Loa	[m]	447,0	150,0	223,5	150,0	223,5
Breadth (B)	[m]	16	48	31	31	31
Draught (T)	[m]	13,0	13,0	13,0	20,0	13,0
C_B	[-]	0,8106	0,8054	0,8367	0,8106	0,8367
L/B	[-]	23,59	4,73	7,45	5,47	7,45
B/D	[-]	0,94	5,72	2,43	1,78	2,43
Displacement	[tonnes]	75 000	75 000	75 000	75 000	75 000

Table 18: Tank configuration selection

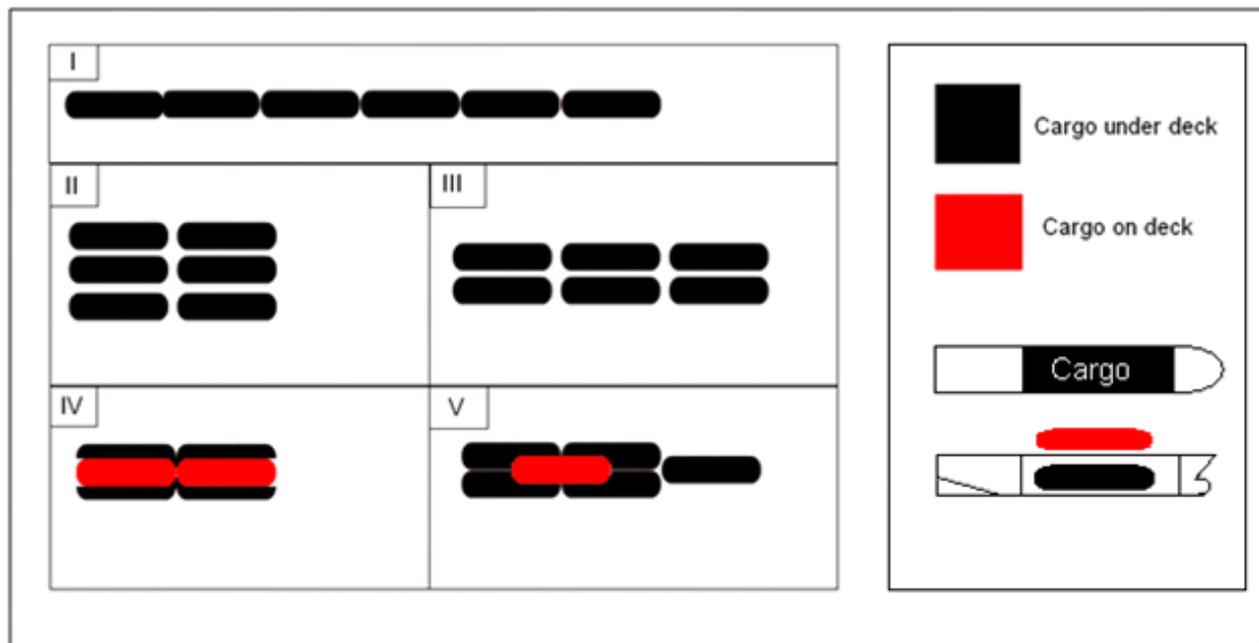


Figure 23: Possible tank configurations

7.2.1.1 *Alternative I*

It is immediately clear that the restrictions regarding length are exceeded, and this version is not considered in the selection process. From a structural point of view the risk of large torsion and bending forces is present.

7.2.1.2 *Alternative II*

The breadth does not exceed the harbour restrictions, but is nevertheless too large. The resistance relative to the other versions will be high. The resulting L/B ratio is unsatisfactory due to the small value.

7.2.1.3 Alternative III

There are no negative aspects seen from the table when comparing with the other alternatives. The L/B ratio suggests a slender hull form.

7.2.1.4 Alternative IV

The tanks should not be placed on deck due to issues with strength (32). Therefore, this alternative is not considered. The draught is also too large. If the length was enlarged instead the surface area would be relatively big and give a lot of friction resistance.

7.2.1.5 Alternative IV

This design is not considered further, of the same reasons as mentioned in alternative IV.

7.2.2 Selection of Alternatives

It is clear from the arguments above that alternative III is the better one, and this is selected for further development.

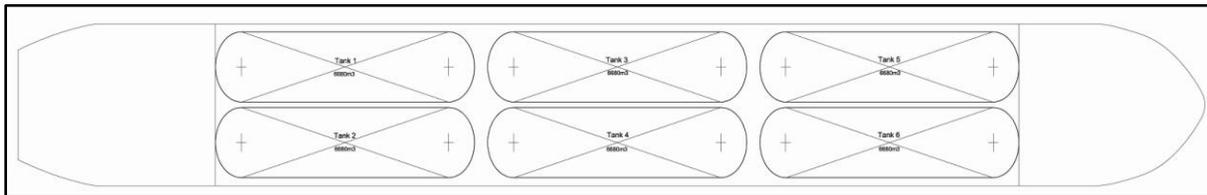


Figure 24: Tanks configuration

7.2.3 Selection of Hull Form

With the main dimensions from the previous subchapter some coefficients can now be calculated. In *System Based Ship Design* (19) many suggested form factors and other data related to the hull can be found.

These are given in graphs with the ship's Froude number as free variable.

With the Froude number of 0.182, the following coefficients were read off from graphs in *System Based Ship Design* (19):

$$C_p = 0.83$$

$$C_w = 0.93$$

$$C_m = 0.98$$

The recommended longitudinal centre of buoyancy (LCB) is given by the Froude number and the slenderness.

$$\text{Slenderness} = \frac{L_{wl}}{(\Delta/1.025)^{1/3}} = \frac{220}{(75000/1.025)^{1/3}} \approx 5.5$$

The displacement will approximately be the same as Clipper Victory, so the actual slenderness will not be much altered.

The recommended LCB can now be found. LCB is two percent of the length of LPP from the centre line (CL). Positive value indicates that the point is located towards the bow.

The properties of the preliminary edition are presented in the following table.

Loa	[m]	223,5
Lpp (L)	[m]	217,0
Breadth (B)	[m]	31
Draught (T)	[m]	13
Displacement	[tonnes]	75 000
Slenderness	[-]	5,5
C_B	[-]	0,8367
LCB	[m]	+4.62
C_m	[-]	0.98
C_p	[-]	0.83
C_w	[-]	0.93

Table 19: Characteristics of preliminary version

7.2.4 Final Ship Version

After having finished one turn of the design spiral, some of the preliminary assumptions were found to be wrong. One error was the presumed displacement of 75 000 tonnes. The weight displacement was found to be 70 000 tonnes from weight calculations. This resulted in some changes. The draught is adjusted to 12.1 meters and LPP is increased to 220 meters. The C_B is now 0.828 so that the deadweight equals 70 000 tonnes.

The length over all has been increased after the work with the general arrangements. This length is now 228 meters. This is described in a separate chapter later in the report.

The midship coefficient C_m , is increased due to the wanted slenderness in the stern (33). The new C_m is approximately 0.9956.



Figure 25: Illustration of final ship version

7.2.4.1 *Principal Characteristics*

Principal Characteristics		
LOA	[m]	228
LPP	[m]	220
LWL	[m]	225
Breadth	[m]	31
Draught	[m]	12.1
Weight Displacement	[tonnes]	70 000
Volume Displacement	[m ³]	68 293

Table 20: Principal Characteristics

Tank Characteristic		
Length, inner	[m]	50
Diameter, inner	[m]	13.5
Maximum plate thickness	[mm]	39
Total cargo space length	[m]	155
Number of tanks	[-]	6
Tank volume	[m ³]	6 680
Total tank volume	[m ³]	40 080

Table 21: Tank Characteristics

Coefficients		
Block coefficient (C_B)	[-]	0.83
Water line coefficient (C_{wl})	[-]	0.92
Midship coefficient (C_m)	[-]	0.996
Prismatic coefficient (C_p)	[-]	0.83

Table 22: Coefficients

8 Function Analysis

The function analysis defines all systems needed in the vessel to perform the tasks demanded in the mission statement and divide them into payload and ship related functions. All spaces are defined and Gross Volume is calculated. The calculations are based in general on coefficients found in "System Based Ship Design" (34) and the comparison vessel (35).

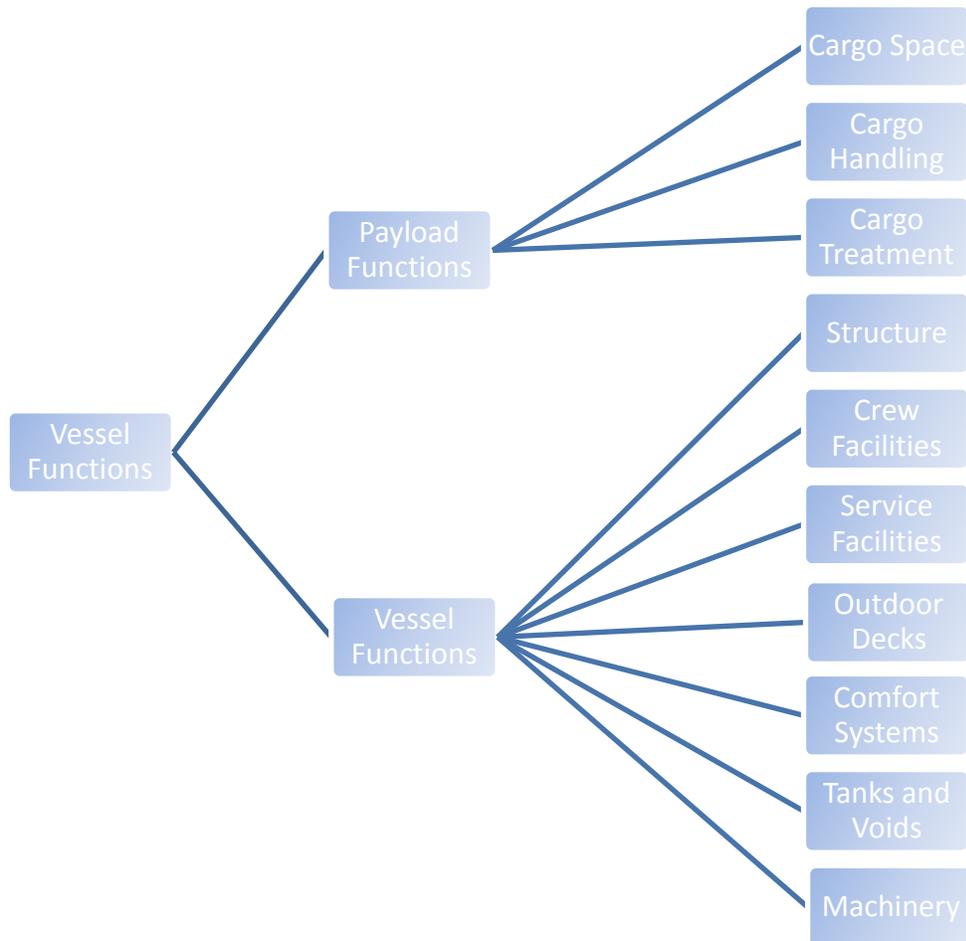


Figure 26: Vessel Functions

It has been a challenge to find a comparison ship which can give accurate coefficients to use in the *System Based Ship Design*. However, since this ship will have a lot in common with ordinary double hull tankers, a comparison ship that has the same deadweight will be a good starting point.

8.1 Ship Service Facilities

The service spaces define areas needed for the ship operation. They are calculated on a basis of coefficients from *System Based Ship Design* (34). The service facilities areas are divided into four

subgroups: Ship service, catering spaces, hotel services and technical spaces in the accommodation. In addition to coefficients from *System Based Ship Design* (34) catering and hotel services are based on number of people on board.

SHIP SERVICE				
Name/ Use of Space	m ² per crew	Height [m]	Area [m ²]	Volume [m ³]
Wheelhouse	11	3	209	627
Offices	4	3	76	213
Sick Bay	3	3	49	138
Ship Service Spaces	18		334	978
CATERING SPACES				
Name/ Use of Space	m ² per crew	Height [m]	Area [m ²]	Volume [m ³]
Galleys	2	3	38	106
Provision Store	2	3	34	96
Garbage	1	3	19	53
Catering Spaces	5		91	255
HOTEL SERVICES				
Name/ Use of Space	m ² per crew	Height [m]	Area [m ²]	Volume [m ³]
Laundry and Linen Store	2	3	29	80
Hotel Store	3	3	57	160
Catering Service Spaces	5		86	239
TECHNICAL SPACES IN THE ACCOMMODATION				
Name/ Use of Space	m ² per crew	Height [m]	Area [m ²]	Volume [m ³]
Air conditioning rooms	4	3	68	192
Deck stores	4	3	76	213
Workshop	4	3	76	213
Catering Service Spaces	12		220	617
TOTAL SERVICE FACILITIES	39 m²/crew		732	2,090

Table 23: Service spaces

8.2 Crew Facilities

Crew facilities include cabins, mess, day room, gymnasium, hobby room, stairs and corridors. The areas are calculated based on the number and size of the cabins. Correction factors are used to determine the cabin corridors area and the wall lining area. The areas for mess- and dayrooms are based on numbers of seats in each space. There are some differences in size and standards based on ranking of the crew. There will be 15 persons on board most of the year. In addition a repair crew of three will be on board for about a week each year.

CREW ACCOMMODATION						
Cabin Category:	No of Cabins	Beds per Cabin	Size [m ²]	Height [m]	Area [m ²]	Volume [m ³]
Captain Suite	1	1	33.0	2.8	33.0	92.4
Officers and Engineers	4	1	20.0	2.8	80.0	224.0
Crew	11	1	12.8	2.8	140.8	394.2
Repair crew	1	3	20.0	2.8	20.0	56.0
Total crew	17	19	16 m²/crew		273.8	766.6
Cabin corridors, wall lining	30% of cabin area			2.8	82.1	230.0
Crew Cabin Area	17	19	20.9 m²/crew		355.9	996.6
CREW COMMON SPACES						
Name/ Use of Space:	Seats	m ² per seat	m ² per crew	Height [m]	Area [m ²]	Volume [m ³]
Officer Mess	5	5.5	1.4	2.8	34.7	97.3
Officer Dayroom	5	5.5	1.4	2.8	34.7	97.3
Crew Mess	14	2.2	1.6	2.8	53.5	149.8
Crew Dayroom	14	2.3	1.4	2.8	51.8	145.0
Gymnasium	0	0.0	3.0	2.8	57.0	159.6
Hobby	0	0.0	3.0	2.8	57.0	159.6
Crew Common Spaces			15.2 m²/crew		288.8	808.6
CREW AND EMERGENCY STAIRWAYS						
Name/ Use of Stair:	Decks	m ² per deck	m ² per crew	D-height [m]	Area [m ²]	Volume [m ³]
Main Stairs	6	18	5.7	2.8	108.0	302.4
Engine Room Stairs	2	15	1.6	3.2	30.0	96.0
Crew and Emergency Stairways			7.3 m²/crew		138.0	398.4
TOTAL CREW FACILITIES			43.4 m²/crew		782.7	2203.6

Table 24: Calculation of crew spaces

8.3 Cargo Spaces

The cargo spaces are dimensioned so that both CO₂ and LPG can be transported in the tanks. The arrangement and numbers of tanks have been set based on price consideration and time consumption of the alternatives.

TANK SPACES							
Tank type:	No of Tanks	Tank Dimensions			Cell guide width [m]	Area [m ²]	Volume [m ³]
		Length [m]	Breadth [m]	Height [m]			
Type "C":	6	50.0	13.5	13.5	1	4,050	54,675
							-
Average additional space needed per tank							
	6	6.0	14.5	14.5			7,569
Tank Spaces						4,050	62,244
CARGO RELATED SPACES							
Name/ Use of Space	No of Units	Unit Dimensions			Cell guide width [m]	Area [m ²]	Volume [m ³]
		Length [m]	Breadth [m]	Height [m]			
Pumps	10	1	1	15	4	150	150
Heater/Compressor	5	3	2	2	4	20	60
Pipings		5,000	0.1	0.1		500	50
Cargo Related Spaces						670	260
TOTAL CARGO SPACES						4,720	62,504

Table 25: Volume and area of tanks and cargo related spaces

8.4 Technical Facilities

Calculations of tank volumes are based on coefficients from *System Based Ship Design* (34) .

MACHINERY, SPEED AND POWER			
Machinery Type:	Medium Speed Lean Burn Gas Engine		
No of Propellers	1		
Operating Scedhule	Trial Condition	Service Condition	On/Offloading
Speed	18.17 knots	16.67 knots	0 knots
Propulsion Power	18,000 kW	15,500 kW	- kW
Load Factor	100 %	86 %	- %
Sea Margin	- %	- %	- %
Shaft Generators	- kW	- kW	- kW
Load Factor	100 %	- %	- %
Auxiliary Power	4,320 kW	2,255 kW	2,611 kW
Load Factor	100 %	52 %	60 %
Total Installed Power	22,320 kW		

Table 26: Machinery capacity and conditions

The Sea Margin is set to zero due to the fact that it has already been taken into account in the resistance calculations.

MACHINERY SPACES						
Name/ Use of Space:		m ² /kW	m ³ /kW	Height [m]	Area [m ²]	Volume [m ³]
Engine- and pump rooms			0.4	-	-	8,928.0
Steering gear		0.004	0.01	6.0	89.3	223.2
Bow thruster room		0.002	0.01	3.2	44.6	223.2
Switchboard rooms		0.002	0.01	2.8	44.6	223.2
Emergency generator, battery room		0.001	0	2.8	22.3	-
	Decks	m ² /deck				
Engine casing, air intakes	3	100.0		3.0	300.0	300.0
Funnel		50.0		10.0		500.0
Machinery Spaces			0.21 m ³ /DWT		500.9	10,397.6

Table 27: Machinery spaces

TANKS AND VOID SPACES						
Name/ Use of Space:	Consump. [g/kwh]	Consump. [tonnes/day]	Range [nm]	Endurance [days]	Margin factor	Volume [m ³]
Fuel	155.1	66.1	5,000	12.5	1.2	1,953.6
Lub Oil	1.5	0.6	5,000	12.5	4.0	13.3
	l/crew/day					
Fresh water	200	3.8		40.0	1.2	182.4
Sewage Holding	75	1.4		40.0	1.2	68.4
BW, side tanks						5,880.0
BW, double bottom						10,690.0
BW, for						4,500.0
VOIDS etc						1,000.0
Tanks and Void Spaces				0.49 m³/DWT		24,287.7

Table 28: Tanks and void spaces

OUTDOOR DECK SPACES						
Name/ Use of Deck:	Length [m]	Breadth [m]	m ² per crew	Covered [%]	Area [m ²]	Volume [m ³]
Mooring deck fore	15	25.0	19.7	100%	375.0	-
Mooring deck aft	10	32.0	16.8	0	320.0	-
Crew deck			2	50%	38.0	57.0
Outdoor Deck Spaces			38.6 m²/crew		358.0	57.0

Table 29: Outdoor deck spaces

8.5 Volume Summary

SPACE ALLOCATION				
	m ² /DWT	m ³ /DWT	Area [m ²]	Volume [m ³]
Tank Space			4,050	62,244
Cargo Related Spaces			670	260
TOTAL CARGO SPACES			4,720	62,504
	m ² /crew	m ³ /crew		
Crew Facilities	41	116	783	2,204
Ship Service	18	51	334	978
Catering	5	13	91	255
Hotel Service	5	13	86	239
TOTAL FURNISHED SPACES	68	193	1,294	3,676
Technical spaces in the accommodation	12	32	220	617
TOTAL INTERIOR SPACES	80	226	1,514	4,294
	m ² /kW	m ³ /kW		
Machinery spaces	0.022	0.466	501	10,398
Switchboard rooms, emergency gen., battery room	0.003	0.010	67	223
Engine casing and funnel	0.013	0.036	300	800
TOTAL TECHNICAL SPACES	0.039	0.512	868	11,421
		m ³ /DWT		
TANKS AND VOID SPACES	-	0.527	-	24,288
	m ² /crew	m ³ /crew		
OUTDOOR DECK SPACES	19	3	358	57
		m ³ /crew		
GROSS VOLUME		2.225		102,563
		GT/DWT		GT
GROSS REGISTER TONNAGE		0.668		30,791

Table 30: Volume Summary

8.6 Lightweight and Deadweight

The ship's weight is divided into lightweight and deadweight. The coefficients used during the lightweight calculations are gathered from *System Based Ship Design* (34). The deadweight calculations are based on actual weights collected and is calculated for both LPG and LCO₂ load condition.

LIGHTWEIGHT				
Weight Group:	Unit	Value	Coeff tonne/unit	Weight tonne
Payload related	Gross Tonnage	30,791.5 GT	0.0025	77
	Pumps number	20.0 -	2.88	58
	Pipings length	5,000.0 m	0.25	1,250
	Tanks tank vol	40,080.0	0.1	4,008
Total Cargo related				5,393
Hull Structure	Hull vol	97,469.5 m ³	0.075	7,310
Deck House	Dh vol	5,093.6 m ³	0.065	331
Interior Outfitting	Area	1,293.8 m ²	0.28	362
Machinery	Pp+Pa	22,320.0 kW	0.105	2,344
Ship Outfitting	Gross Vol	102,563.1 m ³	0.0042	431
Total	Gross Vol	102,563.1 m ³	0.158	16,171
Reserve	%	5 %		809
LIGHTWEIGHT	Gross Vol	102,563.1 m ³	0.166	16,979

Table 31: Lightweight

DEADWEIGHT CO ₂				
Item:	Unit	Value	Coeff	Weight tonne
Hold Cargo	Capacity	46 092,0 ton	1	46 092
Deck Cargo	Capacity	1 843,7 ton	1	1 844
Crew	Persons	19,0 persons	0,10	2
Provision and Stores	Persons	19,0 persons	0,2	4
Fuel	Consumpt.	826,4 ton	1,2	992
Lubrication Oil	Consumpt.	8,0 ton	2,5	20
Fresh Water	Consumpt.	152,0 ton	1,2	182
Sewage in Holding Tanks	Produced	57,0 ton	0,3	17
Ballast Water for Trimming and Antiheeling				-
Ballast Water for Stability				-
DEADWEIGHT				49 152
DISPLACEMENT				66 132
DWT/Displacement				0,74

Table 32: Deadweight CO₂

The displacement was found to be 66 132 tonnes. There will be more equipment related to cargo handling that is not taken into account in these calculations. Since the ship lightweight seems low compared to that of Clipper Victory, the total displacements are corrected, and set to 70 000 and 45 000 tonnes respectively.

DEADWEIGHT LPG				
Item:	Unit	Value	Coeff	Weight tonne
Hold Cargo	Capacity	22 044 ton	1	22 044
Deck Cargo	Capacity	881,8 ton	1	882
Crew	Persons	19 persons	0,10	2
Provision and Stores	Persons	19 persons	0,2	4
FuelOil	Consumpt.	826,4 ton/trip	1,2	992
Lubrication Oil	Consumpt.	7,99 ton/trip	2,5	20
Fresh Water	Consumpt.	152,00 ton/trip	1,2	182
Sewage in Holding Tanks	Produced	57 ton/trip	0,3	17
Ballast Water for Trimming and Antiheeling				600
Ballast Water for Stability				-
DEADWEIGHT				24 743
DISPLACEMENT				41 722
DWT/Displacement				0,59

Table 33: Deadweight LPG

The displacement is assumed to be higher, as mentioned above.

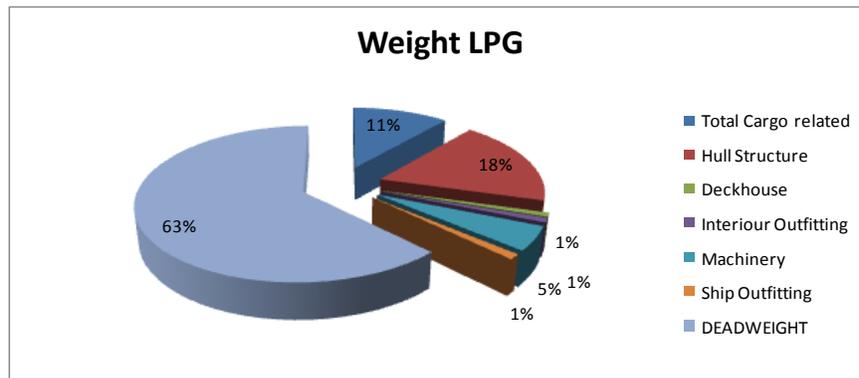


Figure 27: Percent weight distribution when transporting LPG

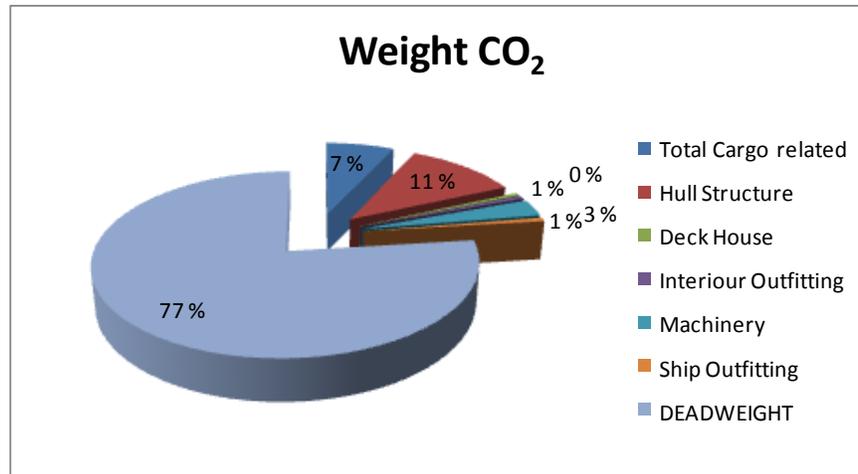


Figure 28: Percent weight distribution when transporting CO₂

8.7 Centre of Gravity

Weight Group:	Weight ton	Center of gravity		Moment tonxm	Center of gravity KG(m)	Moment tonxm
		KG/D	KG(m)			
LIGHTWEIGHT						
Payload related:						
Pipings	1,250.0	1.6	19.0	23,750.0	-	-
Pumps	57.6	0.4	5.0	288.0	-	-
Cargo tank 1+2	1,336.0	0.5	10.3	13,734.1	-48.0	-64,128.0
Cargo tank 3+4	1,336.0	0.5	9.3	12,398.1	3.0	4,008.0
Cargo tank 5+6	1,336.0	0.5	9.3	12,398.1	54.0	72,144.0
Hull structure	7,310.2	0.5	10.4	76,026.2	5.0	36,551.1
Deckhouse	331.1	1.4	27.0	8,939.2	-85.0	-28,142.0
Interior Outfitting	362.3	1.4	27.0	9,781.2	-90.0	-32,604.0
Machinery	2,343.6	0.4	7.5	17,553.6	-80.0	-187,488.0
Ship Outfitting	430.8	1.2	22.0	9,476.8	-	-
TOTAL	16,093.5	0.6	11.5	184,345.3	-12.4	-199,659.0
Reserve	808.5					
LIGHTWEIGHT	16,902.1	0.6	11.5	184,345.3	-12.4	-199,659.0

Table 34: Centre of gravity of the lightweight

DEADWEIGHT (CO ₂)	Weight ton	Center of gravity		Moment tonxm	Center of gravity KG(m) x	Moment tonxm
		KG/D	KG(m)			
Weight Group:						
Hold Cargo	46,092.0	0.4	9.6	443,097.8	3.0	138,276.0
Deck Cargo	1,843.7	0.7	15.0	27,655.2	-40.0	-73,747.2
Crew	1.9	1.2	27.0	51.3	-85.0	-161.5
Provision & Stores	3.8	1.1	23.7	89.9	-80.0	-304.0
Fuel Oil	991.6	0.3	6.8	6,693.6	-90.0	-89,247.9
Lub Oil	20.0	0.0	1.0	20.0	-90.0	-1,797.7
Fresh Water	182.4	0.2	5.0	912.0	-60.0	-10,944.0
Sewage in Holding Tanks	17.1	0.0	1.0	17.1	-90.0	-1,539.0
Ballast Water for Trimming and Antiheeling	-	-	-	-	-115.0	-
Ballast Water for Stability	-	-	-	-	-	-
DEADWEIGHT	49,152.5	0.4	9.7	478,536.8	-0.8	-39,465.4
LEIGHTWEIGHT + DEADWEIGHT	66,054.6	0.5	10.0	662,882.1	-3.6	-239,124.3

Table 35: Centre of gravity of deadweight CO₂

DEADWEIGHT (LPG)	Weight ton	Center of gravity		Moment tonxm	Center of gravity KG(m)	Moment tonxm
		KG/D	KG(m)			
Weight Group:						
Hold Cargo	22,044.0	0.4	9.6	211,916.3	3.0	66,132.0
Deck Cargo	1,843.7	0.7	15.0	27,655.2	-40.0	-73,747.2
Crew	1.9	1.2	27.0	51.3	-85.0	-161.5
Provision & Stores	3.8	1.1	23.7	89.9	-80.0	-304.0
Fuel Oil	991.6	0.3	6.8	6,693.6	-90.0	-89,247.9
Lub Oil	20.0	0.0	1.0	20.0	-90.0	-1,797.7
Fresh Water	182.4	0.2	5.0	912.0	-60.0	-10,944.0
Sewage in Holding Tanks	17.1	0.0	1.0	17.1	-90.0	-1,539.0
Ballast Water for Trimming and Antiheeling	-	0.0	1.0	-	-115.0	-
Ballast Water for Stability	-	0.5	12.0	-	12.0	-
	-	0.5	12.0	-	12.0	-
DEADWEIGHT	25,104.5	0.4	9.9	247,355.4	-4.4	-111,609.4
LEIGHTWEIGHT + DEADWEIGHT	42,006.6	0.5	10.3	431,700.6	-7.4	-311,268.3

Table 36: Centre of gravity of deadweight LPG

SHIP STABILITY FOR SELECTED DIMENSIONS		
Centerer of Buoyancy	KB	6,5 m
Transverse Metacenter	BM	6,4 m
Metacenter above keel	KM	12,9 m
INITIAL STABILITY	GM	2,9 m

Table 37: Initial stability for CO₂

SHIP STABILITY FOR SELECTED DIMENSIONS		
Centerer of Buoyancy	KB	4,5 m
Transverse Metacenter	BM	8,6 m
Metacenter above keel	KM	13,1 m
INITIAL STABILITY	GM	2,8 m

Table 38: Initial stability for LPG

9 Ship Drawings

In this chapter line drawings and the general arrangement of the vessel will be represented. The drawings are made in AutoCad 2009 with assistance from MARINTEK by Erik Lehn, Lars Øien and Erik Henney (36). Section drawings, water line drawings and buttocks, drawings of the mid ship section, the superstructure, the decks and the machinery decks are represented below in separate chapters.

9.1 Hull Design and Line Drawings

In the development of the line drawings, the mid ship coefficient was adjusted up to 0.9956. With this coefficient the ship could be fitted with a rather blunt bow and a slimmer stern. The slim stern gives less drag and better flow to the propeller, while the blunt bow helps obtain the correct block coefficient. In addition it does not give as much drag as a larger stern. This results in an increase of the length of the ship. The LOA was corrected to 228 meter, and LPP to 220 meters.

In the design process of the hull, the use of a generalized SAC-curve (19) for a tanker with the same block coefficient and the breadth, draught and length of the ship as well as line drawings from similar ships (35), the section drawings could be drawn. From these drawings the waterlines and buttocks could be drawn in accordance with the sections.

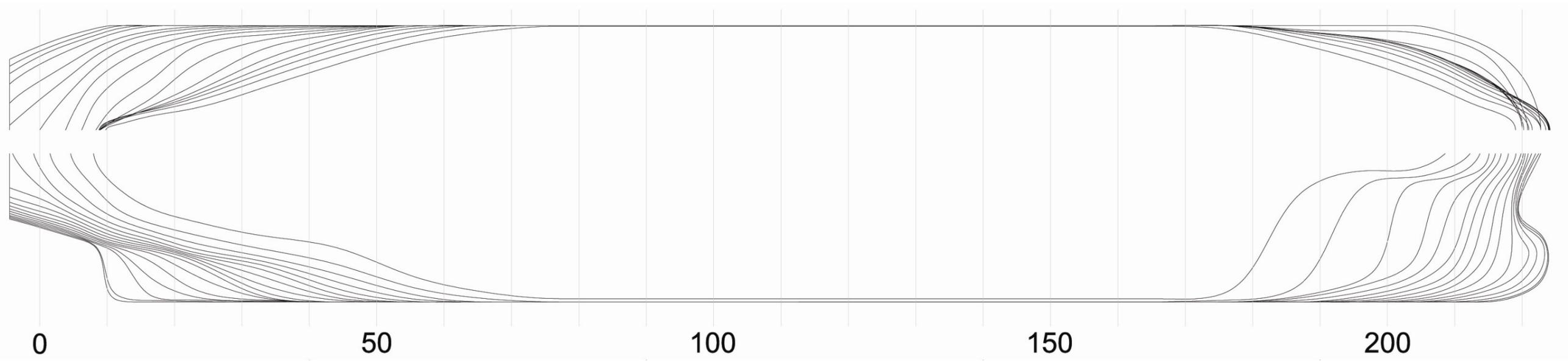


Figure 29: Waterlines and buttocks

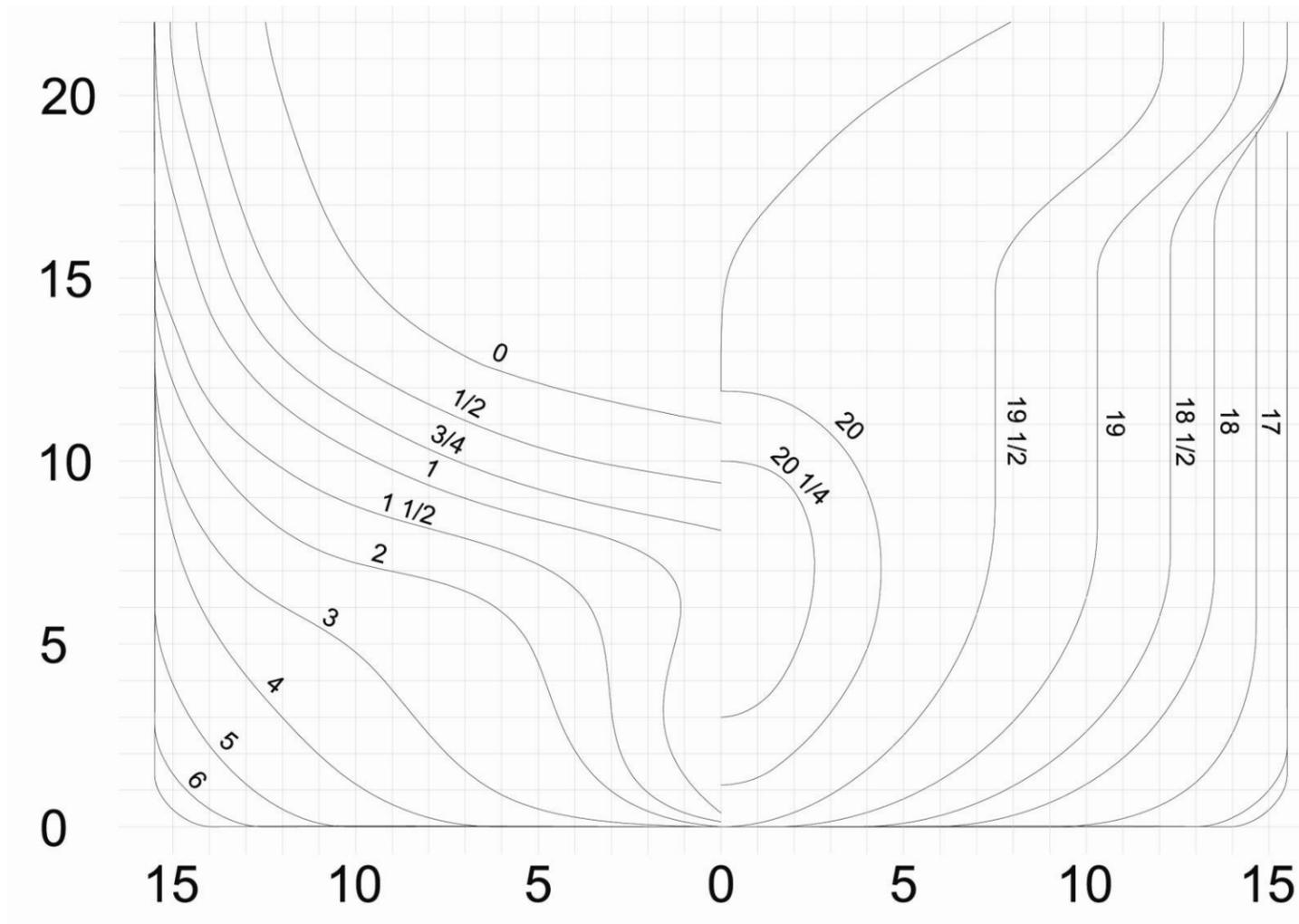


Figure 30: Sections

9.2 SAC and WAC

The Section Area Curve (SAC) represent the area of each section from the line drawings under the waterline. It can be used to find out where the buoyancy of the ship is located.

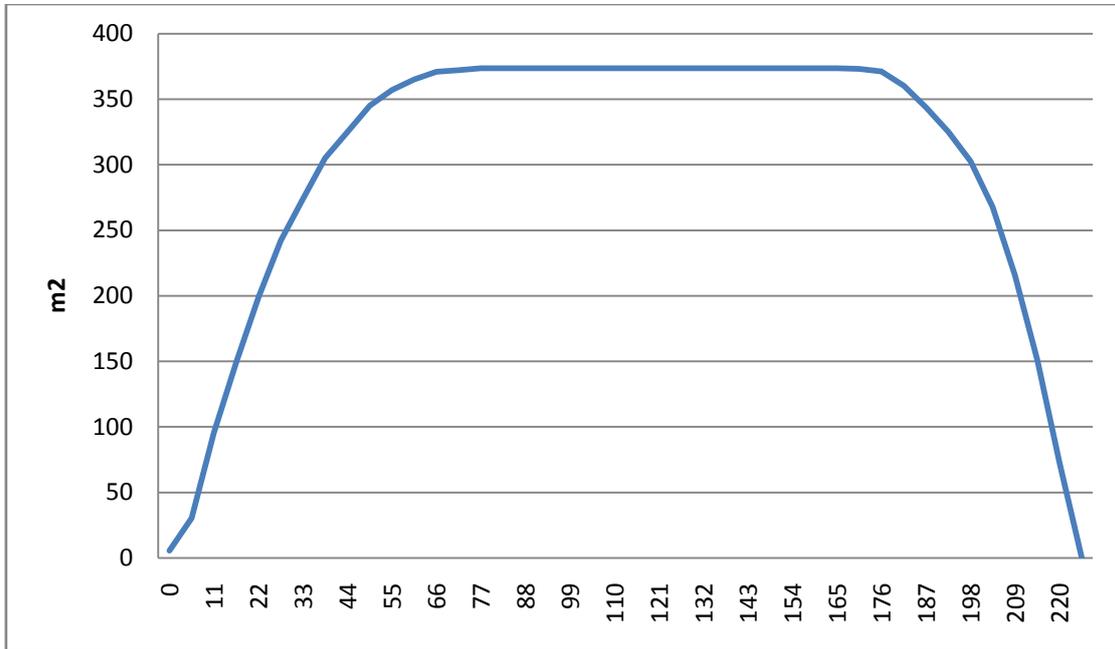


Figure 31: SAC

The Waterline Area Curve (WAC) represents the area of the waterlines of different depths.

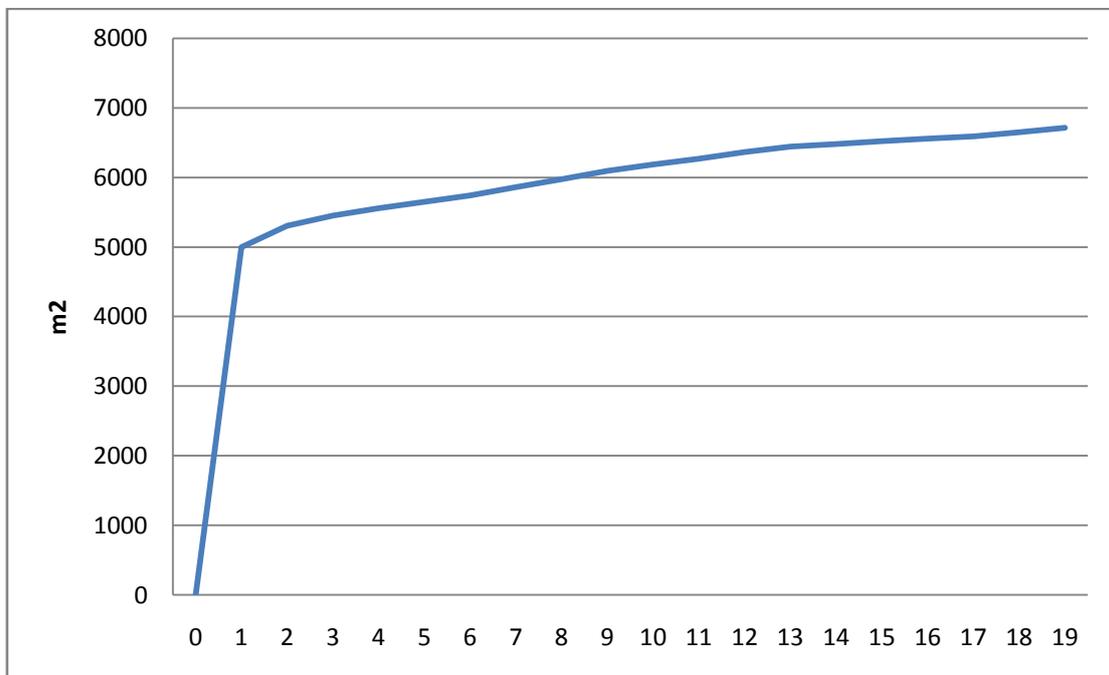


Figure 32: WAC

9.3 General Arrangement

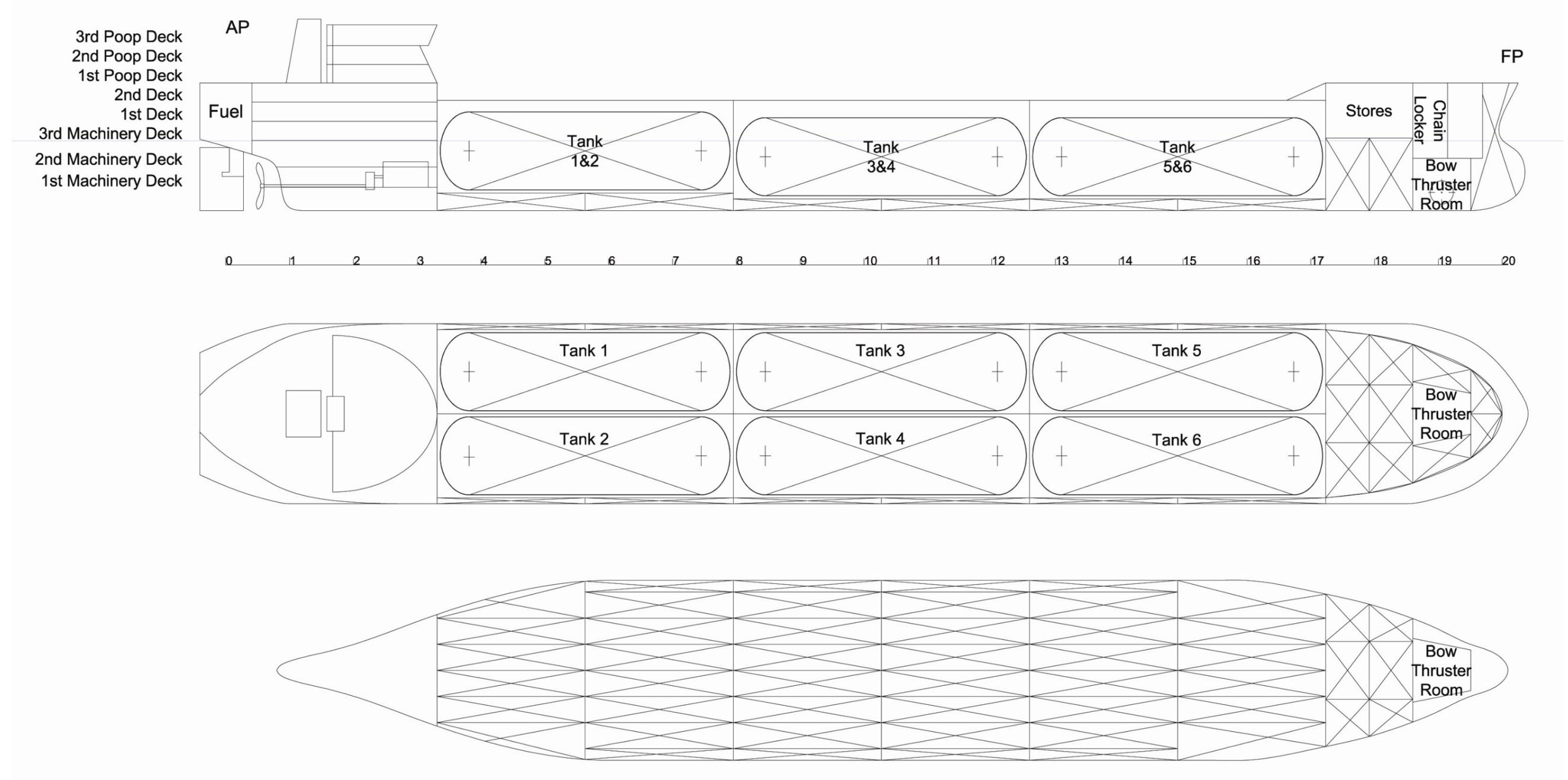


Figure 33: General arrangement from the side, top and tank top

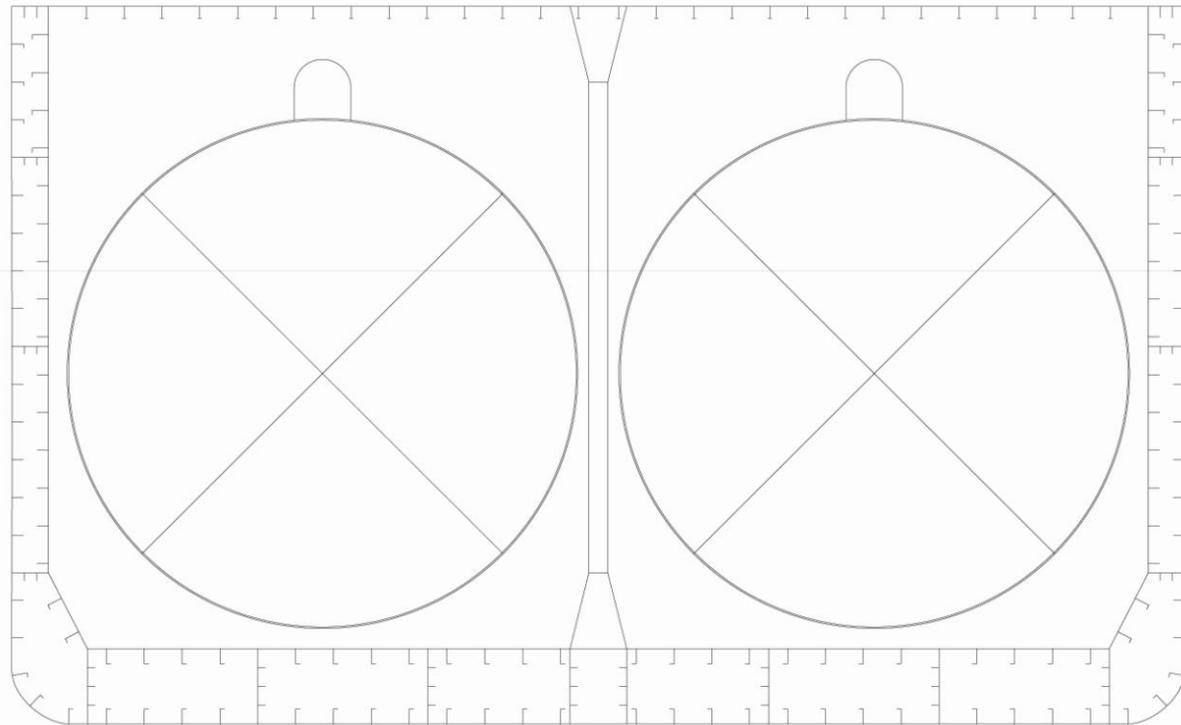


Figure 34: Mid ship section

The general arrangements have been drawn on the basis of the line drawings and the different area and volume requirements. Clipper Victory has been used as a comparison ship (35). All the drawings are made using AutoCAD 2009.

The ship is divided into three sections: Fore section, mid section and aft section.

9.4 Fore Section

In the fore section the bow thruster, deck stores, chain locker and ballast tanks is placed.

9.5 Mid Ship Section

The mid section consists of the pressure tanks, fuel tanks and the rest of the double hull with room for ballast water as well as bulkheads between the tanks. The tanks are placed with enough space between them and the walls to make it possible to inspect the tanks (31 s. Pt.5 Ch.5). The thickness of the insulation, which will be discussed later in this report, has been included in these calculations.

9.6 Aft Section

This section is parted into three: Superstructure, decks and machinery decks.

9.6.1 Superstructure

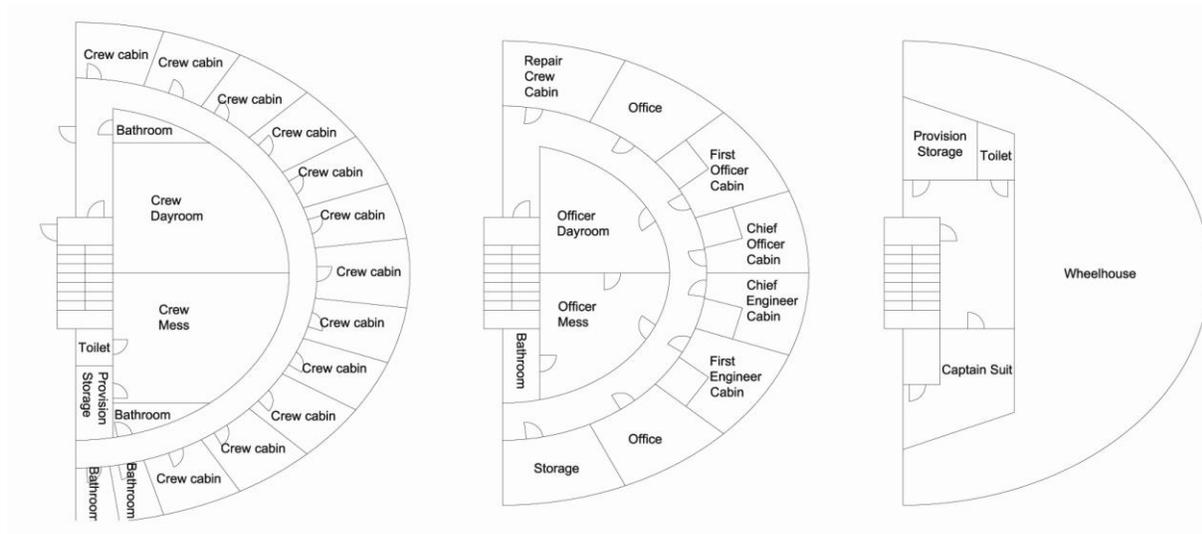


Figure 35: 1st, 2nd and 3rd poop deck

The superstructure contains the living quarters and the bridge. There are cabins for four officers, twelve crew members and three temporary repair crew members, which mean there is one extra bed if it should be needed, and four if the repair crew is not on board.

The wheelhouse has a 360 degrees vision with no large objects in the front of the ship. The field of vision in front is good enough with 333 meters blind zone in front with a waterline of 4.1 meters, all in accordance with the DNV Rules (31 s. Pt.6 Ch.17). Decks

These decks contain most of the ship service facilities such as sick bay, air conditioning, emergency generator and workshops.

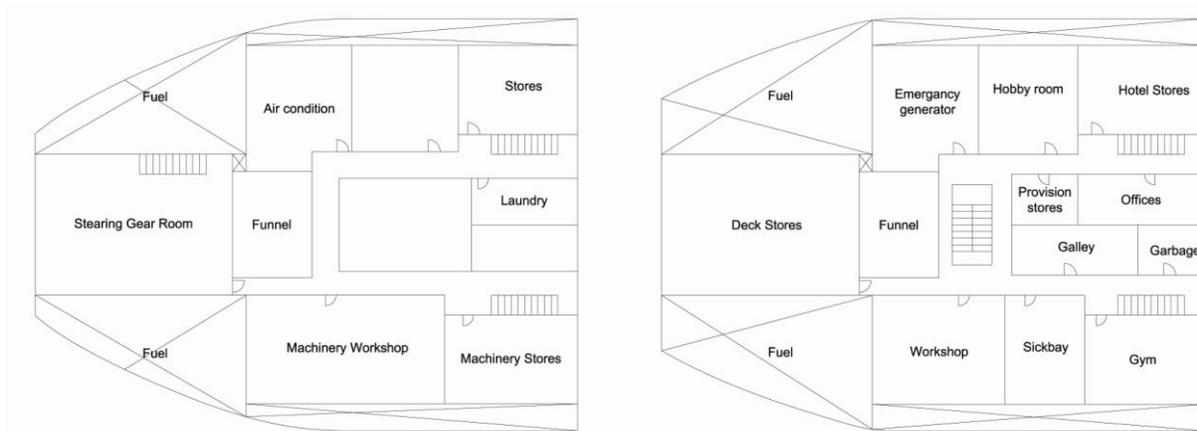


Figure 36: 1st and 2nd decks

9.6.2 Machinery Decks

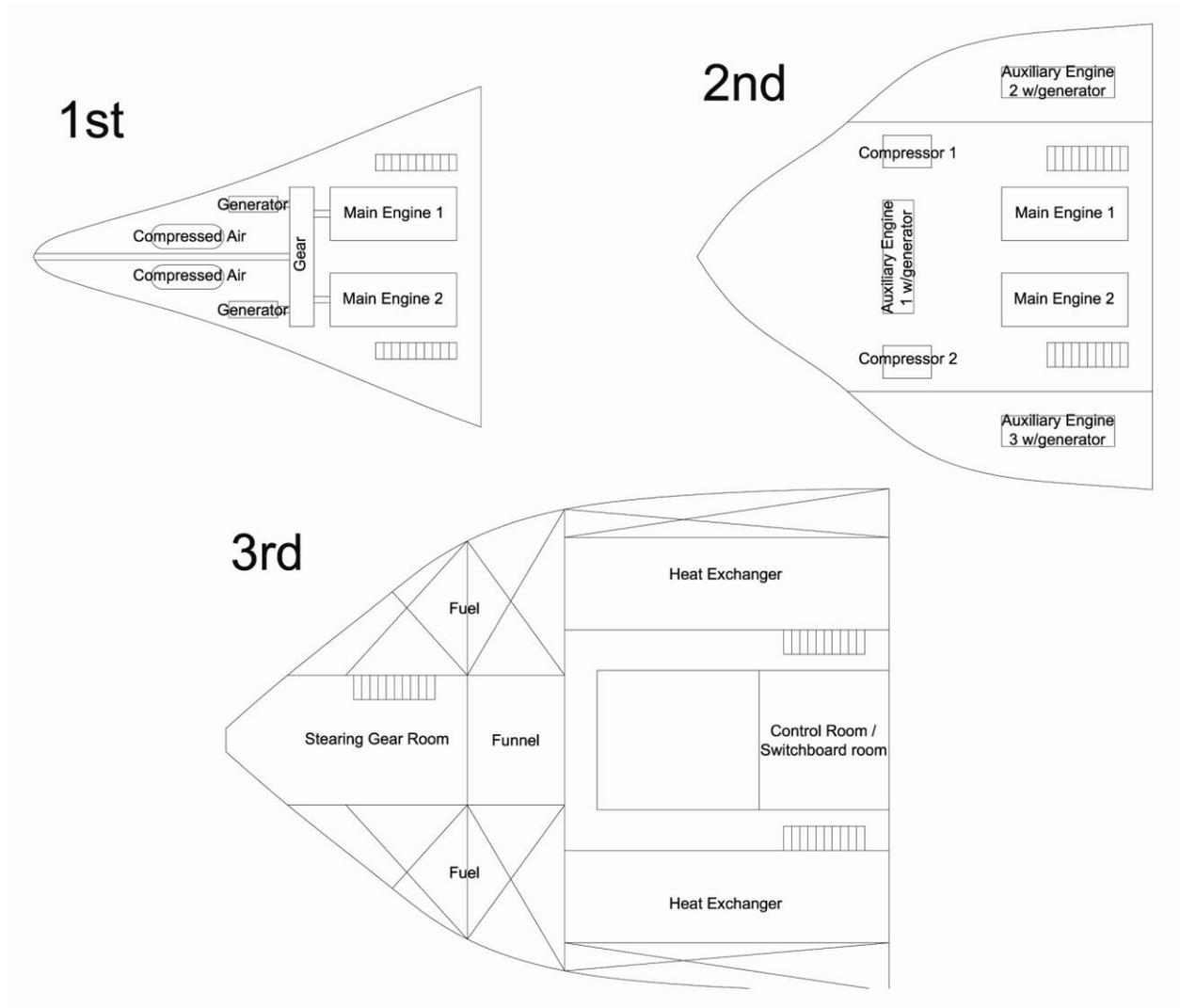


Figure 37: 1st, 2nd and 3rd machinery decks

The machinery space contains three decks. The main engines, the gear, generators and the compressed air are on the 1st machinery deck, while the 2nd deck contains the auxiliary engines with their generators and the compressors as well as the top of the main engine. The upper deck is mostly used for heat exchangers for the payload, and the machinery control room.

Because of lack of space at the bottom of the ship the engine shaft had to be placed about 1 meter above the propeller shaft. This problem has to be adjusted through the gearbox.

10 Stability and Trim

The ship transports two liquids with very different densities. This will cause different stabilities for the two different cases since both liquids are transported in the same tanks. It is therefore important to perform calculations for both conditions.

10.1 Principle

Stability of a vessel is found by evaluating the GZ-curve, which shows the restoring lever as a function of heeling angle. The restoring moment is given by the displacement multiplied by GZ. The GZ (ϕ) curve is given by:

$$GZ(\phi) = GM_T \cdot \sin(\phi) + MS(\phi), \text{ where}$$

MS = rest stability

GM_T = metacentre height above the centre of gravity

ϕ = heel angle.

10.2 Intact Stability

The GZ-curve has to meet the requirements from the Norwegian Maritime Directorate (37 s. 5§14) which are as follows:

- The area under the GZ-curve up to 30 degrees heel shall be at least 0.055 meter x radians, and at least 0.09 meter x radians up to 40 degrees heel angle.
- The area under the GZ-curve is to be at least 0.03 meter x radians between 30 degrees heel and 40 degrees heel.
- The restoring lever (GZ) has to be at least 0.2 meters at a heeling angle equal or greater than 30 degrees.
- The angle at which the restoring lever has it greatest value (GZ_{max}) shall never be less than 25 degrees.
- The initial metacentre height (GM) should be at least 0.15 meters.

To calculate the GZ -curve and other hydrostatic data the program HydroMax Pro has been used. A model of the ship was created in Maxsurf Pro and then imported to HydroMax. The intact GZ-curve was calculated for three main conditions; loaded with CO₂, loaded with LPG, and lightweight. The three GZ-curves are shown in the figure below.

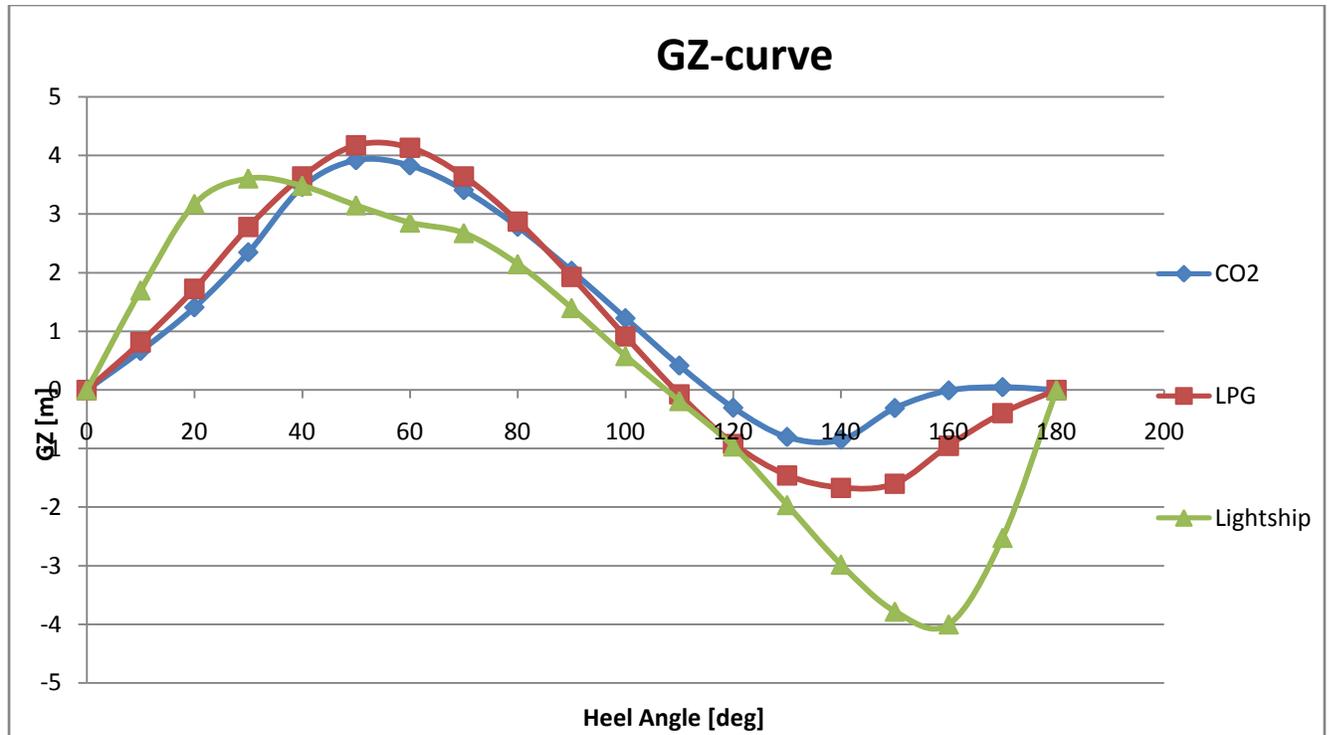


Figure 38: GZ-curve

From these curves we can calculate the values specified in the requirements from the Norwegian Maritime Directorate. The calculation results are shown in the table below.

Case	Draught	Area to 30 degree heel [m*rad]		Area to 40 degree heel [m*rad]		Initial stability (GM) [m]	
		Calculated	Demand	Calculated	Demand	Calculated	Demand
CO ₂	12.1	0.29	0.055	0.55	0.09	3.576	0.15
LPG	8.4	0.35	0.055	0.64	0.09	4.245	0.15
Light ship	4.1	0.61	0.055	0.92	0.09	6.191	0.15

Table 39: Stability calculations

The ship fulfils the given stability requirements.

10.3 Damage Stability

The damage stability is calculated to determine what kind of damage the ship can sustain and still have sufficient stability. In these calculations, HydroMax have been used. The program calculates damage stability by using the lost buoyancy method, which considers all damaged volumes not to contribute to buoyancy. Therefore the ship’s stability geometry will change for different damage conditions.

The ship is divided into watertight compartments as shown in the figure below. For each damage case, hydrostatic calculations are done to check the GM value, trim, draught and heel angle. All calculations are done when the ship is loaded with CO₂ since this will be the most critical load condition.

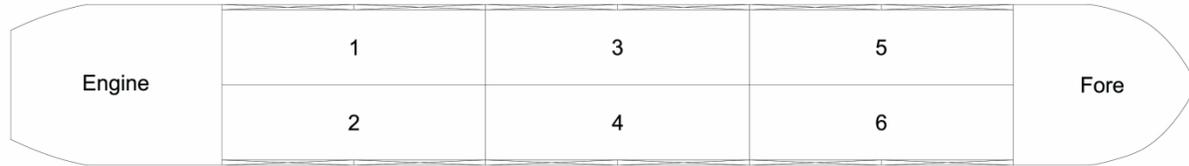


Figure 39: Compartment division

The different damage cases are given in the table below; in each case two adjacent compartments are flooded.

Flooded Compartments	Engine + 1	1+3	3+5	5+fore	1+2
Heel angle [Deg]	12.9	10.2	9	6.9	0
GMt [m]	3.6	3.5	3.5	3.6	3.6
GML [m]	323.8	338.6	338.6	241.2	328.2
Draught AP [m]	25.5	16.4	14.6	11.2	17.3
Draught FP [m]	4.6	9.7	11.3	17.6	8.8
Trim [deg]	5.5	1.8	0.9	-1.7	2.2

Table 40: Damage stability calculations

As seen from the table the ship has initial stability in all damage cases, and the damage stability is good.

10.4 Floodable Length

HydroMax calculates the floodable length curves for permeability values of 85%, 95% and 100%. It shows the maximum length along the ship that can be flooded without flooding the deck. Since the ship has its greatest displacement when transporting LCO₂ the analysis has only been carried out for this condition. The curve is used to control the compartment division. The ship is divided into compartments according to the figure above. The results of the simulations are shown in the figure below.

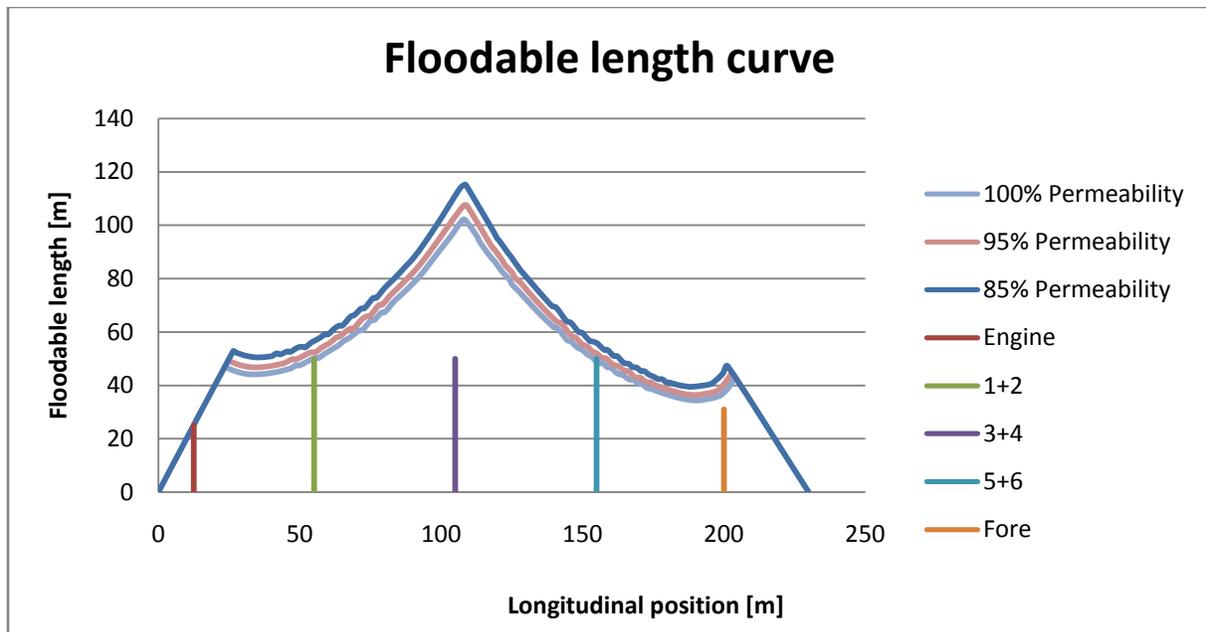


Figure 40: Floodable length curve

From the curves it can be seen that the compartment division is sufficient, this is also the shortest compartment division possible with the given tank arrangement. Compartment 1 and 3 intersects the 100% permeability line, but since the compartments are occupied by large tanks the permeability will be less.

10.5 Trim

Trim calculations are done for the three different conditions: CO₂, LPG and Lightweight. In the LPG load condition it is vital to ensure a sufficient submersion of the propeller. If this is not the case ballast water must be used to correct the trim. The calculations have been done in HydroMax Pro.

In the LPG load condition the initial draught at AP were only nine meters, the calculated sufficient propeller submersion suggested eleven meters draught at AP. 2 000 tones of ballast water at the stern was needed to correct the trim. The results are shown in the table below.

Case	Draft at FP [m]	Draft at AP [m]	Trim angle [deg]
CO ₂	10.13	13.16	0.8
LPG	5.45	10.96	1.5
Lightweight	2.92	5.38	0.7

Table 41: Trim calculations

The trim calculations now show a sufficient submersion of the propeller in the two transport cases. A sufficient submersion is not needed in the case where the ship is not loaded (light ship) as this only will be the case for dry docking.

11 Calm Water Resistance

11.1 Methods

When the speed and powering of the vessel are to be determined, the calm water resistance is central. This resistance can mainly be found from three sources.

- Analytical/numerical methods
- Empirical methods
- Model ship methods

Since this is an early stage of the design process, the use of model ship methods (towing tanks) cannot be done because of the high uncertainty in the hull form. This is expensive and should be investigated in a later stage of the design process. The second alternative is numerical methods based on the geometric form of the vessel and a potential flow method. This method requires more work and is omitted from this report.

The preferable methods at this stage are empirical methods that rely on a large number of previous model test results. The inputs are main dimensions and coefficients. Two methods are compared in this report.

- Hollenbach 98
- Holtrop 84

The calculations have been done in a computer program called ShipX, developed by MARINTEK. The draught has been that of CO₂ since this represent the largest resistance. At this early stage, we assume a mean sea water temperature at 6 degrees Celsius, no trim, and a transverse projected surface of 500 m³. The wetted surface is calculated by built-in formulas. Since the ship will have a bow thruster and bilge keel, the resistance includes appendage resistance of these.

11.2 Hollenbach's Method

This method is based on model trials in one of the oldest ship basins in the world. All data are from the Vienna Ship Model Basin and are taken from the years 1980 to 1995. The method is based on regression analysis of approximately 430 ships with varying main dimensions and form coefficients.

The method includes a larger amount of new hull forms compared to other methods that are based on older ship populations.

If the dimensions are outside the ranges shown in the figure below, the error must be expected to exceed five percent.

	'maximum' and 'mean'			'minimum'	
	single-screw		twin-screw	single-screw	twin-screw
	design draft	ballast draft		design draft	
Ship length L [m]	42.0 – 205.0	50.2 – 224.8	30.6 – 206.8	42.0 – 205.0	30.6 – 206.8
$L/\nabla^{1/3}$	4.49 – 6.01	5.45 – 7.05	4.41 – 7.27	4.49 – 6.01	4.41 – 7.27
C_B	0.60 – 0.83	0.56 – 0.79	0.51 – 0.78	0.60 – 0.83	0.51 – 0.78
L/B	4.71 – 7.11	4.95 – 6.62	3.96 – 7.13	4.71 – 7.11	3.96 – 7.13
B/T	1.99 – 4.00	2.97 – 6.12	2.31 – 6.11	1.99 – 4.00	2.31 – 6.11
L_{os}/L_{wl}	1.00 – 1.05	1.00 – 1.05	1.00 – 1.05	1.00 – 1.05	1.00 – 1.05
L_{wl}/L	1.00 – 1.06	0.95 – 1.00	1.00 – 1.07	1.00 – 1.06	1.00 – 1.07
D_P/T	0.43 – 0.84	0.66 – 1.05	0.50 – 0.86	0.43 – 0.84	0.50 – 0.86

Table 42: Valid ranges of Hollenbach's method (38)

The ship exceeds some of the ranges. The ship length is a bit more than maximum, and the block coefficient exceeds it slightly. This investigation shows that the ship is in the outer range of what the method is accurate for. A statistical variation of more than five percentages must be expected.

By calculations done in ShipX, the following results by Hollenbach's method were obtained.

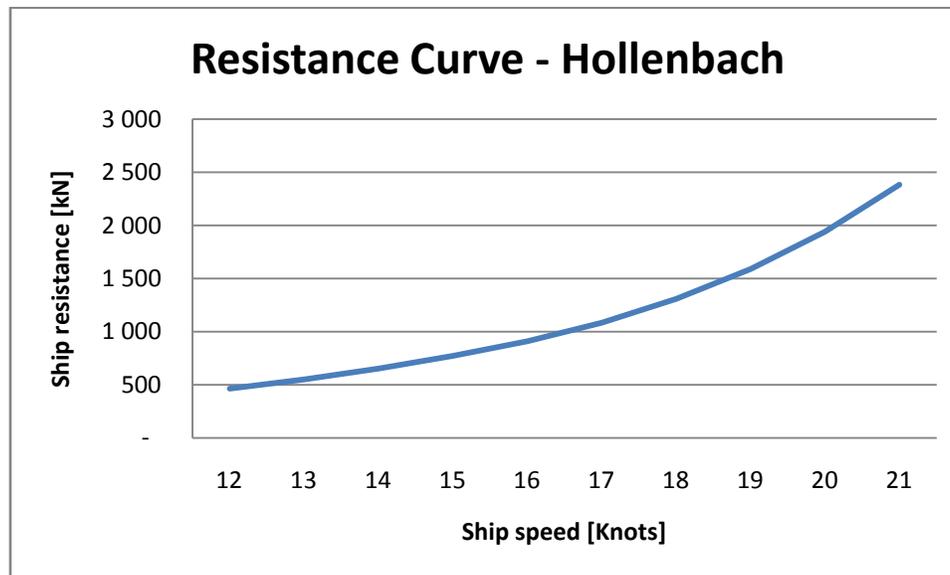


Figure 41: Ship resistance curve by Hollenbach's method (39)

The design speed at 16.7 knots gives a total resistance of 1 031 kN using Hollenbach's method.

11.3 Holtrop's Method

This method forms its basis on results from a more “old fashioned” selection of ships. It is based on regression analysis of historical data of approximately 300 ships with varying main dimensions and form coefficients. Older hull forms are not necessary as optimized in the hull resistance as new vessels. It can therefore be expected some higher results than by Hollenbach's method.

Another weakness by this method is the representation of bulbs. Old ship models were without bulbs and this influence the results.

The method was published by Holtrop and Mennen in 1978. The wave resistance was given by a simplification formula developed by Havelock (1951), but Holtrop transformed the formula and made it more convenient.

The total resistance is expressed as:

$$C_T = C_F(1 + k_1) + C_{app} + C_w + C_b + C_{tr} + C_{bto} + C_a$$

C(subscript)	Denotes
T	total
f	friction
app	appendage
w	wave
b	boulbous bow
tr	transom
bto	thruster opening
a	correlation resistance

Table 43: Explanation of resistance coefficients

The Holtrop method is most reliable within these areas of dimensions:

Ship	Fn (max)	Cp	L/B
Tankers, bulk carriers	0.24	0.73-0.85	5.1-7.1
General cargo	0.30	0.58-0.72	5.3-8.0
Fishing vessels, tugs	0.38	0.55-0.65	3.9-6.3
Container ships, frigates	0.45	0.55-0.67	6.0-9.5
Various	0.30	0.56-0.75	6.0-7.3
Series 64 hull forms			

Table 44: Valid ranges of Holtrop's method

The ship will be in the “Tankers” category. It is in the valid range of Froude number and the L/B ratio. The prismatic coefficient range does not include the ship’s prismatic coefficient. The results obtained will therefore not be accurate.

Calculation by Holtrop’s method in ShipX yields these results.

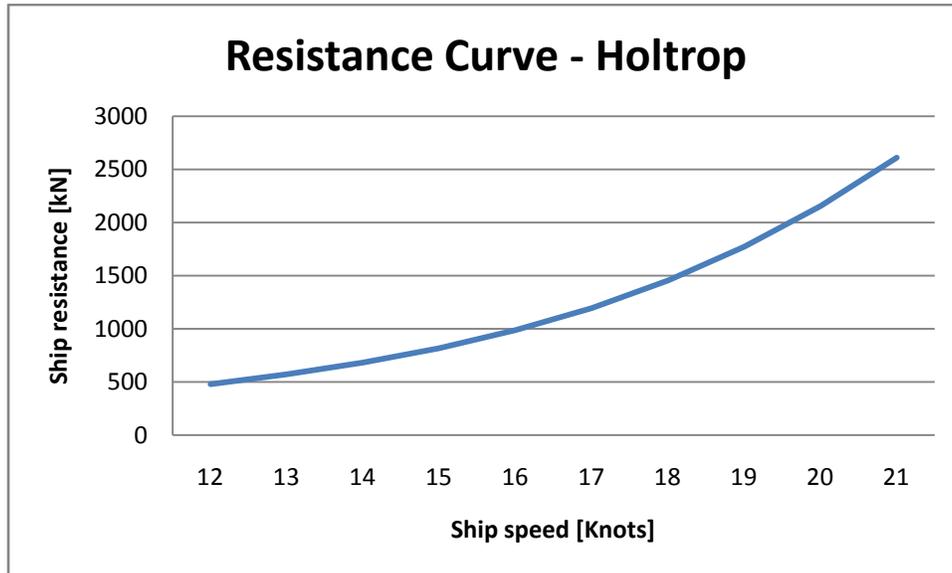


Figure 42: Ship resistance curve by Holtrop’s method (39)

Holtrop’s method results in a total resistance in calm water of approximately 1 131 kN at 16.7 knots.

11.4 Estimation

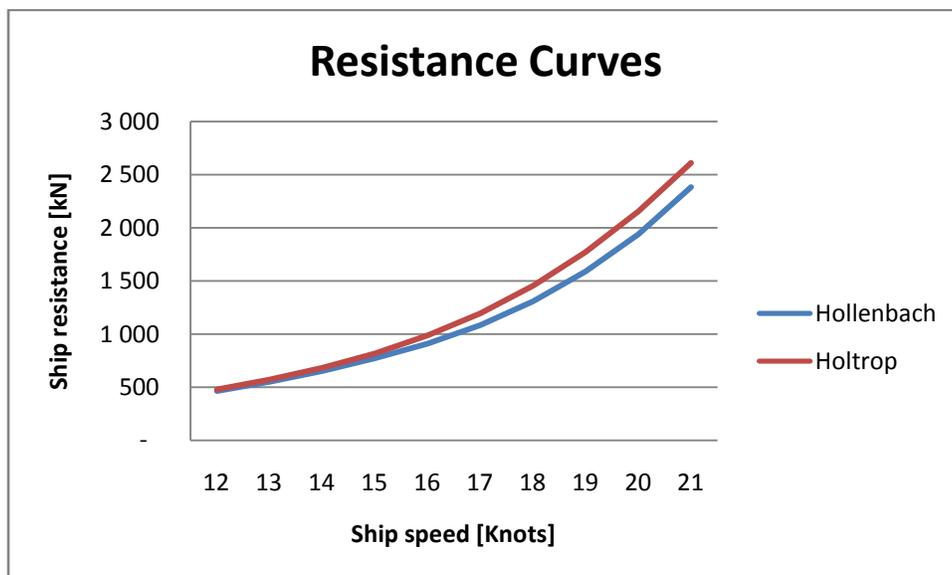


Figure 43: Ship resistance curves (39)

By two different methods we calculated the calm water resistance of the ship. The empiric methods gave different results by the reasons given in the earlier chapters. Hollenbach's method gives a resistance that is more reliable (1 031 kN), since it represents a newer ship population. In further calculations this result is used. The necessary towing power is calculated by using the formula below:

$$P = R_r \cdot V \text{ (kW)}$$

This results that approximately 8 860 kW is the needed power in calm sea. To ensure enough propulsion in head sea, wind, currents, and a higher roughness of the hull due to fouling, a "sea margin" is added. For the mentioned effects the resistance must be increased 25-30 percent for the North Atlantic (40).

The required towing power of the ship will be 11 000 kW.

12 Propulsion and Thrusters

As a CO₂ carrier, an environmental friendly profile is desired. The fuel consumption must be as low as possible, and therefore fuel saving have a high priority. A high efficiency of the propulsion plant is required to ensure this.

A single screw vessel has been selected because of the purpose of the ship. Comparison of large gas ship shows that they tend to have one main propeller. This is also less expensive and results in a less complex installation than twin screw. By these arguments single screw propulsion was chosen.

12.1 Mechanical Driven Propulsion

The main propulsion unit can either be mechanically or electrically driven. Mechanically driven propulsion systems have higher efficiencies since losses in the generator and electric motor are avoided. The engine room should be placed near the main propeller to avoid long shaft lines, with related risk of vibration problems. It is also the case that higher costs are attached when placing the engine room in unconventional areas.

12.2 Generator Driven Propulsion

Electric driven propulsion offers more alternatives when placing the different engines. Since every engine has a generator wired to an electrical system, the need to place the engines together in the same engine room is not present. The system will, however, be more complex and more expensive because of the need for more generators and electrical auxiliary systems.

12.3 Propulsion Units

12.3.1 Propulsion

The submerged propeller must have a clearance to both the ship and the mean water line. The distance from the propeller tip to the hull should not be less than 15 percent of the propeller radius, to avoid large vibrations and pressure impulses on the hull (16). The distance from the propeller tip to the water line should not be less than 30 percent of the propeller diameter, to avoid suction and thrust loss in seaway (41)

Maximum diameter of the propeller is then determined from the line drawings. They allow, due to the considerations above, a diameter of approximately 8.5 meters.

The rudder and the propeller must be regarded as one unit when optimizing the aft end of the ship.

The propulsive efficiency can be expressed as:

$$\eta_D = \eta_R \cdot \eta_0 \cdot \eta_H$$

This is the parameter desired to increase to decrease the ship's fuel consumption. The factors are:

$$\eta_R = \frac{Q_o}{Q_m}, \text{ relative rotative efficiency}$$

$$\eta_0 = \text{open water efficiency}$$

$$\eta_H = \frac{1-t}{1-w}, \text{ hull efficiency}$$

t = thrust deduction coefficient

w = mean Taylor wake.

Approximations for the wake and thrust coefficients must be done. They can either be determined from model tests, numerical calculations or empiric methods. The last method has been used.

The wake is approximated from previous model test results from the British Ship Research Association (BSRA). The block coefficients for these models ranged from 0.65 to 0.80

$$w = -0.478 + 0.790 \cdot C_B - 0.10 + 0.170 \frac{B}{\sqrt{V^{1/3}} \cdot D} - 0.0072 \cdot L_{CB}$$

Calculations done with the maximum propeller diameter of 8.5 meters, yields a wake of 0.35. A decrease in propeller diameter does not influence the wake extensively. Since the C_B –range do not include the projected ship it must be assumed a higher wake. The wake is therefore set to 0.37.

12.3.1.1 Optimum Propeller Dimensions

An optimized propeller is a propeller with the largest possible diameter applied to the ship. Simple mechanics dictates that the largest possible diameter with the lowest revolutions per minute (RPM) is the propeller with the best efficiency. Optimum propeller dimensions are determined when the RPM, ship speed, wake and ship resistance are present. The propeller blade area and pitch determines this optimum propeller, and the highest attainable efficiency.

The input data in the determination of the best propeller is the RPM, mean Taylor wake, and ships design power and speed. The number of propeller blades must also be taken into account. The most used number of blades is 3-5 blades (42). The analyses are therefore done with three, four and five blades.

ShipX has a built-in program called “Optimum Propeller Wizard” which calculates the best dimensions of the propeller by using the Wageningen B-Series. The input factors are the wanted RPM, design propulsion power and ship speed. The program gives out the open water efficiency. The table below is obtained when the RPM is varied, the mean Taylor wake is set to 0.38, and the design diameter is 8.5 meters.

RPM	3-bladed	4-bladed	5-bladed
60	0,61	0,616	0,613
70	0,626	0,608	0,579
80	0,616	0,566	0,521
90	0,557	0,546	0,529

Table 45: Open water efficiency of different propellers

The optimal design is then a three bladed propeller with 70 revolutions per minute.

Optimum propeller	
Number of blades	3
RPM	70
Design diameter [m]	8,5
Pitch ratio	0,829
Expanded blade ratio	0,391
Efficiency	0,626

Table 46: Optimum propeller dimensions

12.3.1.2 *Propel Flow Improvement*

The high wake and thrust coefficients implies that there are a need for improvements. Therefore, the appliances of power reduction units that can reduce the fuel consumption are taken into consideration.

12.3.1.2.1 Mewis Duct

This is a fuel saving propulsion duct that may be applied on large tankers with a relatively high block coefficients. The duct allows either a significant fuel savings at a given speed, or alternatively for the vessel to travel faster for a given power usage.

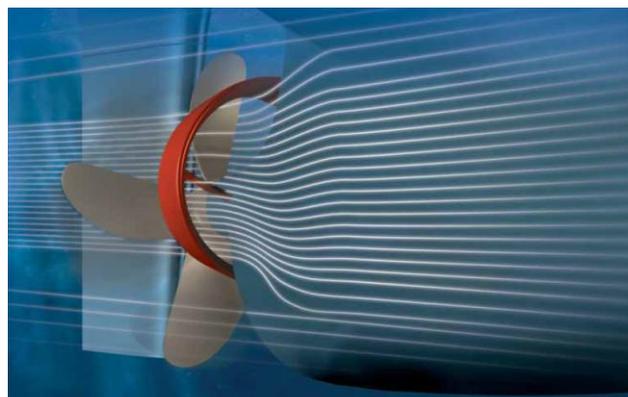


Table 47: Flow with ducted propeller. (43)

The idea is to increase the low propulsion efficiency and lower the risk for cavitations and vibrations. To achieve this, the wake field must be improved by ensuring that the inflow to the propeller is homogenous. The system consists of two elements, a duct positioned forward of the propeller and an

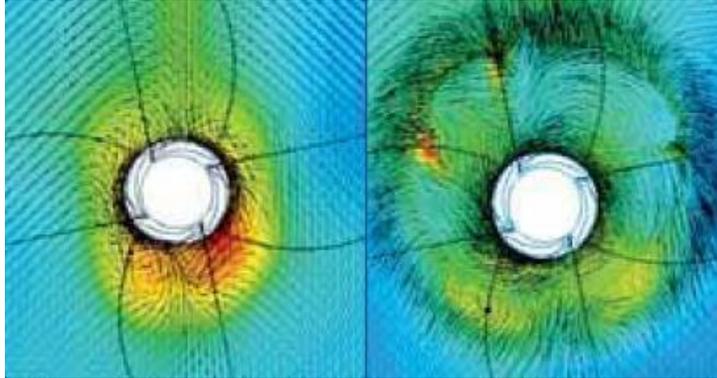


Figure 44: Wake distribution, non- (l), ducted propeller (r)(43)

integrated fin system within the duct. The duct straightens and accelerates the wake flow from the hull into the propeller and produces a net ahead thrust. The fins are asymmetrical profiled to create a homogenous flow distribution. The fins also make a pre-swirl to the ship wake which reduces losses in the propeller slip stream by preventing rotational flow aft of the propeller from occurring. This results in an increased propeller thrust. (43)

The duct increases the water speed at the stern. The wake also becomes more evenly distributed. This increases the efficiency of the propulsion system. A lower risk of cavitations makes it possible to run with a higher thrust without damaging the propeller, and higher propulsion efficiency can be achieved. Since the wake is larger for full ships (which have high block coefficients) the effect will be higher for these types of ships.

The achievable power savings can vary from three percent for small multi-purpose ships, to nine percent for large tankers and bulk carriers. (43)

The ship will have a relatively high block coefficient compared to other LPG-tankers, and the wake will be considerably larger. By these arguments it was chosen a pre swirl duct in front of the propeller. The system is a product of Becker Marine Systems. (43)

12.3.1.2.2 Twisted Rudder with Flap

Water flow improvement to the rudder is here proposed. With a twisted rudder one can take advantage of the induced velocities produced by the vortices made by the propeller. These induced tangential velocities shift direction at top and bottom of the propeller boss. To maintain an optimal in flow direction to the rudder one can lower the drag force on the rudder. This is managed by twisting the rudder at the centre line of the boss.

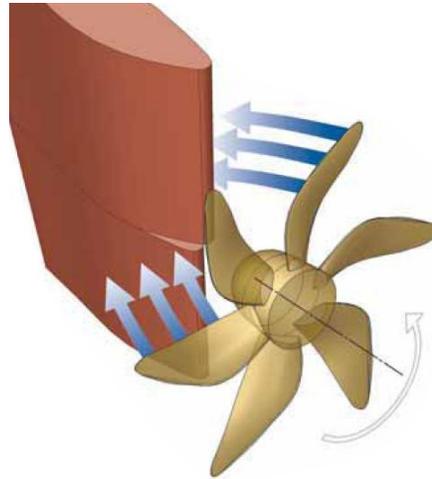


Figure 45: Flow on twisted rudder. (43)

These types of improvements also help to prevent cavitations from forming low pressure areas at the leading edge of the rudder foil. Improved manoeuvrability is also one of the benefits.

The theory behind this improvement is the induced tangential velocities made from the propeller.

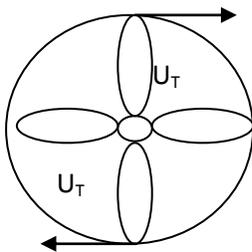


Figure 47: Induced tangential velocities

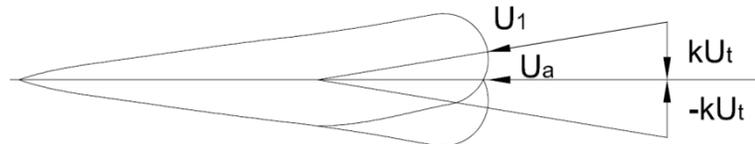


Figure 46: Water velocities at the rudder

A bird perspective of the twisted rudder is seen to the left. Here is U_a , axial water speed, and U_1 is the summed velocity vector. The U_T is scaled with a factor k because of the distance between the propeller and rudder. Since the tangential induced speed changes, it influences the optimal inflow direction to the rudder. With a twisted rudder the drag will be reduced, and the fuel consumption will be decreased.

(42)

This application is suitable for faster vessels like RoRo-vessels, and since the ship velocity is relatively high it will be suited for this kind of enhancement.

12.3.1.2.3 Flaps

A rudder flap is included at the trailing edge to increase the manoeuvrability.

The lift coefficient of a rudder with flaps can be approximated by (41):

$$C_L = \frac{2\pi \cdot (\alpha_{foil} + \eta \cdot \alpha_{flap} - \alpha_i)}{1 + 2 \frac{1 + \tau_G}{Asp}}$$

η = flap efficiency (function of flap / chord ratio)

α_i = self induced velocity angle of the foil

α_{foil} = foil angle

α_{flap} = flap angle

τ_G = correction factor for side ratio



Figure 48: Twisted rudder with flap. (43)

The flap helps to increase the lift of the rudder, as can be seen from the flap angle.

Since the rudder and the propeller must be viewed as one unit, then they are optimized on behalf of each other. With a system based on a rudder with a twisted leading edge and a flap, a good manoeuvrability and a low fuel consumption is maintained. It is therefore chosen this kind of solution as the rudder and propeller design. This will of course influence the optimal diameter and pitch of the propeller since the wake differs, and the velocity of water is speeded up inside the duct. However, the former calculations are pursued as guidelines for further design. In further calculations these adjustments of the diameter must be taken care of.

12.3.2 Bow Thruster

The ship needs good manoeuvrability, especially when navigating in port. A bow thruster is therefore added to improve low-speed manoeuvrability.

It is sufficient to have one bow thruster by the comparison of the similar LPG ship Clipper Victory.



Figure 49: Bow thruster (79)

The hull is implemented with Anti Suction Tunnel (AST) to lower the risk for a large thrust reduction at forward speed. The pressure force that appears at the large hull area when manoeuvring the ship is lowered by an AST.

12.4 Power Prediction

The calculations in the former chapters have found the open water efficiency. ShipX also calculates the total propulsive efficiency. At 70 RPM this is approximately 0.73.

$$P_D = \frac{P_{\text{resistance}}}{\eta_D}$$

The necessary power delivered to the propeller is 11 000 kW divided by the propulsive efficiency, which gives approximately 15 000 kW.

13 Electric Load Analysis

The determination of total necessary motor rating requires an electric load analysis for the major consumers of the ship. The largest and most important consumers are therefore given in this chapter to ensure that the required motor rating is obtained.

A pre-analysis of total power demand has been done to see the possible power of the main engines. LPG ships of similar size have machinery of about 12 000-20 000 kW. A possible solution is to have two 9 000 kW Wärtsila LNG motors. Choice of main engines will be further studied in the next chapter.

Some of the power demands are calculated and some are found from the comparison ship. Solvang's somewhat identical Clipper Victory has been a great reference for finding the right motor rating, efficiency and load factor. Comparisons from Clipper Victory have been done where the efficiency and load factor is set (23)(15). Simple calculations by hand have also been performed.

The following calculations give the required power, efficiency of load machinery, and load factor. These numbers are to be used in the last chapter as the operating profile.

13.1 Brake Power

From the previously chapter the necessary power delivered to the propulsion unit is found. This was 15 000 kW. The assumptions of a mechanical efficiency of 0.97 leads to the brake power. This yield:

$$P_b = 15\,500 \text{ kW.}$$

The graph below gives the brake power as a function of ship speed. The added resistance due to waves and currents are included.

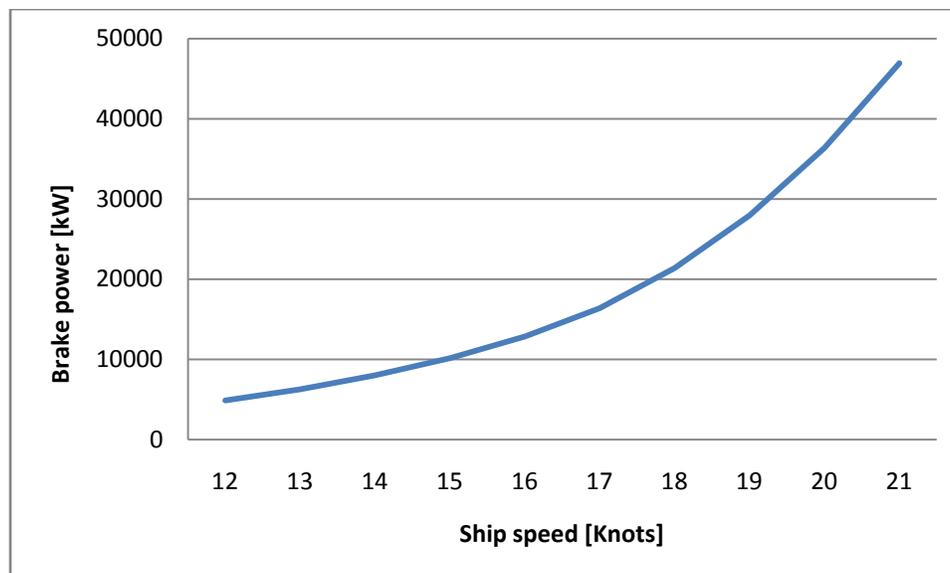


Figure 50: Necessary brake power

13.1.1 Bow Thruster

The power demand of the thruster is set similar to that of Clipper Victory. The need is 1 500 kW. The bow thruster efficiency is set to 0.95.

13.2 Auxiliary System

13.2.1 Fuel System

Since the ship will have an engine that's run on LNG there is no need to have a fuel pump because of the pressure in the LNG tank.

13.2.1.1 LNG Handling

The LNG tanks need to have a pressure larger to that in the engine to ensure a satisfactory flow to the engine. This pressure is kept when we add heat to the tank. The heat can be taken from the engines cooling water by a heat exchanger.

13.2.2 Air Compressor

When starting the engines the air compressor must produce a large torque in seconds to make the combustion process run on its own. This energy must be released in a short time, and it is done with compressed air. From (15) it is said that a motor of 10 000 kW requires compressors that consume 80-100 kW. The ships engine cannot be larger than 9 000 kW. A total of two to three such motors must be installed to ensure enough power. The classification companies demands at least two independent compressed air systems with enough air for twelve starts, without running the compressors. This yields a necessary power demand of 160-300 kW for the ship when multiplying the power with the number of engines. The start motor rating is set to 240 kW.

13.2.3 Lubricating Oil System

The lubricating oil system is used to cool the engine and engine parts. It is also used to lubricate bearings. The lubricating oil must be cooled and cleansed after cycling through the motor. This is done by fresh water that has been cooled off by sea water in a heat exchanger. The required amount for a 10 000 kW engine is 250-300 m³/h circulating oil with four bar pressure (15). This yields approximately for one LNG fuelled medium speed motor of 9 000 kW and a lubricating pump power demand of 30 kW by this formula.

$$P_{\text{lub oil}} = \Delta p \cdot \dot{V}$$

The factors are the pressure differential and volumetric flow.

With two engines the need for 60 kW is present. The efficiency is set to 0.80.

13.2.4 Engine Room Ventilation

The engine room will demand a high amount of low quality air due to heat and exhaust. To ensure good ventilation and circulation for the crew members in this area, we need to apply engine room ventilation. This motor rating can be high due to the amount of needed power demand to the ship.

The pressure differential over the fan is set to 0.02 bar (44). The air will be needed for the combustion of the motor and to cool and ventilate the engine room. The sufficient air flow will be approximately 1.5 times the engine needs (44). If the total power demand is set to approximately 20 000 kW (35) and assuming the specific fuel consumption to be:

$$l_e = 6.5 \text{ kg} / \text{kWh} \quad (44)$$

According to

$$\dot{V} = \frac{P_{tot} \cdot l_e}{\rho_{air}}$$

the necessary airflow is 167 000 m³/h when the density of air at one atmosphere and 20 degrees Celsius is 1.205 kg/m³ (45). The power demand will be derived as follows:

$$P_{fans} = \Delta p \cdot \dot{V}$$

This gives a power demand of approximately 90 kW. The efficiency is set to 0.85. The load factor in transit is set to 80 percent and in port 50 percent, because the the need for ventilation is bigger in transit than in port.

13.3 Gas Handling

13.3.1 Deep Well Cargo Pump

The necessary cargo pump power required is set to 2 000 kW (ten pumps of 196 kW) (25). This is the required input rating. The efficiency is therefore set to one. The load factor is 90 percent because the ship uses only nine pumps at a time.

13.3.2 Inert Gas System

This system is taken care of by the Rapid Purge Technology (12). The necessary power demand is related to the compressor linked to the tank on deck.

Assumptions made by (12) claims that the required power demand will be around 20-30 kW.

The power is set to be 25 kW, with an efficiency of 0.8.

13.3.3 Cargo Compressor

The cargo must be stabilized at the wanted thermodynamic point. There is preferable that the cargo can be regulated by cooling and pressurization. An expected value for the compressor can be found by inspecting existing LPG ships and ethylene ships. Clipper Victory has ca. 600 kW for compressing.

IM Skaugen has some experience on the same topic. For a ship of 8 400 m³ they needed 660 kW for LPG cargo compressing.

From these examples the selected cargo compressing power demand is set to be 1 500 kW, as a worst case study when the power is scaled up to the size of the ship.

13.3.4 Cooling

The needed cooling depends on the heat gain of the cargo. The heat gain can be calculated by making some assumptions to retrieve the respective thermo physical properties.

The heat loss can be expressed by Newton's Law of Cooling. This yield:

$$P_{\text{heatloss}} = \frac{S \cdot \Delta T}{\frac{1}{\chi_{LCO_2}} + \frac{1}{\chi_{\text{air}}} + \frac{t_{\text{iso}}}{\zeta_{\text{iso}}} + \frac{t_{\text{steel}}}{\zeta_{\text{steel}}}}$$

$\chi = \text{heat transfer coefficient}$
 $\zeta = \text{thermal conductivity}$

(46)

The choice of insulation is Polyurethane (PUR), a polymer jointed together with urethane. This is a cell plastic with high strength, low density, extremely low thermal conductivity (47). This is the recommended insulations plastic that is used in low temperature tanks ((48), (46)).

The thermal conductivity of LCO₂ has been neglected because of the complex calculations of internal circulation due to convection.

The other parameters are as follows.

Heat gain coefficients	Unit	Value	Marks
Transfer coefficients of LCO2	[W/Km2]	100	(1)
Transfer coefficients of air	[W/Km2]	10	(2)
Thickness of steel	[m]	0,051	
Thermal conductivity of steel	[W/Km]	25	(3)
Thickness of insolation	[m]	0,15	(4)
Thermal conductivity of polyurethane	[W/Km]	0,03	(5)
Surface	[m2]	2403	(6)
Temperature different of cargo-air	[K]	60	(7)
P-heat gain/tank	[kW]	28	

Table 48: Heat gain coefficients

The following assumptions have been made:

- 1) The tanks are filled 100 percent. No sloshing, or internal movement/velocities in the cargo. The number is from Dittus-Boeltercorrelation for single phase liquid CO₂, and the assumption of some vaporization at the tank wall (49)
- 2) Can be expressed like $\chi_{air} = 5.8 + 3.96 \cdot c$ for air velocities under 5 m/s. It is set to $c=1\text{m/s}$.
- 3) Thermal conductivity for mild steel is 50 W/Km, this number can go as low as 16 W/Km for nickel-chrome alloys. This number do not influence extensively to the heat gain. This is set to 25 W/Km.
- 4) Insulation thickness is set to 15 centimetres as for standard insulations of tanks (48). This must also be in accordance to the general arrangement.
- 5) Thermal conductivity of polyurethane range can be set to 0.03 W/Km (47).
- 6) The tank surface is approximately 2 400 m² by simple geometric calculations.
- 7) The surrounding air of the tanks keeps a temperature of 8 degrees Celsius, two degrees more than the mean sea water temperature.

These assumptions lead to the heat gain of 28 kW per tank. Without insulation the heat gain would be 1 300 kW per tank. So the extra investment is required.

Total heat gain for six tanks is set to 180 kW due to some sloshing and due to that there is no insulation near pumps and piping. This number is also similar to another heat gain case study made out from Hamworthy. The efficiency is set to 0.7.

13.4 Hull Machinery

13.4.1 Ballast System

Approximately 24 000 m³ of ballast water is needed to obtain the design water line when transporting LPG instead of LCO₂. However, there is no need to transport all this ballast water in transit with LPG, because it will cause higher resistance. It plays a supportive role of securing the right trim in transit or in damage stability, and to ensure good stability when the ship is in harbour and during offloading of cargo. The stability is checked in earlier chapters, but the electric load is designed for this amount of water. The pump height is set to 15 meters. The pump time is set to six hours.

$$P_{ballastpump} = \rho \cdot g \cdot h \cdot \dot{V}$$

This gives a necessary power demand of 168 kW when the seawater density is 1 025 kg/m³ and gravity 9.8 m/s²

The efficiency is 0.93 by Clipper Victory ballast pumps. This is a conservative value, and is therefore after a time reduced to 0.85.

13.4.2 Bilge Pumps and Pipes

The bilge pumps need a large suction capacity because of the risk of clogging in the pipes. Some liquid CO₂ can be made solid since the cargo is so near the triple point. Regulations of pipelines diameter and volume flow is done by DNV (31). DNV also imply the meaning of at least two bilge pumps to ensure redundancy. The pumps should preferably be displacement pumps.

The diameter, d , of the bilge pipeline is in mm, the length, breadth and draught are in meters.

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

$$\dot{V} = \frac{5.75 \cdot d^2}{1000} \quad (15)$$

This gives a needed diameter of 195 mm and a required 220 m³/h. The pump head is 15 meter as before. The other constants are the same.

$$P_{bilgeepump} = \rho \cdot g \cdot h \cdot \dot{V}$$

The needed output power demand is then approximately 15 kW. Two pumps imply 30 kW. The efficiency is estimated to be 0.85 (50).

13.4.3 Steering System

The steering system is based on the system on the comparison ship Clipper Victory. This ship is a single screw vessel with the same size. From their electric load data the power demand of the steering gear is set to 45 kW with an efficiency of 91 percent.

13.5 Deck Machinery

13.5.1 Mooring Equipment

The mooring machinery will be four hydraulic pumps connected to winches to ensure safe mooring.

By inspecting the Clipper Victory's need for mooring power, the power is set to 210 kW. The efficiency is set to 0.94. The load factor is approximately 90 percent during the mooring operation.

13.5.2 Manifold Hose Crane

The hoses to transport the CO₂ are supplied by the facilities. But the ship must be able to manage the hoses. Therefore the ship is supplied with manifold hose cranes. The size of the power demand is set similar to Clipper Victory: 45 kW with an efficiency of 0.95 and a load factor of 90 percent during mooring operation.

13.5.3 Provision Crane

Cranes for handle the provision is set to 24 kW with an efficiency of 0.85 like Clipper Victory. Load factor 90 percent in the harbour.

13.5.4 Navigation and Deck Lighting

These numbers is taken from the electric load analysis of Clipper Victory with the efficiency of one. The lighting is set to 70 percent in every stage.

- 27 kW for the total navigation light
- 27 kW for deck flood lights

13.6 Hotel Function

The ship will also have systems related to the hotel functions. These functions are ventilation, heating and lighting among others. These power demands are not big, but they add up to a relative large factor. Heating is ensured by a heat exchanger from the exhaust pipe, it take use of the spill heat from the exhaust. These assumptions presented above are arrived at comparison of the LPG-tanker Clipper Victory.

- 70 kW for the galley
- 300 kW for ventilation and air conditioning
- 120 kW for heating (from exhaust)
- 40 kW for lighting
- 40 kW for the laundry equipment
- 50 kW is miscellaneous

These add up to a hotel power requirement of 620 kW electrical powers. Efficiency is set to one.

13.7 Emergency Load

In case of emergency, the electrical system will experience different power loads due to all the different supply powers to handle the crisis. These power demand consumers will determine the emergency generator motor rating.

Fire pumps, power failure and cold start are happenings that will to some extent rely entirely on the emergency generator. One critical incident in power failure will results in pressure increase in the tanks since the cargo will be heated. This must be avoided due to the risk for steel yield and brittle fraction. Comparison with the emergency load of (35) gives the values presented below. The efficiency set to one in the table.

- 60 kW of emergency fire pumps
- 50 kW of HI-FOG system, fire extinguishing
- 27 kW of emergency lighting
- 180 kW of emergency cargo cooling

- 30 kW of lifeboat winch
- 240 kW of cargo compressing
- 20 kW is assumed for other safety systems (alarm, detectors, etc.)

The HI-FOG system is based on the Marioff system (51) for fire extinguishing inside the deckhouse. They are chosen because of the widely use in the world large ship fleet and cruise ships. The required volume flow is stated to be $0.0025 \text{ m}^3/\text{s}$ when the pressure is set to 150 bars. The power is calculated with a pump efficiency of 0.75. (51)

These emergency loads are presented in the table below.

Emergency loads [kW]	Motor Rating	Fire		Power failure		Cold Start	
Consumer	Input	Load	Power	Load	Power	Load	Power
Fire pumps	60	100 %	60	0 %	0,0	0 %	0
HI-FOG system	50	100 %	50,0	0 %	0,0	100 %	50,0
Air compressor	240	0 %	0,0	0 %	0,0	50 %	120,0
Engine room ventilation	105,9	0 %	0,0	0 %	0,0	25 %	26,5
Steering system	49,5	50 %	24,75	50 %	24,75	0 %	0
Bilge system	35,3	50 %	17,65	10 %	3,53	10 %	3,53
Emergency lighting	27	100 %	27,0	100 %	27,0	0 %	0
Rescue boat winch	30	100 %	30	100 %	30	0 %	0,0
Emergency cooling	257	0 %	0,0	80 %	205,6	0 %	0,0
Others	20	80 %	16	80 %	16	80 %	16,0
Total			225,4		306,9		216,0

Table 49: Emergency loads

In case of fire and cold start the emergency loads are about the same. Power failure is the case of emergency that has the greatest power demand. From this table the needed motor rating of the emergency generator can be established. This power demand is set to be 330 kW.

13.8 Power Requirements

The load factor under is given in percent.

Effect [kW]	Motor rating			In port		On/offloading LCO2/LPG		Maneuvering		Transit	
	Output	Efficiency	Input	Load	Power	Load	Power	Load	Power	Load	Power
Propulsion											
Main propulsion	15500	1	15500	0	0	0	0	50	7750	100	15500
Bow Thruster	1500	0,95	1578,9	0	0,0	0	0,0	60	947,4	10	157,9
Total propulsion			17078,9		0,0		0,0		8697,4		15657,9
Auxiliary System											
Air compressor system	240	1	240	80	192	0	0	50	120	50	120
Lubrication oil system	60	0,8	75	50	37,5	50	37,5	80	60	80	60
Engine room ventilation	90	0,85	105,9	50	52,9	50	52,9	80	84,7	80	84,7
Total			420,9		282,4		90,4		264,7		264,7
Gas Handling											
Deep well cargo pump	2000	1	2000	10	200	90	1800	0	0	0	0
Cargo compressor	1500	1	1500,0	10	150	0	0	80	1200,0	80	1200,0
Rapid Purge Technology	25	0,7	35,7	0	0,0	90	32,1	0	0,0	0	0,0
Coolers	180	0,7	257,1	10	25,7	10	25,7	80	205,7	80	205,7
Total			3792,9		375,7		1857,9		1405,7		1405,7
Hull Machinery											
Ballast system	168	0,85	197,6	90	177,9	20	39,5	0	0,0	0	0,0
Purge air system				90	0,0	0	0,0	0	0,0	0	0,0
Bilge system	30	0,85	35,3	15	5,3	15	5,3	15	5,3	15	5,3
Steering system	45	0,91	49,5	0	0,0	0	0,0	100	49,5	10	4,9
Total			282,4		183,2		44,8		54,7		10,2
Deck Machinery											
Manifold hose crane	45	0,95	47,4	90	42,6	0	0,0	0	0,0	0	0,0
Provision crane	24	0,85	28,2	90	25,4	0	0,0	0	0,0	0	0,0
Navigation and deck lighting	54	1	54	70	37,8	70	37,8	70	37,8	70	37,8
Mooring equipment	210	0,94	223,4	90	201,1	90	201,1	0	0,0	0	0,0
Total			353,0		306,9		238,9		37,8		37,8
Hotel Function											
Galley	70	1	70	30	21	30	21	30	21	30	21
Ventilation	300	1	300	70	210	70	210	70	210	70	210
Heating	120	1	120	60	72	60	72	60	72	60	72
Lighting	40	1	40	80	32	80	32	80	32	80	32
Laundry	40	1	40	10	4	10	4	10	4	10	4
Miscellaneous	50	1	50	80	40	80	40	80	40	80	40
Total			620		379		379		379		379
Propulsion power need			15500,0		0,0		0,0		7750,0		15500,0
Electrical power need			7048,1		1527,2		2611,0		3089,3		2255,4
Emergency generator			330								

Table 50: Power requirements

The highest electrical power need is in the manoeuvring phase. The highest total need is in transit, when the ship requires a total of 17 760 kW.

13.9 Operating Profile

The operating profile is based upon the power requirements in the different phases of the ships transport cycle. The numbers are given for half a cycle. This means the time from leaving one port to finished offloading and loading on the next. The power is presented in different consumer groups. One complete cycle is 5.5 days. With 357.5 operational days per year this yields 65 trips per year. With 46 092 tonnes CO₂ per trip, it sums up to about 3 000 000 tonnes per year. The time consumption for each phase is given below.

13.9.1 In Port

The time in port is dedicated to prepare the on/offloading for CO₂ and LPG. Most of the preparation is done in the manoeuvring phase and in sailing. This time is therefore set to one hour per port.

13.9.2 On/Offloading

On/offloading is a time consuming factor that is the main factor you can decrease with technology aspects. We have selected a total of nine main deep well pumps that can empty the tanks in eleven hours. The time to fill the tanks is estimated to be eleven hours. This implies a total of 22 hours of on/offloading. Of the 24 hours dedicated for on/offloading, the two remaining hours are in case of unforeseen events.

13.9.3 Manoeuvring

This time is estimated to be ten hour per half cycle. This leaves five hours for sailing in and out.

When arriving at Mongstad harbour the ship need a tug boat before entering the jetty. This time is scheduled to 2.5 hours (14). The same time is also presumed at Brünsbuttel. Since the ship must decelerate and may go at lower speeds due to bad weather it is included another 2.5 hour. This is also reasonable if one add the possible delay of the interactive systems at the harbours.

13.9.4 Transit

The route of 500 nautical miles is sailed at a speed of 16.7 knots. This make a transit time of 30 hours, and five and a half day for each cycle.

Operation summary (half cycle)	[hours]	Percent
In port	2	3,0 %
On and offloading	24	36,4 %
Maneuvering	10	15,2 %
Transit	30	45,5 %
SUM	66	100 %

Table 51: Time consumption

13.10 Power Consumption Profile

The operating profile graph is presented here.

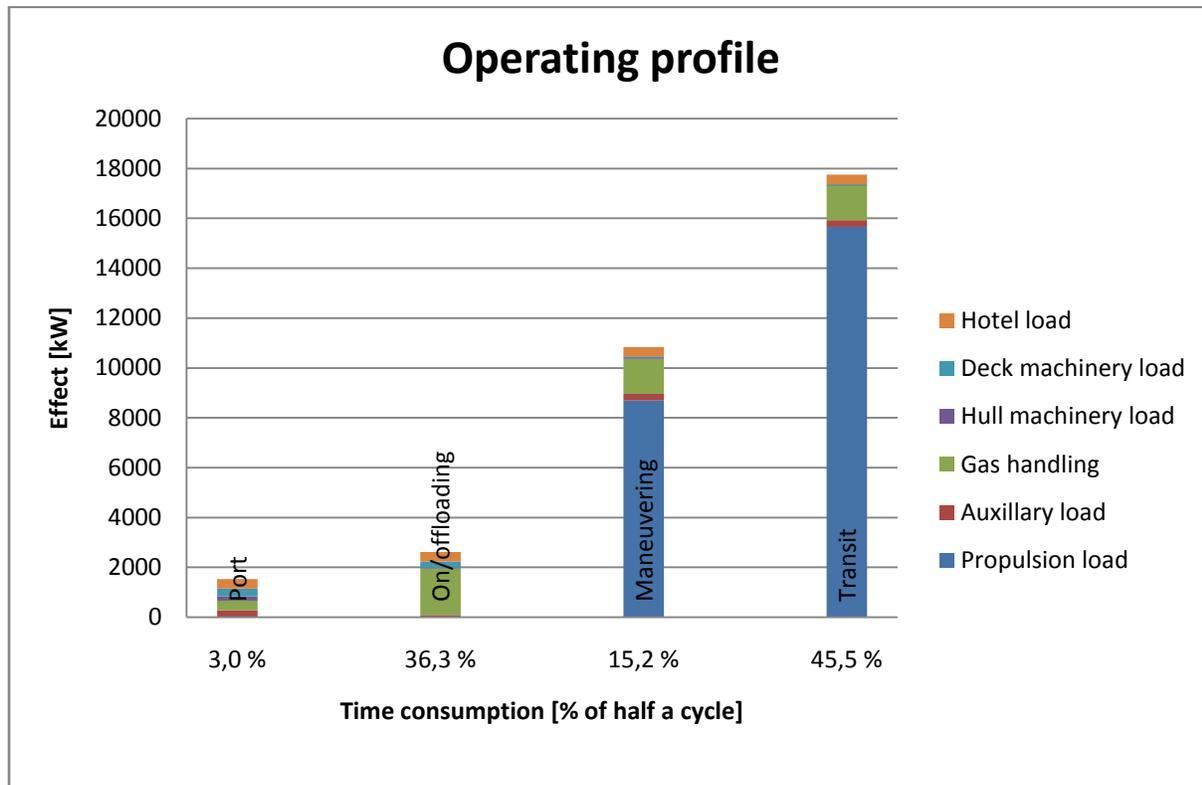


Figure 51: Operating profile

Note that the loading time of cargo is a very time consuming process. Gas handling requires a relatively large percent of the total power demand. The possibility to avoid a large motor rating will influence the economy to a large extent, so it is recommended to do some more investigations in further studies.

14 Power Plant Trade-Off Study

14.1 Main Machinery

The main machinery is very important to optimize. Most of the pollution and fuel consumption is determined by the main machinery and the kind of fuel who is chosen. Here follows a study of possible machinery that can be mounted on the ship.

14.1.1 Gas Turbines

Gas turbines are continuously burning machines, and have a power output ranging from 30 kW to 250 000 kW. They rotate at relatively high speeds (10 000-105 000 RPM). The shaft power is converted to propulsion power through a gear box or an electric generator.

Advantages:

A gas turbine has a great power to weight ratio compared with traditional piston engines, and is often used in ships where the power to weight/volume demands exclude traditional diesel engines, like fast passenger ferries and frigates. Since gas turbines revolves around a central axis at high speed with little to none unbalance it produces far less vibrations than piston engines. They are therefore often used in cruise ships to reduce the noise level. Due to the gas turbines' construction they don't need a complex cooling water system like the diesel engines. Because gas turbines run on higher quality fuel they pollute less than large diesel engines which run on lower quality fuels.

Disadvantages:

Most gas turbines have a larger specific fuel consumption compared with similar sized diesel engines and use more expensive fuel. This is due to the gas turbines' low thermal efficiency. They also have a higher cost than conventional machinery.

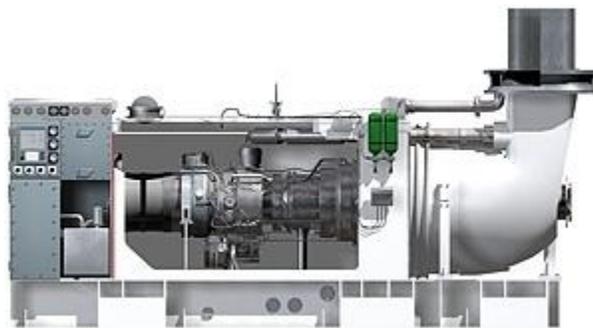


Figure 52: Rolls-Royce gas turbine (52)

There are several types of gas-turbines on the market; the most common for marine applications is the Aero-derived type, as the name suggests they are derived from gas turbines used in the aircraft industry. They are again divided into two sub categories, the simple-cycle and the advance-cycle gas turbine. The simple-cycle turbine does not have additional heat recovery systems or internal waste heat utilization systems like the advance-cycle turbine. These systems help to increase the thermal efficiency of the turbine by utilizing the heat in the exhaust gas. The disadvantage is that these systems are complex and expensive. Here are some performance parameters of the gas turbines:

SPECIFIC DATA	GAS TURBINES	
	Aero-derived	
Process	Simple cycle	Advance cycle
Construction	2-shaft	2-shaft
Output power range [kW]	26 000 - 6 000	24 000
Output speed [rpm]	3 600 – 7 000	3 600
Fuel type	MDF	MDF
Spec. Fuel Cons. [g/kWh]	240 - 280	200
Spec. Air Cons, [kg/kWh]	10 - 15	10.5
Spec. NO _x Emission [g/kWh]	2 - 5	3
Specific Mass [kg/kW]	1.0 - 1.4	1.8
Specific Volume [dm ³ /kW]	2.5 - 4.5	4.1
Specific Cost [Euro/kW]	180 - 280	470

Table 52: Performance parameters of gas-turbines (State of the art 2001) (53)

14.1.2 Diesel Engines

Marine diesel engines may be divided into three main categories; low-, medium- and high-speed engines. These different types have different characteristics represented in the table below.

SPECIFIC DATA	DIESEL ENGINES		
	Low-speed	Medium -speed	High-speed
Process	2-stroke	4-stroke	4-stroke
Construction	Crosshead	Trunk piston	Trunk piston
Output power range [kW]	80 000-8 000	35 000 - 500	9 000-500
Output speed [rpm]	80 - 300	300 – 1 000	1 000-3 500
Fuel type	mostly HFO	HFO or MDF	MDF
Spec. Fuel Cons. [g/kWh]	160 - 180	170 - 210	200 - 220
Spec. Air Cons, [kg/kWh]	9 - 7	9 - 6	7.5 - 5.5
Spec. NO _x Emission [g/kWh]	22 - 14	18 - 10	13 - 7
Specific Mass [kg/kW]	60 - 17	20 - 5	6 - 2.3
Specific volume [dm ³ /kW]	55 - 12	28 - 4	8 - 2.8
Specific Cost [Euro/kW]	420 - 400	Line: 330 - 220 V: 280 - 170	V: 240 - 180

Table 53: Performance parameters of diesel engines (State of the art 2001) (53)

The diesel engines can drive the propeller directly, through a gear box or as a diesel electric propulsion system.

14.1.2.1 *Low-Speed*

As can be seen from the table, the low speed diesel engines have higher NO_x emissions. This is due to the longer combustion process. The low-speed engines have the advantage of low specific fuel consumption combined with the ability to run on cheaper fuel oil. The crosshead construction reduces the wear on the cylinder liner. This means that maintenance needed are reduced. The disadvantage with the low-speed engine is its high mass and volume as well as the high costs.

14.1.2.2 *Medium-Speed*

The medium speed engines can run on a variety of different fuel oils. They can also be converted to run on gases like LNG and LPG. LPG would be a natural choice since this will be the return cargo, but this will be discussed later in this chapter. The medium speed engines can also be constructed with the cylinders in a V-formation which will give a more compact engine.

14.1.2.3 *High-Speed*

High-speed diesel engines are mostly used where the required power output is low and the need for a compact engine is high. Most emergency generators are of this type. Because of the relative light weight the engine can be placed at a higher deck without causing stability problems.

14.1.3 **Dual Fuel Engines**

Some of the medium speed diesel engines are designed to run on other fluids as well as marine diesel oil. These are called dual fuel engines. They mainly use natural gas, although some experiments at the Department of Marine Technology at NTNU show that they can be converted to run on LPG (44). Due to issues with engine knocking while running on LPG it was necessary to reduce the effect by ten percent.

14.1.4 **Lean Burn Engines**

Lean burn engines operate with more air than what is needed for complete combustion; this reduces the peak temperatures, and therefore the NO_x emissions. They also have better efficiencies than conventional diesel engines. These engines usually run on natural gas, but tests carried out by Rolls Royce Bergen (44) show that it is possible to run on LPG with a 30 percent effect reduction to avoid knocking.

The dual fuel and lean burn engines are both medium speed engines. Wärtsilä has gas engines with a power output ranging from 4 000 kW to 17 000 kW.



Figure 53: Wärtsilä 34SG Lean Burn Engine (55)

14.1.5 Fuel Cells

Fuel cell is a relatively new concept in marine power production, and is only in the early phase of testing. The fuel cell converts chemical energy directly into electrical energy. Several different fuel cells are available for use. Most fuel cells use hydrogen and oxygen as reactants in the process, but through a fuel reformer, hydrogen can be obtained from hydrocarbons. The product of the reaction between hydrogen and oxygen is water, depending on the hydrocarbon fuel and the reformer other products are also produced, but in much smaller quantities than for combustion engines and are therefore extremely environmentally friendly.

The most interesting for marine applications is the Solid Oxide Fuel Cell (SOFC) with the ability to run on hydrocarbon fuel by using a fuel reformer. Smaller fuel cells are being tested in ships for auxiliary power production. They have the ability to run on a variety of different fuels such as natural gas, biogas and methanol. Wärtsilä are also looking at the possibility to run on other liquid fuels like low sulphur diesel oils (56)

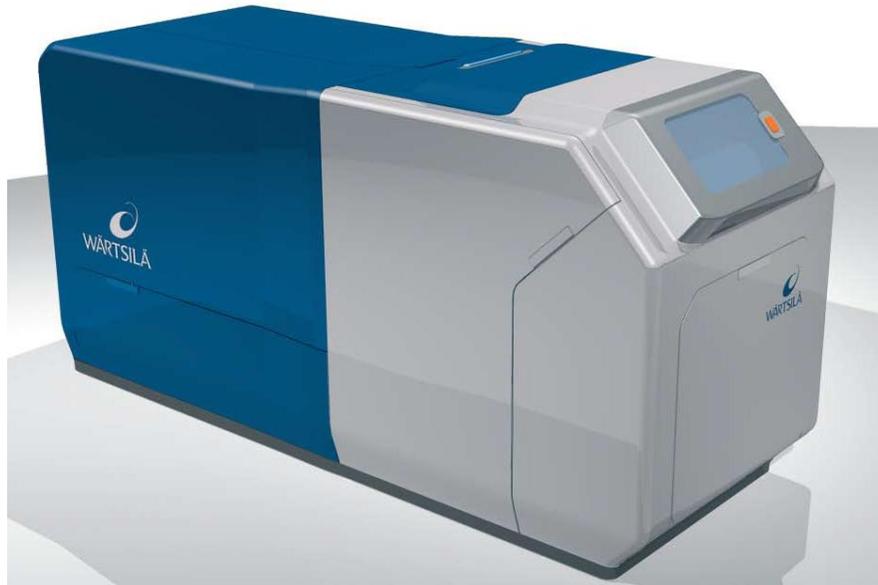
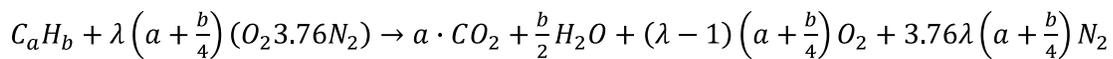


Figure 54: Wärtsilä Solid Oxide Fuel Cell (56)

14.2 Fuel Alternatives

To maintain the ships “green profile” emissions from different fuel alternatives were investigated. As presented above, both the dual fuel engine and the lean burn engine are capable of running on petroleum gas as well as natural gas. Although with a reduced effect to avoid knocking when running on petroleum gas the key aspect to evaluate concerning fuel alternatives is the emission of CO₂.

When an engine burns hydrocarbon fuels it will do so according to the following equation:



(53)

Where λ is the air ratio and $\lambda=1$ when the amount of air is just enough for complete combustion.

A closer look at the equation shows that the amount of CO₂ molecules in the exhaust equals the number of carbon-atoms in the hydrocarbon. Depending on the reaction enthalpy an equation for CO₂ emission per kWh for the different fuel alternatives can be derived.

$$CO_2 - emission = \frac{a \cdot M_{CO_2} \cdot 3600}{M_{C_aH_b} \cdot h_n}, \text{ where}$$

h_n = -reaction enthalpy

M_{CO_2} = molar mass of CO₂ = 44.0095 g/mol

$M_{C_aH_b}$ = molar mass of hydrocarbon

The factor 3 600 is the number of seconds in an hour to get the right dimensions.

This equation does not take the engines thermal efficiency into account, and emissions will therefore vary for different engines.

LNG consists of methane while LPG consists of either propane, butane or a combination of the two. Emission calculations are shown in the table below.

Fuel	Hydrocarbon	Formula	M [g/mol]	h_n [$\times 10^3$ kJ/kg]	CO ₂ emission [g/kWh]
LNG	Methane	CH ₄	16.042	50.01	197.5
LPG	Propane	C ₃ H ₈	44.094	46.353	232.5
	Butane	C ₄ H ₁₀	58.12	45.714	238.5

Table 54: Emission calculations

The reaction enthalpy depends on the phase of the product. In these calculations it has been assumed that all the products are in gaseous phase due to the high temperature during combustion.

The table shows that LPG will have 18-21 percent higher CO₂ emission than LNG, depending on the gas mix. Tests also show the need to decrease the effect when running on LPG. This will reduce the thermal efficiency of the engine contributing to even higher CO₂ emissions. Regular diesel fuels are not included in these calculations since they have a higher carbon number, but not higher reaction enthalpy; they also contain sulphur which gives SO_x emissions in addition.

14.3 Machinery Arrangement

The previous shows that gas engines running on LNG has the lowest CO₂ emissions. The Wärtsilä 34SG has the best thermal efficiency for gas engines today, varying from 46.1- 46.5 percent depending on the cylinder arrangement and load.

Technical Data	[Unit]	9L34SG	16V34SG	20V34SG
No. of cylinders		9	16	20
Nominal speed	RPM	750	750	750
Speed range	RMP	525-750	525-750	525-750
Efficiency	%	46.4	46.4	46.4
Shaft power at 750 RPM	[kW]	4 050	7 200	9 000
Shaft power at 650 RPM	[kW]	3 510	6 240	7 800
Shaft power at 525 RPM	[kW]	2 834	5 040	6 300
NO _x	[g/kWh]	1.3	1.3	1.3
CO	[g/kWh]	1.6	1.6	1.6
Min gas pressure	[kPa]	460	460	460
Lub. oil consumption	[kg/kWh]	0.4	0.4	0.4

Table 55: Wärtsilä 34SG Technical data (55)

For auxiliary power production a smaller engine is needed, and the Rolls-Royce Bergen C25:33L is a good alternative. Here are the main characteristics of the Rolls-Royce lean burn engine for different cylinder arrangements and rpm's:

Engine type		C25:33L6A	C25:33L8A	C25:33L9A	C25:33L6A	C25:33L8A	C25:33L9A
Number of cylinders		6	8	9	6	8	9
Engine speed RPM	RPM	720/750	720/750	720/750	900/1 000	900/1 000	900/1 000
Mean piston speed	m/sec	7.9/8.3	7.9/8.3	7.9/8.3	10/11	10/11	10/11
Max. cont rating (MCR)	kW	1440/1500	1920/2000	2160/2250	1920/2000	2560/2665	2880/3000
Specific fuel cons.	g/kWh	182/183	182/183	182/183	188/189	188/189	188/189
Specific lub. oil cons.	g/kWh	0.7	0.7	0.7	0.7	0.7	0.7

Table 56: Rolls-Royce Bergen C25:33L Technical data REF Rolls-Royce gas turbine (57)

Because of the small variety in gas engines on the market these two were the best choices for the machinery arrangement since other engine had either too high or too low power output.

Engine	Power [kW]	In port		On/offloading		Manoeuvring		Transit	
		Load [kW]	%	Load [kW]	%	Load [kW]	%	Load [kW]	%
Main engines:									
1: Wartsila 16L34SG	9 000	0	0 %	0	0	3 875	43 %	7 750	86 %
2: Wartsila 16L34SG	9 000	0	0	0	0	3 875	43 %	7 750	86 %
Auxiliary engines:									
1: Rolls-Royce Bergen C25:33L6A	1 440	764	53 %	1 306	91 %	1 438	100 %	1 128	78 %
2: Rolls-Royce Bergen C25:33L6A	1 440	764	53 %	1 306	91 %	1 438	100 %	1 128	78 %
3: Rolls-Royce Bergen C25:33L6A	1 440	0	0	0	0	0	0	0	0

Table 57: Machinery arrangement

Using two of the 20V34SG for propulsion the demands of 15 500 kW in transit will be met. To meet the electrical power demands, three Rolls-Royce Bergen C25:33L6A producing 1 440kW at 720 RPM are used. By having one auxiliary engine in reserve most of the time, maintenance can easily be carried out. Maintenance works on the main engines are done in port. It has been estimated an average of 7 days per year for maintenance, classification etc. To further increase the redundancy of the engine system, shaft generators/electric motors will be mounted on the gear box connecting the two main engines to the propeller shaft. They will not be in use in normal operation, but can be used if one or both of the main engines fails, or if some of the auxiliary engines fail. The figure below shows a very simplified sketch of the engine system.

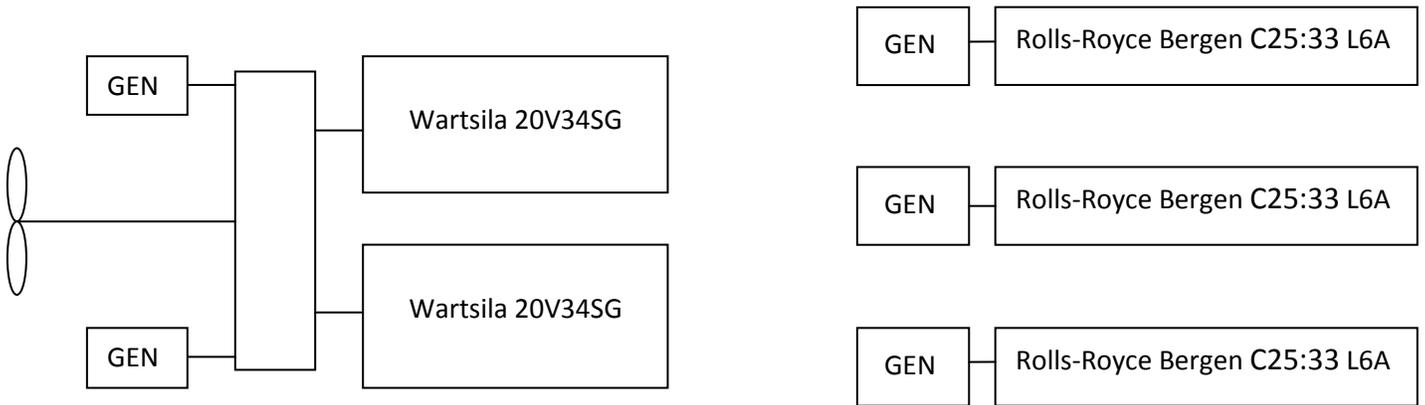


Figure 55: Machinery arrangement

15 Sea Keeping Analysis

Sea keeping is defined as the ship's movement in different sea condition. It is necessary to compute how the vessel responds to the sea states it will encounter. The sea keeping properties are dependent on hull geometry, and in this chapter there have been performed analyses of the sea keeping for this specific vessel. The computer program Maxsurf Seakeeper has been used for all the calculations.

The calculations will be limited to zero and transit velocity because there will be no need for more detailed studies since the ship will most of the time travel with a velocity of approximately 16 knots. Since the ship will carry both LPG and CO₂ it will be necessary to do an analysis for both cargo situations. However, since calculations in Seakeeper shows that there is no significant difference for the two cargo situations the results for the vessel transporting CO₂ will only be represented in the following chapter.

15.1 Sea Keeping Principles

A vessel has six degrees of freedom. There are three linear motions, surge, sway and heave and three angular motions, roll, pitch and yaw. All calculations are measured in the coordinate system described below.

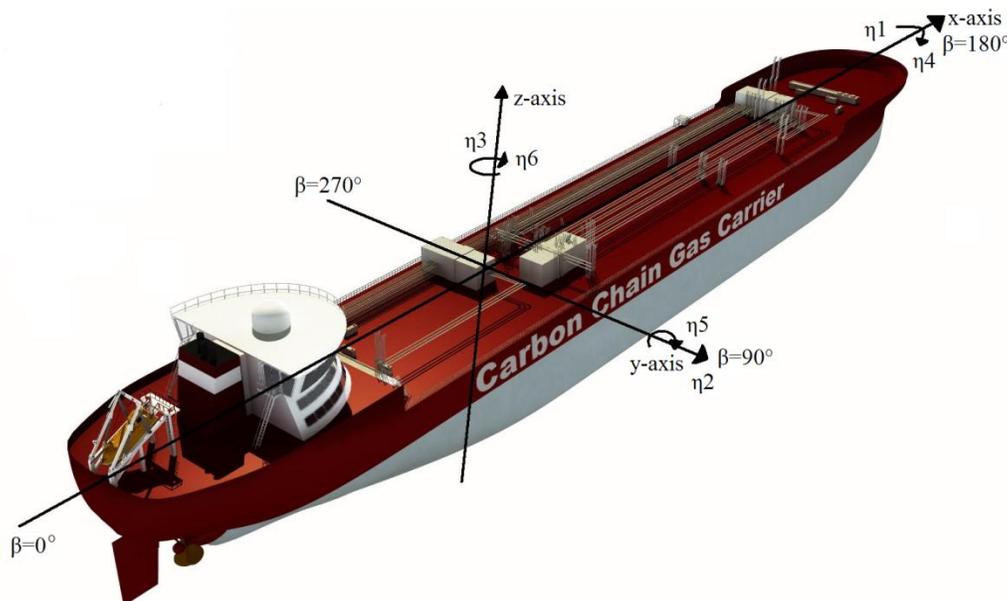


Figure 56: Coordinate system, degrees of freedom

Surge= η_1 Sway= η_2 Heave= η_3 Roll= η_4 Pitch= η_5 Yaw= η_6

The tool used for calculating sea keeping properties is Seakeeper, a 3-degree of freedom (roll, pitch and heave) vessel motion prediction program, a part of Maxsurf Pro software package. The program used strip theory to calculate the coupled heave and pitch response of the vessel in deep water with arbitrary

wave heading. The Response Amplitude Operator where calculated by solving the heave and pitch equations. The roll response is calculated using a user-specified damping coefficient and fixed added roll inertia ratio. Since it would be too time consuming to calculate the user-specified coefficient where the coefficient instead assumed to be 0.1 as a starting point. (58)

The Response Amplitude Operator (RAO), also referred to as a transfer function, describes how the response of the vessel varies with wave encounter frequency. At low frequencies the vessel will simply move up and down with the wave and the RAOs will tend to unity. At high frequencies the effect of many very short waves will cancel out over the length of the vessel and the response tends to zero. It is also typical that the vessel will have a response amplitude peak of greater than unity when the force on the vessel lies on frequency close to the vessel's natural period. Those peaks are due to resonance. Since this response can be damaging it is important to design the vessel so the natural period is not of the same magnitude as excitation forces.

Seakeeper will do an analysis of the vessel when it travels in 16 knots and 0 knots with different wave incidence angles. The angles are represented in the table below.

Wave Incidence Angle
180°, Head Sea
135°
90°, Starbord Beam Sea
45°
0°, Stern Sea

Table 58: Wave Incidence Angles

In Seakeeper it was necessary to define different parameters before the analysis could be run. Mass distribution was one of the parameters which were necessary to define. The values for mass distribution are represented in the table below. Mass distribution was calculated using the centre of gravity and Loa and B for the vessel. The two cargoes will have different mass, and this will influence the pitch gyradius. However, since the centre of gravity of the load is the same for both LPG and CO_2 cargo the pitch gyradius will be the same for both cases and the mass distribution are shown in the table below.

Loa = Length over all

VCG= Vertical centre of gravity

B = Breadth

Mass Distribution LPG and CO
Pitch gyradius(% Loa): 24
Roll gyradius(% Boa): 37
VCG: 10

Table 59: Mass Distribution

15.2 ITTC Wave Spectrum

A wave spectrum indicates the amount of wave energy at different wave frequencies. In Seakeeper JONSWAP, DNV, Pierson Moskowitz, ITTC (international towing tank conference), and 1 Parameter Bretschneid are different spectra which can be used to define the sea state. The ITTC spectrum was chosen in this case because it is a two parametric spectrum. It has a significant wave height, H_s , and zero crossing period, ω_1 , as parameters.

Type	Char. Height	Modal [m]	Average period [s]	Zero crossing period [s]	Peak enhancement fact.	Char. Wind speed [kn]
ITTC	4	9	6,94	6,36	1	9

Table 60: Variables in the ITTC spectrum taken from the wave statistics for the North Sea during 2007

Investigating wave statistics (6) four metres significant wave height were found to be likely in the relevant geographical areas in the North Sea during 2007. The modal period was set to 9 seconds after investigating the wave statistics. The zero crossing period was then calculated by the program to be 6.36s.

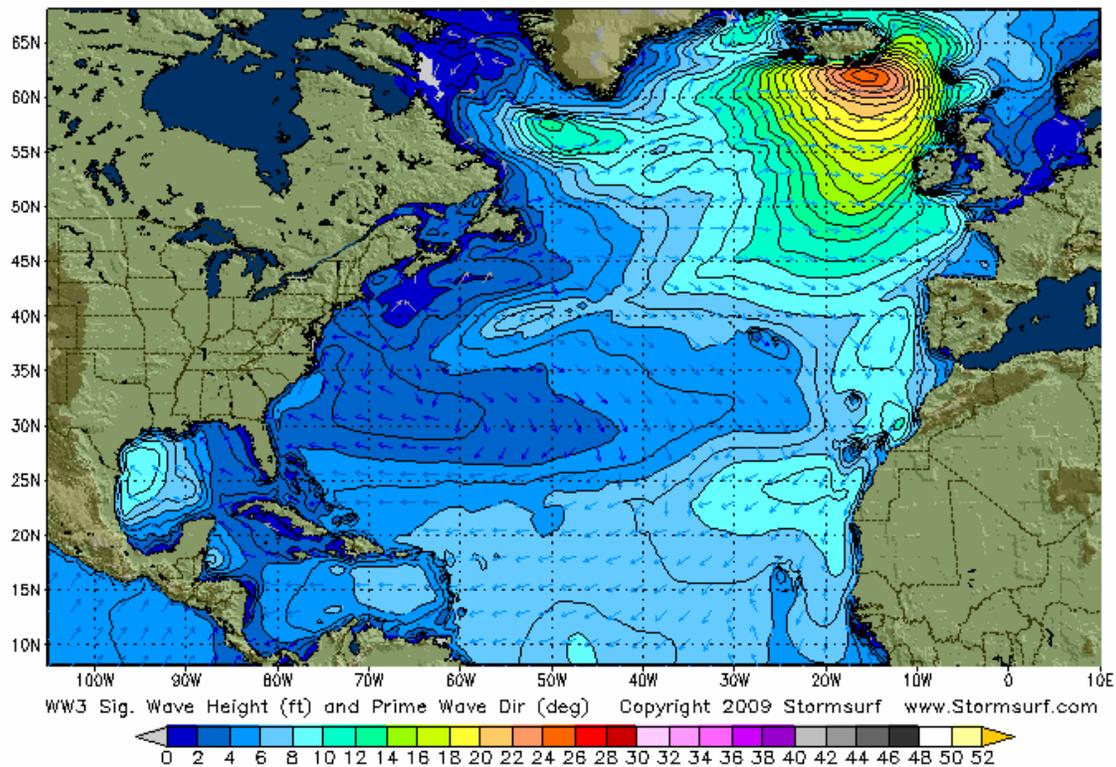


Figure 57: Wave heights in the North Sea

The model from STORMSURF (59) shows that the significant wave height in the North Sea will be from two to six meters. The data from STORMSURF are close to the significant wave height found when

investigating the wave statistics at Karmøy (6), and it is therefore concluded that it is reasonable to set the significant wave height in the ITTC spectra to four metres.

15.3 Natural Periods and Encounter Frequency

The natural period is important for sea keeping since it is not favourable that the force on the vessel is of a frequency close to the vessel's natural period. The response amplitude for heave and pitch should not raise high above one for this type of ships. The roll might be a bit too high. However, this can be reduced by using a bilge keel which is the most common alternative for these kinds of ships.

The encounter frequency;



Figure 58: Wave heading angle

$$\omega_e = \omega + \frac{\omega^2}{g} U \cos \beta$$

ω_e = encounter frequency

ω = natural frequency

β = incident angle

U = vessel velocity

In starboard beam sea, where $\beta \geq \pi / 2$, it is found that $\omega_e = \omega$. However, when $\beta > \pi / 2$, the second part of the equation will be negative. With specific wave frequencies, headings, and velocities can ω_e be both zero and negative. When $\omega_e = 0$ the ship and the wave will have the same velocities and they will follow each other.

15.4 Damping Coefficients

The added mass and damping coefficients must be found. Seakeeper do these calculations, but it is necessary to add a damping factor for roll since Seakeeper does not take into account the viscous effects. According to Seakeeper the typical values for most vessels are between 0.05 and 0.1. However it might be necessary to install a suppression device if the roll is too large. If a bilge keel is installed the damping coefficient might be raised by thirty five percentage according to the compendium Marin Teknikk 3 (60) which states that the roll will be reduced by this factor when installing such a suppressions

device. The damping factor was set to 0.1, and the value was kept constant during the whole analysis. This is due to the fact that it was only interesting to investigate if the sea keeping for this vessel was within the limits of acceptance. It was also further of interest to evaluate the results and come up with a reasonable solution. To do this, it was not found necessary to do several analyses with different damping factors. The next chapters will display the different results graphically for the vessel in transit and zero velocity for different heading angles.

15.5 The Response Amplitude Operator

The curves below show RAO (transfer functions) for different conditions. A response amplitude operator shows response relative to wave height.

15.5.1 Heave

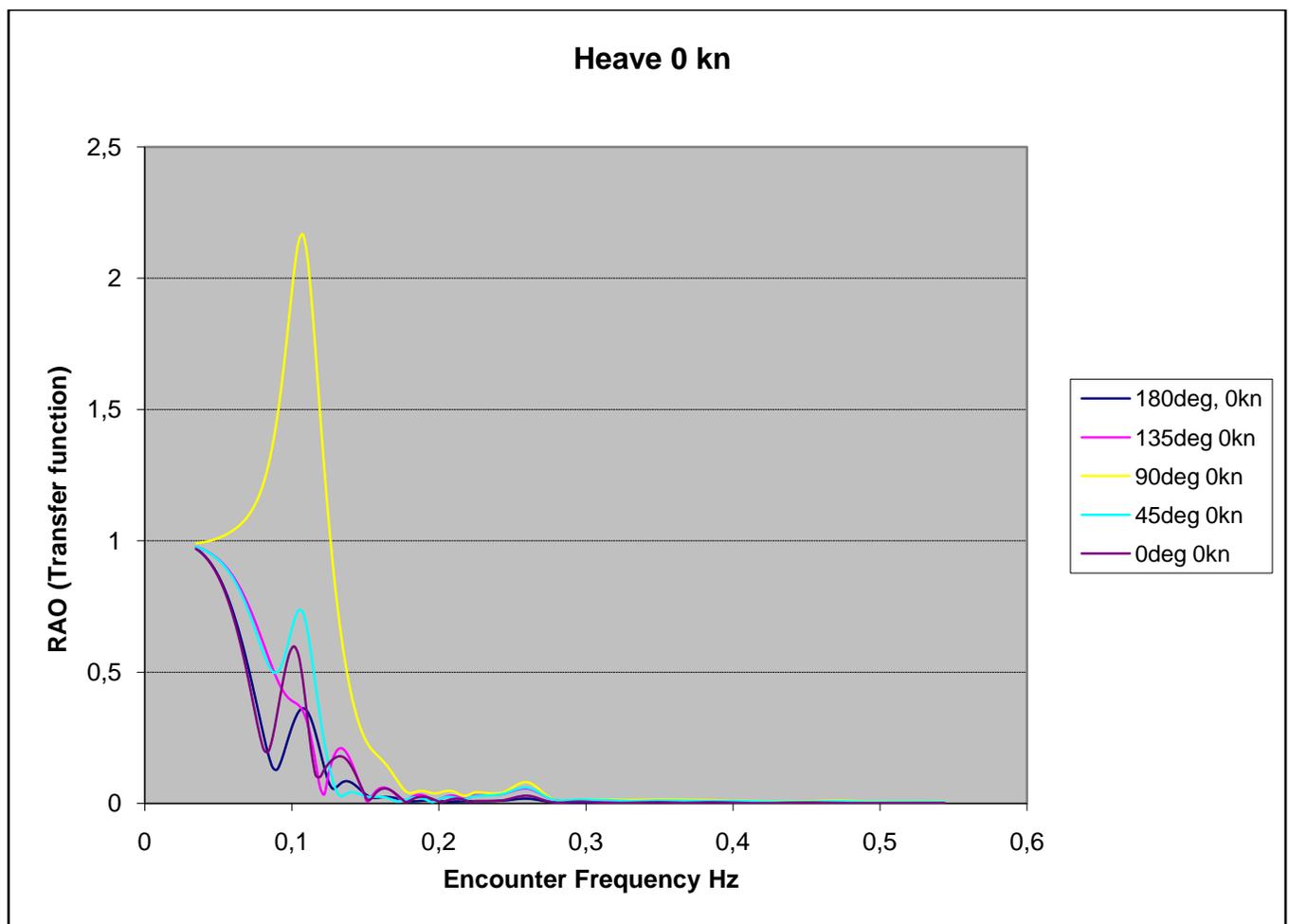


Figure 59: Heave for zero velocity for different heading angles

The largest response amplitude appeared at 90 degrees heading angle as expected. The results displayed in this graph seem reasonable. The response amplitude might be a bit too high for 90 degrees wave heading angle. However, as mentioned earlier it is also typical that the vessel will have a response

amplitude peak of greater than unity when the force on the vessel lies on frequency close to the vessel's natural period. Therefore this peak might be a bit too high. However, it is considered as reasonable and a good enough result.

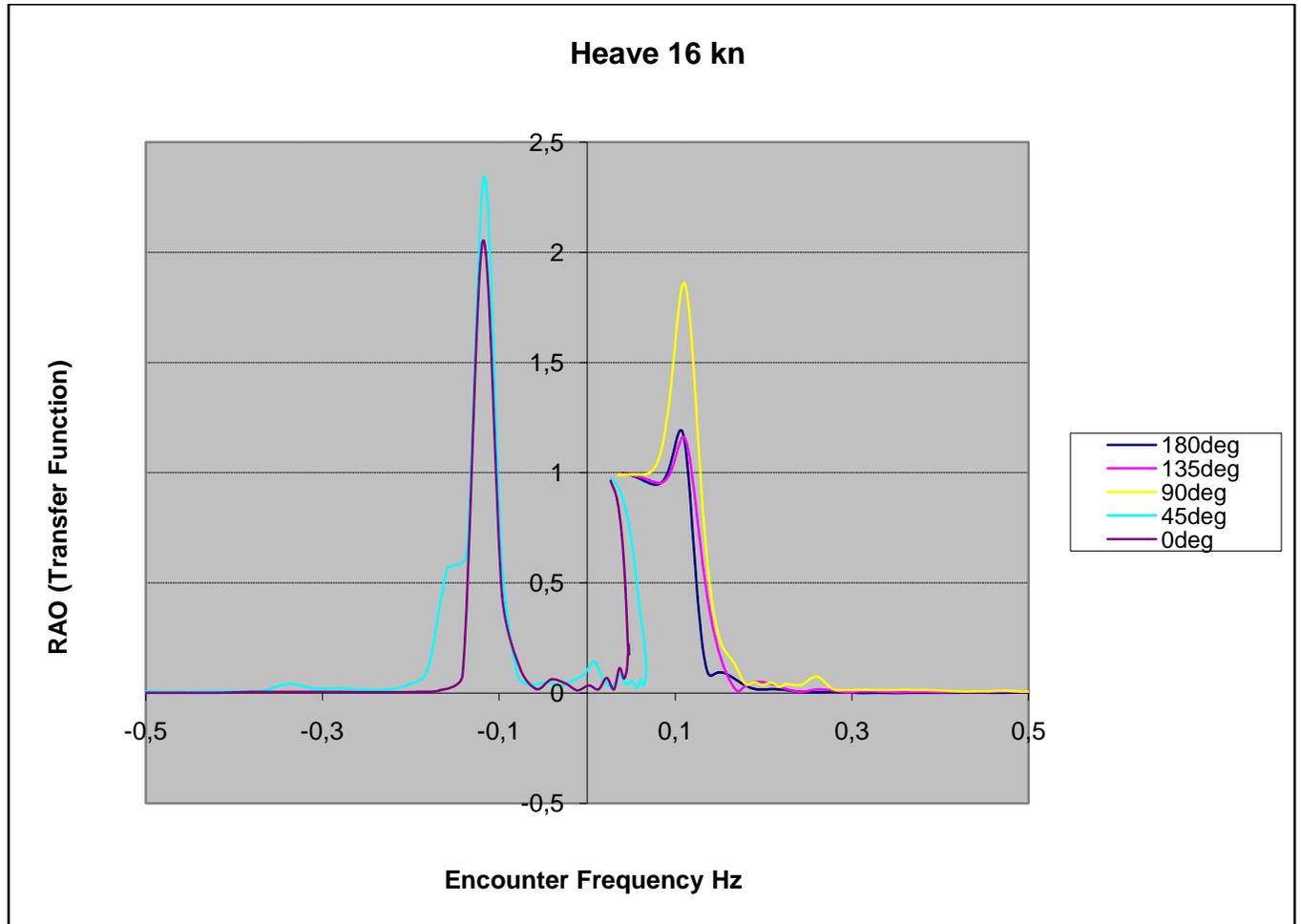


Figure 60: Heave for 16 knots for different heading angles

When the heading angle is zero and 45 degrees the encounter frequency becomes negative because the wave speed is higher than that of the ship. However, since the response amplitude is not higher than 2.5 can it be concluded that the results are reasonable for the same reasons as above. Large roll angles will influence heave and if the roll is reduced the heave might be reduced slightly. These errors would however be insignificant.

15.5.2 Roll

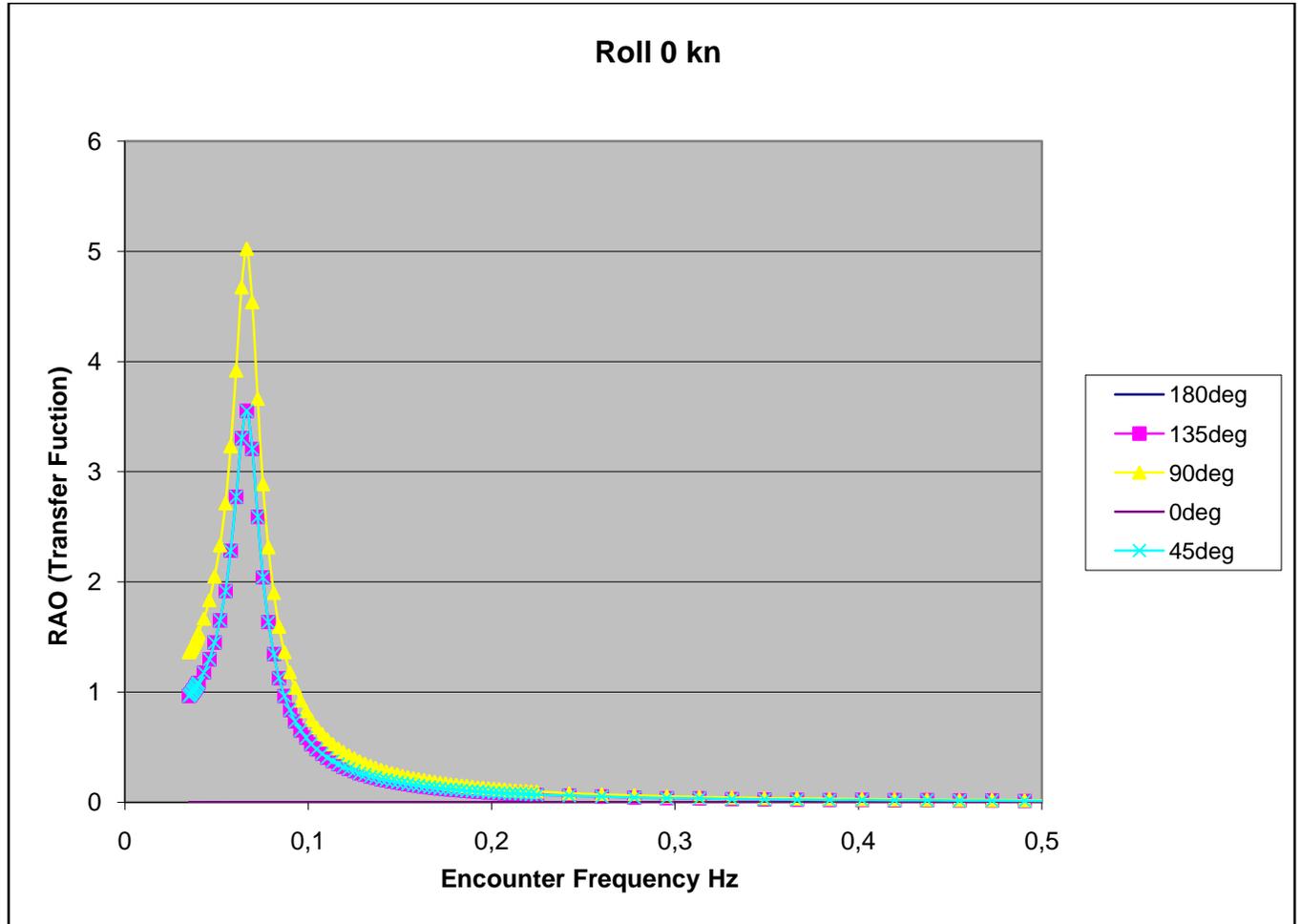


Figure 61: Roll for zero velocity for different heading angles

The results displayed in this diagram are too high. The heading angle of 90 degrees will give response amplitude of five. If it proves necessary to reduce these values a bilge keel may be utilized.

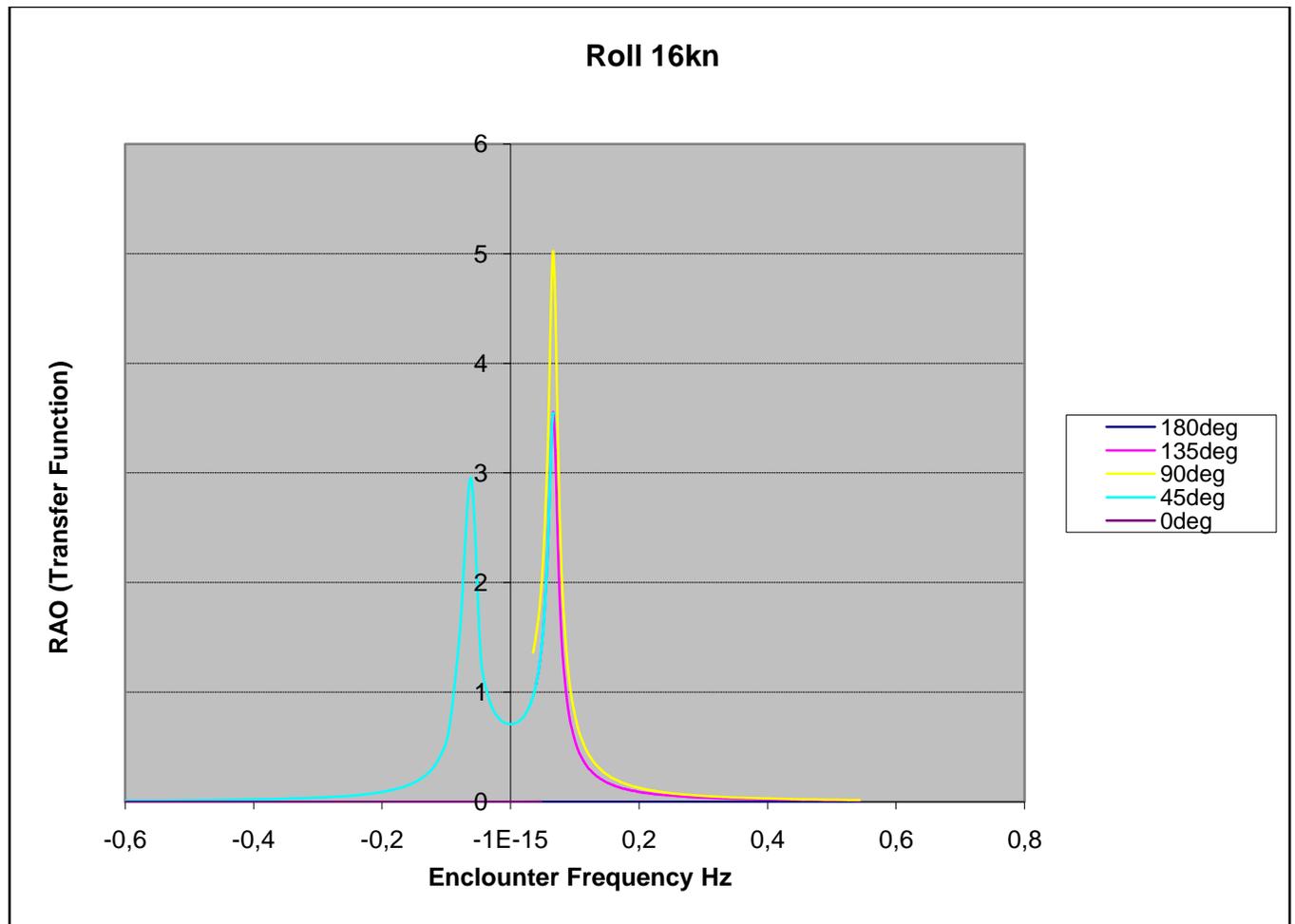


Figure 62: Roll for 16 knots for different heading angles

The roll is too high however by installing a bilge keel the roll can be reduced to a reasonable level as mentioned earlier.

15.5.3 Pitch

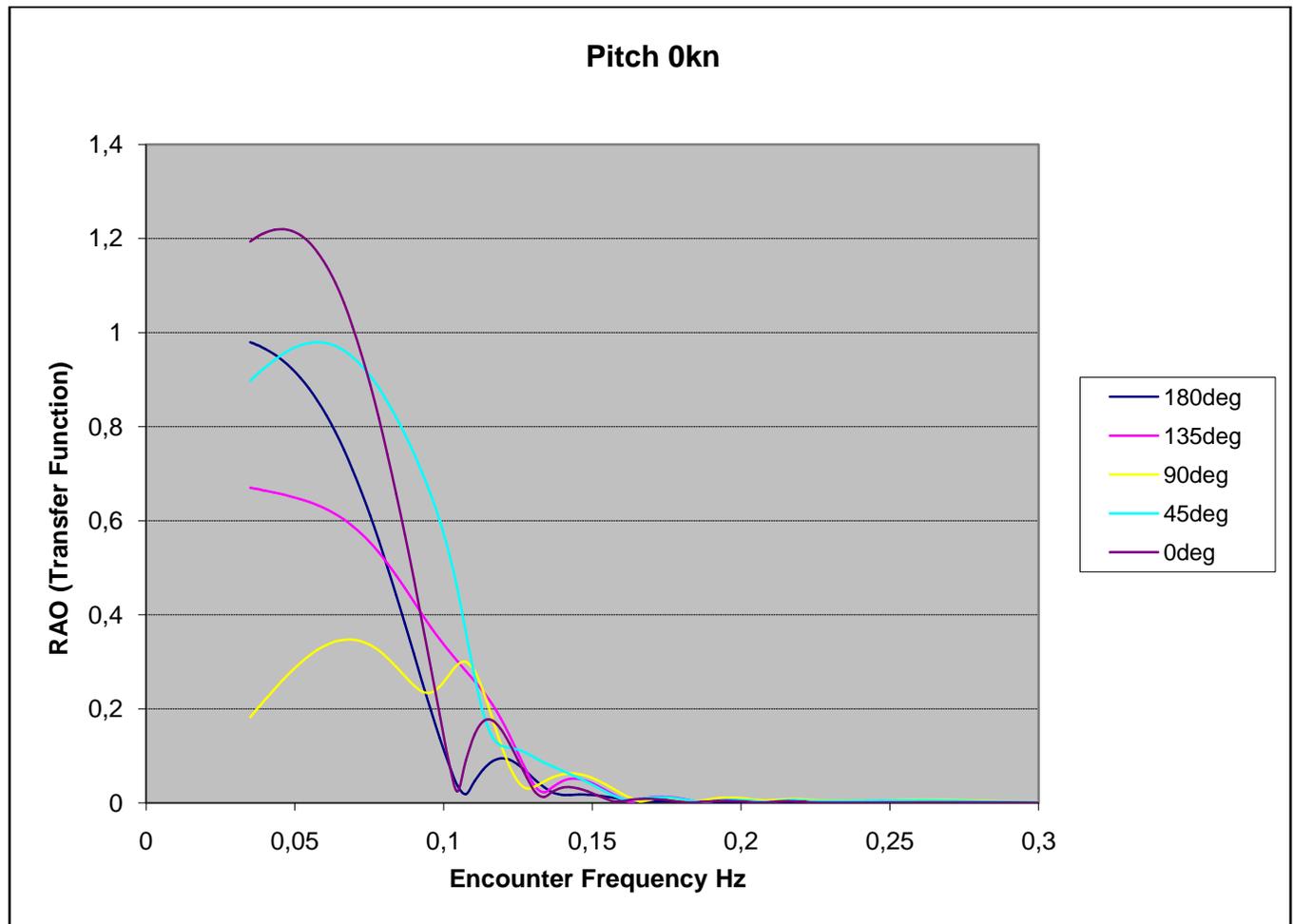


Figure 63: Pitch for zero velocity and for different heading angles

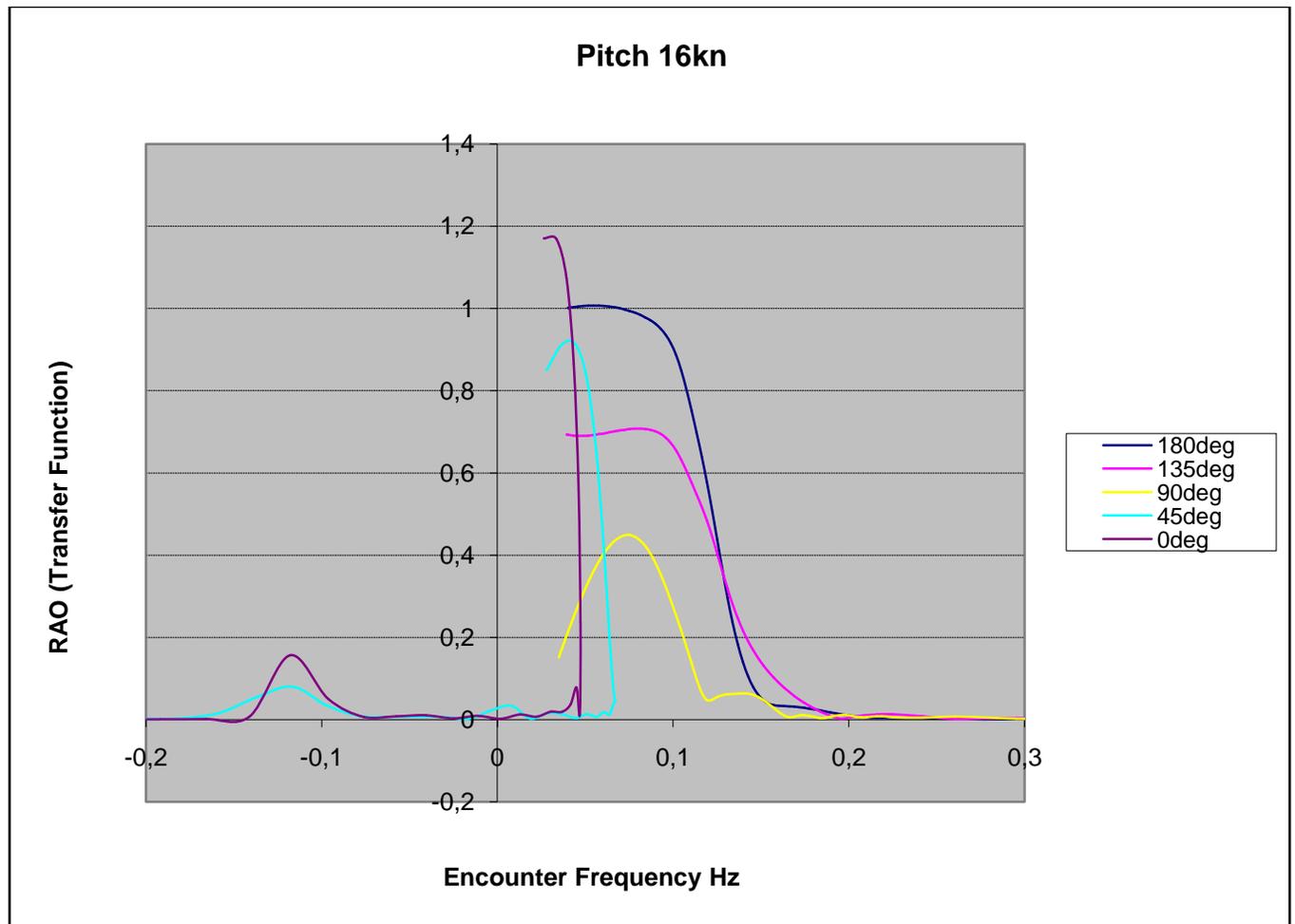


Figure 64: The Pitch for 16 knots and for different heading angles

The response amplitude for pitch for the vessel in transit and zero velocity seem reasonable and within the limits of acceptance.

15.6 Motion Sickness Incidence

The discomfort level on board can be measured by using the motion sickness incidence program in Seakeeper. The bridge is the foremost and highest point where there will always be people working. Its placement will usually make the point of highest acceleration and therefore the worst place in term of motion sickness. The bridge is therefore the place where the level of sea sickness will be calculated.

The motion sickness has also been calculated for midship and keel height. The motions sickness incidence is slightly less at the midship and keel height than at the bridge. This is reasonable.

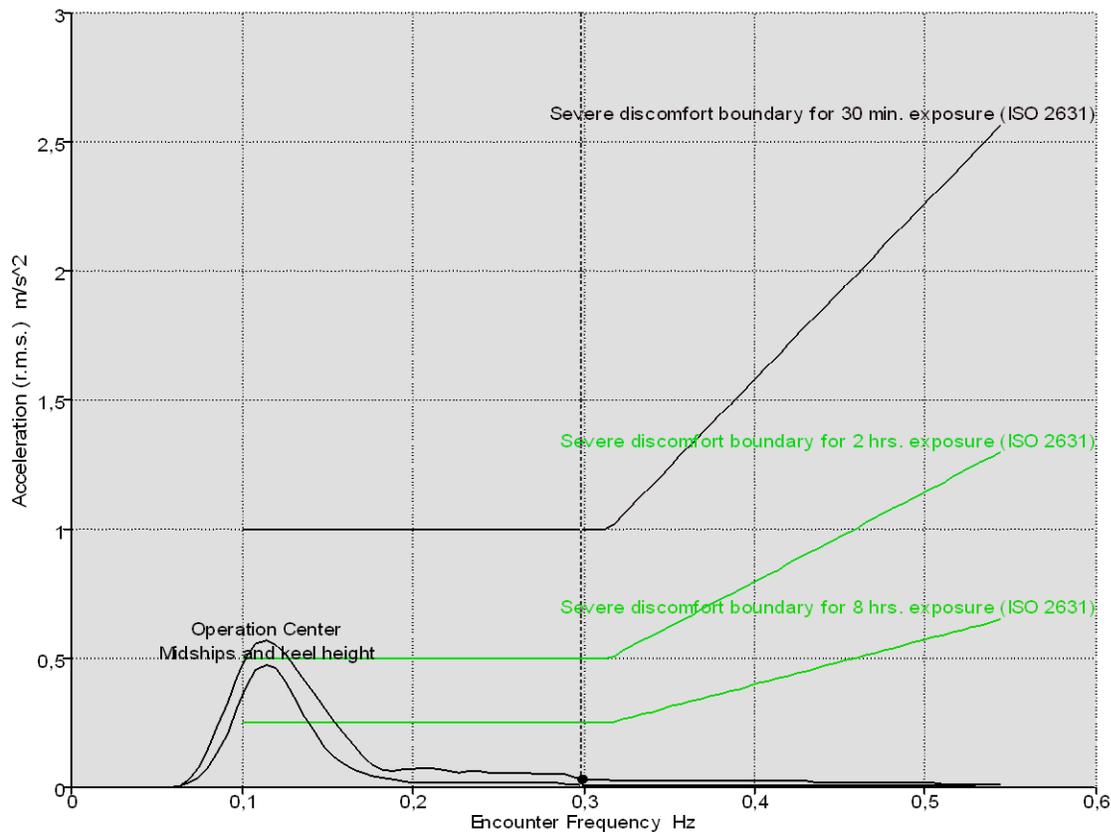


Figure 65: Motion Sickness Incidence curve

When the ship travel in 16 knots and the heading is 90 degrees the discomfort level is highest. This was found out by doing several tests with different angles and velocities. However, only the important values are represented above. As seen in the figure above the MSI curve give out information that the level causes severe discomfort after approximately two hours. However, since it is not likely that the heading angle will stay at 90 degrees for as long as two hours, it can be concluded that the sea sickness level is most likely acceptable.

15.7 Conclusion

After doing the roll analysis it is concluded that it is necessary to increase the damping factors by installing a continuous bilge keel, since the roll seemed a bit too high. The high peaks in the diagrams can be explained by that the forces on the vessel lies on frequencies close to its natural periods, and resonance may occur. A bilge keel stretches out to cover approximately 25 to 50 percent of Loa amidships, with a depth of 3 percent of breadth. According to the compendium Marin Teknikk 3 (60) there will be a 35 percentage roll reduction when installing a bilge keel. However, the bilge keel is vulnerable and maintenance is high but according to the results it will be necessary and also a good alternative. The results show that there are no reasonable hazards except the working condition onboard that shows that the motions sickness rate might be a bit too high.

15.8 Controllability

Special measurements must be performed to ensure sufficient controllability of the ship. The ship must be able to fulfil the IMO requirements of IMO res. MSC. 137(76) given in the table below.

Manoeuvring type	Criteria
Turning ability: Turning circle, port and starboard, 35° or maximum rudder angle permissible at the test speed	Advance < 4.5·L Tactical diameter < 5·L
Initial turning ability: Rudder angle ± 10°. Measure track reach by time the heading has changed 10° from original heading.	Track reach < 2.5·L
Yaw checking ability: -10°/10° zig-zag test	Value of first overshoot angle should be: < 10°, if L/U < 10s < 20°, if L/U > 30s < (5+1/2L/U)°, if 10s < L/U < 30s Value of second overshoot angle should be: < 25°, if L/U < 10s < 40°, if L/U < 30s < (17.5+0.75 L/U)°, if 10s < L/U < 30s
Yaw Check ability: -20°/20° zig-zag test	Value of first overshoot angle to be:<20°
Stopping ability: Full astern stopping corresponding test from speed of at least 90% of the ships speed corresponding to 85% of the maximum engine output	Track reach to be: < 25° However, this value may be modified by the Administration where ships of large displacements make this criterion impractical, but should in no case exceed 20 ship lengths.

Table 61: Standard requirements for satisfactory manoeuvrability (IMO res. MSC. 137(76))

These requirements are tested on sea trials after the ship has been built, or by model testing. Since none of these alternatives are possible at this stage of projecting, the Clipper Victory, comparison ship, will be used as a reference. Since the Clipper Victory fulfil these requirements with a slightly higher block coefficient, higher prismatic coefficient and about the same propulsion power it is assumed that the vessel carbon chain gas carrier also will fulfil the requirements.

16 Structural Analysis

The ship is to fulfil the DNV Rules of classification of ships issued by DNV. In this chapter there has been performed a crude global and local strength analysis of the mid ship section, as well as calculations on the structure of the pressure tanks. Although a thorough analysis like the finite element method is preferable, it is very complex and time demanding. Of the same reason, only the most important load conditions have been considered.

The global and local forces at the mid ship section have been compared with the values calculated from formulae in the DNV Rules, before deciding the dimensions of the plates and scantlings. This has been an iterative process where the cross section has been altered until the requirements were met in a satisfactory way. The computer program DNV Rule Check with DNV Nauticus Hull has been used to confirm all results and present them graphically. The program uses formulae from the Rules. Buckling analysis has been performed by the program only. This is due to the time restraint.

16.1 Material Selection

To fulfil the Rules of Classification the materials have been chosen from, and according to, the Metallic materials section of the already-mentioned DNV Rules (31 s. Pt.2 Ch.2).

The NV-NS steel is chosen for the hull structure. This is a steel often used in ship building, thus making it easy accessible.

The properties are as follows:

E-modulus	Yield stress	Poisson's ratio	Thermal expansion coefficients
[Nmm ⁻²]	[Nmm ⁻²]	[-]	[T ⁻¹]
206,000	235	0.3	10 ⁻⁶

Table 62: NV-NS Steel

The NV 4-4 L steel has good mechanical properties, even in very cold temperatures, and has been chosen for the pressure tanks.

The properties are as follows:

E-modulus	Yield stress	Poisson's ratio	Thermal expansion coefficients
[Nmm ⁻²]	[Nmm ⁻²]	[-]	[T ⁻¹]
206,000	325	0.3	10 ⁻⁶

Table 63: NV 4-4 L Steel

16.2 Longitudinal Strength

The Rules states that the required section modulus is satisfied along the hull girder when calculated for the mid ship section if:

- Scantlings along bottom and deck are unaltered within 0.4L of mid ship section

- Scantlings outside 0.4L of mid ship section are gradually reduced to local requirements.

For satisfactory calculations of the forces working on the ship it is for a liquefied gas carrier, as specified by the DNV (DNV Pt.3 Ch.1), necessary to do so for every relevant loading condition. This includes (DNV Pt.3 Ch.1):

- Homogeneous loading conditions for all approved cargoes
- Ballast conditions
- Cargo condition where one or more tanks empty or partially filled or where more than one type of cargo having significantly different density is carried
- Harbour condition for which an increased vapour pressure has been approved
- Docking condition afloat

The most critical load situations will be with the pressure tanks:

- Empty
- Completely filled with LCO₂

Thus, these are the only load conditions analyzed in this report. In a more thorough study all of the mentioned conditions should be addressed properly.

The loading of the vessel is calculated by dividing the ship into several sections. The load and buoyancy has been considered constant over each of the sections. To get a satisfactory good accuracy, the ship should be divided into a large number of sections (e.g. 100 – 200 or more) (61). As this would be a very difficult and time consuming process, only 26 sections have been used in this report. Every section will then be 11 meters, except for the sections outside 22 and 198 meters from AP which are 5.50 meters.

16.2.1 Still Water Bending Moment

The dimensioning still water bending moment is given by

$$M_s = \begin{cases} -0.065 \cdot C_w L^2 B (C_B + 0.7) [kNm] & \text{sag} \\ C_w L^2 B (0.1225 - 0.015 C_B) [kNm] & \text{hog} \end{cases}$$

$$C_w = 10.75 - \left(\frac{300 - L}{100} \right)^{\frac{3}{2}} \text{ for ships where LPP is between 100 and 300 meters.}$$

C_B = Block coef.

The dimensioning still water bending moment is:

M_s hog	[kNm]	1 654 622
M_s sag	[kNm]	-1 507 076

Table 64: Dimensioning still water bending moment

The longitudinal shear force and bending moment are found by using the following formulae (61):

$$Q(x) = - \int_{-L/2}^{L/2} q(s) ds$$

$$M(x) = - \int_{-L/2}^{L/2} Q(s) ds$$

s = Integration variable

q(s) = Force (weight) distribution.

However, as the load distribution is not continuous, it is necessary to perform a numerical analysis.

The trapezoidal rule equation can be used, and gives:

$$Q(x_k) = - \sum_{i=1}^k \frac{q(x_{i-1}) - q(x_i)}{2} \Delta x_i$$

$$M(x_k) = - \sum_{i=1}^k \frac{Q(x_{i-1}) - Q(x_i)}{2} \Delta x_i$$

Using this equation, the bending moment is found.

At mid ship, this does not exceed the value set by the DNV rules, and it is sufficient to use the dimensioned values from the Rules.

16.2.2 Wave Bending Moment

The dimensioning wave bending moment is given by

$$M_w = \begin{cases} -0.11 \cdot C_w L^2 B (C_B + 0.7) [kNm] & \text{sag} \\ 0.19 \cdot C_w L^2 B C_B [kNm] & \text{hog} \end{cases}$$

This results in:

M_w hog	[kNm]	2 402 890
M_w sag	[kNm]	-2 550 436

Table 65: Dimensioning wave bending moment

16.2.3 Mid Ship Section

Vertical longitudinal bending stress is found by the equation

$$\sigma = \frac{M_s + M_w}{I} \cdot y = \frac{M_s + M_w}{Z}$$

DNV allows maximum bending stress $\sigma_1 = 175 f_1$ [N/mm²], where $f_1 = 1.0$ for NV-NS steel.

The required section modulus is then found by

$$Z = \frac{|M_s + M_w|}{\sigma_1}$$

The Rules states that the moment of inertia about the transverse neutral axis shall be no less than

$$I = 3 \cdot C_w L^3 B \cdot (C_B + 0.7)$$

The required and actual values are then

	Unit	Required	Actual
I	[m ⁴]	153.0	251.1
Z _{deck}	[m ³]	23.2	23.3
Z _{bottom}	[m ³]	23.2	30.5

Table 66: Section modulus and moment of inertia

The actual values are calculated from the cross section presented in the end of this chapter. The required plate thicknesses are calculated and confirmed through a scantling generation in DNV Rule check. The requirements of the thickness of the plates vary over the cross section, and only the maximum values are shown in this chapter.

Plate	Unit	Required	Actual
Bottom	[mm]	19.0	20.0
Bilge	[mm]	14.8	15.0
Side	[mm]	18.5	25.0
Strength deck	[mm]	16.8	25.0
Inner bottom and inner side	[mm]	16.9	20.0

Table 67: Plate thicknesses

The final mid ship cross section is shown in the figure below.

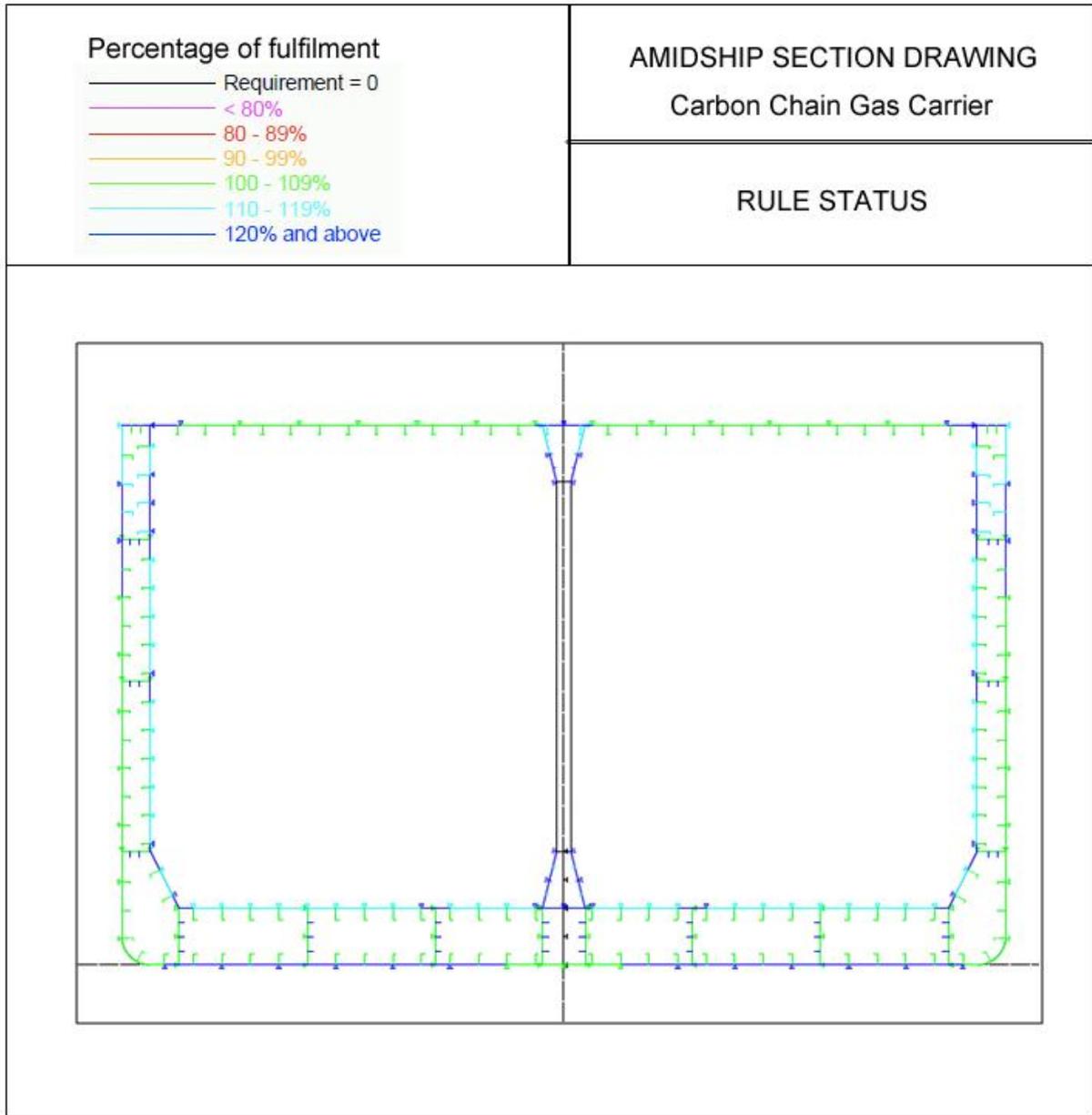


Figure 66: Mid ship section

16.3 Pressure Tanks

The tanks are independent type “C” as specified by the DNV. They are freely supported to avoid loads from ship deflection, and arranged such that the weight is evenly distributed all over the inner bottom. They are expensive and it is important that the amounts of material, and thus the thickness of the tanks, are minimized. As the tanks are cylindrical (except for the ends, where they are ellipsoidal), simple mechanics dictates that the stress experienced can be written as follows:

$$\sigma_{\theta} = \frac{r}{t} p$$

$$\sigma_z = \frac{r}{2t} p = \frac{\sigma_{\theta}}{2}$$

It is obvious that the stress in the theta-direction are the most critical as they are double the stress in z-direction.

As mentioned, the ends of the tanks are ellipsoidal. The ellipsoidal shape in question almost resembles that of a sphere. The stress of a sphere is lower than that of a cylinder in tangential direction:

$$\sigma = \frac{\pi r^2 p}{2\pi r t} = \frac{rp}{2t}$$

The stress in the ends will be somewhat higher, but still not exceed that of the cylinder in tangential direction. Therefore, no more calculations are required.

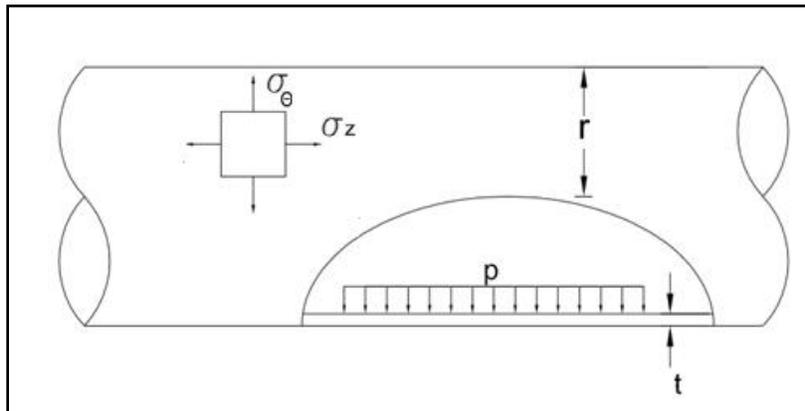


Figure 67: Pressure tank

With a fixed shape and maximum gauge pressure, the thickness is the variable in the stress function, and the required thickness must be found.

The calculations have been done using several assumptions and simplifications, thus making them somewhat inaccurate. It has been deemed necessary to do so because of the time restraint. The calculations made have been done in accordance with the Rules.

The following assumptions have been made:

- Same thickness over the entire tank
- Circular cross sections
- No sloshing.
- No vibrations: Loads from vibrations have been neglected due to the complexity of such calculations. They should be included in further studies.
- No loads from ship deflection: The tanks are assumed to be freely supported, and will therefore not experience any minor deflections of the ship.
- Loads from tanks, insulation mass, equipment, towers and attachments have been neglected.

Note that this is a rather crude method, and a thorough FEM (Finite Element Method) analysis is preferred and should be included in further studies. As the loads vary over the length and height of the tank, the thickness should do so as well. However, only the required thickness at its maximum has been calculated in this chapter. LCO₂ is heavier than LPG, and has been used in the calculations.

16.3.1 Strength Analysis

Even though the tanks are not part of the hull structure, they are exposed to a variety of loads. As the already-mentioned rules from DNV dictates, the calculations should include the following (where the above-mentioned assumptions and simplifications have been taken into consideration):

- Internal pressure
- External pressure
- Dynamic loads from ship motion
- Thermal loads
- Cargo weight (static pressure)

16.3.1.1 Internal and External Pressure

The internal and external pressure will give a resultant internal gauge pressure p_g . This will give global strains on the tank as explained from the formulae above.

According to required design pressure found in Chapter 3, the design gauge pressure is set to:

$$P_g = 0.5 \text{ Nmm}^{-2}$$

16.3.1.2 Dynamic Loads

Dynamic loads take into account the long term distribution of wave-induced motions that the ship will experience during its lifetime. The dynamic loads are calculated from formulae in the mentioned DNV Rules.

Dynamic pressure is found from

$$p_{gd} = \frac{a_{\beta} Z_{\beta} \rho}{1.02 \cdot 10^4} \text{ [bar]},$$

Where:

a_β = maximum dimensionless acceleration found from formulae in the Rules.

Z_β = largest liquid height at measured point.

ρ = maximum density of cargo.

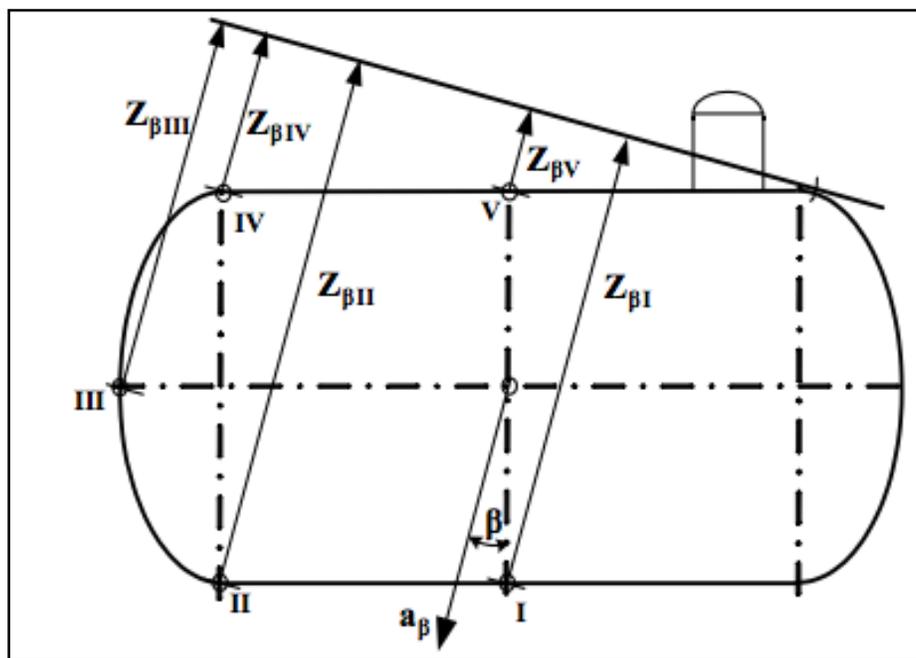


Figure 68: Liquid heights Z_β in β direction [(62)]

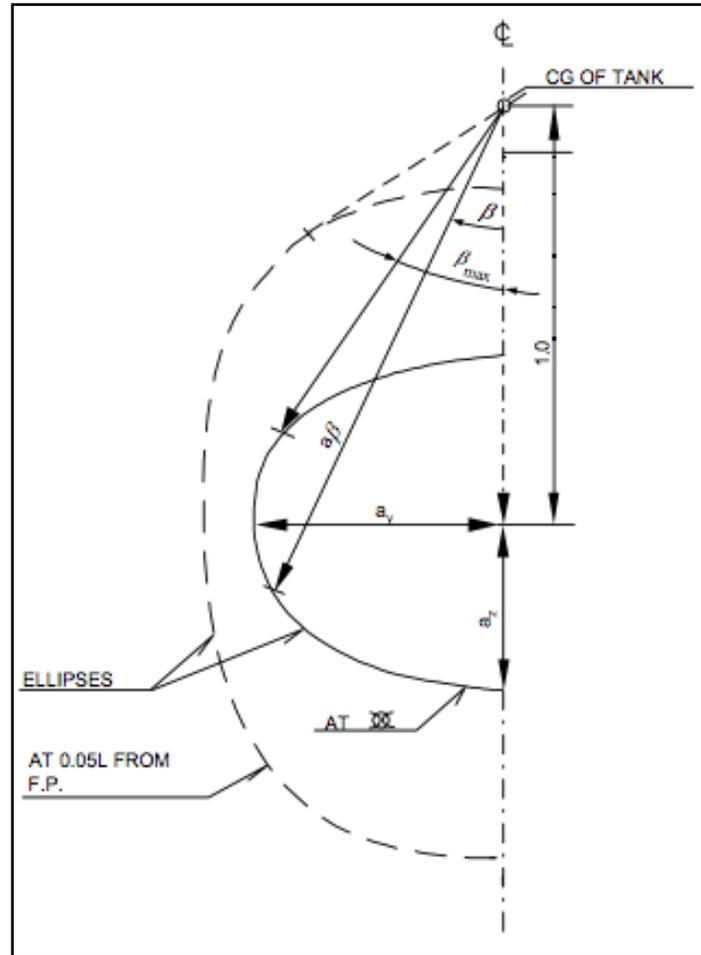


Figure 69: Resulting acceleration (static + dynamic) a_{β} in arbitrary direction β [DNV Rules 2008 Pt. 5, Ch. 5, Section. 5]

Using Figure 66 as well as simple mathematical formulae for ellipses, the acceleration a_{β} is found. The largest liquid height Z_{β} has been calculated for $Z_{\beta II}$ as the pressure is at its maximum there. This is shown in Figure 66.

The maximum dynamic pressure is then found to be:

$$P_{gd} = 0.30 \text{ Nmm}^{-2}$$

16.3.1.3 *Weight of Cargo*

The total mass of the cargo will cause additional static pressure p_s .

$$p_s = \rho g \cdot 2r$$

$$p_s = 0.16 \text{ Nmm}^{-2}$$

$$\rho = \text{cargo density} = 1150 \text{ kgm}^{-3}$$

16.3.1.4 *Thermal Loads*

The thermal expansion (or rather, contraction in this special case) is decided by the thermal expansion coefficient α of the material in question as well as the difference in temperature, dT . The temperatures are given as the ones outside and inside the tanks respectively. The thermal induced strain ε^T is then given by:

$$\varepsilon^T = \alpha dT$$

$$dT = 60$$

α = thermal expansion coefficient

This strain will in turn give the thermal induced stress.

$$\sigma^T = \frac{\varepsilon^T E}{1 - \nu^2} (1 + \nu)$$

With a negative temperature change dT , the thermal induced stress is found to be:

$$\sigma^T = 17.66 \text{ Nmm}^{-2}$$

16.3.1.5 *Total Stress*

The total stress is given by the formula mentioned above.

$$\sigma_{\theta, total} = \frac{r}{t} p_{total} + \sigma^T$$

$$p_{total} = p_s + p_{dg} + p_g$$

Total stress in theta-direction is then:

$$\sigma_{\theta, total} = 183.7 \text{ Nmm}^{-2}$$

16.3.1.6 *Dimensioning Thickness*

Unforeseen events and neglected elements have been taken into consideration by a safety factor f . This has been set to:

$$f = 2$$

Both von Mises and Tresca yield criteria have been considered. Von Mises is more conservative, and has been taken as the dimensioning criterion.

Von Mises criterion is given by:

$$\sigma_j = \sqrt{\sigma_{\theta}^2 + \sigma_z^2 - \sigma_{\theta}\sigma_z}$$

With the safety factor taken into the equation, the required thickness at its highest is then found to be:

$$t = 39.0 \text{ mm}$$

17 Endurance Calculations

In Owner's Requirements it is mentioned that the vessel should be able to transport captured CO₂ from Europe to Norwegian geologic formations for storage. The distance between Brunsbüttel and Mongstad equals 500 nautical miles and the service speed of the vessel is 16.7 knots per hour. Earlier estimations show that one round trip will take approximately 5.5 days. The calculations are based on the fact that there will be no on/offloading offshore. Since the ship's route may be altered by different owners during the vessel's lifetime, the endurance will be set higher than 5.5 days. It is also preferred that the ship can transport cargo to other countries and different harbours than Mongstad and Brunsbüttel since this will give the ship a bigger market to operate in. Minimum endurance is set to be 12.5 days due to the arguments above.

The safety factors and the values are taken from the compendium (62).

The fuel tanks are designed for an endurance of 12.5 days. A sea margin of 20 percent is added to the power calculated for trial condition to get the power needed for the same speed in actual operation. The specific fuel consumption of the medium speed lean burn gas engine was given to be 155 g/kWh. The density of the fuel is assumed to be 423 kg/m³. The fuel tanks are to be placed in the aft of the vessel. Necessarily space for fuel was found to be approximately 2 000 cubic meters.

To calculate the necessarily capacity of the lubrication oil tanks an endurance of 12.5 days has been used. A safety factor of 4 is added. The specific consumption of lubrication oil is 1.5 kg/kWh. The lubrication oil tank is placed in the double bottom in the aft ship. Necessary space for a lubricant oil tank was found to be 13 cubic meters.

17.1 Fresh Water

The fresh water tank capacity was calculated using an endurance of 40 days, a fresh water consumption of 200 litres per day for each crew member, and a safety factor of 20 percent. The total fresh water consumption was found to be 3.8 tonnes per day. The necessary space for a fresh water tank was found to be approximately 180 cubic meters.

17.2 Sewage Holding

Sewage tank capacity was calculated for an endurance of 40 days. Estimated daily sewage production for each crew member was set to be 75 litres per day. A safety factor of 20 percent was added, and necessarily space for sewage was found to be approximately 70 cubic meters.

TANKS AND VOID SPACES						
Name/ Use of Space:	Consump. [g/kwh]	Consump. [ton/day]	Range [nm]	Endurance [days]	Margin factor	Volume [m ³]
Fuel	155,1	66,1	5 000	12,5	1,2	1 954
Lubrication Oil	1,5	0,6	5 000	12,5	4,0	13
	l/crew/day					
Fresh water	200	3,8		40,0	1,2	182
Sewage Holding	75	1,4		40,0	1,2	68
BW, side tanks						5 880
BW, double bottom						10 690
BW, for						4 500
VOIDS etc						1 000
Tanks and Void Spaces			0,49 m ³ /DWT			24 288

Table 68: Showing tanks and void spaces

18 Manning

Safe manning is, according to IMO resolution A.890(21)

“a function of the number of qualified and experienced seafarers necessary for the safety of the ship, crew, passengers, cargo and property, and for the protection of the marine environment.”

The manning shall satisfy the IMO regulations. (63).

In the table below is an overview of the manning assumed to be needed on board during transit.

Position	Onboard	Ashore
Captain	1	1
Chief officer	1	1
First Officer	1	1
Chief Machine Engineer	1	1
First Machine Engineer	1	1
Cook	1	1
Temporary Repair Crew	3	0
Subordinate	10	0
Total manning	19	6

Table 69: Manning onboard during transit and manning needed ashore.

The manning estimates are based on the comparison ship “Clipper Victory”. Clipper Victory is sailing with 22 crew members, and has a three watch system (four hours on/eight hours off) and are sailing in international waters. It can be concluded that the manning on board for this ship will be less. This is due to the fact that it will only travel in European coastal waters and may have a two watch system (six hours on/six hours off).

It is assumed that this ship will have a European captain and European engineers and officers since factors as skill, experience and training are important, and European engineers and Officers have a good reputation in this respect. However, as the manning expenses are less for Indian/Philippine crew will all subordinates be of Indian/Philippine nationality.

The repair crew will be one week on board each year, and they will work 12 hours each day (32)

A port captain is needed in each port since the on/offloading procedures are complicated (32).

Since this ship will sail with both European and Indian/Philippine crew, English will be the working language to ensure effective crew performance in safety matters. According to IMO each seafarer shall be required to understand, give orders and instructions, and report back in English.

19 Health and Safety

To comply with class regulation, safety systems onboard must meet specific standards set by the International Maritime Organization (IMO), International Convention for Safety of Life At Sea (SOLAS) and DNV regulations.

In case of emergency an emergency generator of 330 kW is required. The generator will give power to emergency fire pumps, HI-FOG system, emergency lighting, emergency cargo cooling, lifeboat winches, air compressor and other safety systems.

Transporting CO₂ will make the vessel more vulnerable and special types of tanks as mentioned earlier must be used for transporting the liquid. A port captain is placed in each harbour to control the on/offloading procedures so that it is kept under strict control. In addition the bellows inside the tanks used for Rapid Purge, and the tank themselves will be regular controlled to increase the safety during on/offloading.

19.1 Fire Fighting Devices

The engine rooms, cargo area, deckhouse, compressor rooms and workshops are the main risk zones for fire. The safety in the cargo area is especially important and inert gas regulations must be taken into account as mentioned in an earlier chapter. To avoid rapid uncontrollable spreading of fire, the ship must be divided into fire zones according to SOLAS class divisions. "A" class and "B" class fire doors are to be installed where they are needed and according to the regulation. (64)

Cargo ships are required, by SOLAS regulations, to be equipped with fixed fire detection and fire alarm system and/or an automatic sprinkler, fire detection and fire alarm system. As an alternative to sprinkler systems, a Hi-Fog system produced by Marioff will be evaluated. (51)

Fire alarms, detection systems and fire announcement system are to be installed so to rapidly detect the onset of fire. They will be directly connected with control panels at the navigation bridge and the engine room or in a place where a responsible member of the crew is on duty if the bridge is unmanned. (65).

Smoke detectors shall be installed and escape routes, stairways and exit shall be marked by emergency lighting according to SOLAS regulations. (66)

Furthermore there shall be at least two jets of water and two hydrant and four independent driven fire pumps on board. Twelve fire hoses will be provided since this vessel carries dangerous goods. In addition will the vessel contain five portable fire extinguishers and fixed deck foam fire-extinguish system. Since this ship is a tanker there will be need of two fire fighters outfits in addition. (67).

Hi-fog water mist fire protection is an alternative by Marioff used for extinguishing fire in all compartments. The Hi-fog system suppresses fire quickly by utilizing water of high pressure up to 140bar so that water and smoke damage is minimized. It also prevents flashover and re-ignition. The advantages of the Hi-Fog systems are that they only triggers in the affected room and will minimize fire and water damage. (51)

19.2 Life Boats

This ship has a manning of nineteen persons and will have

“[...] one free-fall lifeboat capable of being free-fall launched over the stern of the ship and rescue the total number of persons on board”

(68)

One MOB will be placed near the bridge to be launched quickly in an emergency situation.

Six life rafts will be placed on deck in according to SOLAS regulations.



Figure 70: Showing the vessel with the free-fall lifeboat at the stern

19.3 Additional Safety Equipment

Lifebuoys, lifejackets, communication and medical equipment will be kept on board in accordance with SOLAS and IMO.

19.4 Overview over Safety Equipment

In the table below is a simplified overview over all the Safety Equipment on board. Only the most important safety equipments have been mentioned.

Fire Fighting Device	Boats	Additional Safety Equipment
HI-FOG sprinkler system	1 MOB	Lifebuoys
Fire detection/Fire alarms	1 Free-fall lifeboat	Life Jackets
Smoke detectors	6 Life Rafts	Communication Equipment
Emergency lighting		Medical Equipment
2 Hydrants		
2 Jets of Water		
12 Fire Hoses		
5 Portable fire extinguisher		
Fixed deck foam fire extinguisher		

Table 70: Overview over safety equipment

20 Cost Analysis

The cost calculations are based on coefficients found in *System Based Ship Design* (19) and coefficients found in the reports “Rapid Response Vessel” (69) and “WindFlip” (70) submitted to earlier Dr. Lisnyk Student Ship Design competitions. In this chapter an analysis of the operating- and the building- costs will be evaluated. All the calculations will take into account that the exchange rate between USD and NOK is 6.45 USD. (71)

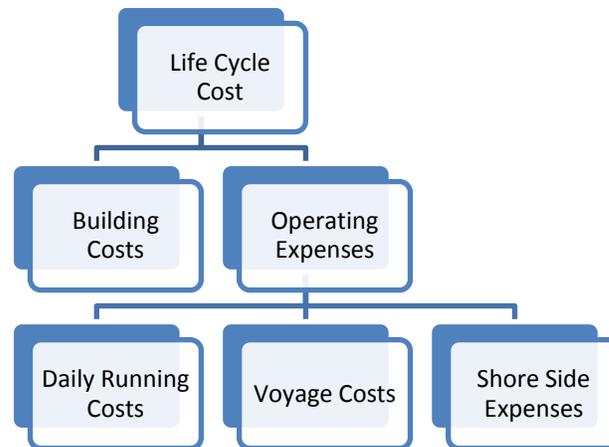


Figure 71: Life Cycle Cost

The cost analysis is divided into two main categories: Building costs and operating expenses. Operation expenses are divided into three sub groups: Daily running costs, voyage costs and shore side expenses. Each subgroup will be evaluated separately in the following chapters.

The first category that will be evaluated is the building costs since it is one of the main cost groups.

20.1 Building Costs

The building cost consists of design cost, material cost, production cost and financing costs.

Financing costs are then divided into the following four sub groups;

Financing sub groups	Percentage of total cost	Unit
Profit	5	[%]
Loan	3	[%]
Broker fees	1	[%]
Building time financing costs	-	[%]

Table 71: Financing groups

Percentage values are taken from (19). Building time financing costs are not calculated from percentage values, and it's therefore not shown in this table.

Materials are divided into several different costs groups. Payload related costs are costs related to pumps, piping and tanks. The values related to payload are assumed numbers which seemed reasonable. Other important material cost groups are: Machinery costs, interior outfitting costs, deckhouse costs, hull structure and general costs related to the lightweight. When evaluating the price for steel is it assumed that tank steel will cost twice as much as steel used in the hull.

In addition a safety margin of five percentages will be added to the total material and labour costs.

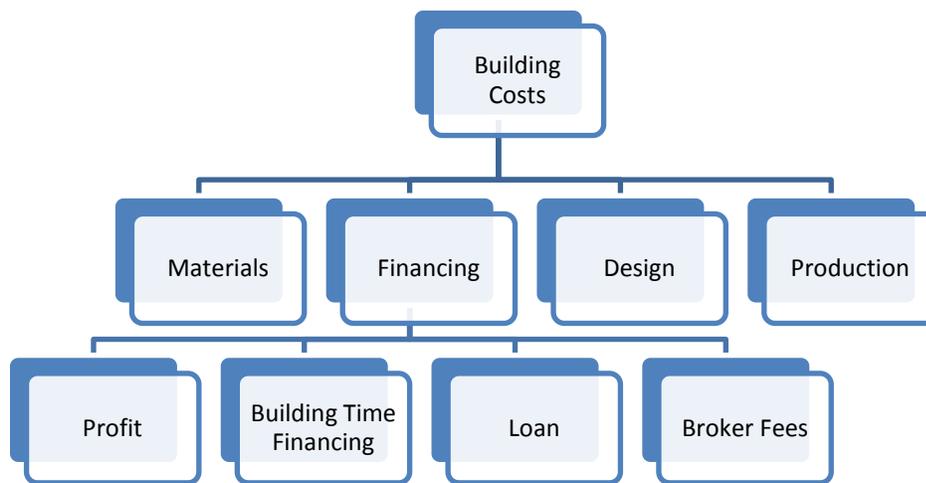


Figure 72: Building Costs

In the table below is an overview over the total cost related to the material and the labour costs. All coefficients except those related to payload are based on coefficients found in "System Based Ship Design" (19). The price for 11 deep well cargo pumps are set to 930 233 USD (72). The price for 10 deep well pumps are then calculated to be approximately 850 000 USD. The time for installing one deep well pump is assumed to be approximately 50 hours.

The coefficients related to payload are only approximately assumed numbers.

MATERIALS AND LABOUR							
Exchange rate [USD/NOK] = 0,155							
Cost Group:	Unit	Value	Coef [NOK/unit]	Coef [h/unit]	Material [USD][10 ³]	Labour [1000 h]	
General	LWT [ton]	16 979	2 000	5	5 265	85	
Payload related	Pumps	No	10	545 454	50	846	10
	Pipings	length [m]	1 000	2 500	30	388	30
	Tanks	tank weight [ton]	4 008	5 000	30	3 107	120
Hull Structure	Hull WT [ton]	7 310	6 000	30	6 800	219	
Deckhouse	Dh WT [ton]	331	6 000	50	308	17	
Interiour Outfitting	Area [m ²]	1 294	15 000	25	3 009	32	
Machinery	Pp+Pa [kW]	22 320	3 000	2	10 381	45	
Ship Outfitting	Gross Vol [m ³]	102 563	100	0	1 590	21	
Total	LWT [ton]	16 979	2	34	31 694	578	
Reserve	[%]		5 %	5 %	1 585	29	
MATERIALS AND LABOUR	LWT [ton]	16 979	2	36	33 278	607	

Table 72: Building cost. Materials and Labour

The table above gives an overview over the total material and labour costs. They are set to be approximately 71 million USD. The total amount of hours needed are set to be approximately 0.6 million hours. In addition, the financing payment, broker fees and profits will be represented and taken into consideration when estimating the final building price. As it can be seen in the table below the final building price is estimated to be approximately 91 million USD.

PRICE ESTIMATION					
	[h/LWT]	[Hours]	[USD/h]	Price [USD][10 ⁶]	Price [USD/kg]
Design	10	169 790	54	9	0,54
Labour + Over Head	36	607 418	62	38	2,22
Material				33	1,96
Building time financing (Interestxtime/2)	6 %	months:	18	4	0,21
Total Production Cost				84	5
Profit	5 %			4	
Loan	3 %			3	
Broker fees	1 %			1	
BUILDING PRICE			Price	91	5
			Price/DWT	1 856	[USD/ton]
			Price/GT	2 963	[USD/GT]

Table 73: Building costs

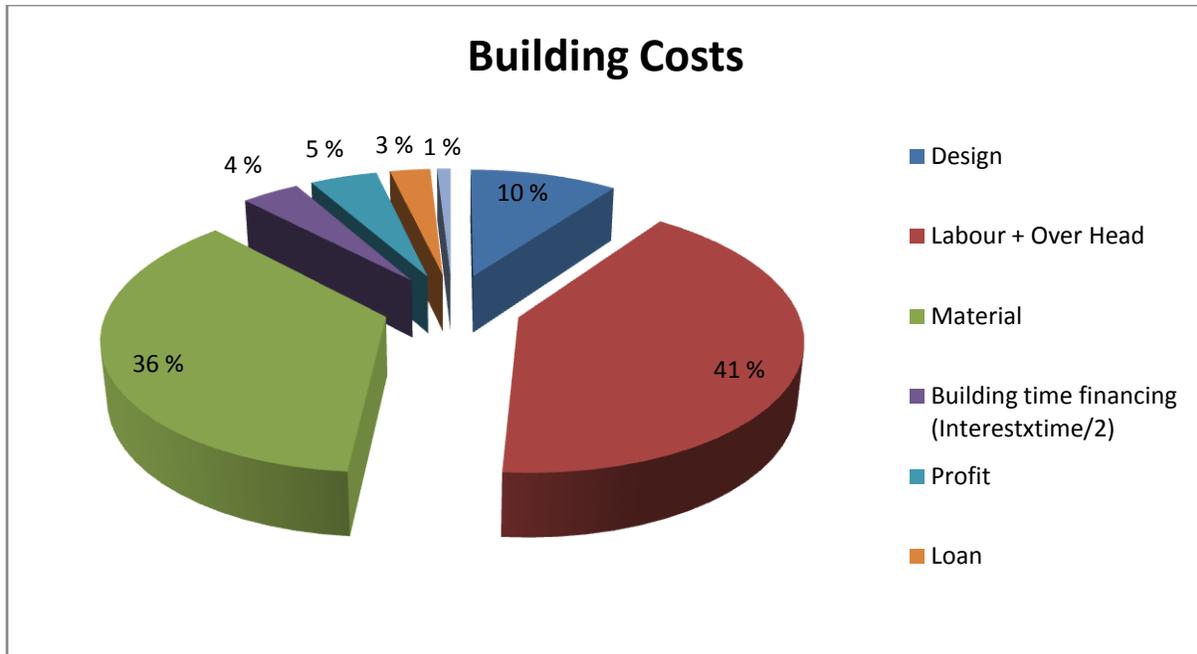


Figure 73: Building costs

The cost of labour and over head contributes to somewhat over a third of the total cost. Material costs contribute to almost half of the total building costs. Building time and financing, profit, financing payment, broker fees and design contributes to the remaining percentages.

20.2 Daily Running Costs

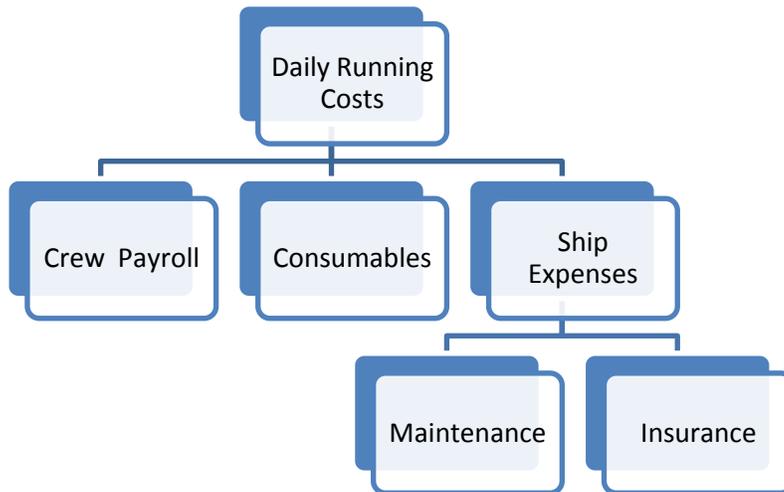


Figure 74: Daily Running Costs

The operating expenses consist of three groups as mentioned earlier. One of these groups is the daily running costs, which will be studied further in this chapter. The daily running costs includes: Crew payroll, consumables and ships expenses. Costs are estimated using a crew of 19, and three additional temporary repair workers. Two men will be needed for each position of captain, officers, engineers and

cook. The expenses of having an additional temporary repair crew from time to time are included. The additional costs in this case will be set to 30 percent (32). Ten subordinates are assumed. As they will be 11 months on board, and one month off, the additional costs are assumed to be 40 percent (32).

In the table below the annual salaries are represented for each position. The salary for the repair crew is assumed to be 50 000 USD (32) assuming they will be on board one week per year. The column additional cost, which is set to 30 percent, includes social costs, consumables, insurances, travel expenses etc. The annual salary for the crew has been estimated by use of statistics presented at the international site for Statistics Norway. (73) The total cost for the ship crew will be approximately 2 million USD annually.

MANNING EXPENSES					
Position	Onboard	Ashore	Annual Salary [USD]	Additional Cost	Payrol Total[USD]
Captain	1	1	130 000	30 %	338 000
Chief officer	1	1	130 000	30 %	338 000
First Officer	1	1	85 000	30 %	221 000
Chief Engineer	1	1	90 000	30 %	234 000
First Engineer	1	1	90 000	30 %	234 000
Cook	1	1	65 000	30 %	169 000
Repair Crew	3	0			50 000
Subordinate	10	0	20 000	40 %	280 000
Total chip crew	19	6			1 864 000

Table 74: Manning Expenses

It is assumed that the vessel will operate 357.5 days per year. The remaining 7.5 days of the year it will be necessary with maintenance which is one of the sub groups under ship expenses. The second group under ship expenses are the insurance. From *System Based Ship Design* (19) the maintenance and insurance was set to 0.5 and 0.8 percent of the total ship building price respectively. As it can be seen in the table below an additional sub group called "other costs" has been added to the ship expenses as a safety factor. This safety factor is set to be 0.5 percent of the ship building price. It results in 1.5 million in total ship expenses. The total daily running costs are then estimated to be approximately 3.5 million USD.

SHIP EXPENCES		
Cost item	%of ship building prices	Cost [USD]
Maintenance	0,5 %	456 229
Insurance	0,8 %	729 967
Other Costs	0,5 %	456 229
Total Ship Expenses		1 642 425

Table 75: Summary of Ship Expenses

20.3 Voyage Expenses

The second group in under the operating expenses is the voyage expenses. Voyage Expenses includes fuel expenses, port charges including on/offloading expenses.

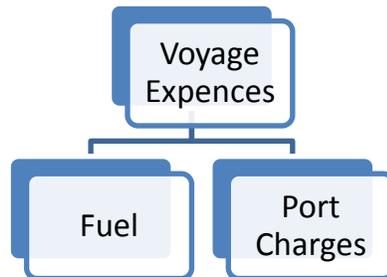


Figure 75: Voyage Expenses

In the table below is the power needed and the total annual fuel and lubricant oil consumption in port, during manoeuvring, during transit and on/off loading represented. In addition the total annual fuel, lubricant and total bunker costs are estimated. The total bunker cost is estimated to be approximately 2.5 million USD.

ANNUAL BUNKER COSTS				
Consumer	Consumption [g/kWh]		Price [USD/ton]	
Fuel oil	155,14		155	
Lub Oil	1,5		2 800	
	In Port	Manoeuvring	Transit	On/Off loading
Power [kW]	1 527	10 839	17 755	2 611
Hours per year	256	1 280	3 840	3 072
Annual fuel oil consumption	61	2 152	10 578	1 244
Annual lub oil consumption	1	21	102	12
Annual fuel oil costs	91	333 632	1 639 517	192 877
Annual lub oil costs	1 642	58 272	286 358	33 688
Total Bunker costs				2 546 079

Table 76: Annual Bunker Costs

The estimated fuel price is set to 155 USD/ton based on information from Jon Båtbugt (74). The lubricant oil price is assumed to be approximately 2 800 USD per ton (16)

PORT FEES		
Calls pr year	Costs pr call [USD]	Costs pr year [USD]
128	52 000	6 656 000

Table 77: Port Fees

The estimated annual port fee is based on the fees at Mongstad harbour (81). The estimated annual port fee includes a pilot, tug boats, mooring boats, mooring/unmooring, harbour dues and agency fee.

20.4 Shore Side Expenses

It is assumed that one port captain is needed in each harbour since the on/offloading procedures are complicated (32). The coefficients are taken from "Rapid Response Vessel" (70) and compared to coefficients from *System Based Ship Design* (19)

SHORE SIDE EXPENSES				
Cost item	Personnel	Annual salary [USD]	Overhead	Annual cost [USD]
Wages	2	62 776	0 %	125 552
Social costs			30 %	37 665
Administration			100 %	125 552
Offices cost			50 %	62 776
Total Annual Shore side expenses				351 545

Table 78: Shore side Expenses

20.5 Required Freight Rate

This chapter is a summary of the cost calculations in the earlier chapters and the total annual operating costs. The building costs are represented in a table below. The present value (P) is calculated using the equation below.

$$P = Investment + AnnualCost \cdot \frac{(1 + p')^n - 1}{p'(1 + p')^n}$$

P= Present Value

p`= real interest rate

n= time in years

The real interest rate is estimated using the equation below.

$$p' = \frac{(1 + p_{Inf})}{(1 + p_{Int})} - 1$$

p_{Inf}=Inflation rate

P_{Int}=Interest rate

The necessary income per year (F) is estimated using the equation below.

$$F = \frac{LCC}{\frac{(1 + p')^n - 1}{p'(1 + p')^n}}$$

F = Necessary income per year

LCC= Life circle cost

TOTAL COSTS		
REQUIRED FREIGHT RATE		Unit
Building costs	88	[10 ⁶ USD]
Annual costs	13	[10 ⁶ USD]
Time	20	[years]
Inflation	3 %	
Interest	10 %	
Real interest rate	7 %	
Present value	228	[10 ⁶ USD]
LCC Life Cycle Costs	316	[10⁶ USD]
F Necessary income per year	29	[10 ⁶ USD]
Number of trips per year	65	[trips]
Annual transported CO ₂	3	[ton]
Annual transported LPG	1,4	[ton]
Total annual transported	4	[ton]
RFR Required Freight Rate CO ₂ og LPG	7	[USD/ton]
RFR Required Freight Rate CO ₂	10	[USD/ton]

Table 79: LCC and Required Freight Rate

The required freight rate is estimated to be 7 USD/ton if return cargo is taken into account in the estimations.

If the ships only carry CO₂ the required freight rate increases to 10 USD/ton. Therefore can it be concluded that the expected required freight rate will be somewhere in between 7 and 10 USD/ton.

20.6 CO₂ Prices

The Intergovernmental Panel on Climate Change (IPCC) have stated that the price of CO₂ must be at least 80 USD/ton for a sufficient carbon emission reduction.

The importance of high CO₂-prices is that the coal fuelled power plants will implement CCS instead of buying CO₂ quotas. This would be both economically and environmental friendly. The price of CO₂ should therefore be higher than that of CCS.

The CO₂ prices estimated by Deutsche Bank for CO₂ are estimated to be approximately 65 USD/ton in 2015 (Euro = 1.36 USD). In the figure below is the cost of CCS and CO₂ is shown.

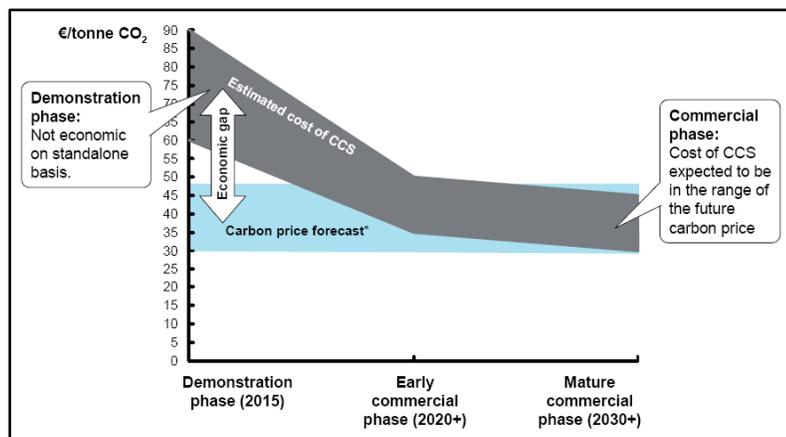


Figure 76: CCS profit intersection curve (80)

20.6.1 The Carbon Chain Cost

The total costs of CCS can be grouped into the following when considering shore to shore ship transport.

- Liquefaction
- Regasification
- Buffer storage
- Terminal
- Storage
- Transportation

Earlier studies done by Audun Aspelund and Dr. Gelein De Koeijer (7) showed that the ship transportation in the North Sea with more than 2 000 000 tonnes annually would cost 20 - 30 USD/ton.

The maximal price of ship transportation can be calculated by subtracting the costs related to liquefaction, regasification, buffer storage terminal and storage from the CO₂ price estimate from Deutsche Bank.

In-house studies done by Teekay, for a vessel in the North Sea transporting 4 000 000 tonnes annually, the total cost besides transportation is estimated to be 46 USD/ton (10). This leaves 19 dollar for ship transportation.

20.6.2 CCGC and the Carbon Chain Cost

From the analyses and estimations done in the earlier chapters it is shown that CCGC will, with its RFR of 7-10 USD/ton, fulfil the required ship transportation cost. The ship can transport CO₂ with a good margin to the maximal transportation cost, and its RFR is reduced compared with the expected RFR of 20-30 USD/ton.

With this RFR the CCGC have shown that low cost ship transportation of CO₂ is viable, environmental friendly and efficient.

21 Risk Assessment

The following risk assessment is performed to find the risks related to the choice of design based on the owner's requirements. The risk mitigation and evaluation will be included in the analysis.

The analysis is divided into three parts to sort out the risks of the project: Identification, analysis, and prioritization.

A review of the entire project plans and areas of uncertainty have been identified. The analysis defines how the identified areas of uncertainty could affect the performance and success of the project in terms of cost, quality, scope and duration. The risks have been divided into intolerable, tolerable, and negligible risks.

All these steps must be repeated and reviewed in an iterative process until all of the risks and possibilities are detected and recorded. Because of time restraint only the main risks and possibilities have been processed in this analysis.

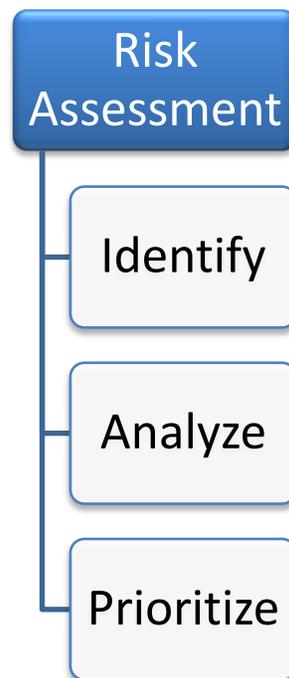


Figure 77: Risk assessment hierarchy

21.1 Risk Categorization

Risks have been differentiated into internal and external risks. Internal risks are the ones the project team can control or influence. External risks are those that lie outside the project team's control.

Risks are classified in nine different degrees, and they can be set into a two-dimensional table like the one below. The y-axis represents the project impact, and is ranged from low to high. The x-axis represents the likelihood of a risk to occur, and is also ranged from low to high.

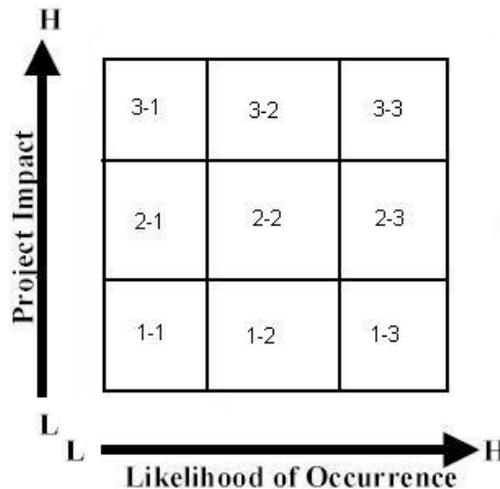


Figure 78: Risk classification (78)

From this classification risk categories 3-2, 2-3 and 3-3 are intolerable. Categories 3-1, 2-2 and 1-3 are tolerable, and 2-1, 1-2, and 1-1 are negligible risks.

The risk factor is given when the project impact number and likelihood of occurrence number is multiplied. Hence, the results is evaluated with six different risk factors; 1-2-3-4-6-9.

21.2 Risk Analysis

The risk analysis points at the risks that arise when the projected ship is compared to those demands set in the owner requirements. The following risk analysis points at such areas of risks. They are graded from a negative point of view - aspects which the project must avoid.

21.2.1 Liquid Condition of CO₂ is not Optimal

The CO₂ could be transported as solid, liquid or gas. Gas phase is excluded because of the low density. Solid phase CO₂ has a slightly higher density than the liquid condition, but requires lower temperatures. The LCO₂ has some properties similar to that of LPG. Since large-scale transport of LPG is established, it seems desirable to transport CO₂ in liquid phase. Solid CO₂ requires more energy in transition of gas to solid, so the entire chain cost will be higher. However, the ship might be able to transport more.

The choice of liquid condition for CO₂ has a low project impact, and a low likelihood of occurrence. The risk category is set to 1-1.

21.2.2 The Time Table is Too Tight

Harsh weather, winter storms and malfunction of ship's machinery or transfer auxiliary systems may delay the ship operation. From the time calculations it is said that the ship will travel 65 roundtrips per 357.5 days, which gives 7.5 days for maintenance annually. The time table also has a relatively good safety margin in the manoeuvring phase, so the ship can gain some time at this stage.

The time for purging the tanks is estimated. There is no real experience on this time consumption. Since the preparation time of the RPT at port is set to two hours, some time could be saved here.

The project impact for not being able to satisfy the time table is set to medium. The likelihood of occurrence is set low. 2-1.

21.2.3 Malfunction of Rapid Purge Technology

The Rapid Purge Technology is not a proven technology and therefore the project impact is set to high. The Rapid purge technology has the advantage that it reduces the on/offloading time by several hours. Without the technology the number of trips per year will decrease and make the ships operating costs higher.

The risk of a malfunction of this technology is that damage to the bellows during transfer may occur. This would have a low impact since it just makes it necessary to purge the tanks in standard ways. However, the likelihood of occurrence is set to high since it's a non-proven technology. The total project risk is set to 2-3.

21.2.4 Inappropriate On/Offloading Facilities

The two different harbours have restrictions on the vessel's length, breadth and draught. This has been taken into account, but it has also been made some assumptions. There are currently no facilities for on/offloading of CO₂, but it is assumed that these will be made. The basin of the LPG jetty at Mongstad is not deep enough, but it is assumed that a new jetty will be available.

At both harbours the need for intermediate storage is present. Storage compartments for LCO₂ are currently not present at the harbours. At the CO₂ capturing plant at Mongstad, Test Centre Mongstad (TCM), and full-scale TCM in 2014, there are no plans of establishing an intermediate CO₂ storage. This will only be done if transport by ship is present (1).

Both harbours must be able to handle 3600 m³/h of gas flow and have the necessary pipelines for further transport. Handling the cargo on offshore facilities is a possibility. Offshore structures can be loading buoys connected to subsea wells, like the Submerged Turret Loading buoy (76) or External Turret Production (77). As the ship is equipped with coolers and compressors besides own pumps, offshore on/offloading is possible.

The appropriate on/offloading facilities have a large project impact. The Owner's Requirements are not fulfilled if the cargo cannot be delivered. The likelihood of occurrence is set to low as transfer buoys can do the job if the jetties are too small, and the assumption that the harbours itself will satisfy the demand can be set. Total risk set to 3-1.

21.2.5 Risk Review

The risk areas above are summarized in the table below. The risk factor is calculated.

Risk areas	Risk type	Project Impact (PI)	Likelihood of Occurrence (LO)	Risk factor
Liquid condition of CO2 is not optimal	Internal	1	1	1
Malfunction of Rapid Purge Technology	Internal	2	3	6
Not appropriate loading/offloading facilities	External	3	1	3
The time table is too tight	External	2	1	2

Table 80: Risk overview

From the table it is clear that inappropriate on/offloading facilities along with malfunction of RPT are the biggest threats to the project with risk factors of 3 and 6 respectively. Risk mitigation is needed to reduce the risks before the ship design is moved into a second cycle.

21.3 Risk Mitigation

Risk mitigation is actions that are proactive to prevent a risk from occurring, and makes a severe impact on the project (78). It is important to identify the necessary actions that can be carried out in advance for the different risks, in order to reduce or eliminate the impact. The different risks can be mitigated by elimination and reducing the probability of occurrence or the impact on the project. Risk mitigation can be divided into two components: Reducing probability of occurrence, and hazard impact. Both would decrease the risk factor.

Risk mitigation is done for the risk areas that are a threat to projective phase. The external risk types are hard to deal with, and time and resources may therefore be difficult to canalize. The internal types are more controllable. The efforts to change the highest risk factor are presented below, since the others risk were of negligible form.

This study must be taken into account in further development of the ship. Improving the flexibility for cargo and ports is the main and easiest actions that can be done for reducing the project risk.

Risk mitigation	Not appropriate on/offloading facilities	Rapid Purge Technology
Type	External	Internal
Project Impact (PI)	3	2
Likelihood of Occurrence (LO)	1	3
Suggested mitigations	Reduce PI	Reduce LO/PI
Reducing actions	Use a smaller ship	
	Build a more flexible ship	Do more tests on the RPT
	Change the harbours	

Table 81: Risk mitigation

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