Monohull vs. Semi-submersible

For offshore heavy lift crane operations

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for offshore heavy lift crane operations

By

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Preface

I started as a student at the Aerospace Engineering faculty at the TU Delft. This was an interesting and challenging bachelor, which ended with an intense group project where we designed an electric helicopter. During my bachelor I also did the minor ”Sailing Yachts” at the Maritime Engineering faculty.

During this minor I began to feel my passion more lies in designing vessels, rather than air- and spacecraft. This is mostly due to the smaller scale projects, which felt like there is more freedom for different designs.

Thus, I decided to switch over the maritime world and do my masters in ship design. From a selection of companies and research possibilities the differences between the Monohull and Semi-submersible felt like a fun and challenging assignment.

I would like to thank Vuyk Engineering Rotterdam and all its employees which made it possible to work on this assignment. In particular my daily supervisor, Sander Bot, who helped me a tremendous amount. Without his help and insight this assignment would not be possible. In addition, I would like to thank Johan van den Berg, Gijs van de Wiel, Alex van der Zee and Ben van der Kleij for their help on specific research parts that required expertise. Martijn Scheltes, who was also a graduate student, also helped me in the early stages of my assignment and we spend quite some afternoons brainstorming about our assignments.

Finally I’d like to thank my girlfriend, Susanne Gillig, for her help on proofreading select chapters and correcting my grammar.

E.A. Schouten
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Summary

Heavy lift crane vessels are used to install the heaviest and largest projects offshore. Interestingly, there are two significantly different vessel types that are often used for similar projects. The monohull, popular due to its low capital and operational expenditures, is often used for smaller projects. The semi-submersible is used for larger projects, and although it has higher capital and operational costs, it has a much higher workability, enabling the vessel to operate in rougher sea states. However, there is currently no clear distinction made between these two vessels in terms of total operational performance and cost.

There appears to be a turning point between these two vessel types, after which the semi-submersible is the more favourable vessel type to carry out a lifting operation. This is investigated for several performance aspects and a total cost is calculated. These performance aspects are; Resistance & propulsion, stability, ship motions and dynamic positioning.

In order to calculate these differences, three concept vessels are designed for each vessel type. Based on these designs, each performance aspect is calculated and a total cost is derived for every vessel. A relation is then made to estimate the turning point in two different sea states; the North Sea and the coast of West Africa. These sea states can be considered the extremes of the world; The North Sea has high and short period waves, while the sea at the coast of West Africa is characterized by swell waves.

It was found that the resistance of the semi-submersible is almost double compared to the monohull, due to the high wetted area and the vessel's bulky shape. With respect to the stability the semi-submersible performs well and has a high initial stability. In addition, it has a lot of room for ballast and the least ballast is required to level the vessel. Because the semi-submersible has relatively high natural periods for their motions, it performs especially well in the North Sea environment. The workability is almost double compared to a monohull. Interestingly, due to the swell waves near West Africa, the workability for the semi-submersible drops in this environment. The workability of the monohull increases, but is still lower than the semi-sub. In general, the power required to keep position during dynamic positioning is lower for the monohull. This is mostly due to the well shaped hull and low exposed area to wind and current.

The total cost per vessel is calculated by combining the results of each aspect and their effect on the capital and operational cost. The workability is taken into account by estimating the income and calculating the effect of operational delay. By using a calculation method that also takes interest and inflation into account, the turning

points for both environments are calculated. For the North Sea the turning point is around 6100 mt, while for the coast of West Africa it is reached at 8500 mt.

Contents

1

Introduction

In the year 2000 the heavy lift vessel Thialf lifted a 11,600 metric tonnes topside structure with its two cranes and installed it offshore [29]. With its sophisticated dynamic positioning system the ship was capable of holding its position very precisely, while lifting and installing this structure a whole. This means that the weight of 1000 school busses was lifted and positioned with a precision in the millimetre range which is already a challenging operation when in fact on shore. Yet this remarkable operation is just one of many examples of the massive offshore projects that have become normal within the last decades. Not only is there an increasing need to install new offshore structures, also the decommissioning of old and inactive structures has become a lucrative field.

There are many different vessels that can be used to lift and transport heavy cargo. An elaborate explanation of all relevant vessels for the offshore lifting market can be found in appendix A. Each of these vessels has its own unique advantages and disadvantages and can be used for a range of heavy lifting projects in the offshore market. Interestingly, two fundamentally different vessel types are in use, although there is a large overlap in their fields of application.

One might thus wonder: **"Which vessel type should be chosen for a certain operation?"** Before trying to answer this question, a general description about heavy lift crane vessels is given to better understand the vessels that will be compared.

The heavy lift vessel

A heavy lift crane vessel consists of a floating hull (or a set of hulls) that is fitted with a large crane that is capable of lifting objects that cannot be lifted by regular ships due to their extreme weight. A ship falls into the category of "heavy lift vessels" if its crane has a capacity of at least 200 metric tonnes (mt).

Monohull crane vessel

The most common crane vessel is the monohull. Due to its simplicity, the monohull is most commonly used as a starting point for the design of a suitable vessel for a new application.

As early as in the 14th century the first floating crane was constructed that was able to lift objects in port waters [21]. Nowadays there are all sorts of crane vessels, such as the derrick barge, jack-up crane vessel, heavy cargo crane vessels to name a few [11]. The focus of this thesis will be on the heavy lift crane vessels that use extra ballast for more stability and large revolving cranes to lift objects.

This vessel type is capable of performing a wide range of operations in the current market, such as the installation of monopiles, wind turbines and laying pipes. The offshore market and a selection of typical operations are explained in more detail in appendix B. An example of a heavy lift crane vessel can be seen in figure 1.1.

Figure 1.1: The Oleg Strashnov vessel, fitted with a single 5000 mton capacity crane.

The monohull crane vessel is chosen for most general applications that do not require an exceptional lifting capacity or a high workability in rough sea states. It is known for its good transit performance, especially due to its relatively low fuel consumption. It is exceptionally popular for the installation of offshore wind turbines which is a growing market.

Semi-submersible crane vessel

At some point there was a need for even higher crane capacities and the semi-submersible crane vessel was developed as an innovative solution that also improved the workability in rough sea states. In transit mode the semi-submersible uses floaters, but in lifting mode the vessel takes in a lot of ballast water to significantly increase its draft. The vessel is therefore more stable and the workability is improved significantly as only the columns break through the surface of the water. Although the semi-submersible vessel is more costly to build and in many ways also more expensive to operate, the design proves to be invaluable for operations under more difficult conditions, e.g. in the North Sea. Two examples of semi-submersible vessels can be seen in figure 1.2.

Figure 1.2: Two multihull semi-submersible ships carrying out an operation. On the left the Thialf with a combined crane capacity of 14200 tonnes, and on the right the Sleipnir, which is still under construction, with a combined crane capacity of 20000 tonnes

The semi-submersible crane vessel is often used for the lifting of especially heavy and bulky objects, such as topsides like e.g of oil platforms. Due to its increased width relative to the monohull, it is often equipped with two cranes that can perform tandem lifts. This is exceptionally useful for lifting heavy and large objects, as the attached load is more stable and taller objects can be lifted due to its multiple attachment points. However, it has a relatively high fuel consumption due to its increased resistance.

The problem

Today there is still a need for a wide range of crane vessels. As the monohull vessels are mostly used for lower crane capacity operations, while the semi-submersibles are used for higher crane capacity and better workability, there appears to be a turning point between these two vessel types. It is thus a logical step to compare these vessel types with each other and further specify this turning point. There are several approaches to comparing the two different designs. It is essential, however, to choose an approach that allows the selection of the proper vessel type in an early phase of the design process. To provide the data needed for such a preliminary evaluation is the aim of this Master thesis.

This will give the company a unique advantage as there is currently no clear difference made between the operational characteristics of monohull and semi-submersible. There are comparisons for, e.g. the workability for a set of vessels, but no clear distinction in total operational performance or cost. The objective of this thesis is thus to get a clear overview of the performance differences and the total cost that a vessel has during its lifetime. In chapter 2 this problem is further analysed and a research plan is developed.

2

Research plan

In order to answer the question stated in the introduction a research plan should be made. The problem should be further analysed and enclosed. This is an important step in order to formulate clear and proper conclusions.

First the problem is further analysed in section 2.1 to determine what drives the problem. Then the general approach is explained to find a solution for the problem in section 2.2, followed by a look at the operational profile that further defines the scope of the thesis in 2.3. Section 2.4 continues the approach and looks at important aspects that determine the performance and cost. Then the research questions are determined in section 2.5. Finally, in 2.6 the general structure of the work is explained.

2.1. Problem analysis

There are many factors that influence the decision of the ship hull. It is an option to compare the vessels on the resistance, the ship motions in a given environment, workability and many more. In the end, the industry is interested in the most important parameter of all; Cost, the total amount of money that is required to build and operate the vessel in its lifetime. Two different types of costs can be defined:

- Capital Expenditures (CAPEX): This includes the cost of the vessel with all fixed items on board, which consists of the hull construction cost, but also all lightship content such as the engines, propulsion system, crane(s), furbishing etc. As the vessel still has a certain value at the end of its lifetime, the CAPEX can be defined as the value difference between the vessel when purchased and at the end of its lifetime (also known as the depreciation), which also includes investment costs and interest.

- Operating expenses (OPEX): The costs to operate the vessel is included in the OPEX. This includes for example the crew wages, fuel consumption and maintenance and repair.

These costs are influenced by a few key parameters. The CAPEX can be summarized as:

- Hull & superstructure: A large component of the capital expenditures can be related to the structural weight of the vessel. Besides the (raw) materials used for the vessel, the construction cost is also significant for ship construction. This is thus an important parameter and should be determined with caution.
- Other lightship components: There are many other parts on a ship that have a fixed cost. The crane, machinery, outfitting and interior fitting are included in the building cost of the vessel and should be taken into account.

The OPEX can be divided into the next components:

- Fuel consumption: During transit and dynamic positioning the vessel consumes fuel to overcome its resistance. The main influence is due to the hull shape, propulsion system, vessel speed, travel duration and environmental influences.
- Crew: To operate the vessel, crew is required. The crew wages are the most important aspect, but the supplies play a role as well.
- Workability: Although it is not part of the OPEX, it does indirectly cause additional costs during the vessel operational life. The workability is the percentage amount of time a vessel can continue its operation. A lower workability means the vessel has to be shut down earlier and this results in a higher duration of the operation. This can be seen as a slower profit, or a loss due to penalties in the contract. When the vessel cannot continue operations, the crew still has to be paid and less contracts can be completed during the vessel's lifetime.
- Maintenance and repair: During the vessel lifetime maintenance and repair is necessary. Some systems on board could get damaged which have to be repaired or replaced, which costs money. Scheduled maintenance inspections also cost time and money.
- Remaining items: There are many other components which have a smaller influence, especially when the difference between vessel types is considered. This includes the port and canal fees, administration cost, tug assistance and many more.

Operational profile

These items listed above, especially those of the OPEX, depend on the operational profile of the vessel. A detailed operational profile is obtained via Vuyk, that describes several different operations performed by a number of heavy lift vessels in a year. Since this is classified information, no detailed information can be given. A general division of phases that are relevant for this thesis can be seen in figure 2.1.

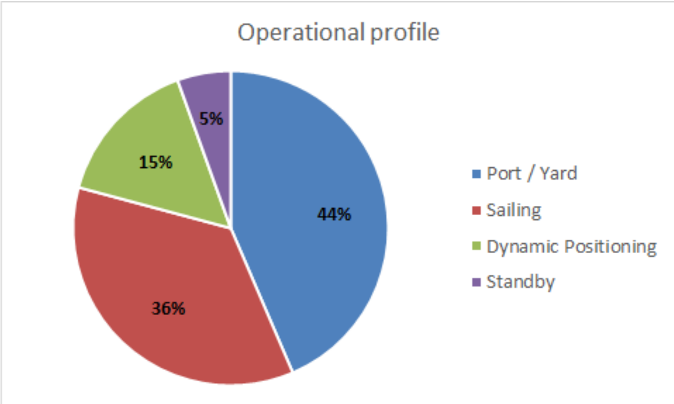

Figure 2.1: General heavy lift vessel operational profile.

- Port / Yard: The largest part of the time the vessel is in port or yard. This is mostly due to (de)mobilization, but also consists of maintenance & repair and a weather delay.
- Sailing: Another large part consists of sailing between ports and offshore locations.
- Dynamic positioning: This phase consists of the installation time where dynamic positioning is required and the crane is used to lift objects.
- Standby: During standby the vessel is waiting offshore before it can begin its operation. This could be because the vessel has to wait for another vessel to complete its operation, the weather is too severe to start operating, or some operations on deck have to be completed first. There are many more examples that can be thought of. In this phase the vessel is mostly weather vaning, in order to reduce the power consumption.

Using the above operational modes, various literature research [17][33], company experience and general ship knowledge, a general cost division can be made. Figure 2.2 can be used to get a general cost overview. Note that this division is only to give a first general impression of the involved costs. It can be seen that the largest part of the CAPEX is the construction of the hull and superstructure. The machinery also has a relatively high cost due to a high power demand due to the many thrusters and its large crane and crew size. In addition, since there is a large crane on deck, this has a high cost as well. The largest OPEX part is the crew and supplies due to the large crew that is required to operate the crane vessel. Since fuel is required for almost all operational modes (figure 2.1), the fuel cost is high. As with many ships, the maintenance & repair cost is substantial. The workability is left out from this chart as it is not a direct operational cost. It is also not yet clear what kind of impact the workability has for the two vessel types.

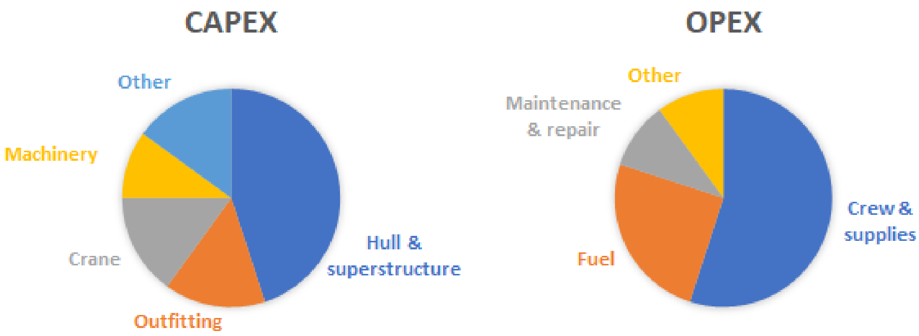

Figure 2.2: Estimated capital (left chart) and operational expenditures (right chart) for a heavy lift crane vessel. (citations)

Before the above mentioned costs can be determined, a selection of vessels has to be made which can be compared with each other. The next section will show a general approach that will be used as a starting point.

2.2. General approach

The objective is to get a clear overview of the performance and total cost of the vessel types during their lifetime so that the vessel types can be compared to each other. In order to determine this, several vessels will be designed and their performance and costs analysed. The starting point and strategy is explained below.

It can be said that the most important requirement of heavy lift crane vessels is the crane capacity. The purpose of a heavy lift vessel is to lift an object and a higher capacity enables the vessel to accept a wider range of contracts. It also has a large impact on the main geometry of the vessel as sufficient stability is required.

As the monohull often has a lower crane capacity compared to the semi-submersible which is explained in the introduction, it is interesting to determine where the turning point lies, the point where a semi-submersible is better than a monohull in overall performance in terms of CAPEX and OPEX. This is roughly displayed in figure 2.3.

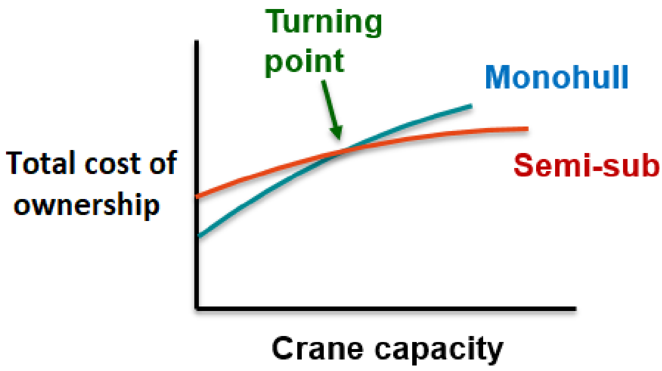

Figure 2.3: Schematic impression of the cost (Operational and capital) that depends on the crane capacity of the vessel.

For this approach multiple designs have to be made so that the cost for each vessel type can be determined at various crane capacities. It is therefore chosen to make 3 designs per hull type at 3 crane capacities. By combining the results for each crane capacity, a relation can be found between the total cost and crane capacity of the vessel. The two relations, one for each vessel type, can then be used to find the turning point.

By looking at reference vessels, it is expected that the turning point could be around 6000mt. From this point, it is expected that the monohull will become too large to remain the cheaper alternative to semi-submersible vessels. In order to get a good relation around this point, crane capacities of 3000 and 9000mt are also chosen. There will thus be 3 designs for each ship type at 3000, 6000 and 9000mt crane capacity.

Before more decisions can be taken, a closer look at the operational profile and scope of the thesis have to be taken which define boundaries and other requirements for the vessels.

2.3. Scope

In order to get a total cost estimate, the operational profile of the vessel should be known. As an example, a vessel that spends a different amount of time in transit mode will have a different OPEX (e.g, due to fuel consumption, fresh water supply and perhaps even a different propulsion system) and thus has a different total cost. Also, if a vessel is designed to carry a smaller load, the CAPEX will be lower. In order to get a clear difference between the two vessel types, several choices have to be made. Some aspects are fixed or constant as their influence on the results is limited, not relevant or out of scope. These are explained in subsection 2.3.1. In order to determine the overall investment and operational performance, some aspects are considered in depth in this research and are explained in subsection 2.3.3.

2.3.1. Constants

The following operational aspects are taken as constant. A more detailed list of the operational profile decisions can be found in appendix C.

- Function: In order to compare the vessels effectively the function should be constant. Multiple functions could cause the vessel to have a different operational profile and have a lot of minor details that take a lot of time. Also, it is expected that a vessel with a different function, but with the same capacities, does not have a significant difference in capital and operational cost. There are many different structures and objects that have to be lifted and installed offshore. These all have a different weight and size, requiring the vessels to be versatile. The function of the vessels will be the lifting of common offshore structures with a weight equal to the design crane capacity, such as monopiles and wind turbines for the lower crane capacities and jacket and topside structures for higher crane capacities.
- Environment: The environment is an important aspect as it significantly influences the workability of the vessels. This could cause a large difference in operational costs, so ideally multiple environmental conditions should be looked at. First the North-sea environment will be investigated, after which the coast of west Africa is also looked at. These two spectra have a large difference in wave system, as the North Sea mainly consists of short high waves, while West Africa has a lot of surface gravity waves.
- Payload: For the comparison the payload, thus the cargo that can be carried on deck, will be constant for the design points. By keeping it constant, the difference in required ballast and stability will become clear.
- Maintenance & repair: The vessel has to undergo scheduled maintenance and sometimes part of the vessel has to be repaired, which results in a cost. There are methods to estimate these costs, e.g. a relation with the construction cost and installed power. Because this aspect is very hard to determine, especially in a concept stage, it is assumed as a percentage of the construction cost [1]. Detailed aspects of maintenance & repair are thus out of scope of this thesis.
- Vessel speed: The vessel speed is an important parameter as it influences the required power, fuel consumption and thus OPEX cost. However, to get a clear difference between the two ship types, it should be taken constant so that the power and fuel consumption differences become clear. In addition, different vessel velocities could cause contract (dis)advantages. A higher vessel speed would increase the fuel consumption, but the client could be willing to pay more as the operation is completed faster. More contracts can then also be completed within the vessel lifetime. Looking at reference vessels it can also be concluded that both vessel types could have the same vessel

speed, although the average monohull does sail a bit faster. This can be seen in the reference vessel list in appendix D.

- Transit distance: As the function is taken as constant, it makes sense to also select the same distance for the transit phases for the two vessel types. This is merely expressed as an amount of days per year that the vessel is in transit mode, as determined earlier in the operational profile.
- Operational duration: It is expected that for the two vessel types with the same function and crane capacity, the operational duration is almost equal in ideal circumstances. There are of course small differences, but these are too specific to take into account and will have a small influence on the total performance and cost. The duration does differ as both vessel types will have a different workability due to the influence of the environment. It is important to realise that the operational duration is only equal for both vessels when the workability is equal. It is thus assumed that the vessel types can operate equally efficient in a certain time window where the operation can be continued.
- Crew: The offshore heavy lift operations are complex, so a large crew is required. The crew wages and their supply consumption is an important expenditure. However, it is expected that the ship type does not have a large influence on the crew size, as the function of the vessel is the same. This is because crew is mostly required to operate the vessel equipment. As the vessels will have the same function and operational profile, the required crew will likely be (almost) equal. By using reference vessels it can also be seen that the accommodation capacity can be related to the revolving crane capacity and no large deviations between the vessel types are found.
- Vessel lifetime: It is assumed that the lifetime of both vessels is equal.

2.3.2. Crane

The crane is of course one of the most important aspects of a crane vessel. For a clear comparison it is thus important to make a valid choice for the crane(s). In light of this aspect one might think of the number of cranes, the position (where the crane is located on deck) and the crane type (mast or tub-cranes for example).

The above mentioned aspects are described shortly below. For an extensive analysis and reasoning, please refer to appendix F.

Crane type

There are many different crane types which are used in the offshore industry. The two types that are viable for the heavy lifting industry are the mast and tub-crane. Each of these cranes have advantages which make both cranes a viable option for the designs, which becomes clear in the study of Kamp [17]. In this study a trade-off is done between the crane types and these cranes result with similar scores.

However, since the tub-crane is already used for a wide range of crane capacities (whereas the mast-crane is not yet used for crane capacities above 5000mt), this crane type is chosen for all designs. It is expected that the calculations are more accurate and more data is available for this crane type.

Crane number

The two options are a single or dual crane configuration. More than two cranes are only used for large cargo vessels which focus on transporting as much as possible rather than performing offshore installation operations. A single crane is usually used for a monohull, whereas the semi-sub most often has two. This is mainly due to the hull shape (width difference) and a tandem crane configuration has better lifting stability in various environmental conditions which better fits with the semi-submersible workability.

For a good vessel type comparison it is important to choose one configuration for both vessels, so that a lot of differences that are not vessel-type related, are eliminated. By looking at several possible configurations for both vessel types and discussion at Vuyk it is chosen to use a single crane per vessel. This is mainly decided after weighing the applicability of each configuration for practical use, their integration complexity and importance for the comparison.

One might wonder if a single 9,000 mt crane design is a feasible concept. The Sleipnir is currently being build which has two 10,000 mt cranes on its deck, which are tub-cranes. It can thus be concluded that such a large crane is feasible, but not commonly applied. The practical application of a single 9,000 mt crane for a heavy lift operation is therefore unknown. The other option would be to fit two 4,500 mt cranes on the vessel. However, this would cause a lot of implications, e.g. the operational profile will change since a tandem lift is now possible, but a lift where a revolving crane is required is restricted to 4,500 mt. In addition, it will become clear in chapter 10 that the difference in cost between a single 9,000 mt crane and two 4,500 mt cranes is small. In addition, both crane number options have a very limited effect on the other calculations, such as the workability and dynamic positioning. It can therefore be concluded that the amount of cranes does not have a significant impact and should be kept the same for all designs so that the vessels are compared for one clear operational profile.

Crane position

The most difficult decision is the positioning of the crane. The reader is directed to appendix F for a detailed discussion. A short summary is written below.

For both the monohull and semi-sub there is looked at the following aspects that largely influence the position of the crane:

- Lifting operation: This consists of two items, the outboard and inboard reach. This is thus largely related to the capability of reaching and installing the cargo.

- Ballasting: As the name implies, for this aspect there is looked at the required ballast to level the vessel.
- Structural design: For this aspect there is looked at the structural design in the vessel to carry the crane and its load. For instance, positioning the crane in a corner will cause extra stresses that have to be compensated by additional structure.
- Deck arrangement: The crane takes a lot of space and each position will have an affect on the total unobstructed deck area and the possibility for additional equipment to be on board.

By looking at these aspects for each vessel type and performing a trade-off, it becomes clear which position should be chosen and what effect it has on the operational profile. For the monohull a crane which is centered at the stern is the most ideal location as it scores well in each aspect. The semi-submersible is less specific and the crane can be positioned either on the side or centre at the stern. A centered crane requires less ballasting and has a good reach on deck. If the crane is at the side, less adjustments to the structural design are required as the crane is positioned on top of a column and the unobstructed deck area is the largest.

For a clear comparison it is chosen to position the crane for both vessel types at the same location. The cranes are positioned at the centreline at the stern, to perform lifts over the stern.

2.3.3. Performance aspects

The following aspects will be looked at in detail and are summarized below for a clear overview. A more detailed explanation can be found in section 2.4

- Resistance & propulsion: The resistance and required power will be calculated which has a large influence on the OPEX (fuel consumption) and CAPEX (machinery investment).
- Stability: The stability of the vessel is determined so that the design is checked for required criteria. It is of importance as the design has to be feasible and if the stability criteria are not met, the design has to be adjusted.
- Hydrodynamics: In order to determine the workability, which has a large influence on the cost, a motion analysis will be performed.
- Dynamic positioning: During the lifting operation, the vessel often uses the DP system to stay in position. This phase consumes fuel, which is an important part of the OPEX. With a high power demand, the CAPEX will increase as well.

2.3.4. Vessel characteristics

The vessel characteristics are defined the following:

- Function: The function of the vessels will be the same; the lifting of common heavy lift objects equal to the crane capacity. For the smaller crane capacities this could be the installation of monopiles, which are especially interesting due to its relatively large vertical size. For the mid and high crane capacities a wide range of operations is available, such as jackets and topsides. The main importance is that each vessel has a sufficiently large crane so that it can perform a wide range of operations for that crane capacity. Use is made of reference vessels and its crane(s) to choose crane characteristics that meet this demand.
- Environment: The vessels will perform their operation initially in the North-sea. The second sea state could be the west coast of Africa, which has significantly different characteristics.
- Speed: The constant vessel speed for all designs is chosen to be 10 knots, by looking at reference vessels (appendix D). A vessel speed is chosen that is the most reasonable for both vessel types.
- Crane: All vessels will operate with a single revolving tub-crane which is centered at the stern for lifting operations over the stern.
- Crew: By looking at reference vessels and using company experience it is chosen to use a crew of 300 for all vessels, which is found to be sufficient to perform the offshore operations.
- Lifetime: A common vessel lifetime is 25 years. This is chosen the same for all designs.
- Payload: The payload carrying capacity of the vessels is chosen to be 1.5 times the crane capacity for the 3 designs. This means the vessels can carry a deck load of 4500, 9000 and 13500 mt respectively. This is determined by looking at reference vessels.
- Endurance: The endurance has to be estimated so that the fuel weight can be determined. This is thus not related to the operational phase durations, but merely to estimate the fuel weight and tank size in the concept designs. An endurance is estimated by looking at reference vessels, which is often around 40 days in transit mode. As an indication, this means the vessel can sail back and forth between the Netherlands and Angola without refuelling (in perfect conditions and no lifting phase). Using the same data, it is found that the fuel consumption in DP-mode is often a bit less than in transit mode and could be estimated at 50-60 days endurance. However, it could be that there is a difference in ratio between the transit and lifting phase for different vessel crane capacities (thus, vessel size). A more detailed analysis with varying ratios would be too complex and it is decided this is out of scope of this research. However, there will be a sensitivity analysis (chapter 11) so that the impact of variations in these phases will become clear on the OPEX and CAPEX.

2.4. Performance & cost

Now that the scope and operational profile of the vessels are determined, a closer look can be taken at the determination of the performance and cost. Starting from the crane capacity and payload requirement, the concept designs will be made. Use is made of reference vessels and parameter relations to get to a first estimate of the geometry definition. Followed by a weight-based design and establishing the hull shapes, the concepts are defined.

These 6 concept designs have to be compared with each other. In particular, the total cost over its lifetime will be estimated for each design. This is done by looking at the most important performance aspects, which will be explained in more detail below. Each of the below mentioned aspect has a large influence on the OPEX and/or CAPEX. By combining the results of each aspect, the total CAPEX and OPEX can be estimated so that the total lifetime cost can be determined.

2.4.1. Resistance & propulsion

In this phase the required power will be calculated that a ship needs to overcome its resistance. This is required to calculate part of the OPEX, as the power required results in a fuel consumption. A higher required power will also influence the CAPEX, as more expensive thrusters will have to be installed. For the monohull the well known Holtrop & Mennen method is used [13, 12]. The semi-submersible has two floaters for which this method cannot be used, as these vessels often use barge shaped hulls. Instead, the method from Holtrop, Mennen & van Terwisga [16] is used. As the multihull consists of two hulls next to each other, a factor is used to take the hull-hull interaction into account. The calculated resistances have to be validated by applying these methods to reference vessels and using a correction factor. This will be done for calm water conditions to get a relatively quick impression. A detailed explanation and intermediate results can be found in chapter 5.

2.4.2. Stability

In order to make sure that each vessel can perform its function and meets its criteria to stay afloat in numerous conditions, there is looked at the stability of the vessel. The purpose is mainly checking that the concept design fulfils the stability criteria and that the concept is not infeasible. No results of this section will influence the costs, unless it is found that the vessel design is not stable enough and the geometry has to be changed. In this phase some general stability parameters are calculated such as GM values and heel angles. There will also be looked at important special circumstances such as strong (storm) winds and when the vessel unexpectedly loses its load during a heavy lift. In addition, the differences in required ballast for heavy lifting will become clear for the two vessel types. Chapter 6 describes all criteria, loading conditions and the results.

2.4.3. Hydrodynamics

The workability of the vessel is an important aspect as it determines a large part of the OPEX. With a lower workability the lifting phase would take longer. In case of an estimated profit for a certain job, the lower workability could be translated into a loss as less profit can be made in its lifetime. This could be included into the OPEX so that it is taken into account. Also, the crew on-board still has to be paid, more fuel is consumed during DP-mode and less contracts can be completed during its lifetime.

To obtain the workability a motion analysis is done. There will be looked at sea spectra and how these conditions affect the performance of the vessel. This analysis will initially be done for one sea spectra. If significantly differences in OPEX are found, a second sea spectra with different characteristics can be used. It could be that these two different environmental conditions show clear differences between the two vessel types. This analysis and all its aspects are further explained in chapter 7.

2.4.4. Dynamic positioning

During an offshore operation it is often necessary to stay in position. Mooring systems can be used, but in many cases use is made of the Dynamic positioning system. A configuration of thrusters is used to automatically keep the vessel stationary which is constantly affected by environmental forces. In DP mode the vessel also consumes fuel that is important for the OPEX. Similar to the resistance and propulsion, it also has an influence on the CAPEX due to the required power. There will be looked at a general thruster configuration that is often applied to these types of vessel. This is not the same for each ship type, and the resistance will be different as well, resulting in a clear difference in OPEX for each ship type. More detailed information and results can be found in chapter 8.

2.5. Research questions

The following main research question can be defined:

When is the semi-submersible the preferred vessel type for heavy lift crane operations compared to the monohull, in terms of performance and total cost?

The following set of research questions relate to the exploration of solutions and in which manner the problem should be solved. Some have already been answered in previous sections, but are listed for the sake of completeness.

- What are the characteristics of heavy lift vessels that are currently used?
- For which operational profile should the ship types be compared and what are the requirements?

- Which aspects should be further investigated to obtain clear differences between the vessel types and to calculate the total cost?

The next set of research questions is aimed at the concept design of the vessels:

- What are the geometry and capacities of the vessels?
- What does the general arrangement look like for the designs?

The following questions should be answered during the performance comparison of the concept designs:

- What is the (difference in) resistance and propulsion power for the concept designs during transit?
- Which loading conditions should be investigated and what are the stability differences between the selected hull shapes?
- What effect does the environment have on the vessel workability?
- How much is the power consumption during DP operation and how much power has to be installed?

Finally, some questions concerning the analysis and recommendations that come from the comparisons:

- How do the determined performance differences relate into the costs?
- What can be said about the CAPEX and OPEX costs of the vessel type?
- What effect does the workability have on the total cost?
- How sensitive are the results to changes in the assumptions made?

2.6. Research structure

To get a clear overview of the thesis, in figure 2.4 the general structure of the thesis can be seen. The figure can be read from left to right, which are packages that have to be completed.

The work can thus be divided into the following stages:

- Reference ships: In this stage a list of reference vessels is made. For each vessel there is looked at their geometry and characteristics. Using these parameters, some relations are made so that it can be used in the concept design phase. The reference vessels and a concise selection of parameters can be found in appendix D.

- Concept design: In order to compare the ship types, several concept designs are made. First the requirements have to be defined, after which several design choices are made. The result of this phase is the vessel geometry and their characteristics. This can be found in chapter 3 for the monohull and chapter 4 for the semi-sub.
- Design performance: The concept designs are evaluated in this phase. The resistance and propulsion power can be determined at the given vessel velocities. This results in an estimate for the fuel consumption. The stability is also determined and a look will also be taken at the dynamic positioning. The workability of the vessel designs are determined by looking at the vessel motions and DP capability. Finally, the installed power and power consumption per operational phase are calculated. These steps can be found in chapters 5 through 9.
- Analysis: The final package is the analysis, in which the results of all packages are combined to make an estimation of the total cost. A sensitivity analysis is also performed. Finally, the conclusions and recommendations are given. This can be found in chapters 10 through 12.

Figure 2.4: General structure of the thesis work.

3

Monohull concept design

In this chapter the concept design process for the monohull is described. The main goal is to design three monohull concepts that are able to operate as crane vessel. The geometry, weight, hull shape and general arrangement are determined which are important as starting point for the following calculations in the next chapters.

First a list of reference vessels is made in section 3.1, which is used to determine the main geometry of the designs in section 3.2. An estimate for several other parameters such as the deadweight and displacement is also given. The hull shape design is explained in section 3.3. Then in section 3.4 the main weight groups are calculated. Section 3.5 describes the strategy for the general arrangement and the placement of the weight items. The sketches are then shown in section 3.6.

3.1. Reference vessels

In order to get a good impression of monohull vessels that are currently used, an extensive reference matrix is made. In this matrix, monohull heavy lift vessels are listed with important characteristics such as their geometry, displacement, deadweight, transit speed, installed power and many more. In addition, these reference vessels are used for nearly all other chapters in this report to get a general idea of common characteristics.

The reference vessels are shown in appendix D. There is tried to gather as much reference vessels as possible for which enough information is available. In order to get a relevant matrix the vessels have to comply to the following requirements:

- Monohull heavy lift vessel: Of course the vessel has to be a monohull and be capable of lifting objects with a crane.
- Self propelled only: There will only be looked at self-propelled vessels.

3

- Minimum crane capacity: In order to have a concise overview of heavy lift vessels, the vessel's crane (combination) needs to be able to lift a minimum of 1000 tonnes.
- Revolving crane: In order to carry out a wide range of offshore operations, the crane has to be of the revolving type. This means the crane can turn relative to the vessel, so that cargo can be lifted from several different angles and can be carried onto the deck.

3.2. Main parameters

The selection of the main parameters is done by using the reference vessels as described above. These parameters include e.g. the length, width, draft, depth, displacement and geometry ratios such as L/B. These are determined as a starting point for the concept designs. A detailed analysis of the parameter relations and decisions can be found in appendix E. A summary of the parameter selection is described below. Figure 3.1 can be used as a general guideline.

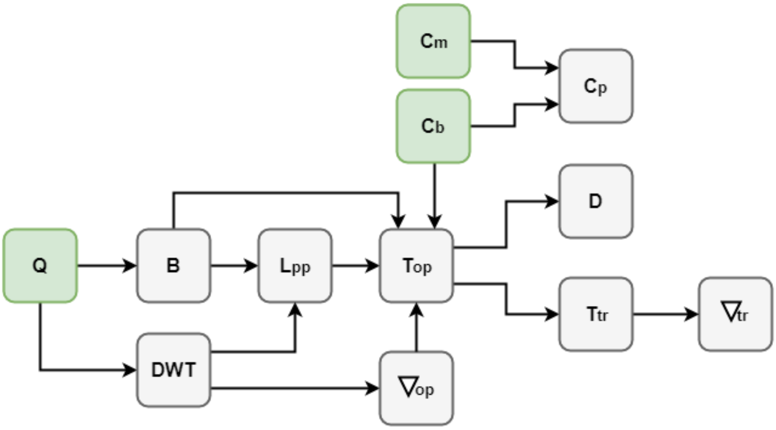

Figure 3.1: Main parameters selection flow for the monohull designs. Green boxes mean it is either an input or chosen parameter.

Starting from the crane capacity, several clear relations are found that determine a few parameters such as the deadweight and operational displacement. It makes sense that these relations exist, as a vessel that can lift more is likely to be able to carry more and have a higher displacement. The width is determined by taking the relation between the revolving crane capacity and the width to the third power (B^3), as this is the main influence on the vessel stability. For some parameters not all reference vessels could be used, as the differences are too large. E.g. the block coefficient is determined by looking at vessels that have a ship-shaped hull (thus no barge types). For the remaining parameters high correlation relations are used, by also keeping an eye on parameter ratios such as L/B, B/T, T/D etc. In figure 3.2 the results are shown.

	#1	#2	#3	
Q	3000	6000	9000	
LPP	159	185	211	[m]
B	41.8	47.7	53.6	[m]
D	12.8	14.8	15.9	[m]
T_tr	5.9	7.4	8.2	[m]
T_op	8.3	10.3	11.5	[m]
C_b	0.81	0.81	0.81	[-]
C_m	0.98	0.98	0.98	[-]
C_p	0.83	0.83	0.83	[-]
DWT	25312	46179	68148	[mt]
∇_tr	30899	53515	76131	[mt]
∇_op	46992	81386	115781	[mt]

Figure 3.2: Main parameters that are selected for the monohull designs.

3.3. Hull selection

The hull shape is taken from DELFTship [5], which has a standard form for a heavy lift crane vessel. The shape is scaled and manually edited to obtain the calculated main dimensions and block coefficient. An important observation is that this vessel has no bulb, while various monohull designs do have one to reduce their resistance. However, at the Froude number and block coefficient of the designs, the bulbous bow is expected to have limited effect for the design block coefficient and transit speed[34]. In the sensitivity analysis a decrease in resistance is investigated to see if the bulb can have a significant effect.

3.4. Weight & placement

In this section the main weight groups are determined and there is discussed where these weight items should be placed on the vessel.

3.4.1. Lightship weight components

There are several components that are included in the lightship weight. The lightship weight is the finished vessel, excluding variable components such as crew, fuel, cargo, provisions etc. It is common to divide this weight into three main parts; steelweight, machinery and outfitting [34]. A margin is then applied, which takes any uncertainties into account. The crane is taken into account separately, as it is not present in this standard weight model and is a relatively large part of the total weight.

3

Steel-weight

The steel-weight is estimated by using 2 basis ships, provided by Vuyk. No detailed information can be given as this is sensitive information. The steel-weight is estimated by calculating a coefficient that relates the weight to the deck and section area. Equation 3.1 is used. In this equation, n is the number of decks and m the number of longitudinal sections. This equation is often used by Vuyk, which has proved to give a reasonable estimate. The length is squared as it cannot be taken linearly. This is due to the high longitudinal bending moment that has to be carried by the structure which has a strong non-linear effect on the structure weight.

$$K_{st} = \frac{W_{st}}{n \cdot L^2 \cdot B + m \cdot L^2 \cdot D} \tag{3.1}$$

In addition to this area approach, the steel weight is also calculated by looking at the total inner volume of the hull and the approximation by Watson [34]. This is mainly done to double check the obtained weights on the order of magnitude and similar values are obtained. In table 3.1 the calculated steel-weights are shown.

Table 3.1: Quick list of calculated steel-weights for the monohull. All values in metric tonnes. Will be adjusted!

Vessel name	Area approach	Volume approach	Watson
Q=3000	7914	9332	9672
Q=6000	12594	14974	14224
Q=9000	18097	20616	19640

The steel-weight of the accommodation is also calculated using these two basis ships, which both have a similar accommodation capacity.

Machinery

The machinery mainly holds items such as the (main) engines, propellers, generators, pumps, pipework etc. Watson gives approximation equations for this weight group [34], which estimates the engine and remainder weight with the installed power and engine type. The engines are assumed to be medium speed diesel engines, which are commonly used for crane vessels and diesel-electric power plants. Diesel-electric power plants are often used as a large portion of the installed power is used for crane operations and DP. The installed power is initially estimated using references vessels and adjusted in a later project stage when the installed power is estimated more accurately. The estimation equations for cargo vessels are used, but since crane vessels are equipped with more thrusters for DP, an extra weight is added by estimating the thruster weight.

This weight component is divided into four parts; the engines, thrusters, electrical machinery and remainder weight. The remainder weight is assumed to be spread out over the ship (so the centre of gravity is in the middle), while the other weight groups depend on the general arrangement (explained in section 3.5).

Outfitting

The outfitting is a large group of smaller components, such as the HVAC system, deck equipment, lights, paint, winches etc. Watson [34] gives approximation values that depend on the length and width of the vessel. It is important to note that this weight group does not include the heavy lift crane, which will be described in the next section.

Since the outfitting is determined using cargo vessel approximations with a relatively small crew, some weight has to be added. As some weight is located in the accommodation, such as furnishing, HVAC systems, lifts, windows, lighting, stores and electrical work, a percentage of the hull outfitting weight is assigned to the c.g. location of the accommodation. This is assumed to be 10% after looking at a few weight distributions of various vessels at Vuyk.

Crane

As the crane of a heavy lift vessel is such a large and heavy object, it is taken individually. By looking at cranes for which the weight is available, an estimation can be made. About 7 tub-cranes from Huisman[14] were used and it showed a clear trend that indicated that the weight is approximately equal to the crane capacity. A relation is made to calculate the crane weight per crane.

The crane consists of two main parts; the pedestal and boom. The pedestal is a fixed weight on the vessel, while the boom can be rotated. The boom weight is estimated at 20% of the total crane weight [17]. It is thus important to note that the crane boom weight c.g. location is variable, as it rotates when the crane is slewing.

3.4.2. Variable weight components

There are several components that have to be added to the lightship weight so that the vessel can begin its voyage. These components are described in this section.

Fuel

The fuel weight depends on multiple factors such as the engine type, main propulsive power and the endurance. This is the time that a vessel can sail in perfect conditions at its design speed and full bunker capacity (it is often calculated with 90-95 % bunker capacity). Fuel is required to power the propellers to obtain this transit speed, so an estimation of the main propulsion power is made by using reference vessels. Watson[34] estimates an averaged fuel consumption of 0.184 kg/kWh for medium speed diesel engines which results in the fuel weight.

Crew & provisions

The crew requires fresh water and provisions for the duration they are on board. The weight of the crew including their luggage is assumed to be 120 kg. The amount of fresh water is estimated at 90 kg/person/day for personal use, shower, cleaning and cooking [33]. These fresh water estimates are considered a minimum, but as it is nowadays common to use distillation systems on board that produce fresh water, it is assumed to be enough. The weight for provisions is assumed to be 10 kg/person/day [33]. Using the endurance duration the total weight is calculated.

Cargo

By looking at reference vessels, it is found that it is common to be able to carry 1.5 times the crane capacity as deck cargo. This results in a deck capacity of 4500, 9000 and 13500 mt respectively for the designs. This is taken the same for both the monohull and semi-sub designs.

Stability & hoisting operation

The ballast water that is required for the stability and during a hoisting operation is not accurately determined (yet) as it varies for every loading condition. An estimation of the ballast water capacity (volume) is made by looking at reference vessels, which can be used for the general arrangement in the next section.

In figure 3.3 the weight components per vessel can be seen for the monohull.

Monohull		[mt]	[mt]	[mt]
Steelweight		10233	15829	22752
	Hull	9354	14950	21873
	Superstructure	879	879	879
Fixed components		9449	14431	19760
	Crane	2501	4756	7012
	Crane boom	625	1189	1753
	Machinery	2425	3078	3733
	Engines	447	603	761
	Thrusters	414	581	854
	E-machinery	116	140	157
	Remainder	1449	1754	1961
	Outfitting	3544	4916	6602
	Outfit superstr	354	492	660
LIGHTSHIP WEIGHT		19683	30260	42512
Dead weight		7264	12007	16716
	Fuel	928	1171	1380
	Crew	36	36	36
	Fresh water	1500	1500	1500
	Provisions	300	300	300
	Cargo	4500	9000	13500

Figure 3.3: Weights of the monohull.

3.5. General arrangement

In this section the location of the weight items is determined and the general arrangement is made. At first some general notes are explained that have to be taken into account when making the general arrangement. Then the ship is divided into a number of decks and sections. The crane position is an important decision and is explained elaborately. Then the remaining weight items are discussed and a general arrangement is shown.

The general arrangement should reach the following:

- A weight distribution that results in the least initial heel and trim, so that ballasting is minimized.
- The placement of weight items that have to interact with each other, such as the engine room and fuel tanks, should be chosen so that the extra weight and complexity to make this interaction possible (by means of pipes, valves, etc) is minimized.
- Regulations and criteria by IMO and DNV-GL should be taken into account that restrict the placement of some items.
- Heavy weights should be placed in the lower parts of the hull to keep the centre of gravity as low as possible, which is mainly for better stability.

3.5.1. Decks and sections

Before the main weight components inside the hull can be divided over the ship, a number of decks and sections should be made. A crane vessel should be fitted with a double bottom according to the SOLAS convention [7]. A pipelay crane vessel usually has 3 decks due to the large additional equipment required for these operations, whereas a crane vessel intended only for lifting operations requires less space and 2 decks are sufficient. It is therefore chosen to have two decks; a tank top and a main deck.

The amount of longitudinal sections is determined by looking at reference vessels. An averaged spacing is determined and the ship is divided in a number of sections with equal length. The number of sections also complies with the DNV regulations [7].

The transverse sections are chosen to fit two ballast tanks at the side of the vessel and two midship compartments for additional space for e.g. engines, thrusters, supply space etc.

3.5.2. Accommodation

The accommodation unit is located far forward on deck to compensate for the trim caused by the crane weight and so the bridge crew has a good view. The accommodation dimensions are estimated by looking at accommodations of reference vessels and their crew capacity. A total floor area is estimated together with the number of decks and height. The accommodation is then assumed to be a block that is positioned on the vessel.

3.5.3. Machinery weight

The location of the engine room is shifted more forward to compensate for the trim caused by the crane weight. The thrusters are located for optimum propulsion and DP performance and can thus not be shifted to compensate for heel or trim angles. Crane vessels are often equipped with multiple thrusters that are at the aft and front of the vessel. By looking at reference vessels some front and aft compartments are assigned for thrusters. The remaining weight is assumed to be in the middle of the vessel.

3.5.4. Ballast tanks

As stated before, ballast tanks will be located at the sides of the vessel. These are mainly used to counteract the heeling moment caused by the crane boom and cargo. In addition, most compartments in the double bottom can also be filled with ballast water. The size of these tanks is initially determined using reference vessel data. Stability calculations are carried out and changes can be made, mainly to the height of the double bottom and width of the transverse ballast sections.

3.5.5. Fuel tanks

As the fuel weight reduces during a mission, it should be located close to the centre of the vessel. Locating it further aft or forward (or even at the side of the vessel) would mean that constant ballasting is required. In addition, since 2007 it is no longer an option to store fuel in the double bottom tanks due to a MARPOL regulation [22]. The DNV-GL guidelines also state a requirement for the fuel tank distance from the side hull plating [7], but since wide ballast tanks are located on both sides of the vessel, this distance is always met. It is therefore chosen to locate the fuel tanks in the middle of the vessel. Some space is left open around the fuel tank so that personnel can move to adjacent compartments.

3.5.6. Fresh water tanks

It is common practice to use distillation systems on board to produce fresh water. In many situations it is therefore common to have full fresh water tanks. In these circumstances the water tanks can be considered a fixed weight and no constant ballasting is required. It is therefore chosen to place the fresh water tanks forward in the vessel, in the double bottom tanks. As the main consumers of fresh water are in the accommodation, the position is favourable as well.

3.5.7. Provisions

The space for the provisions have no hard requirements and are fitted in the remaining compartments.

3.5.8. Additional remarks

There are quite some other required compartments that have to be located on the vessel, but are too light or small to have their own weight item. Therefore some additional remarks are made:

- Pump rooms: These rooms are for pumps that mainly transfer ballast water, but also to distribute fresh water around the vessel and pump fuel from the fuel tanks to the day tanks for example. Usually a room is located in the front of the vessel, and some pumps are integrated into a thruster room aft. Some compartments should thus be assigned for these pumps.
- Electrical rooms: For diesel-electric power plants a switchboard room is a large part that has to be taken into account, as it requires quite some space. There is also often an electrical storage compartment. The weight is relatively low, but some space has to be assigned for these.
- Engine related: The engine control room takes (a small) additional space, and some space is assigned for the workshop and stores.
- Day tanks: A smaller fuel tank is located near the engine room that is filled regularly from the main fuel tanks. This enables the engines to keep running in case the fuel treatment or pump system goes down. The fuel also passes by additional filters to make it cleaner. As the fuel tanks are located in the middle of the vessel (and not in the double bottom) close to the engines, the shift in fuel c.g. is small. Day tanks are thus not taken into account.
- Waste water: Fresh water that is consumed has to be filtered before it can be disposed into sea. Ships are often equipped with sewage treatment plants. There will thus be some tanks for waste water and a treatment system has to be installed. However, it is assumed that this is a low weight and does not take a lot of space on-board relatively to other weight items.
- Other: There are even more items that could be taken into account, such as chain lockers, staircases and cofferdams. Instead of looking at these (minor) details, it is assumed that enough spare space is available for all these remaining items. In addition, the permeability of the ballast and fresh water tanks are lowered to account for these items that could limit the tank volume.

3.6. General arrangement sketches

The general arrangement for the monohull design can be seen in figures 3.4-3.6.

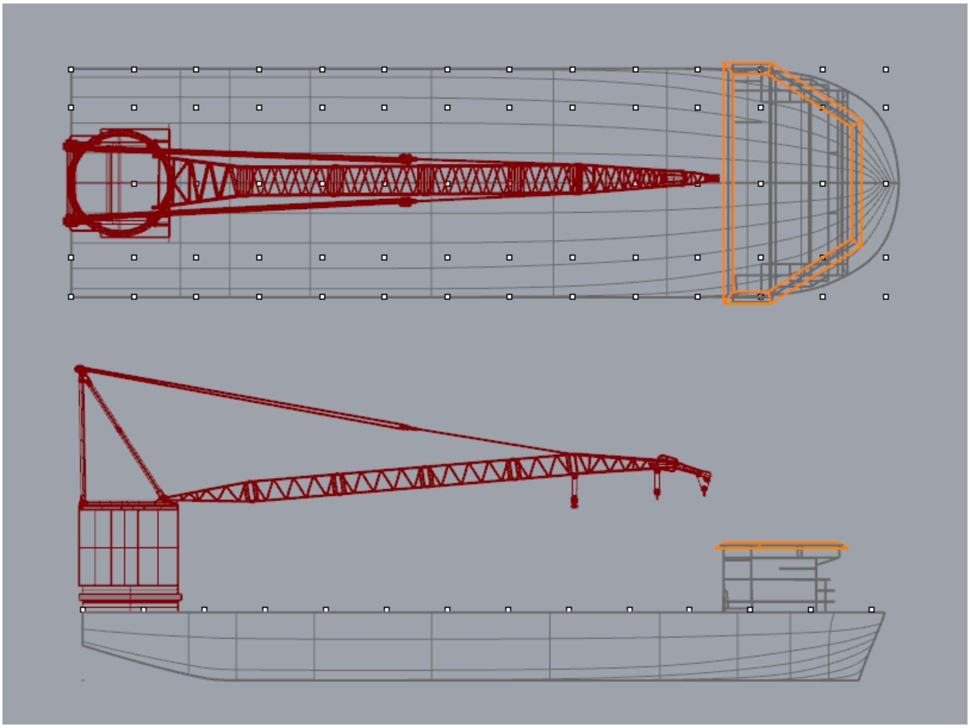

Figure 3.4: Monohull general arrangement drawings of the main deck.

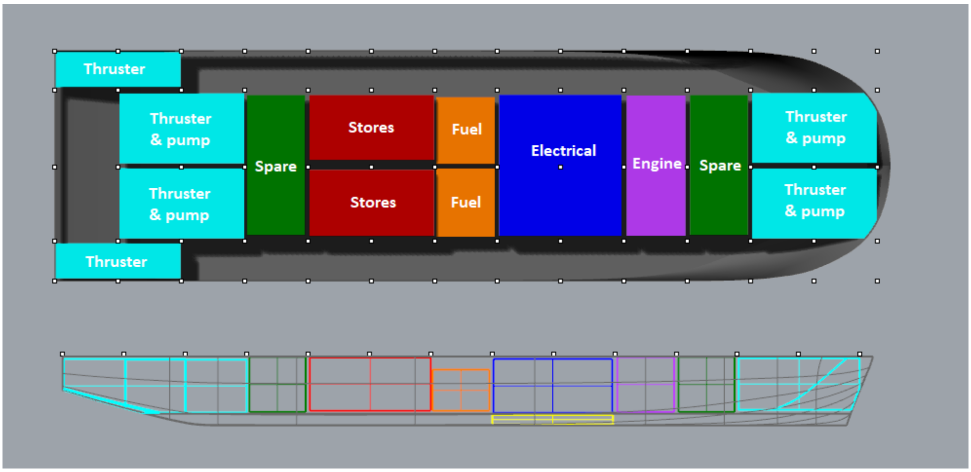

Figure 3.5: Monohull general arrangement of the internal components, except the ballast tanks. All items are separately indicated. The fresh water tanks can be seen in the bottom picture below the other items.

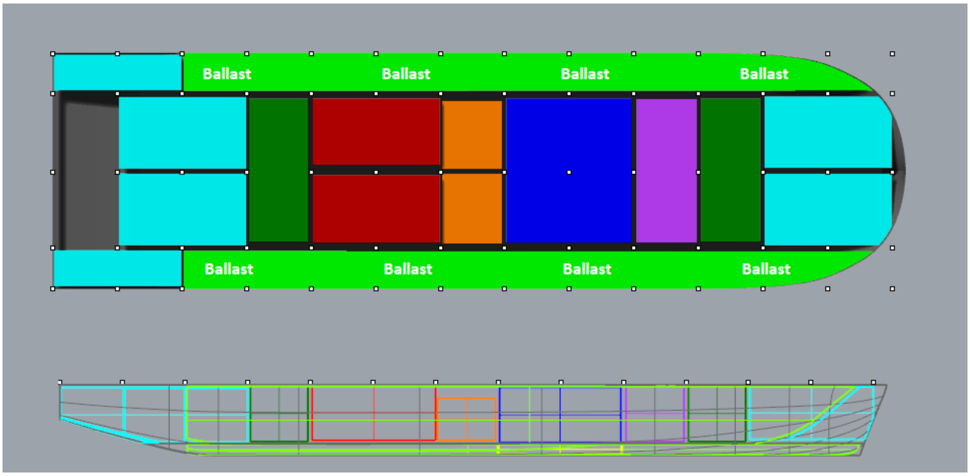

Figure 3.6: Monohull general arrangement of the internal components, including the ballast tanks.

4

Semi-submersible concept design

In this chapter the concept design process for the semi-submersible is described. The main goal is to design three semi-sub concepts that are able to operate as crane vessel. The geometry, weight, hull shape and general arrangement are determined which are important as starting point for the following chapters.

First the main geometry is determined by using reference vessels in section 4.2. An estimate for several other parameters such as the deadweight and displacement is also given. The hull shape design is explained in section 4.3. Then in section 4.4 the main weight groups are calculated. Section 4.5 describes the strategy for the general arrangement and the placement of the weight items. The sketches are then shown in section 4.6. Many more details for the semi-sub concept design can be found in appendix E.

4.1. Reference vessels

Also for the semi-submersible vessels a reference matrix is made. In general the same data is collected which is used for the parameter selection, but also for several other sections and chapters. The list can be found in appendix D.

4.2. Main parameters

Similar to the monohull design, several parameters can also be directly determined using the crane capacity. There are, however, important differences. The general flow can be seen in figure 4.1. It must be noted that this figure is only a small part of the total parameter flow chart. Since the semi-sub has floaters and columns, there are many more parameters to take into account. This figure gives a general idea of the most important parameters. These are influenced by several other design loops such as for the floater width, column spacing and deck clearance in both transit and the operational phase. These are described in more detail in the following sections.

The width is less important for the vessel stability compared to the monohull, and finding a relation for the semi-submersible designs proved to be difficult. Instead, by using sketches of the reference vessels, an estimation of the moment of inertia at operational draft at the waterline is made. Combined with the assumptions and boundary conditions described in the next section the length and width is determined. An important note is that the deadweight is initially kept the same as for the monohull designs. This is mainly done due to the lack of data and the struggle to find good relations between parameters. Since the deadweight is an important parameter that could cause additional differences in cost due to contractual requirements and client preferences, it also makes sense that it is equal. During the weight calculations the fuel is calculated separately and this will cause a small difference in dead weight between the vessels again. For the remaining parameters a similar approach is used as for the monohull designs, where clear relations between parameters are used, while trying to keep common parameter ratios, such as L/B, similar to the reference vessels. A more elaborate explanation can be found in appendix E. The result can be seen in figure 4.2.

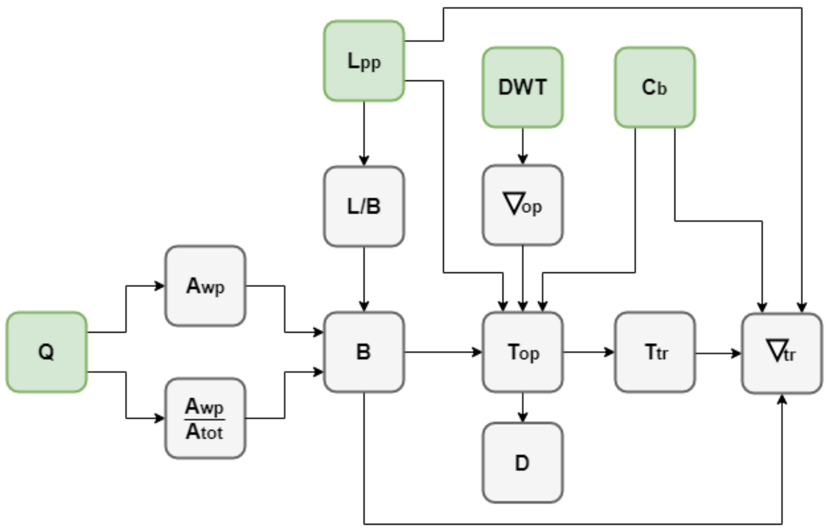

Figure 4.1: Main parameters selection flow for the semi-sub designs. Green boxes mean it is either an input or chosen parameter.

	#1	#2	#3	
Q	3000	6000	9000	[mt]
LPP	110	132	151	[m]
B	76	83	88	[m]
D	32	37	39	[m]
T_tr	8.5	9.9	10.9	[m]
T_op	22	26	28	[m]
C_b	0.90	0.90	0.90	[-]
DWT	25312	48278	71245	[mt]
∇_tr	32155	48886	65617	[mt]
∇_op	58706	89252	119797	[mt]

Figure 4.2: Main parameters that are selected for the semi-sub designs.

4.2.1. Semi-sub floater & column design

The semi-submersible reference vessels show a great variety in design, so several assumptions and decisions have to be made. In general these are simplification assumptions that barely affect the final results. As the goal is to make general designs to be used for the comparison, these assumptions are acceptable.

- Symmetry is assumed around the x-axis, i.e. the width of the floater and columns on both sides are equal. Asymmetry is especially interesting for designs with a single crane or cranes on the side of the vessel. It does make the design much more complex, while it is expected that the difference in performance is relatively small. It is therefore decided it is out of scope of this thesis.
- Some reference vessels have larger columns below the cranes which makes sense for several reasons, but for simplicity it is assumed that all columns are of equal size. Similar to the above, it is expected to have a relatively small influence on the performance.
- The width of the columns and floaters are assumed to be equal. This is also the best weight and cost saving solution according to the research of Kampen [18].
- The structural support beams between the columns are important for the structural design but are neglected from calculations in this thesis. The added displacement due to these supports is only a small fraction and thus has a small influence.

4.2.2. Columns

The column design has a large influence on many aspects, such as the structural design, stability and general layout. The used reference show a great variety in the column design, although it can be concluded that all columns are box-shaped. Attention is paid to a few important aspects.

Column spacing

Each reference vessel has its own spacing between the columns. No clear trend is found that could be used for the designs, although it does show a range of viable options. E.g., the Hermod has a small longitudinal spacing with the columns being 2.0 times as long as the spacing. The column-spacing width ratio is 0.7. Compared to the Saipem 7000 these two ratios are 0.9 and 1.1 respectively, showing a great difference. These two ratios are taken within the range of the reference vessels for the designs.

Number of columns

By looking at reference vessels a number of columns of 4 to 8 is common. The number of columns depends on many aspects, which will be discussed in this section. The main difference is between 4 and 6 columns, while an 8 column design is similar to 6 column designs.

A 4 column design is often used for semi-submersibles with a hotel function (i.e relatively small deck loads). This is seen with the Gretha, Prometheus and Serooskerke vessels. Due to the high spacing of the columns, the deck length that is unsupported by the columns is large. For crane vessels that are designed for heavy deck loads, this is a disadvantage. It could be an option to shift the columns closer to the centre of the vessel, but this significantly affects the stability and increases the required ballast. In addition, if a crane is fitted in the deck-box corner, it is advantageous to have a column directly below for structural reasons.

6 column design is more often seen for vessels that are designed for heavy deck cargo, such as the Hermod and Saipem 7000. As there is now extra columns in the centre of the vessel, the deck-box is well supported and there is a column for every deck-box corner. In addition, ballasting is minimal due to the corner positions. The centre columns can be used to counter heel angles. A disadvantage is that the steel-weight is likely to be higher.

The number of columns for the designs is selected after an iterative process that calculated the required column dimensions after an estimation is made of the required moment of inertia, calculated by using reference vessels. Overall, it showed that for most designs the 4 column option would result in unrealistic column dimensions compared to the reference vessels. Since the 6 column design showed results that fit well with the reference vessels and has the above stated advantages, this column number is chosen. Only for the smallest semi-sub (3000 mt capacity) the 4 column design is an option, but it is on the edge of feasibility. Therefore it is decided to take 6 columns for every design.

4.3. Floater design

The floater is, compared to the monohull design, more of a barge shaped hull. As it is only used for transit and water on on top of the floater has no significant disadvantage since everything is watertight, the free-board is relatively small. In addition, in order to sink to a certain depth, additional free-board would require more ballasting. The floater shape of the Saipem 7000 is used as it gives a good general shape of a barge shaped hull used for a semi-submersible. It is modelled in Rhinoceros after which it is scaled to the dimensions of the designs.

4.4. Weight & placement

Many weight items are similar to the monohull design. For detailed calculation explanations, see section 3.4. Below the weights are summarized and several additional notes are given:

Lightship weight components

- Steel-weight: The steel-weight is estimated using a basis ship, provided by Vuyk. The floaters, columns and deck-box are estimated separately by using a similar area approach as for the monohull design. By relating the amount of decks, sections and their areas with the steel-weight an estimate can be made. The internal volume of the structures are also related to the steel-weight and showed similar results.
- Machinery: This weight is determined using the same relations as for the monohull, but with different installed power, which is determined using reference vessels.
- Outfitting: This weight group is assumed to be equal to the monohull designs. There are of course small differences, but it is decided this is out of scope of this thesis and makes a relatively small difference in the results.
- Crane: As an identical crane is used, the weight is the same. The crane is positioned at the aft at the centreline, which is also similar as for the monohull. This is explained in more detail in appendix F.

Deadweight components

- Fuel: The same method is used, but with a different main machinery power.
- Crew & provisions: As the crew size is equal, this weight is the same.
- Cargo: As stated before, the payload is taken equal for both ship types.
- Ballast water: This is a major difference as a lot of ballast water is required to sink the vessel to operational draft. The ballast capacity is initially estimated using reference vessels.

In figure 4.3 the weight components per vessel can be seen for the semi-sub.

Semi-submersible		[mt]	[mt]	[mt]
Steelweight		18380	26526	33835
	Hull	17501	25646	32956
	Superstructure	879	879	879
Fixed components		10146	15029	19827
	Crane	2501	4756	7012
	Crane boom	625	1189	1753
	Machinery	2558	3259	3952
	Engines	480	652	823
	Thrusters	414	581	854
	E-machinery	123	150	169
	Remainder	1541	1876	2107
	Outfitting	4056	5295	6464
	Outfit superstr	406	530	646
LIGHTSHIP WEIGHT		28526	41554	53662
Variable components		3620	3850	4021
	Fuel	1784	2014	2185
	Crew	36	36	36
	Fresh water	1500	1500	1500
	Provisions	300	300	300
	Cargo	4500	9000	13500

Figure 4.3: Weights of the semi-sub.

4.5. General arrangement

In general the same items have to be placed on the semi-sub as for the monohull. Many of the previously discussed aspects have to be taken into account, such as placing the engine-room close to the fuel tanks. Some important deviations will be discussed.

4.5.1. Decks and sections

Similar to the monohull a double bottom also has to be placed, which is located in both floaters. On top the double bottom two longitudinal sections are placed to provide a gangway and to use the space for ballast tanks. The transverse sections are determined by looking at reference vessels. Besides ballast tanks, there are two pump rooms located in each floater. The permeability of the ballast tanks is lowered a bit to account for the thruster placement.

The columns and deck-box do not have an additional deck, but are divided into sections directly. The deck-box sections and compartments are made by looking at several reference vessels. It was found that, compared to the monohull, placement of weight items was not much of an issue because the available space is much larger. The columns have multiple sections in order to fit three ballast tanks in each column.

4.5.2. Accommodation

The accommodation unit has the same dimensions as for the monohull design and is placed in the opposite corner of the crane to compensate the trim and heel caused by the crane weight. This also gives the crane the largest unobstructed deck area for large cargo.

4.5.3. Deck-box weights

Multiple weight items are placed inside the deck-box. This includes the engine room, stores, fuel tanks and electrical rooms. In general, similar room sizes are assumed as for the monohull and distributed over the deck-box so that the initial heel and trim angles are partly compensated. The machinery weight is also split up in three components and additional thruster weights are added.

Compared to the monohull quite some space is still available after placing the main weight groups. It would therefore be an option to partly fit the accommodation unit inside the deck-box. This would reduce the required accommodation deck-house on the main deck, which could increase the available deck area or decrease the height of the accommodation unit. Both options would be beneficial, but since the monohull also has some empty space, it is decided to keep the original assumption (that the accommodation unit is equal for both vessel types) standing. During the sensitivity analysis the effect of a slightly lower steel-weight can be calculated, which would be the case if some part of the accommodation would be fitted in the deck-box.

4.5.4. Ballast tanks

Ballast tanks for the semi-submersible vessels are located in the floater and columns. The floater can be entirely filled with ballast, except for the fresh water tanks and two pump rooms per floater. Since space is required for staircases, hallways and thrusters, the permeability of the ballast tanks (in the calculation program) is lowered to account for these losses. The columns usually require space for thrusters, chain lockers and storage. Some compartments are left empty so these can be used for these items.

4.5.5. Fresh water tanks

Similar to the monohull design, the fresh water tanks are assumed to be full due to distillation plants. The tanks are positioned in the floater so that the centre of gravity is low and positioned forward to compensate the initial heel and trim.

4.6. General arrangement sketches

The general arrangement for the semi-sub design can be seen in figure 4.4. The general arrangement in the deck-box is not shown because space was not an issue. The weight items are fitted in the deck-box on the opposite side of the crane to counter-act the initial trim and heel. Figure 4.5 displays the floater general arrangement. Most compartments are used for ballast tanks, besides the 4 pump rooms and 2 fresh water tanks. In figure 4.6 the ballast tanks in the columns are shown.

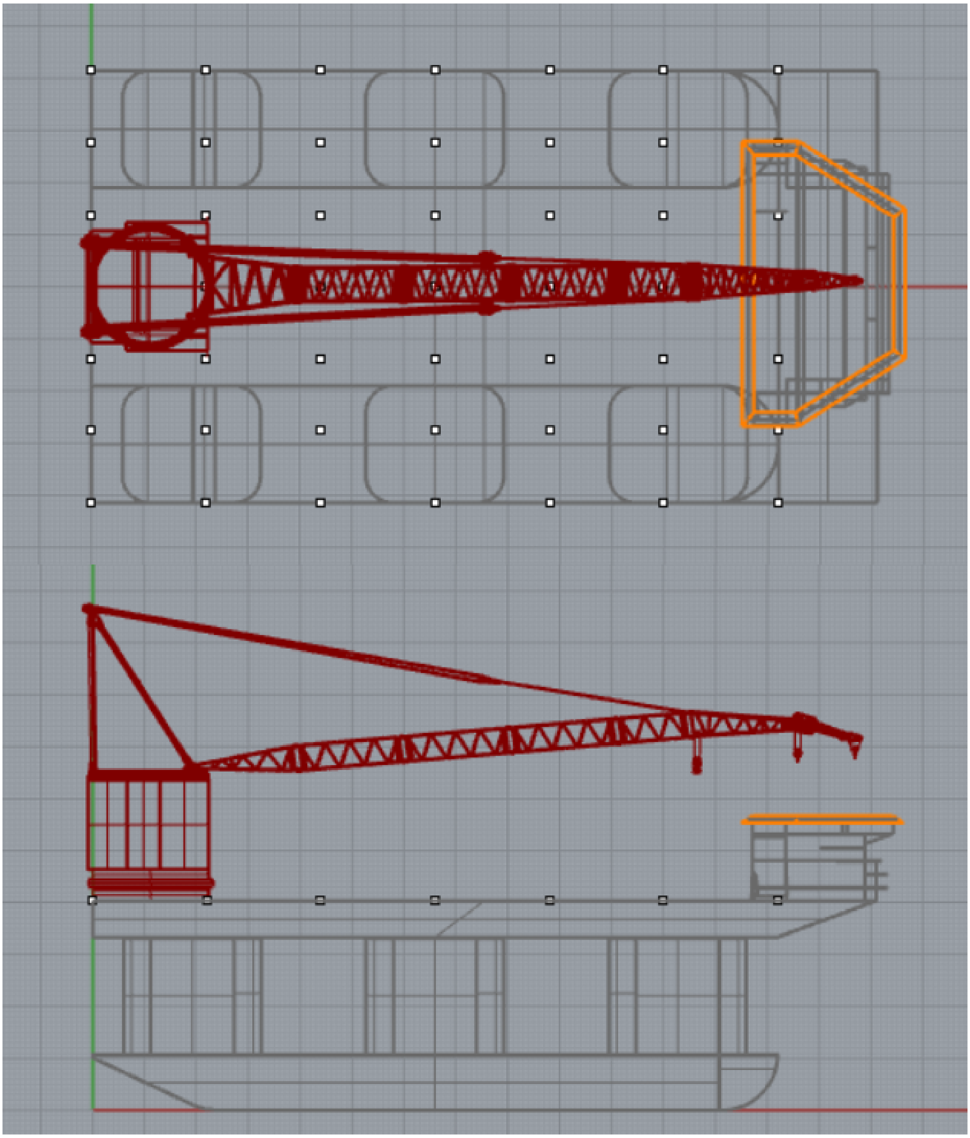

Figure 4.4: Semi-sub general arrangement drawings of the main deck.

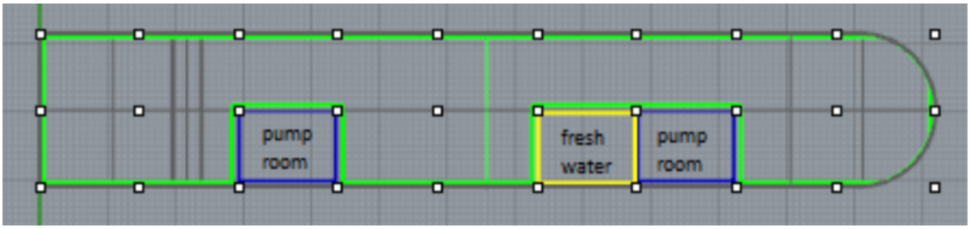

Figure 4.5: Semi-sub general arrangement of the floater. All space is used for ballast tanks unless otherwise indicated for the pump rooms and fresh water tank.

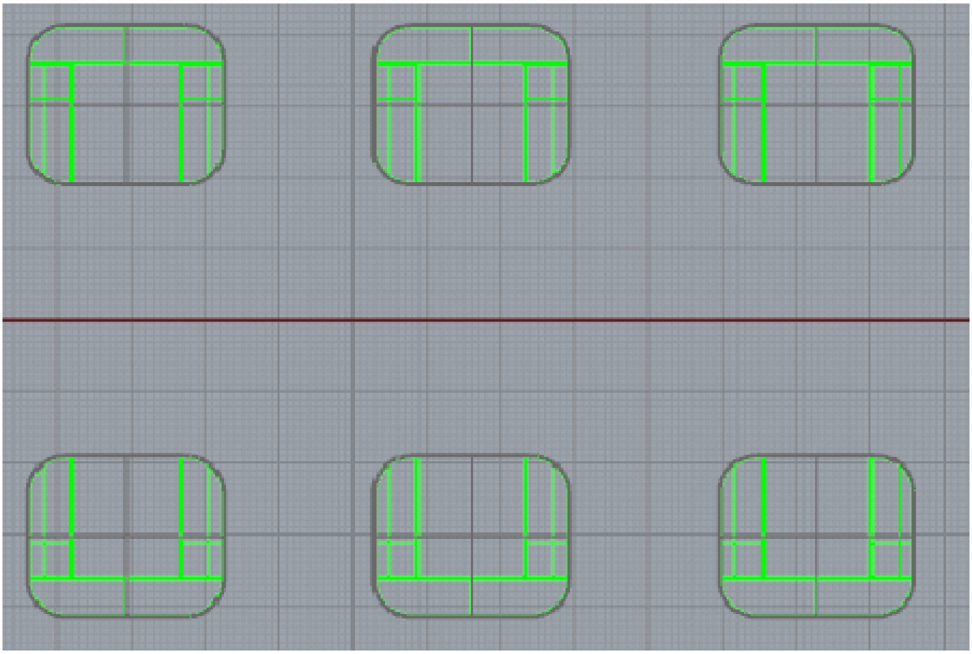

Figure 4.6: Semi-sub general arrangement of the columns. There are three ballast tanks fitted per columns. The rest of the space is reserved for other items such as thruster rooms and chain lockers.

5

Resistance & propulsion

In this chapter the resistance and required power is determined during the transit phase. This is done in order to calculate the fuel consumption which will have a large influence on the operational costs. The total required power is also calculated so that the installed power for the main propulsion thrusters can be determined, which will have an effect on the capital costs. For the calculations empirical relations will be used together with data from reference vessels.

In section 5.1 the resistance is calculated for both vessels, after which the propulsion system is further elaborated in section 5.2. Section 5.3 then further determines the required power.

5.1. Resistance

Since the resistance prediction method is different for both vessel types, these are described individually.

5.1.1. Monohull

Multiple methods are used to estimate the resistance of the monohull. One of the most commonly used and well known methods is from Holtrop & Mennen [12]. However, this method does not predict the resistance accurately as the vessel designs are very wide. The width-to-draft ratios are well out of applicable range, indicating that the resistance predictions have a high error and uncertainty. The designed vessels have a B/T-ratio of around 7 while the empirical relations should be used for a B/T-ratio between 2 and 4.5. It is therefore decided to look at other methods and try to obtain a prediction with a lower uncertainty.

Multiple other methods are found in literature, of which most also have a B/T-ratio restriction. The ones that did not have a B/T-ratio restriction were either for much smaller vessels or had other ratios that did not fit well with the designs. These would therefore also impose a high uncertainty and error. Only the Holtrop, Mennen & van Terwisga empirical method could be applied, but this is mainly for barge shaped vessels. It can, however, be used as it forms the upper limit of the resistance prediction. Since one of the methods predicts the resistance too low (due to the B/T ratio) and one too high (as it is for barge shaped hulls) the real resistance is likely to be somewhere in the middle.

At Vuyk a study is performed to get a better idea of wide vessel resistance. This is sensitive information so no details can be given. By comparing the design geometry, an approximation can be obtained that is more accurate than both methods mentioned above. The resistance is first calculated by the H&M method, after which an additional allowance factor is added. This is determined to be +20% on average for the designs. The barge resistance prediction is also calculated, as this should be higher for all cases and serves as a boundary condition. The final resistance for all monohull vessels can be seen in figure 5.1.

5.1.2. Semi-submersible

The resistance of the semi-submersible is predicted using the 1990 Holtrop, Mennen & van Terwisga method[16]. This is an empirical prediction method for barge shaped vessels. All parameters of the floaters are within the practical range of the method. The resistance predictions are shown in figure 5.1.

Hull-hull interaction

Since the semi-submersible has two floaters, a certain interaction between the two will occur. Not much information in literature was found, although according to (to be added) the wave interaction is minimal and can be neglected. This makes sense as the wave resistance is already only a small part of the total resistance, since the vessel speed is relatively low.

Another effect can cause a contribution, which is the tunnel effect. Once the water hits the floaters, it is compressed between the floaters and causes an increase in flow velocity. Due to the higher flow velocity the frictional resistance increases. However, since the floater depth is finite the water can flow away which reduces the flow velocity again. Due to this it is difficult to determine how much the frictional resistance increases.

In order to get an idea of the tunnel effect, an ideal case calculation is performed. It is assumed that the water can not flow away and is compressed between the two floaters. Use is made of the continuity equation. For this scenario, the increased velocity causes an increase of 34% to the frictional resistance component. For the 6000 mt semi-sub design this accounts for an increase of 9% to the total resistance.

It would be unrealistic to assume the above interaction, but also to assume it does not exist. Since no approximation methods are found and a detailed analysis is deemed out of scope, it is decided to take the middle road. It is assumed that the frictional resistance increases by half of the ideal case, thus about 17%. This gives a small increase due to the hull-hull interaction that can later be adjusted in the sensitivity analysis.

5.2. Propulsion

Heavy lift crane vessels are often equipped with a twin screw configuration, which is mainly due to the high manoeuvrability and redundancy requirement for DP. In addition, the vessels are usually very wide but have limited draft, which makes the use of two propellers more advantageous.

The propulsion type is chosen to be ducted azimuth thrusters. This is, similar to the above, due to the DP performance. A ducted propeller has a much better bollard pull efficiency, greatly increasing the performance during the DP phase. These thrusters have a higher cost compared to conventional propellers, but by looking at reference vessels it can be concluded that these type of thrusters are almost always used, indicating that the extra capital investment is worth it.

Choosing the best propeller is a trade-off between many aspects such as limiting draft, noise, vibrations, cavitation and efficiency. The latter depends on other aspects such as the hull interaction, open water efficiency and even some of the previously mentioned aspects. Instead of looking at all these aspects in detail, it is chosen to choose one efficiency that applies to both vessels. The hull shapes that are chosen for both vessel types have a similar stern shape. This means that, for similar thruster configurations, the efficiency is close to each other when the vessel speed is the same. For a more detailed design the shape of the stern and propulsion system can be adjusted to better fit the specific requirements, such as a different vessel speed, thruster characteristics, structural aspects etc. This is, however, not taken into account and a propeller efficiency of 0.65 is selected, which is a common average for many propellers[34].

In addition, it is assumed that cavitation does not occur and that enough draft and distance to the water surface is available for the thrusters to operate without additional losses. This is done by drawing 3D sketches to position the thrusters. The thrusters are selected from manufacturer Wärtsilä[25], which has a wide arsenal of thrusters of each type. By selecting existing thrusters it is ensured that the resulting power, size and weight are reasonable and have a good efficiency.

5.3. Power

Now that the resistance and propeller efficiency is determined, the required power for transit can be calculated. In figure 5.1 an overview of the resistance, efficiencies, margins and resulting power is shown. As can be seen, the resistance of the monohull is approximately only half of the semi-sub resistance. This seems rather extreme but can be explained since the semi-sub has two barge shaped hulls, whereas the monohull has a better streamlined body. In addition, the semi-sub displacement is a bit higher.

For the transmission and generators a common efficiency is used. A sea margin is also applied, which estimates the effect of environmental influences during transit.

MONOHULL						SEMISUB				
	#1	#2	#3				#1	#2	#3	
R_total	382	482	568	[kN]		R_total	808	912	990	[kN]
P_E	1966	2481	2923	[kW]		P_E	4157	4693	5092	[kW]
eta_D	0.65	0.65	0.65	[-]		eta_D	0.65	0.65	0.65	[-]
P_D	3025	3817	4496	[kW]		P_D	6396	7221	7833	[kW]
eta_trm	0.95	0.95	0.95	[-]		eta_trm	0.95	0.95	0.95	[-]
eta_gen	0.96	0.96	0.96	[-]		eta_gen	0.96	0.96	0.96	[-]
Sea margi	0.8	0.8	0.8	[-]		Sea margi	0.8	0.8	0.8	[-]
P_B	4146	5232	6163	[kW]		P_B	8766	9897	10737	[kW]

Figure 5.1: Resistance, efficiencies and resulting power for both vessel types.

6
Stability

Although the stability has no direct influence on the performance and vessel cost, it has to be checked to make sure that the vessel design is feasible and is able to lift the heavy objects. Therefore in this chapter the stability of all vessels is checked and analysed for several loading conditions. In addition, some noticeable differences are shown and discussed.

First the loading conditions are described in section 6.1, after which design considerations that influences the ballast capacity are discussed in 6.2. The class notations and stability requirements are explained in 6.3. The results are shown and discussed in section 6.4

6.1. Loading conditions

Multiple loading conditions are defined and checked on their stability. The basic loading conditions that are defined by the DNV-GL regulations[7] during transit are:

1. Unloaded departure: No deck cargo is loaded, but the fuel and fresh water tanks are full.
2. Loaded departure: The vessel is loaded with full deck cargo distributed over the deck and the fuel/fresh water tanks are full.
3. Unloaded arrival: No deck cargo is loaded, and the fuel/fresh water tanks are assumed to be at 10%.
4. Loaded arrival: Full deck load and the fuel/fresh water tanks are at 10% capacity.

6.1.1. Operational loading conditions

It is chosen that both vessels have to be able to lift a cargo piece equal to the maximum crane capacity and maximum outreach, over the stern. In reality this could be the case when a piece of cargo is lifted off a barge and installed directly offshore. The vessel should be capable to ballast until a trim and heel angle of zero is reached. It is common for heavy lift vessels that the crane can lift more in one fixed position. Designing a vessel that can lift the cargo in all possible angles would result in a costly over-design especially since the maximum crane capacity is rarely used. Also, (very) small pitch and heel angles could be acceptable, which could increase the revolving capacity a bit. For many operations the vessel is also free to move itself in position with numerous headings.

In general, this boundary condition ensures that the vessel will be capable of performing many heavy lifts with a fully revolving crane, even though at some angles it might not be possible to lift a piece of maximum capacity.

There are two main disadvantages when lifting cargo that affect the stability. First, due to the increased vertical height of the cargo the combined centre of gravity shifts upwards, resulting in a lower GM. Secondly, by moving the cargo in the horizontal plane, a heeling moment is induced. Therefore it makes sense to look at several combinations of crane-tip height and cargo weight. It is therefore decided to look at the following additional loading conditions:

5. Lifting aft - Maximum outreach: The vessel is assumed to be lifting the maximum design weight (i.e. a weight equal to the maximum crane capacity) over the stern of the vessel. The outreach is maximal for a maximum bending moment.

6. Lifting aft - Topped up: This is similar to the previous loading condition but with minimum outreach for which the maximum capacity can still be lifted. This will cause the maximum reduction of GM.

7. Lifting starboard: The vessel is assumed to lift an object at maximum outreach, but not with maximum capacity. As explained before, this would result in a costly over-design (especially for the 6000 and 9000 mt designs). It is therefore chosen, after discussion at Vuyk and by looking at reference vessels, to take the maximum revolving capacity at 80% of the maximum crane capacity. This equates to a 2500, 5000 and 7500 mt lift for the designs.

6.2. Ballasting efficiency

The vessel can be fitted with many additional measures to increase the ballast capacity. For example; The forecastle can be raised so that an additional deck layer is available for extra ballast tanks. Extra ballast tanks can be fitted in the spare compartments on the tank-top. The longitudinal section width can be increased so that more ballast is available on both vessel sides. There are many more examples of design considerations that are worth investigating, but for this thesis this is considered out of scope and the above boundary condition should be reached with the basic layout as described earlier.

All loading conditions are initially ballasted to a trim and heeling angle of zero. This is done by first using the ballast tanks in the double bottom so that the centre of gravity is the lowest. The ballast tanks at the side are then filled until equilibrium is reached. In addition, it is tried to use the least ballast as possible by filling the tanks the furthest away from the centre of gravity first.

6.3. Class notations and criteria

The designs are checked according to the following guidelines:

- General vessel stability: DNV-GL Part 3 Chapter 15 [6]. These include the general stability criteria that apply to all vessels according to the IMO IS 2008 code.
- Special operation vessels, crane vessel: DNV-GL Part 5 Chapter 10 [8]. These hold several criteria that apply for heavy lift operations, such as additional free-board requirements, heeling angle criteria and the loss of hook load.
- Special purpose ships (SPS): DNV-GL Part 6 Chapter 5 [9]. The monohull vessels have to comply with these criteria as a large crew is on-board. Essentially this means that some passenger vessel criteria also have to be met. In general it is found that most criteria are not applicable or are easily met. This is mainly because of the relatively low crew weight compared to the vessel displacement.
- Mobile Offshore Drilling Units (MODU): The semi-submersible vessels have to meet the criteria specified by IMO MODU. As these criteria cannot be used together with SPS, only these criteria are used for the semi-subs.

After consulting all class notation documents and the requirements, a generalised list of criteria is shown below that summarizes the criteria per vessel:

- General intact stability: This set of criteria tests the vessel at the righting arm properties and their resulting righting areas. It also states requirements for maximum heel angles and the initial metacentric height. These criteria apply to all loading conditions.
- Weather intact stability: The vessel (with appendages) is tested by loading it with a (static and gust) wind heeling lever and tests the stability by several criteria. Only loading conditions 1-4 are tested with this criteria.
- Damaged stability: Vessels are usually also analysed on their performance in case compartments are damaged. These compartments can flood and the stability declines. DNV-GL gives a minimum amount of transverse sections for a given vessel length, which is used as a first check. After consulting all class

notations the most important damaged stability criteria is the flooding of two adjacent compartments. There is looked at several possible combinations of compartment flooding and if the vessel remains stable and within a maximum defined heeling angle. These criteria are applied to all loading conditions. Only tanks that are expected to have a significant impact on the stability are taken into account.

- Loss of hook load: During an operation, the crane might lose its cargo. In such an event, the sudden shift of centre of gravity will cause the vessel to rapidly heel to the other side. DNV-GL has numerous criteria that analyse this event and its one of the most important criteria for a heavy lift vessel. This criteria is only applied to loading condition 6. Loading condition 5 is not considered as the lift is aft of the vessel and DNV-GL states no criteria for this configuration.

6.3.1. Weakest axis theory

For the monohull vessels it is reasonable to assume that the longitudinal axis is the weakest axis. This is the axis for which the stability is the lowest. In other words, the axis where the stability of a vessel is pushed to its limits. This is also known as the weak axis theorem[27]. Since the semi-submersible is more box shaped the weakest axis lies somewhere between the longitudinal and transverse axis. By adjusting the position of the crane hook load several axes are investigated and it is found that the longitudinal axis is still a reasonable approximation. Since the semi-sub still has a significantly larger length than width (L/B=1.6) this is deemed plausible.

6.4. Results & discussion

The complete results for all vessels can be found in the specially provided appendix. Since this consists of 176 pages of data, it is not included in the regular appendix. In this section the most important results are discussed. In tables 6.1 and 6.2 the results are shown for a lift over stern with maximum outreach and lift over the side. These loading conditions are deemed the most important and interesting for the comparison. The lift over stern which is topped up, does not show additional differences and is similar to the maximum outreach condition. It must be noted that all loading conditions are modelled and tested, and meet all criteria.

Table 6.1: Stability results for a lift over the stern at maximum outreach.

	Lift over stern - Maximum outreach					
	Monohull			Semi-sub		
Hook load [mt]	3,000	6,000	9,000	3,000	6,000	9,000
Displacement [mt]	38,567	70,910	103,877	46,856	73,539	101,548
Required ballast [mt]	13,174	30,170	47,441	12,375	21,835	33,155
Draft [m]	6.4	8.8	9.6	13.0	15.8	18.3
GM upright [m]	9.6	8.3	12.5	23.0	21.5	21.2
All criteria passed?	Yes	Yes	Yes	Yes	Yes	Yes

Table 6.2: Stability results for a lift over starboard at maximum outreach.

	Lift over starboard					
	Monohull			Semi-sub		
Hook load [mt]	2,500	5,000	7,500	2,500	5,000	7,500
Displacement [mt]	36,680	68,716	106,261	39,970	63,987	102,556
Required ballast [mt]	11,287	28,176	49,825	6,489	13,283	33,163
Draft [m]	6.1	8.6	9.8	11.5	14.1	18.5
GM upright [m]	10.1	10.0	14.7	26.8	25.2	23.6
All criteria passed?	Yes	Yes	Yes	Yes	Yes	Yes

Several conclusions can be drawn from these results:

- Total displacement: For almost all designs the total displacement is almost equal. Only for the 3000mt lift a significant difference can be seen, which could be explained as the semi-sub has to ballast significantly to obtain a certain draft and stability.
- Required ballast: Interesting results can be seen for the required ballast. For all designs the semi-sub requires the least ballasting, which makes sense as the large columns can hold a lot of ballast which have a large arm. The monohull is comparatively limited in this aspect and quickly has to fill tanks that are close to the centre of gravity.
- GM upright: The upright stability figure GM is the highest for the semi-sub. This is as expected, mainly due to its larger width. One might conclude that the semi-sub is thus more stable in general, but it is also important to realise that a higher GM means that the vessel is more rough. Higher accelerations will occur which might be uncomfortable for personnel on-board.

In addition some observations during the performed work are listed:

- Meeting criteria: In general it was easier to obtain a semi-sub design that passed all criteria compared to the monohull. This is mainly due to the high initial displacement and stability properties, such as a high GM upright. For the monohulls a constant balance between tank and compartment size had to be found to find a feasible design.

- Most critical criteria: For both vessels the drop load criteria was the most critical and showed that the designs are pushed to their limits. Compared to other criteria the margin was much lower.
- Easiest criteria: In contrast to the above, the general intact and damaged stability were the easiest to pass. This makes sense as the vessels are initially already very stable (since they have to perform heavy lifts) and enough compartments are used by looking at reference vessels.
- Damaged stability: The damaged stability is manually checked for the 6000 mt monohull and semi-sub. Several scenarios are investigated where the largest impact on stability is expected, but both vessels easily met all criteria. It is therefore decided to not investigate the damaged stability for the other 4 designs, as it became clear these were not critical criteria.

7

Motions

The workability is largely affected by the vessel motions. Since the workability has a large effect on the contractual profit of the vessel, the vessel motions are investigated in detail. In addition, in order to understand these differences better the individual motions are investigated and discussed as well.

Section 7.1 first describes the criteria for which the vessel can still operate. The method is then described in section 7.2. Finally, the results are discussed in section 7.3.

7.1. Criteria

The workability of a heavy lift crane vessel can be determined by limiting the crane-tip motion according to several studies [2, 31, 37]. These state specific performance criteria for a heavy lift crane vessel, which are listed below. If any of these criteria are exceeded, the work on the vessel has to be stopped and the lifting operation cannot be executed.

- Vertical displacement: The significant vertical displacement amplitude at the crane-tip should be lower than 0.45 meter.
- Roll: The significant roll amplitude should be lower than 3 degrees.
- Pitch: The significant pitch amplitude should be lower than 1 degrees.

7.2. Method

The following procedure is followed to get from the vessel and environmental conditions to the workability:

1. The motions of the vessel are obtained by using the program AQWA which calculates the RAOs of the vessel.
2. The motions at a specified point on the ship that are relevant for the criteria are calculated. e.g., combining the heave, pitch and roll motions to obtain the vertical motion at the crane-tip.
3. The motion response is translated to significant wave heights by the use of the JONSWAP spectrum.
4. The maximum allowable wave height for the significant vertical displacement, roll and pitch amplitudes are calculated for every heading and wave period combination.
5. The lowest maximum wave height of every motion criteria mentioned above is determined, to establish which criteria is exceeded first.
6. For every heading, significant wave height and wave period there is checked whether the critical criteria is exceeded (workability = 0) or within bounds (workability = 1).
7. The workability for every combination is multiplied by the occurrence of the specific waves to obtain a workability rate for every heading, by using a scatter diagram of the North-sea waves.

7.2.1. Semi-sub damping

After running the initial calculations for the semi-submersible vessels, it turned out that the damping is not sufficient as very large resonance amplitudes are found for the 3 motions. This is a common problem for column stabilised units, SWATHs and other non-conventional shapes that have a small waterplane area.

To solve this issue, the critical damping is calculated. A 5% percentage of the critical damping is then added as extra damping which reduces the peak amplitudes. This is according to Vuyk a small percentage that can be added without detailed calculations to significantly reduce the high peaks that occur. In the case that a higher damping is required, extensive calculations have to be done to support the addition of such a high additional damping. For the vessels in this thesis this was not deemed necessary. E.g. for the 6000 mt design the heave and pitch maximum amplitudes changed from 10, 22 and 150 to 1.8, 2.4 and 2.6 (m/m or deg/m) respectively with 5% additional damping.

7.3. Results

First the vessel response and characteristics are discussed after which the determined workability is shown.

7.3.1. Vessel response

One of the most important characteristics that influence the motions of the vessel are the natural frequencies. At these frequencies the vessel will respond very strongly, which will cause the workability to drop significantly. In table 7.1 the natural frequencies for both vessel types are shown for the heave, roll and pitch motion.

Table 7.1: Natural periods in seconds for the monohull and semi-sub.

	Heave		Roll		Pitch	
	Mono	Semi	Mono	Semi	Mono	Semi
Q3000	8.6	13.3	9.0	15.8	7.9	17.4
Q6000	9.5	14.2	10.0	16.9	8.6	17.9
Q9000	10.2	14.8	10.5	17.8	9.2	18.2

It can clearly be seen that the monohull has lower natural frequencies compared to the semi-submersible vessel. This is as expected because the semi-sub has a much lower waterplane area compared to its underwater volume. It will also be the main reason for the difference in workability.

The actual response of the vessels in an irregular sea-state can be plotted as well, which is shown in figures 7.1 and 7.2. A few interesting things can be observed:

- Most peaks can be explained by the natural frequencies for that particular motion. Especially for the roll and pitch high peaks can be observed.
- For the semi-submersible several additional peaks can be observed around wave periods of 6 and 10 seconds. This could be explained by the fact that there are 2 floaters with a wave system in between them. In some circumstances there will be a wave crest at one floater and a wave trough at the other. This will cause a resultant force which could explain the motion.
- For some motions another peak can be observed which is caused by another motion. This is most easily seen for the heave amplitudes, which have a relatively small increase in amplitude caused by the pitch and roll motion. This is also known as the pitch and roll induced heave motion.

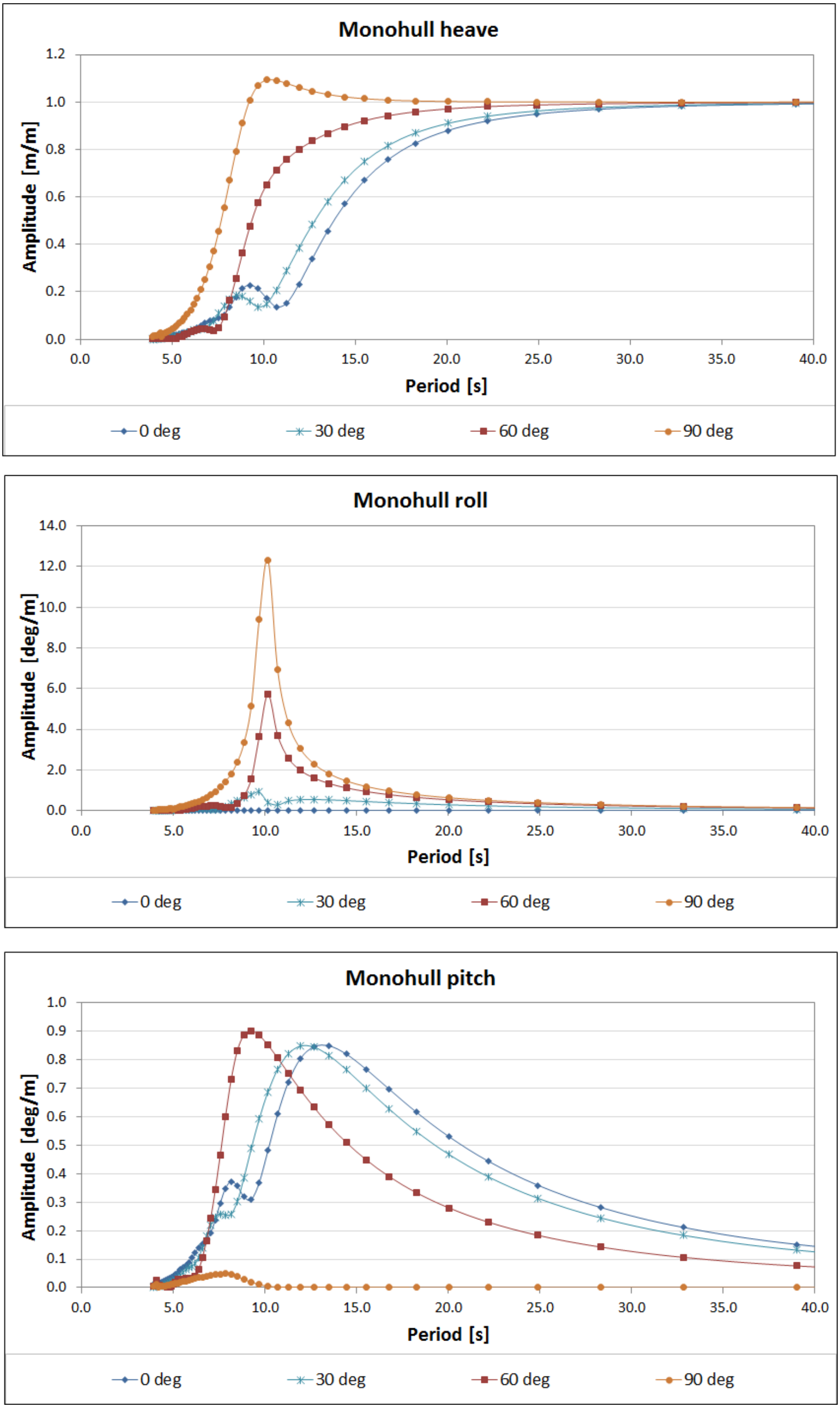

Figure 7.1: Monohull RAOs for heave, pitch and roll in the North-sea.

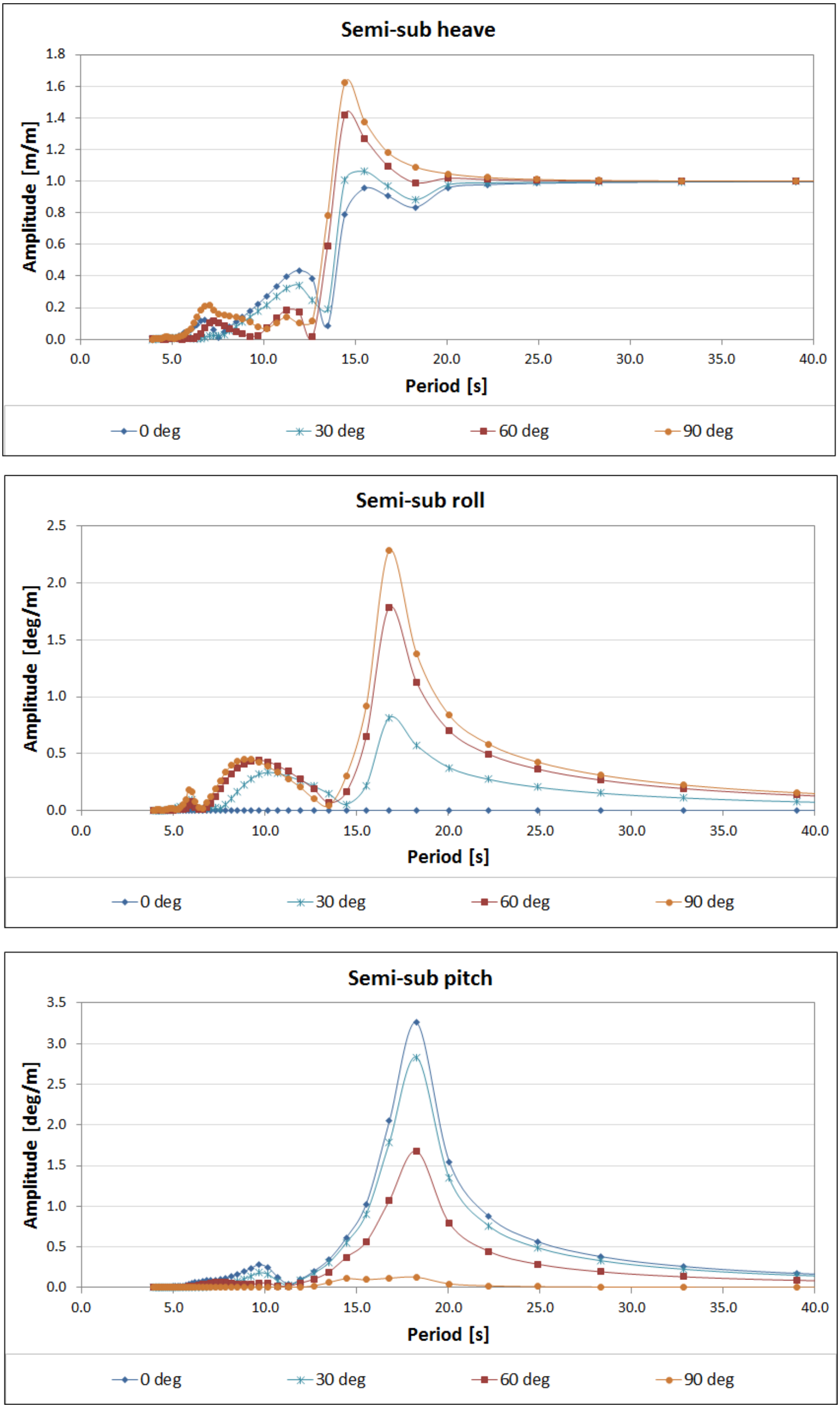

Figure 7.2: Semi-sub RAOs for heave, pitch and roll in the North-sea.

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Not much can be said about the difference in peak amplitudes, since the damping of the semi-submersible is not accurately calculated in AQWA. The manually added damping gives a good approximation to calculate the workability, but the RAO results are not good enough to give precise differences. A more accurate calculation for the damping is possible by modelling the semi-submersible as a simple underwater cylinder and calculating the forces using the Morison equation, but as this would significantly increase the amount of work and has a limited effect on the workability, it is decided this is out of scope of this thesis.

7.3.2. Workability

For the north-sea environment (all-year), table 7.2 shows the workability rates for lifting over the stern & over the side. These are averaged workability rates, as the workability depends on the heading. For a closer look at the effect due to the heading, table 7.3 shows the workability rates for several headings for the 6000 mt design. In the studies by Zheng and McGregor [37], the workability of a ship-shaped vessel and semi-submersible is calculated to be 0.41 and 0.84 respectively for general north-sea conditions, giving a first indication that the results are plausible.

Table 7.2: Workability rates for all concept designs for two different hoisting conditions (North-sea).

	Lift over stern	Lift over side
Mono Q=3000	0.37	0.41
Mono Q=6000	0.44	0.48
Mono Q=9000	0.50	0.54
Semi Q=3000	0.79	0.77
Semi Q=6000	0.84	0.80
Semi Q=9000	0.87	0.82

Compared to the monohull, the workability rates are much higher for the semi-submersible. This makes sense and is as expected due to the higher natural frequencies. As the high motion amplitudes occur at longer waves, the vessel is less affected by a large part of the sea spectra.

There are also differences between the different sizes of the same vessel type. In general the smaller vessels have a lower workability, which makes sense as the vessels have a lower displacement and size resulting in reduced motions. The difference for the semi-subs are relatively lower as these vessels have differ less in displacement and size.

Table 7.3: Workability rates for the 6000 mt designs for various heading angles (North-sea).

Heading	Lift over stern		Lift over side	
	Mono	Semi	Mono	Semi
0°	0.46	0.68	0.54	0.77
30°	0.42	0.79	0.63	0.83
60°	0.30	0.87	0.38	0.76
90°	0.51	0.88	0.30	0.80
120°	0.37	0.91	0.38	0.80
150°	0.54	0.87	0.56	0.81
180°	0.56	0.79	0.72	0.86
Average	0.44	0.84	0.48	0.80

For the monohull it can thus be seen that for both lifting conditions the best heading is 180 degrees. This makes sense due to the bow shape and as there is no roll motion. For the semi-sub lifting over the side this is also the case. However, for a lift over the stern the most favourable heading seems to be in beam waves. This makes sense as the longitudinal distance to the crane tip from the centre of gravity is the longest for a lift over the stern. This induces a large pitch motion response in head or following seas. The reason this is not the case for the monohull, is due to its very large roll amplitude response, making head or following seas the most favourable for both lifting operations.

Due to the difference in longitudinal distance to the crane tip, the workability is higher for a side lift. This is caused by the pitch motion which has a strong influence on a lift over the stern. Also, since the lift over the stern is performed at the centreline the effect of roll is much less compared to the starboard lift. This causes the significant differences near a heading of 90 degrees, where the pitch motion is close to zero.

7.3.3. West Africa

Besides the detailed analysis for the motions and workability in the North-sea, it is also interesting to compare the vessels in a whole different sea spectrum. The sea conditions at the coast of West Africa are well known for their swell waves. Compared to the North-sea spectra, the average wave system consists of lower and longer waves, which could have a large impact on the workability.

In figure 7.3 a rough representation of the wave systems for both the North-sea and West Africa coast[23] are displayed. In addition, an averaged vertical motion RAO is shown for both vessels, which has proved to be the most critical motion during earlier analysis. This figure gives a clear overview and it's easy to see why the workability of one vessel might be higher (or lower) in a certain sea spectra.

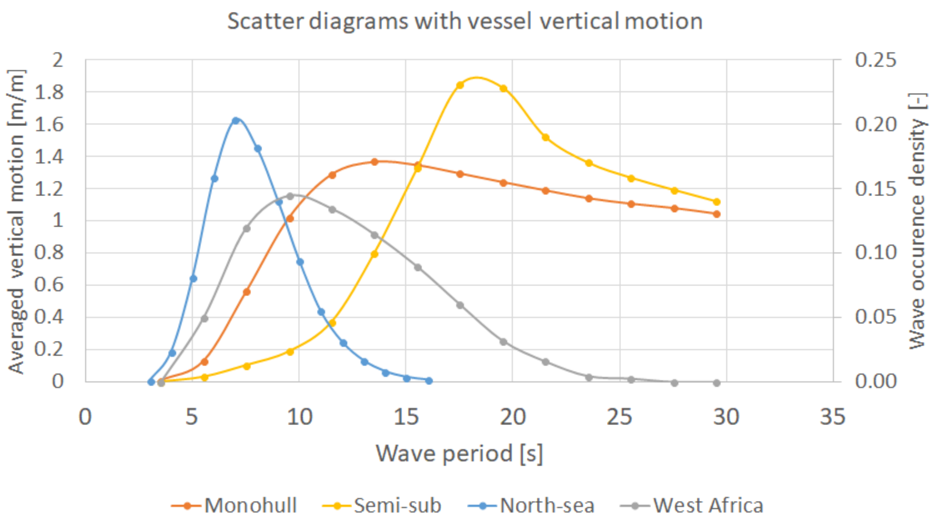

Figure 7.3: The north-sea and West africa wave scatter plot (blue and grey lines), together with an averaged vertical motion plot for both 6000 mt vessels.

In table 7.4 the calculated workability rates for the North-sea and coast of West Africa are compared for a lift over the stern. It can clearly be seen that the workability for the monohull is relatively higher, while the semi-sub is performing a bit worse.

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Table 7.4: Workability rates for the North-sea and coast of West Africa for a lifting operation over the stern.

	North-sea	West Africa
Mono Q=3000	0.37	0.47
Mono Q=6000	0.44	0.50
Mono Q=9000	0.50	0.53
Semi Q=3000	0.79	0.67
Semi Q=6000	0.84	0.69
Semi Q=9000	0.87	0.72

8 Dynamic positioning

Dynamic positioning is the ability to keep position during various environmental conditions. While performing a heavy lift this is critical, as a sudden failure can result in high risk situations. Therefore a high redundancy is required which will have a high impact on the installed machinery and thrusters. It is thus necessary to look into this aspect in more detail. In general the purpose of this analysis is the determination of the required power, to calculate a fuel consumption and determine the capital investment of the thrusters and machinery.

First the general DP system and failure conditions are explained in section 8.1. The method is then elaborated in section 8.2. The propulsion system is elaborated and thrusters are chosen in section 8.3. In section 8.4 the environmental forces are determined. A closer look at the thruster losses is taken in section 8.5. Finally in section 8.6 the results are shown and discussed.

8.1. DP system

It is common for heavy lift vessels to have a DP-3 system, as it is important to keep position while in operational mode and failure of positioning equipment occurs. This is especially true when a heavy lift is performed, where a small position error can have large consequences. DP-3 is thus often used to significantly reduce the involved risks. It is therefore decided to design a DP-3 system for all concept designs. The main difference compared to DP-2 is that in DP-3 there has to be enough redundancy in case a whole compartment (e.g. a switchboard or engine room) floods or caught fire and has to be shut down.

There will be looked at two situations:

- Intact condition: In this situation there is no failure of a DP system component and the DP system works properly without any failures. This is mainly an influence on the OPEX as it results in a consumed power for a certain environmental condition.

- Worst case failure (WCF): Here it is assumed that the worst possible single failure happens. For DP-3 this means a complete loss of an engine room, due to fire or flooding. In this thesis a standard layout is assumed for both vessels, without taking into account the many options for optimization. Examples of such solutions are by having more engine/switchboard rooms or even innovative solutions such as the Wärtsilä Low Loss Concept[35] that divides the main switchboards into 4 separate sections, each with a generator connection. The WCF mainly has an influence on the CAPEX as it determines the total amount of power that has to be installed. More information about the propulsion system can be found in section 8.3.

8.2. Method

In this thesis a quasi static equilibrium of the vessel is used that is influenced by constant environmental forces. Dynamic behaviour is thus not determined, but a constant factor is applied to compensate for this effect. The required thruster forces are then calculated to reach equilibrium. These forces translate into a required power, that determine the CAPEX (installed power) and OPEX (required power during DP operation).

In general it is necessary to design a DP system that does not have a too limited workability and also is not too over-designed so that the cost is much too large. It is a trade-off between workability and cost. Performing an operation in extreme weather is also not possible even when the installed power is high enough, as several other limitations restrict the workability. This is already partly taken into account in chapter 7. A high wind load on the cargo that is lifted will result in high risk situations and high crane bearing loads. The cargo might even start swinging. Also, at a certain environmental condition the situation for the crew on deck might become dangerous and there is a high risk of losing equipment.

8.2.1. Design condition

Based on reference vessels and discussions with engineers at Vuyk, it is decided to design the DP system so that the vessels can withstand wind conditions up to Bft 6 for a DP-3 operation. This means that these conditions have to be withstand in WCF condition (loss of engine room), as the vessel has to continually be able to keep position with a worst case failure. If the vessel operates in DP-2 mode and the WCF is less severe, the capability increases as less power is lost. This is because the WCF might be the loss of an engine or thruster and more connections can be made since no completely redundant DP system has to be made that has to be isolated. The 6 Bft condition is set to be the minimum requirement for every heading. The required power is initially estimated by hand calculations for beam waves where the environmental forces are in general the largest. At the other headings the capability is thus higher, which is particularly the case for head and following seas.

8.2.2. DNV-GL web-app tool

For the calculations a web-app tool is used from DNV-GL. At the time of the thesis work a finished tool is available for monohull vessels. A tool for other vessel types is in development and it was possible to gain access to the beta version to perform the calculations. A valuable side-effect is that more insight knowledge is obtained by testing the tool and communicating with DNV-GL.

In the tool the thruster configuration, switchboards, engines and different types of thrusters can be entered to model the ship with its propulsion system. The environmental forces have to be entered as wind, current and wave-drift force coefficients. The tool then calculates the capability of the vessel for the installed power.

8.2.3. DP consumed power

The DP-tool is unable to calculate the required power for a specific environmental condition as it only calculates the capability at maximum installed power. However, in order to estimate a fuel consumption during the DP-phase, the required power depend on the heading and environmental conditions. In order to obtain these data points, multiple runs are performed in the DP-tool where the total installed power is reduced systematically. By combining the results a relation can be made between the environmental conditions and required power.

8.2.4. Procedure

The general procedure is as follows for a clear overview:

1. A DP requirement is set that states for which conditions the vessel should be able to stay in position. This is done for beam sea conditions to get a first good estimation of how much power should be installed for DP purposes.
2. A manual first calculation for beam sea conditions is done to get an initial power estimation.
3. The vessel is modelled in the DP-tool with its thruster configuration, explained in section 8.3.
4. Thrusters are selected according to the Wärtsilä [25] selection.
5. The environmental forces are calculated and the force/moment coefficients are entered in the tool, explained in more detail in section 8.4.
6. The DP tool is used to calculate the DP capability for every heading.

7. Multiple calculations for alternating installed power are run to get a consumed power estimation for various environmental conditions.

8.3. Propulsion system

In this section the propulsion system is explained. First a general propulsion system is chosen followed by the thruster and position selection.

For a good comparison it is chosen to keep the general system structure equal for both vessels. This means that the amount of thrusters and switchboards are equal. In reality additional engine/switchboard rooms could be used to decrease the loss of power during a WCF. For this thesis it is chosen to assume a basic layout to compare the two vessels with. A total of 2 engine rooms and 2 switchboard rooms are selected. For the current DP-3 system this means that, in case of a WCF, a switchboard room has to be shut down, thus disabling about half of the thrusters.

The general structure can be seen in figure 8.1. It can be seen that the thrusters are divided equally over the switchboards. The specific thruster allocation will be discussed in the next subsection. In reality, power also has to be supplied to e.g. the crane and hotel facilities. The power of these additional users does need to be determined, but for the DP comparison it is not relevant. More information about these additional power users is explained in chapter 9. An emergency generator is mandatory and the vessels have to be equipped with one. However, these are not taken into account as these do not have a large impact on the comparison between these two vessel types.

In addition, since the WCF is an engine-room failure, additional wiring (a detailed single-line diagram) to reduce the power loss during a single component failure is not investigated, as it does not matter for the DP-3 capability.

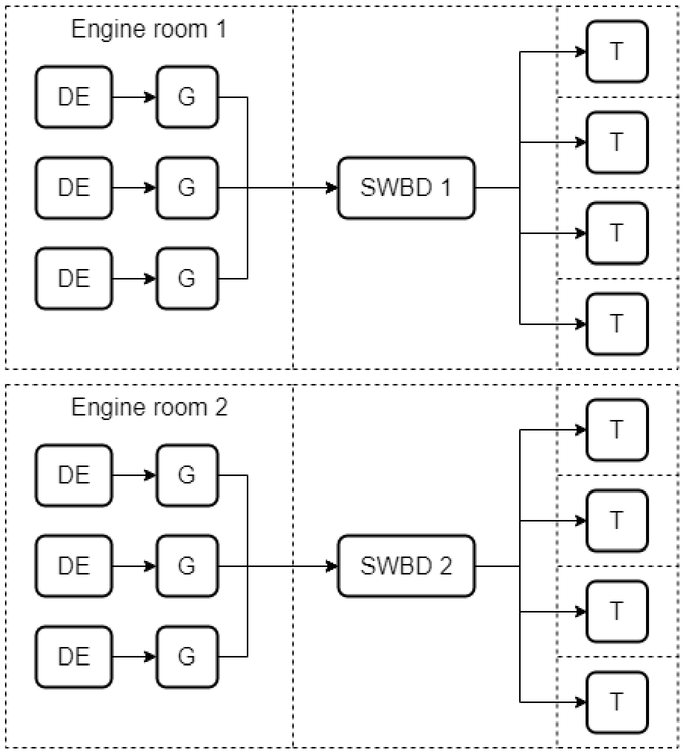

Figure 8.1: Sketch of the propulsion system layout. DE=Diesel Engine, G=Generator, SWBD=Switchboard, T=Thruster

8.3.1. Thruster configuration

A standard thruster configuration is chosen by looking at reference vessels, the vessel characteristics and discussion at Vuyk.

Monohull

For the monohull design 8 thrusters are used:

- 2 Azimuth thrusters that are used for both transit and DP operations, at the stern of the vessel.
- 4 Retractable azimuth thrusters that can be deployed for DP operations, of which 2 at the stern and 2 at the bow of the vessel.
- 2 Tunnel thrusters for additional transverse thrust at the bow, supporting manoeuvrability in depth-limited areas, such as in harbour.

After some testing it is found that a good thruster division over the switchboards is as displayed in figure 8.2. The averaged DP capability for WCF is the highest for this configuration. It could be argued that the optimal division also depends on other aspects, such as the required wiring, but this is not taken into account as it is expected to only have a small effect.

Semi-sub

For the semi-sub 8 thrusters will be used:

- 2 Azimuth thrusters that are used for both the transit and DP phase, at the aft of the vessel.
- 6 Retractable azimuth thrusters that can be deployed for the DP phase, of which 4 forward and 2 aft of the vessel.

Since the resistance of the semi-submersible is higher, it is possible that the required power for transit is higher than is practical for the propellers. It would therefore be necessary to use multiple retractable thrusters to reach the desired transit speed. There could be argued that 4 azimuth thrusters should be installed, but with the current hull design this would mean one of the following:

- If 2 (or more) of the retractable thrusters are replaced by azimuth thrusters without relocating them, this means the minimum draft is significantly increased. This could cause issues while in depth limited areas such as in harbour.
- If, for the above situation, the thrusters are relocated to the aft of the vessel, more thruster-thruster interaction will occur. This is therefore also not ideal.

It is therefore decided to stick with the current thruster configurations and assume that the vessel will, while at open sea, deploy 2 (or more) retractable thrusters to reach the desired vessel speed.

The switchboard-thruster division is similar to that of the monohull, as displayed in figure 8.2.

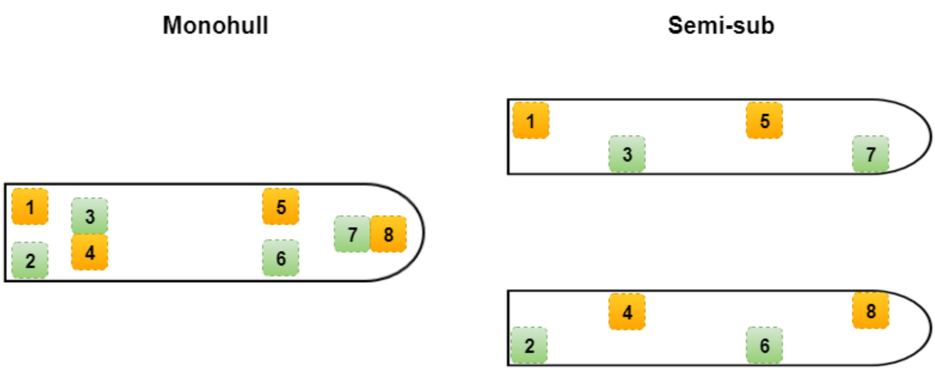

Figure 8.2: Monohull (left) and Semi-submersible (right) thruster division over the switchboards.

8.4. Environmental forces

The environmental forces are a combination of the wind, current and wave-drift forces.

8.4.1. Wind & current

The wind & current forces are calculated using the DNV-GL estimation functions[10], which estimates the forces with a frontal and lateral surface, drag coefficients and trigonometric function. The calculations for the monohull are straightforward and relatively easy to implement in the tool. Since DNV-GL also has a pre-programmed calculation for monohulls, it was possible to verify the calculations for the wind and current forces. For the semi-sub the same functions are used, but with an additional shielding factor as will be explained below.

Shielding effect

For the semi-submersible designs two simple approaches for the wind and current forces can be used:

- The frontal and side projected area can be calculated, which are then used for the force & moment calculation. This is similar to the monohull.
- All separate structures can be taken separately. This results in a side and frontal area that is almost twice as large, since there are 2 floaters and 6 columns.

Both solutions are not accurate enough to make a good estimation of the forces and moment. It is therefore chosen to take a shielding effect into account, since the floaters and columns are spaced at a significant distance, but not enough to be outside the wake of another structure.

Literature gives no clear estimations to take this effect into account. However, after reading several papers and articles it seems to be a reasonable assumption that the wind velocity is reduced to 50% for objects that are in a wake. This is thus merely supported by various literature citations that seem to point in this direction. However, this gives a rough estimation of its effect and is deemed more accurate than the two methods mentioned above. The reduction in velocity is, of course, not simply calculated as it depends on, e.g. the structure size, spacing, velocity, wake-shape, viscosity, temperature and many more. A mathematical expression is derived that describes the additional area that is outside the wake ($V/V_\infty = 1$) and inside the wake ($V/V_\infty = 0.5$) as function of the heading.

The general calculation method is to derive a mathematical expression as function of the heading. The projected area and wake-area is then adjusted by this expression. For both the floaters (underwater) and columns (underwater & above water) an expression is derived. For the columns it is assumed that the additional projected area is a single surface rather than 3 separate ones (but with equal area) to keep the equations simple, but still giving a good estimate.

In figure 8.3 the projected area coefficients of the floater can be seen. It must be noted that the coefficients for the columns are nearly identical due to the simplification assumptions. It is therefore not shown.

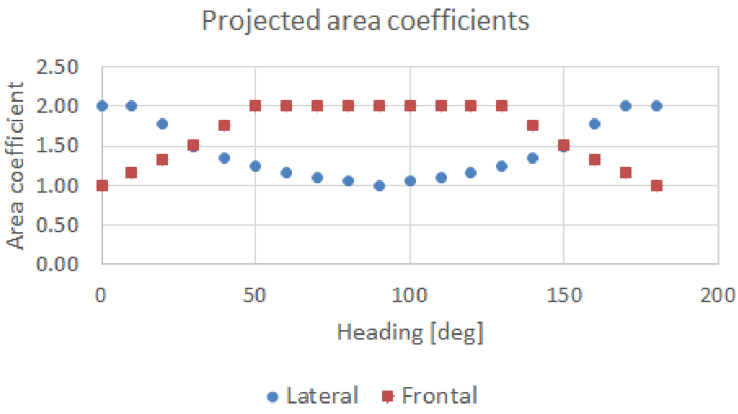

Figure 8.3: Projected area coefficients for the floater.

8.4.2. Wave-drift forces

The second order wave forces are calculated using AQWA. Since this is an output that is calculated with the work performed in chapter 7 it is an easy task to extract it and upload it into the DP-tool. The only difference is the different axis system, so the data first has to be reorganized before it can be entered in the DP-tool.

8.5. Thruster losses

Thrusters experience all sorts of losses, such as flow interaction with other thrusters, skegs and the ship hull. These have to be taken into account as it can have a significant contribution to the loss of thrust. DNV-GL estimates the produced thrust according to the thruster input parameters (Diameter and power) and thruster type. The thrusters are chosen according to the Wärtsilä[25] selection which have a wide arsenal of various thrusters to ensure that feasible propeller designs are used. Thruster losses are estimated to be 10%[10] and the forbidden zones are determined using DNV-GL recommendations[10]. These zones are primarily due to flushing and skeg interaction, which normally results in a thrust loss. Thrusters in reverse are also assumed to have a lower efficiency. Specifically for tunnel thrusters the inlet shape is taken into account by the inlet efficiency.

In some situations thrusters can have high losses which do not fall within the specified 10%. E.g. thrusters number 5 & 6 (see figure 8.2), which are far forward of the vessel, will have high thrust losses when producing thrust in positive surge direction as the flow speed underneath the hull is higher along a large part of the hull length, causing an additional friction loss. In addition for the semi-sub, a thruster that causes its outflow to hit the other floater, will also have higher thrust losses. Thruster losses due to these effects can be as high as 30% [15]. However, these losses can partly be mitigated by giving the thrusters a slight angle so that the flow is directed slightly downwards. This causes a small loss due to the extra vertical thrust component, but is a large improvement for the above described high loss situations. Since the flow is partly directed back towards the boundary due to an under-pressure, this strategy has a limited effect.

Although in some scenarios a substantial loss can occur, it can thus be partly compensated by adjusting the flow angle. It is therefore decided that the 10% assumed thruster loss by DNV-GL is sufficient to compare the vessels.

8.6. Results

In this section the most important results are discussed. First some intermediate results are shown, which will give better insight in the effect of several environmental forces. Then the total performance and capability is discussed.

8.6.1. Wind & current loads

Since the vessel shapes are quite different, the loads are of course also different. This depends on various aspects such as the projected area, the shape coefficients and height above the waterline. Of special note is the shielding effect of the semi-submersible design that significantly increases the forces at certain headings (compared to the method with a frontal and side projected area).

In figure 8.4 the wind and current force/moment coefficients for both vessels are shown. It can be seen that the wind coefficients are quite similar. The main reason for this is that a lot of the objects above the waterline are similar for both the monohull and semi-sub such as the crane, cargo and accommodation unit. The monohull has a more streamlined hull and is slightly less affected by the wind compared to the semi-sub. In addition, the shielding effect is also partly the reason for this difference, as the coefficients are determined using the projected area as stated by DNV-GL and as used in the DP-tool.

A clearer difference can be observed for the current loads. Especially for the frontal current coefficient the semi-sub is much higher. This is mainly due to the hull shape. The semi-sub has large block-shaped structures below the waterline, while the monohull has a streamlined body. The shielding effect can clearly be seen as the Cx coefficient has an odd shape for the semi-sub. This is because a small heading deviation in head or following seas adds a relatively large area that is outside the wake.

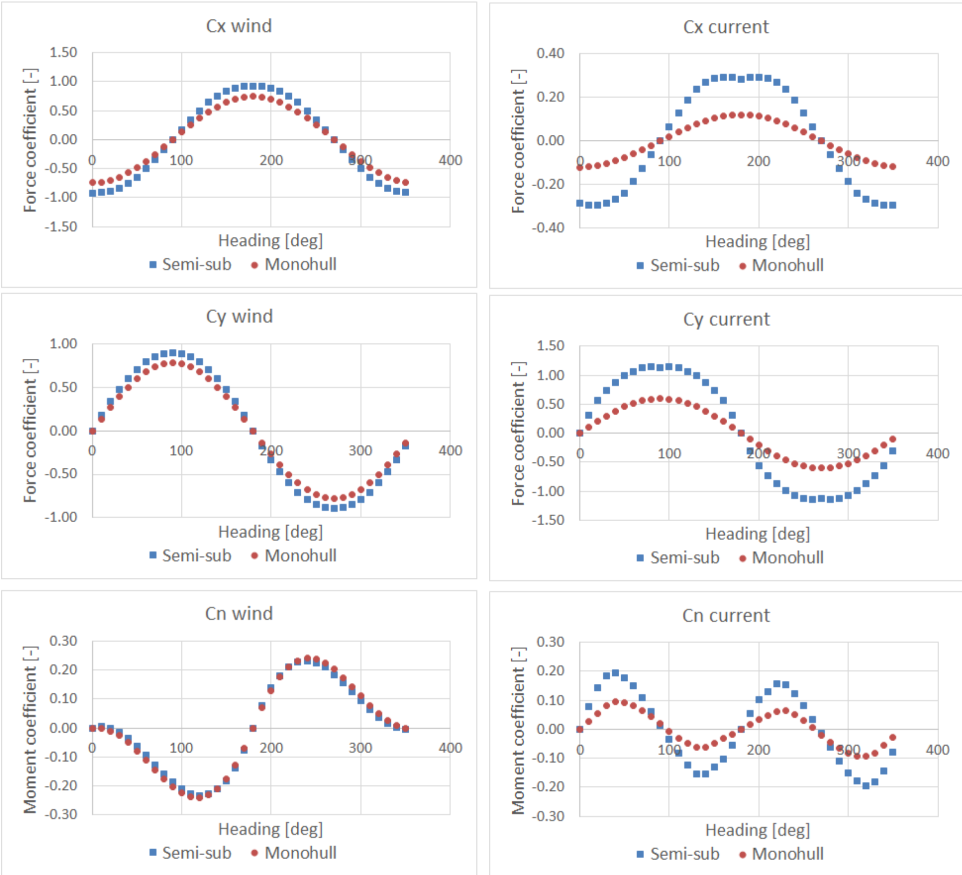

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Figure 8.4: Wind and current force/moment coefficients for the monohull and semi-sub 6000mt design.

8.6.2. Wave loads

The waves induce a second order wave force/moment in surge, sway and yaw direction. In figures 8.5-8.7 these forces and moment are displayed for a selection of wave periods. These wave periods are manually selected to at least include the highest peak values for both vessels and a few intermediate values. The values mentioned in the graph legend are the wave peak periods (T_p) in seconds. For all figures it holds that forces and moments converge to zero for wave periods higher than the maximum wave period displayed in the figures. This makes sense as these are long waves.

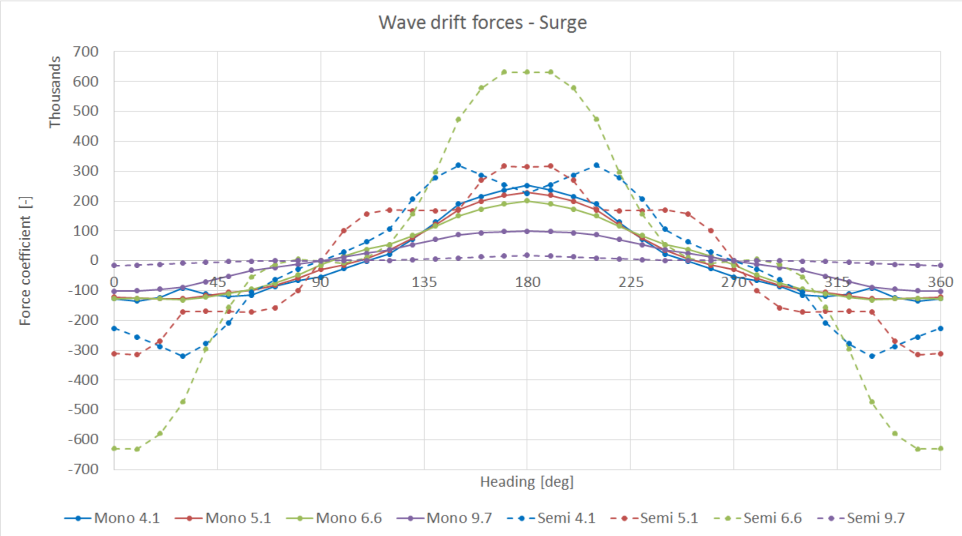

Figure 8.5: Wave drift surge forces for both vessel types for a selection of wave periods.

Some interesting conclusions can be drawn:

Surge

- The surge forces are limited for the monohull due to the small frontal area of the vessel. For the semi-submersible higher peaks are present.
- In beam seas the semi-submersible barely has any surge force due to a nearly symmetrical design around the y-axis. This is not the case for the monohull, as the bow is significantly different compared to the stern.
- The semi-sub surge forces seem to fluctuate a lot more compared to the monohull, which has a nearly perfect harmonic distribution. This could be explained by the fact that the semi-sub has multiple columns which have a much different flow around them. (Needs work)

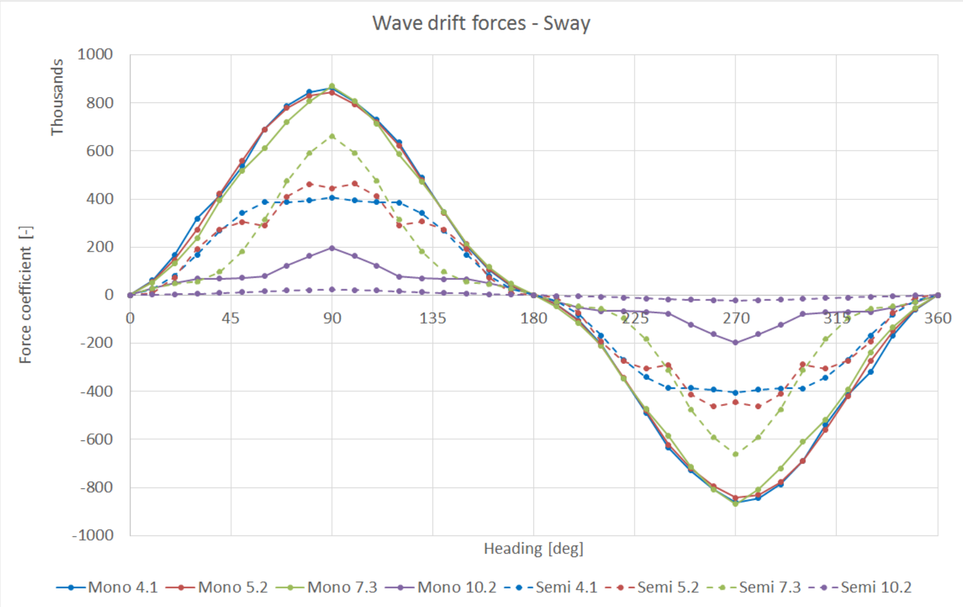

Figure 8.6: Wave drift sway forces for both vessel types for a selection of wave periods.

Sway

- In contrast to the surge forces, the semi-submersible has lower sway forces for all headings and wave periods. This can be explained by the fact that the semi-submersible is more box-shaped (and it can be seen that the order of magnitude of surge and sway is almost the same) while the monohull is much longer and has a large lateral area. This difference in surge and sway forces for the monohull is thus significantly larger.

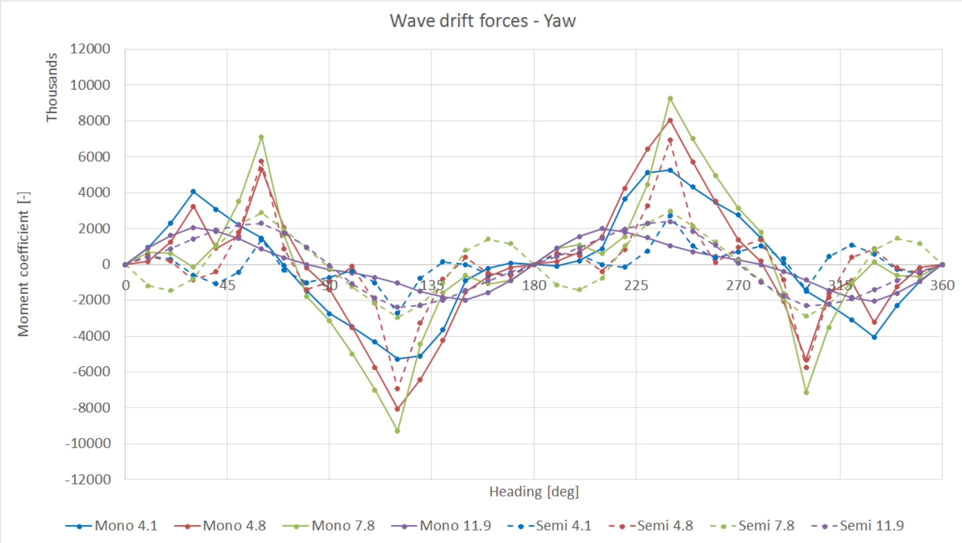

Figure 8.7: Wave drift yaw forces for both vessel types for a selection of wave periods.

Yaw

- In general it can be said that the monohull yaw moments are almost always larger than the semi-sub. This makes sense as the sway forces are larger as well. In addition, it must be noted that higher yaw moments does not necessarily mean that the vessel performs worse in this specific aspect. Since the distance between the thrusters and centre of rotation is higher for the monohull, it can counteracts these moments more effectively. The arm to the thrusters is on average 69 m for the monohull compared to 40 m for the semi-sub, meaning that the monohull has a 1.73x longer arm to compensate the yaw moment. This means that for some situations the monohull is better in compensating the yaw moment, but there are still plenty of situations where the semi-sub performs better.
- As mentioned earlier, the yaw drift moments and sway drift forces are closely related. It thus also makes sense that the yaw moment distribution can be modelled by a harmonious relation that has double the frequency. Since near 90 and 270 degrees the arm is close to zero and near 0 and 180 degrees the force is zero. This is why at 90 and 270 degrees the moment changes its sign.

8.6.3. Environmental loads combination

One might wonder at this point how the wind, current & wave loads work together and which vessel is more ideal for a range of combined environmental conditions. As the wind & current loads are usually lower for the monohull, but the wave loads are sometimes higher, it is interesting to show a comparison of the combined load.

For this comparison the various runs at different installed power can be used that was initially run to determine the OPEX. In figure 8.8 a plot is made that shows the required power for a range of environmental conditions for three headings. It can be seen that for almost all conditions the monohull has a lower power consumption. Only when the environmental conditions are really high the semi-submersible consumes less power. It is thus safe to say that the monohull concept consumes less power compared to the semi-sub in practical terms, as the vessels will not be in DP mode in these extreme wind conditions. This is also practically impossible as not enough power is installed.

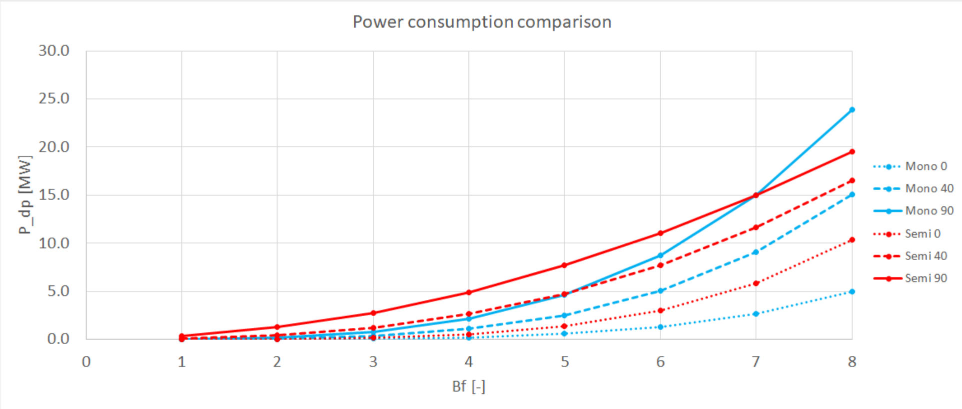

Figure 8.8: Required DP power for a range of environmental conditions for both vessel types, for a few headings.

It is interesting to see that there is a major difference between the two vessel types at various environmental conditions. One might expect that, since the projected areas of the vessels do not change, the differences between the vessel types should be nearly constant. This does make sense for the wind and current forces, but after some testing and data analysis it turned out that the wave drift force causes a major difference between the two vessel types. The following is observed from the data analysis:

- At relatively calm environmental conditions the wave drift forces of both vessels are close to each other. This holds for up to Bft 5 approximately. For some headings the semi-sub even has a slightly higher wave drift force.
- Once the environmental conditions deteriorate (Bft 6 and up) the monohull vessel is much more affected by the waves and the force increases rapidly.
- For the semi-sub the wave drift force also increases but with a much lower increment compared to the monohull, causing the large difference.
- For all environmental conditions the semi-submersible has a higher wind & current force as explained before.

- Due to this large difference in wave drift force, the semi-sub eventually performs better than the monohull in rough environmental conditions for most headings.

8.6.4. DP capability

The required installed DP power for each vessel can be found in table 8.1. It can thus be seen that the required DP power for WCF is generally higher for the semi-submersible vessel.

Table 8.1: Installed thruster power & DP capability score per vessel

Crane capacity [mt]	Installed DP power [MW]	
	Mono	Semi
3000	19.6	21.6
6000	24	26
9000	29.4	34.4

The capability plots for both vessel types (6000 mt designs) are shown in figure 8.9. It can thus be seen that the monohull has the classic oval shape, while the semi-sub is more round. This makes sense as the semi-sub is relatively more square compared to the monohull. In addition, for the WCF condition (DP-3 criteria) it can be seen that the minimum 6 Bft criteria is reached at 90 degrees, where the environmental forces are the largest. Although the semi-submersible has a higher installed power, the DP capability is less for all other headings. There could be argued whether the semi-submersible requires an additional penalty in workability due to this difference, but since the crane is limited at 6 Bft, this is not required. However, there is a different effect on the workability that will be explained in the next section.

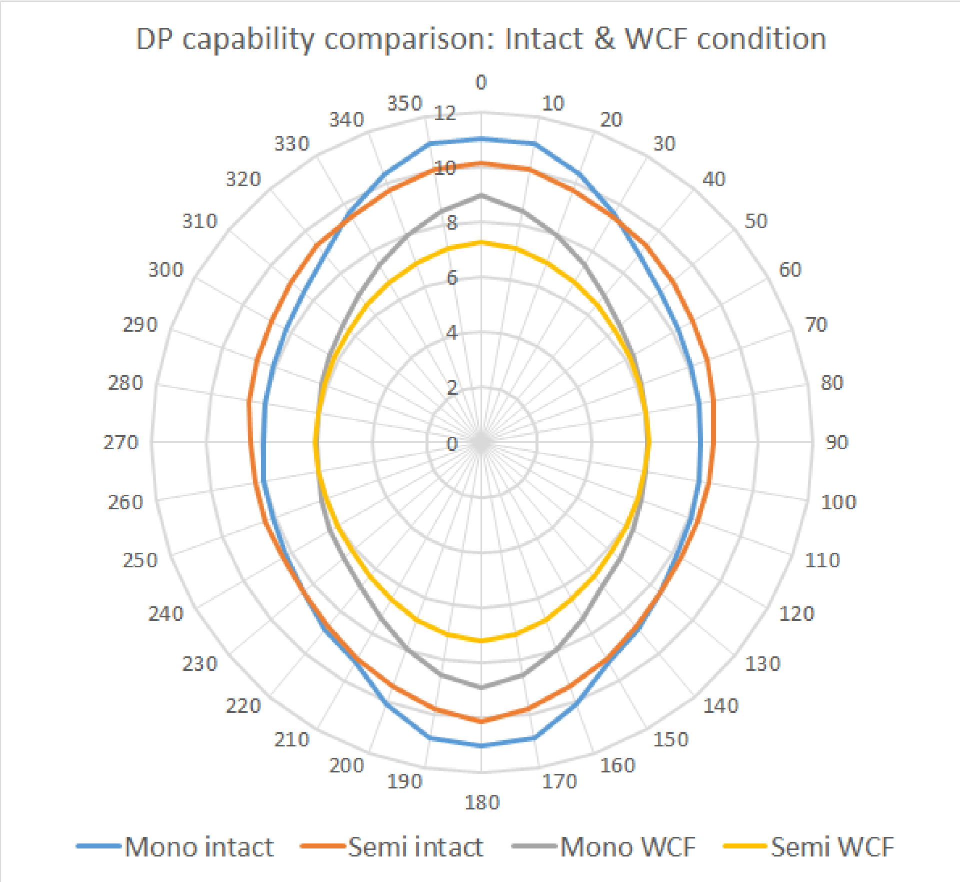

Figure 8.9: DP capability comparison

8.6.5. Workability

Besides the workability as determined in chapter 7, dynamic positioning also implies restrictions on the operability. Since a limited amount of power and thrusters is installed on board the vessel (to maintain position up to 6 Bft), the position cannot be maintained for all possible environmental conditions. There should therefore also be looked at the DP workability.

Since the dynamic positioning is determined for a world-wide spectrum using the DNV-GL tool, while the motion analysis is done for the North Sea and coast of West Africa, it is tricky to combine both to a single workability figure. There is, however, enough data available to make a reasonable combination of both.

For each investigated sea spectra, there is looked at how many waves are outside the 6 Bft range. This is an estimation using the DNV-GL environmental prediction table[10], which relates the Bft number, wind speed, significant wave height, peak wave period and current speed with each other. For the waves that fall above the 6 Bft criteria, the workability is assumed to be 0. This thus causes an additional workability drop that is caused by the lifting criteria. Although for some headings (e.g. the monohull in head waves) the DP capability is much higher than 6 Bft, it is assumed that a lifting operation is not possible due to the strong winds. Therefore the above method is used for all headings.

In table 8.2 the additional workability rates are displayed that are caused by the lifting criteria, which thus limit the operation below 6 Bft. It must be noted that these workability rates are for an operation that is already within the lifting criteria due to the motions. It thus also makes sense that the workability rates are lower for the semi-sub, as these have a much higher motion-workability.

Table 8.2: Extra workability limitations due the to lifting criteria (6 Bft).

Vessel	3000 mt	6000 mt	9000 mt
Monohull	0.988	0.987	0.970
Semi-sub	0.883	0.854	0.835

9

Required power

In this chapter the required power per vessel is determined. This means the installed power per design is established and the consumed power per operational phase is calculated. This has a large impact on the costs, as a required power will return in a fuel consumption. In addition, the installed power has an impact on the capital costs.

First the total installed power and the power consumption per power consumer is determined in section 9.1. In section 9.2 the E-balance, or load chart, is then made to determine the power consumption per operational phase.

9.1. Power consumers

In the previous chapters the required power for transit and dynamic positioning is calculated. It is, however, important to realise that multiple power consumers are present in the vessel and a good balance has to be made. Installing enough power to enable all functions at full load at all times will result in too much installed power that will rarely be used. This will add significant costs that are simply put not worth it. Some important decisions thus have to be made, which is further elaborated in this chapter. Besides the thruster power, there are two main additional power consumers; the hotel and crane.

9.1.1. Hotel functions

The power consumption of the accommodation is estimated by using reference values from Vuyk. The estimation for a crew of 300 is 1500 kW, with an approximate load factor of 75%.

9.1.2. Crane

The power consumption of the crane highly varies during an operation. There are three different crane manoeuvres possible; luffing, slewing and hoisting. Each of these phases have their own power consumption, which depend on various characteristics such as the cargo weight and manoeuvre speed. The cranes are also often designed to carry out two manoeuvres at once (e.g. slewing and hoisting can be done at the same time). It is therefore not possible to determine a power consumption that applies to all cranes, as each will have their own characteristics that satisfy the client. It is, however, important to have a reasonable estimate so that the total installed power and fuel consumption can be determined. Since the cranes are identical for each vessel type, a rough estimation suffices.

A few reference values are obtained from Vuyk. This is sensitive information, so no detailed information can be given. The data consists of important characteristics for two heavy lift cranes, such as the hoisting weight, hoisting speed, slewing speed and power consumption. In addition, some crane information is also found for the 7000 mt crane on the SAIPEM 7000 [17].

A relation can be made between the power consumption and crane characteristics by looking at equation 9.1. This simple equation gives a rough estimate of the power required to hoist an object. The efficiency is assumed to be equal for all cranes.

$$P_{crane} = \frac{W_{cargo} \cdot V_{hoist}}{\eta} \tag{9.1}$$

A good correlation is found when this equation is applied to the reference cranes, where the theoretical calculated power to the indicated maximum consumed power ratio is close to each other. It is therefore relatively easy to obtain a good power estimate if the weight and hoisting speed are known.

However, there appears to be no clear relation between the weight and hoisting speed. This appears to be very specific for an operational profile, so an estimation is made. The values are not too far apart, so the average is taken to get a good estimation. This is taken constant for all cranes.

Table 9.1: Maximum power consumption per crane.

Weight [mt]	Hoisting speed [m/min]	Slewing speed [rpm]	P_{max} [kW]
3000	3.9	0.2	4295
6000	3.9	0.2	8590
9000	3.9	0.2	12885

It can be said that the 3000 and 6000 mt crane are well within the range of the reference cranes. However, the results for the 9000 mt crane are outside the data range and is questionable. The main question that arises is if such a large crane actually does perform the two manoeuvres at once. Because of the increased level of risk for this calibre of crane, the hoisting speed could also be lower. Since both vessel types use the same crane and an error in the power estimation for the cranes will not significantly impact the results, it is found sufficient enough.

9.1.3. Total installed power

Now that all power consumers are known and their load calculated, some decisions have to be made. Enough power has to be installed in order to reach the design criteria, however, it is costly to install more power than is necessary.

Since the dynamic positioning system requires the largest part of the power, and the criteria is to remain at position with environmental conditions at 6 Bft in DP-3 mode, this is the lower limit of the total installed power. Installing less than this amount will reduce the DP capability that is designed for. In addition, since the crew on-board during these conditions need power for on-board systems and HVAC, the accommodation power has to be included.

Another important power consumer is the crane. Since in chapter 8 the lifting limit is set at 6 Bft, this means that at 6 Bft environmental conditions, the crane is theoretically at its limits. It does mean that it should be possible to operate the crane, as this is the design criteria. Therefore, the crane power also has to be included. In table 9.2 the installed power per vessel is shown.

Table 9.2: Installed power per vessel.

	Q=3000 mt	Q=6000 mt	Q=9000 mt	
Monohull	21.9	30.2	37.0	[MW]
Semi-sub	27.2	35.3	43.3	[MW]

The extra power that has to be installed due to the crane also enables the vessel to start dynamic positioning in slightly worse environmental conditions. It could, for instance, be that the vessel starts its dynamic positioning phase while the conditions are approximately 6.5 Bft. If it is expected that the weather gets better, this saves a lot of valuable time.

9.2. Load chart

In order to estimate the power consumption per phase, load charts (or E-balance) are made. In figure 9.1 the E-balance for the monohull 6000 mt design can be seen. The other balances can be found in appendix G. Most power consumers are already discussed, but in order to complete the picture some extra consumers are added. For the deck equipment an estimate is made using balances provided by Vuyk. There are many more smaller consumers that are not included in detail, but an estimate is added as remainder. 2% of the other power consumers showed to be a decent approximation. By summing up the power consumption per phase, this can be used to approximate the fuel consumption, which is used for the OPEX calculations.

9.2.1. Operational phases

- Transit: For the monohull the two main thrusters are used. The load is increased until enough power is available to overcome the vessel's resistance, as calculated in chapter 5. As explained in chapter 8, two retractable thrusters are used for the semi-sub in addition to its two main thrusters, as the resistance is significantly higher.
- DP Normal: In this phase the averaged power consumption that is calculated in chapter 8 is used. For every vessel this meant a load of around 20% per thruster.
- Standby: This phase is only a short time of the total operational profile. It does, however, need to be taken into account. Since the vessel can mostly weather vane in this phase, its power consumption is likely to be lower. In addition, the accuracy in DP mode can be less, so less power is consumed. It is assumed that the power consumption is approximately half of the averaged DP normal power consumption. Although it is unclear if this is accurate, it is deemed sufficient for the goal of this thesis.
- Harbour: In this phase it is assumed that the vessel is docked and the thrusters are off. The crane can still be operated, but operations might also be carried out by onshore cranes. The load is therefore assumed to be lower, at 20%. The accommodation power consumption is also assumed to be slightly lower, at 50%, since the crew can board the vessel. The deck equipment is assumed to be higher, because the vessel has to (de)mobilize (for its next operation).

Additional phases such as manoeuvring and survival mode are thus not taken into account, as these have a small effect on the results. This is mainly because the time spend in such a phase is relatively small.

	Nominal output	Efficiency	Power consumption	Transit			DP Normal			Stand-by			Harbour		
				Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	4000	0.9	4444	0.52	1	2303	0.2	1	889	0.1	1	444	0	0	0
2. Main thruster	4000	0.9	4444	0.52	1	2303	0.2	1	889	0.1	1	444	0	0	0
3. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
4. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
5. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
6. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
7. Tunnel thruster	2000	0.9	2222	0	1	0	0.2	1	444	0.1	1	222	0	0	0
8. Tunnel thruster	2000	0.9	2222	0	1	0	0.2	1	444	0.1	1	222	0	0	0
Crane	8590	0.9	9544	0	0	0	0.5	1	4772	0	0	0	0.2	1	1909
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	750	0.9	833	0.5	1	417	0.5	1	417	0.5	1	417	0.75	1	625
Remainder	697	0.9	774	1	1	774	1	1	774	1	1	774	1	1	774
Total power			39485			7048			12546			5108			4141

Figure 9.1: E-balance sheet for the 6000 mt monohull design.

9

10 Total cost

In this chapter all results and findings of previous chapters are combined. The result is an estimated total cost for each vessel and the expected turning point, in order to answer the research question.

In section 10.1 the CAPEX is determined. The OPEX calculations can be found in section 10.2. The influence of the workability is described in section 10.3. By combining all costs an NPV calculation is done to compare the vessel types with each other in section 10.4.

10.1. Capital costs

The capital costs can be subdivided into multiple components which are determined separately below. In addition, the end-of-life vessel value has to be taken into account and is used to estimate the depreciation. The final capital cost figures for every vessel are shown in the last subsection for a better overview.

10.1.1. Hull

The total costs for the bare ship hull consists of the material and man hour cost. The general formula is shown in equation 10.1, with K_{hull} the total bare hull construction cost, K_{st} the cost of steel per ton, W_g the gross steel weight, K_l the man-hour cost and h_m the total amount of man hours required.

$$K_{hull} = K_{st} \cdot W_g + K_l \cdot h_m \tag{10.1}$$

Material cost

The material is mostly independent of vessel size and can be estimated to cost \$950 per ton gross steel[1]. This includes purchased steel, conservation and an allowance for special materials. Kerlen[18] gives an expression (equation 10.2) to estimate the scrap weight which has to be included in the purchased steel weight.

$$W_g = W_{st,net} \cdot \left(1 + \frac{12 + \left(\left(\frac{W_{st,net}}{1000} + 100 \right)^{-5.3} \cdot 54 \cdot 10^{10} \right)}{100} \right) \tag{10.2}$$

The scrap percentage is mostly between 16 and 22% and is lower for larger vessels.

Labour cost

The amount of man-hours per ton steel is also approximated by using the approach of Kerlen[18]. Kerlen states that the amount of man-hours is mainly depending on the size and block coefficient of the vessel. In general this means that a smaller vessel has relatively more required man-hours. A lower block coefficient also increases the amount of man-hours. This is mainly as it increases the complexity and the steel sheets are thinner that has to be worked with. A relatively full and large vessel has large heavy steel sheets that can be formed and welded quickly. The required man-hours per ton steel are determined for every monohull design according to equation 10.3. The cost per man-hour is set at 45 \$ [1].

$$\frac{h_m}{W_g} = C \cdot f \cdot \left(45.36 \cdot \left(\frac{L \cdot B \cdot D}{1000} \right)^{-0.115} + 3.5 \right) \qquad f = 0.866 \cdot C_B^{-1/3} \tag{10.3}$$

For the semi-submersible design a different strategy has to be applied, since the equation cannot be directly used. Since most semi-sub shapes are less complex compared to the monohull, it is expected that the amount of man-hours per ton steel is slightly less compared to the monohull. To take this into account, the same approximation equation is used but the block coefficient is set at 0.90 and the total volume is used rather than the LBD term. Note that the total volume is simply taken as if the block coefficient is 1, thus that all shapes are simple rectangular shapes (which is thus a similar strategy as taking LBD).

The combined hull cost for the monohull and semi-sub is 36.3 and 55.8 million respectively (for the 6000mt design).

10.1.2. Accommodation

The costs for the accommodation can be divided into two main components; the steel and interior costs.

Steel costs

Unfortunately no suitable literature is found to estimate the steel costs for constructing the accommodation. It is therefore decided to use a similar approach as for the hull. This will still give a reasonable result, and since the accommodation is equal for all vessels, it does not have to be very accurate.

Due to the simple shape the scrap weight is assumed to be relatively low and is set at 15%. The cost per ton steel and man-hour cost is assumed to be equal as for the hull.

Interior

The accommodation interior cost can be estimated with the total floor area according to Aalbers[1]. The formula used can be seen in equation 10.4, with K_i the interior cost, K_m the material cost per m^2, A_f the total floor area and K_l the man-hour cost.

$$K_i = K_m \cdot A_f + 250 \cdot A_f^{0.55} \cdot K_l \qquad (10.4)$$

Total accommodation cost

With a net steel weight of 802 mt for the accommodation unit, the total steel-weight related cost is 0.88 million. The interior cost sums up to 4.05 million for a total floor area of 3972 m^2. The total cost for the accommodation is thus 5.93 million.

10.1.3. Outfitting

The outfitting cost can be estimated by the method from Levander [20]. In this prediction method the gross volume (LBD) is used, which makes sense as the outfitting items are spread out over the vessel. Similar to other estimation tools, a division between material and labour cost is made. In total the material costs are estimated at 25.36 \$/m^3 and the amount of labour work at 0.4 h/m^3.

10.1.4. Crane

There are many different cranes, many of which are uniquely designed and fitted for a specific operational profile. Since there are so many subtypes and options for a range of applications, it makes it difficult to make an accurate cost estimation. Especially since the specific purpose and details are not known. Huisman is a well known crane designer and manufacturer. Thanks to their cooperation a few rough cost estimates for heavy lift tub mounted cranes were obtained. This made it possible to make a cost estimation for the cranes. The costs are estimated to be 30, 65 and 100 million for a 3000, 6000 and 9000 mt crane respectively. Since the cranes are assumed to be equal for both vessels, this rough estimation suffices.

10.1.5. Machinery

The machinery cost can be split up in multiple parts again. For a clearer comparison, the thruster costs are taken individually in the next section. The engines and remainder is describes below.

Engines

According to Aalbers[1] the engine cost can be estimated at 200-300 $ per kW. This is confirmed by literature[20] that estimates the engine cost at 250 $ per kW.

Since a fully diesel-electric power plant is used on the vessels, the engines and generators are combined into a generator set. Since the above costs are for only the engines, an additional cost has to be taken into account for the generators. A few price indications are obtained[36] which are used to establish a relation for the generator sets. The cost per kW for a generator set is approximately 380 $ per installed kW.

Remainder

The cost for the remaining machine items are estimated using the method from Aalbers[1]. In this group many different items are taken into account, such as the fuel system, lubricating system, cooling system, air system and the exhaust system with pumps, heaters, coolers, compressors, ventilators, separators, filters, controls, electrical cabling and many more [1]. The used expression also includes the E-machinery cost, so this is also included. It is thus no surprise that a large cost is involved and this is an important part of the total machinery cost.

The cost is split up into several components consisting of the propulsion plant and their systems. The material and labour are also determined separately. A single function is then derived that describes the cost per ton, as shown in equation 10.5. The cost per ton thus gradually reduces with a larger (heavier) propulsion system, which makes sense.

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$$C/W = 26111 \cdot P_B^{-0.08} \tag{10.5}$$

10.1.6. Thrusters

Thanks to the cooperation of Wärtsilä some price figures were obtained for a range of tunnel, azimuth and retractable thrusters. No precise costs can be shown, but there are some noticeable general differences. As expected, the tunnel thrusters are the cheapest alternative. These thrusters are only about half the cost of main azimuth thrusters. Retractable azimuth thrusters are the most expensive, with an increased cost of nearly 50% compared to the main azimuth thrusters. This results in a very large difference between tunnel and retractable thrusters.

10.1.7. Scrap value

A contribution to the capital costs that one might not immediately expect is the end-of-life value of the vessel. After a certain operational lifetime the vessel reaches the end of its life. This could mean that the vessel is scrapped, refitted or has an extensive maintenance to increase its lifespan. If the vessel is scrapped, the vessel is disassembled (either by beaching or in a dry-dock) and all of its items and material are sold. If there is decided to extend the vessel lifetime, additional money has to be put into the vessel.

There are two main things that are important. In the event of scrapping, the materials are sold and the vessel thus still has a value at the end of its lifetime. Therefore, a certain amount of money is obtained at the end. In the event there is decided to extend the vessel lifetime, money has to be invested to give it several additional operational years.

In both situations it is important to realise that the vessel has a scrap value, and extending the vessel lifetime does not necessarily have to increase the cost per unit time.

It is therefore decided to assume a certain percentage of the total investment costs that the vessel still has at the end of its life. This percentage is hard to determine, as it differs per ship type, the current market and shipyard. For example, heavy lift vessels have an expensive and large crane on deck. It is unknown how much this crane is still worth after its lifetime or that it also has to be completely scrapped after the vessel lifetime. By looking at several reference vessels an idea of the scrap value is obtained [30][28]. 470$ per metric ton of the light ship weight is taken, which amounts for approximately 5-7% of the new building value.

10.1.8. Total capital costs

To summarize all cost calculations for the capital expenditures, table 10.1 shows the individual items and the total capital cost per vessel.

Table 10.1: Capital expenditures for all vessels.

	Monohull			Semi-submersible			Unit
Crane capacity	3000	6000	9000	3000	6000	9000	[mt]
Hull	28.5	47.4	72.1	56.6	63.4	68.0	[$10^6$$]
Accommodation	6.4	6.4	6.4	6.4	6.4	6.4	[$10^6$$]
Outfitting	4.5	7.9	11.4	8.3	10.3	11.3	[$10^6$$]
Crane	30.0	65.0	100.0	30.0	65.0	100.0	[$10^6$$]
Machinery	27.5	36.9	42.3	38.6	41.3	45.8	[$10^6$$]
Thrusters	5.6	8.3	10.7	10.4	11.5	12.9	[$10^6$$]
Building cost	102.5	171.9	242.9	150.3	197.9	244.4	[$10^6$$]
End-of-life value	9.3	14.2	20.0	13.4	19.5	25.2	[$10^6$$]
CAPEX/year	3.7	6.3	8.9	5.5	7.1	8.8	[$10^6$$]

10.2. Operational costs

10.2.1. Crew

Since the crew on both vessels is the same, the accuracy of the crew wage hour estimation does not need to be very precise. It does, however, influence the importance of other cost differences. In the paper by A. Aalbers[1] a crew cost estimation is given for a mixed crew, giving a yearly salary of 50,000 $. An upper roll factor of 1.5 is included for personnel that is on holiday or requires sick-leave and other circumstances. A total salary of 75,000 $ is then estimated that includes the wages, costs, travelling expenses and casualties. For a total crew of 300 persons, a total yearly cost of 22.5 million is calculated.

10.2.2. Fuel consumption

The cost of fuel depends on the different operational phases. For the total lifetime of the vessel, the total time per operational phase is calculated to determine the fuel weight that is required. The power is determined using the load chart in section 9.1. A specific fuel consumption of 0.17 kg/kWh is assumed, which is a common consumption for Wärtsilä engines[36]. These medium-speed diesel engines can burn heavy fuel oil which typically has a price of 375 $ per mt[26]. However, it is questionable if this fuel can still be used in the future, mainly due to its environmental impact. Another commonly used fuel is Marine Diesel Oil (MDO), which is cleaner but more expensive (425 $ per mt[26]). It is therefore decided to take the average, which is 400 $ per mt. Using the required power and these characteristics the total fuel price per unit time can be calculated.

10.2.3. Insurance

The costs for insuring the vessel is estimated to be 1% of the new building value of the vessel[1]. Different types of insurance are possible, but the above is a rough estimate to take it into account.

10.2.4. Maintenance & repair

Many different factors influence the maintenance & repair costs as it depends on e.g. the vessel type, fuel type, age and quality of the building yard. Similar to the insurance costs, a rough estimate is given based on the new building price[1].

Besides the occasional failure of components and system maintenance the vessel also has to dry-dock regularly for inspection. This usually happens 3 times in its lifetime after 3, 8 and 13 years after delivery[1]. A special survey also has to be conducted after 5, 10 and 15 years. A total cost estimate of 11.8% of the new building price is used which takes all previous mentioned events in account.

10.2.5. Management

The cost for management is estimated at 0.5% of the new building price[1].

10.2.6. Total operational costs

By combining everything, the total operational costs can be seen in table 10.2.

Table 10.2: Operational costs for all vessels.

	Monohull			Semi-submersible			Unit
Crane capacity	3000	6000	9000	3000	6000	9000	[mt]
Crew	22.5	22.5	22.5	22.5	22.5	22.5	[$10^6$$]
Fuel consumption	2.5	3.1	3.7	3.6	4.4	5.1	[$10^6$$]
Maintenance & repair	0.8	1.3	1.8	1.2	1.5	1.8	[$10^6$$]
Survey	0.3	0.6	0.8	0.5	0.7	0.8	[$10^6$$]
Insurance	1.0	1.7	2.4	1.6	1.9	2.4	[$10^6$$]
OPEX/year	27.1	29.1	31.3	29.5	30.9	32.5	[$10^6$$]

10.3. Workability

Now that the capital and operational costs are determined, a closer look at the effect of the workability can be taken. A vessel that is the cheapest to operate does not necessarily mean it is the best vessel for an operation. It could be, that the workability is so low, that the extra expenses to wait for a weather window are so high that the other vessel is better overall. It is therefore necessary to take a close look at how the workability affects the operational profile and an estimate of the extra expenses should be made.

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10.3.1. Profit margin

The vessels are used to complete contracts that give an income to cover the vessel expenses. The workability could impair the profit of a vessel, or the client can induce penalties if the contract is fulfilled too late. Even if the client gives a contract that continues payment in case the vessel cannot start the operation due to severe weather, the client could be willing to pay more if the vessel is more likely to complete a contract earlier (In other words; the client is likely to pay more for a vessel with a higher workability rate).

It is common for a business to have a profit margin of 10%, which is confirmed by S&P 500 which determined the average profit margin of 11%. This means that the income is 10% higher than the averaged cost per year. This estimated income can then be influenced by the workability. By assuming a fixed income, some voyage costs can be ignored such as the harbour costs, loading and offloading, canal fees, commissions, claims and many more. These are all nearly the same for the vessel types, and since the income is fixed, it does not have to be determined.

10.3.2. Income

With a profit margin of 10%, the income is approximately 110% of the total cost expenditures of the vessel. However, it does not make sense that the semi-sub vessel with higher cost expenditures compared to the monohull, has a higher income if the function and operational profile is equal. Therefore, the income is averaged for vessels of equal crane capacity. This means that the income is fixed for an equal operation, which requires a certain crane capacity and operational profile. This also gives both vessel types an equal basis that can be used for the comparison.

10.3.3. Operational profile

Now that the income is determined for each vessel, a closer look has to be taken at the operational profile. The workability has an effect on the duration of the operational phases, which can induce an additional loss. For example, a monohull with 40% workability compared to a semi-sub with a workability of 80% will take much longer to complete its lifting operation as it has to wait for certain weather conditions. This could be seen as a slower rate of income, as additional time is required to complete a contract. However, it is important to realise that for this comparison a clear basis is also required. For this, the operational profile that is established at the beginning of this thesis is used.

The operational profile that is established in chapter 2 is for monohull vessels that operate throughout the world. It is therefore important to realise that also this operational profile already has a certain workability rate that sometimes causes the vessels to stop their operations. Since these are monohull vessels that are used across the globe, the workability rate for this operational profile is set at 50% [37]. This means that a vessel with a lower workability will result in an additional loss, while a higher workability vessel will gain additional profit. This may seem like a very rough guess, but is done after the motion analysis is performed, to have a good estimate of the monohull workability. In combination with the classified information of the operational profile from Vuyk, the 50% workability rate estimation is deemed plausible.

10.3.4. Effect on cost

The difference in workability causes two effects on the total cost. Each one will be described individually.

Effect on income

First, due to the difference in operational duration there is an effect on the income. It is assumed that the additional time required to complete an operation, the vessel is in standby mode. The operational duration in this case is the DP-phase. This means that, if the workability is lower, the standby phase increases. For example, the DP-phase consists of approximately 55 days per year. If the workability of a vessel is 25%, the time added to the standby phase is 55 days as it takes double the time to complete the DP operation. This is of course an extreme example, but shows the general approach. An example for a low and high workability vessel can be seen in figure 10.1.

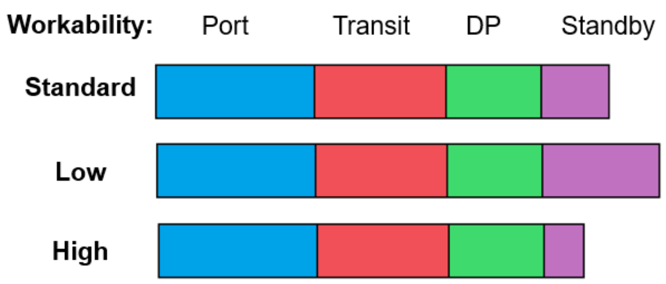

Figure 10.1: Illustration of the workability effect on total income.

It is then assumed that the income per completed contract (or operation) is the same. This means that the money that is generated for a certain mission is fixed. If a vessel has a low workability, and requires more time to complete an operation, the income over time would be lower. This can then be seen as an additional loss.

For the established operational profile this can be seen as the following. Applying the phase durations for a single averaged mission, it is clear that the total operational duration for this mission is extended if the workability is lower (due to the added time in standby). If the income per mission is then fixed, the income per day is then lower. Using this new income per day for every vessel, some clear differences between the vessels arrise.

Effect on operational cost

One also has to realise that the workability has an effect on the operational costs. Since the workability causes additional waiting time, the standby phase will deviate from the base operational profile. Since the power consumption is different in each phase, the operational costs will be different. Similar to the income, the operational costs over this new timespan are determined by averaging the costs over a year. Since the power consumption in standby is the lowest, the operational costs will be slightly lower if the vessel has a low workability rate. Example normalized operational profiles can be seen in figure 10.2.

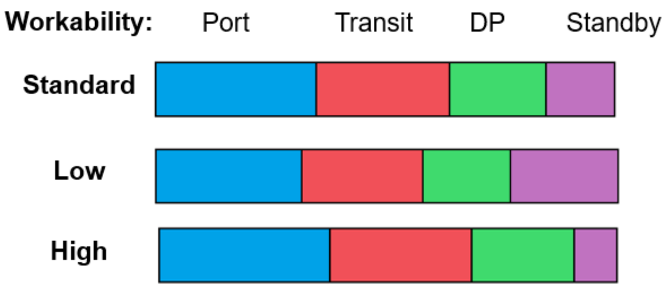

Figure 10.2: Illustration of the workability effect on the operational costs.

10.4. Net Present Value

Now that all expenditures and the effect of the workability is determined, the total vessel value can be calculated.

A common economical strategy to determine if an investment is profitable, is to calculate the net present value (NPV). This figure determines the difference in cash inflow and outflow over the vessel lifetime. Equation 10.6 shows how to calculate the NPV.

$$NPV = \sum_{t=1}^{T} \frac{C_t}{(1+r)^t} - C_0 \tag{10.6}$$

In this equation multiple important financial aspects are taken into account. First, the total cashflow in a year is calculated, which is indicated by C_t. This includes the operational costs and income. Then the money value in the future has to be determined, which is done with the parameter r. This is the real discount rate and includes the rate of other investment options with similar risk, which investors might obtain somewhere else. This is set at 5% which seems like a good estimate according to several readings[19][24]. The total is then summed over the operational lifetime. Finally, the initial investment cost, C_0, is subtracted to obtain the net present value. Overall, a positive NPV means the project is worth investing in and a negative value does not. Since in this thesis a comparison between vessels is done, the number itself is not important, but the point where the two vessel types have equal NPV.

The NPV results can be seen in figure 10.3. The relation seems nearly linear, which is also why a linear relation is used to calculate the turning point. This is also true for the West Africa environment. The following turning points are then calculated:

- North Sea: 6095 mt
- West Africa: 8482 mt

These are interesting results and show the clear effect of swell waves. In West Africa more swell waves occur, which reduces the workability and thus decreases the potential profit of the semi-submersibles. This causes the significant increase of the turning point.

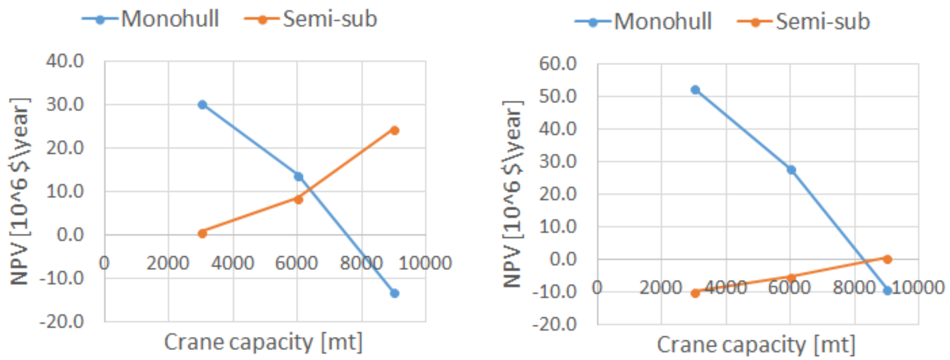

Figure 10.3: NPV calculations for the North Sea(left) and West Africa(right) environment for both vessel types.

For a final clear overview the combined cost calculations with the workability and turning point can be seen in tables 10.3 and 10.4 for the North Sea and West Africa environment, respectively.

Table 10.3: Total costs per year, workability and turning points for the North Sea environment.

	Monohull			Semi-submersible			Unit
Crane capacity	3000	6000	9000	3000	6000	9000	[mt]
CAPEX/year	3.7	6.3	8.9	5.5	7.1	8.8	[$10^6$$]
OPEX/year	27.1	29.1	31.3	29.5	30.9	32.5	[$10^6$$]
Workability	0.37	0.44	0.49	0.70	0.72	0.79	[-]
Income	34.6	40.0	45.3	38.1	42.8	48.2	[$10^6$$]
Turning point	6095						[mt]

Table 10.4: Total costs per year, workability and turning points for the West Africa environment.

	Monohull			Semi-submersible			Unit
Crane capacity	3000	6000	9000	3000	6000	9000	[mt]
CAPEX/year	3.7	6.3	8.9	5.5	7.1	8.8	[$10^6$$]
OPEX/year	27.4	29.5	31.8	28.9	30.2	31.7	[$10^6$$]
Workability	0.47	0.51	0.51	0.60	0.60	0.59	[-]
Income	36.1	41.0	45.6	37.4	41.9	46.6	[$10^6$$]
Turning point	8482						[mt]

11

Sensitivity analysis

In this chapter the influence of certain deviations is investigated. This is important because certain deviations can have a large effect on the results. It is thus critical to realise which deviations this might be, and if the assumptions made are accurate enough. For this analysis it should become clear whether the obtained results have a decent accuracy, or that additional research is necessary to verify and validate the results.

It is important to note that not all sensitivity analysis is discussed here. There are many calculations performed that are individually subjected to manual testing. An example is described below:
Since the resistance is determined using empirical relations, it depends on many different input parameters. There is looked at the influence of parameters on the resistance. In some cases it was found that a parameter value, e.g. a certain curve radius, seemed out of range of applicability and showed some odd results, which could either significantly decrease or increase the resistance. These peaks are avoided in order to get a realistic resistance for both vessels.

The following sections will discuss some aspects which are found to be the most important or those that affect the results the most.

11.1. Workability criteria

Although the workability criteria consist of 3 separate ones, the vertical displacement showed to be the most important one. This criteria is often limiting the workability the most. There is therefore looked what happens if the vertical displacement is significantly lower or higher. By setting it higher, the roll and pitch criteria start to play a more important role, which could affect the results significantly.

The significant amplitude of the vertical motion is changed from the original 0.45

to 0.35 and 0.55 meter. The results are shown in table 11.1. It can be seen that the change in criteria seems to have the most impact on the monohull designs, which makes sense as a lot of waves are near the criteria limits for these vessels. It therefore also makes sense that the turning points are significantly different. The vertical displacement amplitude is thus a sensitive criteria which should be selected with utmost care.

Table 11.1: Workability rates & resulting turning points for different vertical displacement amplitudes (North-sea). The middle column is the original criteria.

	$Z_a = 0.35$	$Z_a = 0.45$	$Z_a = 0.55$
Mono Q=3000	0.31	0.37	0.42
Mono Q=6000	0.38	0.44	0.49
Mono Q=9000	0.44	0.50	0.55
Semi Q=3000	0.74	0.79	0.83
Semi Q=6000	0.80	0.84	0.87
Semi Q=9000	0.83	0.87	0.89
Turning point [mt]	5583	6095	6502

The same analysis can be done for the West Africa calculations. It is expected that the change in turning point is less, as more waves are also located around the criteria limits of the semi-sub. Since the difference in workability is less between the two vessel types, the resulting deviation in turning point should also be less. This is confirmed by a quick rough calculation, which is not shown in this report.

11.2. Resistance

Another interesting deviation could be the resistance. In the current designs there is a large difference, where the monohull has approximately only half the resistance of the semi-sub. This makes sense, due to the relatively low transit speed and well shaped single hull. It is important to check what effect the difference in resistance could mean for the turning point.

The semi-sub resistance is reduced by 20% to account for possible errors in the resistance calculations and design alterations that improves the flow around the hulls. At this point the resistance is approximately equal to the resistance as if the semi-sub floaters are shaped as the monohull. The turning point then drops from 6095 to 5780 mt. This makes sense as a reduction in the operational costs for the semi-sub would make the semi-sub more favourable at a lower crane capacity.

A similar deviation is to add a bulb to the monohull, reducing the monohull resistance by maximal 10%[34]. This increases the resistance difference between the two vessel types, and the turning point increases to 6174 mt. This is thus only a small difference, most likely because the monohull resistance is already so low compared to the semi-sub.

It must be noted that such a resistance reduction is possible by design, but is also likely to increase the building cost. More complex curves increases the amount of man-hours, which is not taken into account in the above sensitivity analysis.

11.3. DP power

The same can be done for the power consumption of the dynamic positioning phase. The thruster power consumption for all vessels is increased by 20% which showed a minor difference in the turning point, ranging from 6080 to 6111 mt. This shows that if the DP-criteria are set otherwise (for instance 5 Bft instead of 6), the effect on the results would probably be low.

In addition, in order to investigate the differences between the vessel types, there is looked at what happens if the power consumption of the semi-subs is increased while the monohull is kept the same. By increasing the semi-sub DP power by 10%, the turning point increases to 6229 mt. This is a relatively low impact as a 10% deviation is quite large, since a lot of effort has been put into modelling and calculating this aspect in detail.

11.4. Operational profile

The operational profile consists of multiple phases, as explained in chapter 2. It is interesting to look at the effect on the turning point if the division of these phases is changed. Each phase duration is extended by 20% and the turning point is calculated. The results can be seen in table 11.2.

Table 11.2: Operational profile phases extended by 20% one at a time, with resulting turning point for the North-sea environment.

	Original	Port +20%	Sailing +20%	DP +20%	Standby +20%
Port/Yard	44%	48%	41%	42%	43%
Sailing	36%	33%	40%	35%	35%
DP	15%	14%	14%	18%	15%
Standby	5%	5%	5%	5%	7%
Turning point [mt]	6095	6300	6499	5414	6160

The port and standby phase do not have a large effect on the turning point. The important and interesting phases are during transit and DP. For transit it makes sense that the turning point increases when this phase is longer, since the semi-sub has a higher resistance. The difference in DP is largely because an extended DP-phase has a large effect on the income. This is due to the high difference in workability. This thus means that if the DP-phase is longer than initially assumed, the semi-sub is earlier the more favourable vessel type.

11.5. Operational profile workability

The workability rate for the initial operational profile was assumed to be 50%. This is a reasonable assumption, but also induces an uncertainty. If the workability for this operational profile is actually lower, the difference in income will become even larger between the two vessel types. This will make the semi-subs more favourable and the turning point will increase.

At a workability rate of 40% the turning point is 6580 mt. At 60% this reduces to 5665 mt. These are substantial differences, but it is unlikely that either of these extremes are true, since the vessels in the operational profile are mostly monohulls and are operating world-wide. The workability would therefore lie somewhere between the two investigated sea spectra, which are 0.44 and 0.52 for the 6000 mt monohulls, indicating that a workability of 0.50 is a reasonable assumption.

11.6. Discount rate

Deviating from the assumed discount rate (5%) in chapter 10, will cause a different balance between the capital and operational costs. It is therefore also interesting to know how the turning points differ for other discount rates. In figure 11.1 this relation is shown. It can be seen that for a 0% discount rate, the turning points are significantly different, with 5000 for the North Sea and 8800 for West Africa. These are thus the turning points if investment interests is not taken into account.

Another interesting note is an increasing slope for the North Sea, but decreasing for West Africa. Apparently due to the different workability rates, the difference in income causes a shift for the two regions. Although such a large deviation from the assumed 5% is not likely, it is interesting to see what the lower and upper turning points could be. It is not expected that the real discount rates change more than 2%.

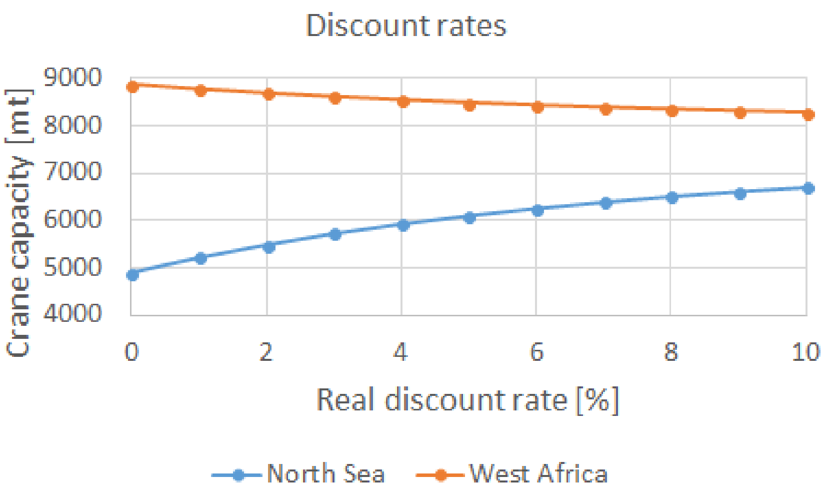

Figure 11.1: Relation between the discount rates and turning points for both sea spectra.

11.7. Others

There are some other variations that are investigated, but turned out to have a relatively small effect on the turning points.

11.7.1. Profit margin

The profit margin, which has been set on 10% initially, is varied to investigate its sensitivity. It turned out that it has almost no effect on the turning point. E.g. when the profit margin is set at 20%, the turning point would not shift further than 100 mt.

11.7.2. Lifetime

Varying the lifetime between 20 and 30 years has a limited effect on the turning point. Similar to the income margin, the turning points would not shift further than 100 mt.

11.8. Uncertainty

The calculated turning points have an uncertainty, which became especially clear in the above sections, where in some cases a small difference can cause a significant change in the outcome. It is, however, hard to say how large this uncertainty is.

Not only do the calculations depend on so many variables and aspects, most data and calculations are not validated.

The largest uncertainty is in the cost calculations. Due to the severe lack of validation data it was impossible to check the calculation methods with actual real numbers. Compared to this aspect the other calculations have a relatively low uncertainty, which had well established methods and enough data was available to check if results are within reasonable bounds. In addition, the performance aspects could be checked and discussed within Vuyk.

After the sensitivity analysis it has become clear that a certain uncertainty exists, but it is less than initially expected. No methods have been used to establish an error, because this is impossible with the available resources (mostly for the cost calculations). There can, however, be made an estimate by combining all research done so far. It is expected that the uncertainty is about 500 mt, which means that the turning point for the two vessel types lie somewhere between 5595 and 6595 mt for the North Sea and 7982 and 8982 mt for the coast of West Africa. If it is assumed that these sea spectra are the extremes, it can be said that for any location on earth the turning point lies between 5595 and 8982 mt.

12
Conclusion & Recommendations

The research questions as defined in chapter 2 have been answered in the previous chapters and some general conclusions and recommendations can be drawn. First some general conclusions about the comparison and their results are discussed, after which some individual aspect conclusions are discussed that further zooms in on the performance aspects. Finally, some recommendations are given in order to improve the accuracy and validity of the work.

12.1. Conclusions

The two vessel types, monohull and semi-sub, are compared for two environmental conditions. First, the North Sea, which consists of mainly short and high waves. The coast of West Africa is known for its many swell waves and proved to get much different results.

For both environmental conditions it can be said that the monohull is ideal for lower crane capacities. This is mainly due to its low capital cost, as the vessel is relatively cheap to build, and low operational expenditures. For the heavier lifts the semi-sub starts to pay off, when the workability starts to outweigh the increased capital and operational expenditures. The turning points at which this happens is different for each sea spectra.

North Sea

Due to the short and high waves, the North Sea spectra proves to be a challenge for the monohull. The monohull workability is low (approximately between 37 and

49%) which results in a relatively low turning point (6095 mt). Because the semi-sub has much lower natural frequencies for their motions, it is capable of operating in many conditions on the North Sea (workability of approximately 70 to 79%).

West Africa

In this sea spectra the swell waves caused a significant difference between the two vessels. The workability of the monohull (47 to 53%) increased, while for the semi-sub it reduced (67 to 72%). This is due to the natural frequencies of the vessels. Since there are now more low frequency waves, which affect the semi-sub relatively more, the workability of the semi-sub is reduced. In addition, since more waves are now further away from the monohull natural frequencies, the workability increased. Due to this difference, the turning point increases to 8482 mt.

Uncertainty

As described in detail in chapter 11, the uncertainty is estimated. It is expected that the turning point is likely between 5595 and 6595 mt for the North Sea and 7982 and 8982 mt for the coast of West Africa. This is purely done with an educated guess and knowledge of all the calculations and methods used.

12.1.1. Resistance & propulsion

The determined resistance for both vessels show some interesting differences. The semi-sub has approximately double the resistance compared to the monohull, as it has a large wetted surface, the floaters have a high block coefficient and the sailing speed is relatively low. For the semi-sub a hull-hull interaction is taken into account, which causes an approximate 17% increase in frictional resistance. This is due to the tunnel-effect, which increases the flow velocity between the two hulls.

For both vessels the propulsion system is kept the same. This means the efficiency to overcome the resistance is equal. This is done because the differences in this aspect are small and will barely impact the comparison. The main difference between the vessel types is thus the resistance and not their propulsion system.

12.1.2. Stability

Both vessels are designed to meet the stability requirements. In general it is found that it is easier to meet the criteria for the semi-sub, because it initially already has a lot of free space that can be used for ballasting and to get an optimal weight distribution. The semi-subs also have a high initial GM upright.

The required ballast is optimal for every semi-sub. Especially for the larger vessels the semi-sub proves to be optimal, as it can store a large amount of ballast in its columns which are far away from the centre of gravity. The monohull is limited to smaller tanks and it requires to fill tanks closer to the centre of gravity relatively quickly.

12.1.3. Motions

The vessel motions are highly related to the workability rates. As has been said before, the semi-sub has much higher natural periods, which causes high workability rates.

For the semi-sub an interesting behaviour is found, where additional (smaller) peaks are found in the vessel RAOs. This is most likely due to the multiple columns and their individual resonance with the environmental waves.

12.1.4. Dynamic positioning

Due to the large surface areas of the semi-sub, the required power to maintain position is usually higher compared to the monohull. However, in more extreme weather conditions, the wave drift forces of the monohull become higher and the required power also increases for the monohull (relative to the semi-sub). For the criteria set, the monohull requires less power on average to maintain position.

12.2. Recommendations

In order to improve the accuracy of the results in this thesis, multiple recommendations can be given. This is also listed per chapter, which gives a clear overview.

Concept designs:

- General arrangement: The arrangement of weight groups is not done in much detail. It would be interesting and recommended to see what a more detailed (and more optimized) general arrangement could mean for the change in turning point and performance.

Resistance & propulsion:

- Vessel shape: For both vessel types a commonly used shape is chosen. It is, however, very well possible that for either (or both) vessels a more optimal shape should be selected. For the semi-sub the floaters can be more streamlined so that the resistance is reduced.
- Resistance prediction: Since empirical relations are used, which for the monohull also fell outside its applicable range, the uncertainty for the resistance prediction is large. Although a correction is applied using validation data, both vessels require additional research so that the resistance is predicted better. It is advised to use more real test data of crane vessels to validate the calculation methods.
- Propulsion system: For both vessels the same propulsion system is used. This seems like a reasonable assumption, but it is advised to investigate what possible differences this could cause between the vessels. It is, however, expected to have a low impact on the results.

Stability:

- General arrangement: For both vessels a standard layout is assumed and the weight blocks are distributed. However, some vessels could have special modifications that significantly improve their stability and performance. For example, the monohull could be fitted with a heightened fore castle, which makes it possible to fit extra ballast tanks. This makes ballasting easier and it might alter the design in such a way that the costs are lower.

Motions:

- Criteria: In the current thesis work 3 criteria are assumed, which are well documented in literature. It is, however, advisable to establish more elaborate criteria, which also take into account velocities and accelerations. This is shortly looked at during the thesis work, and seemed to have limited effect. But it is recommended to further investigate and verify that these 3 criteria are enough.

Dynamic positioning:

- Other DP systems: Besides the DP-3 system, it could be interesting to compare the vessel types for DP-2. This makes the single-line system more important and alternative innovative solutions could be implemented to reduce the loss of thrust in case of a failure. This would give a large difference in required installed power and could significantly affect the results.
- Shielding effect: This effect is taken into account roughly. It is however recommended to include this effect in more detail. This could, for instance, be done by a CFD analysis.
- Thrusters: it is advised to look further into the amount of thrusters and their type. For example, by replacing some retractable thrusters in the semi-sub by tunnel thrusters could reduce the cost that alters the turning point. Another example could be the use of 4 main fixed azimuth thrusters (instead of 2) for both vessels at the aft of the vessels.

Costs:

- Applicability: For quite some cost calculations, it is questionable if the empirical relation can be applied accurately for the vessel types. Since in many aspects the vessel types are so different, it might be required to establish more accurate and individual cost estimations for both vessels. In general more effort should be put in to make the cost estimations more accurate.

Other:

- Additional aspects: In this thesis the most important aspects are taken into account that give a difference between the two vessel types. It is however advisable to look at additional aspects and put effort into possible additional differences. Example: What if the semi-sub requires more crew due to its complexity? Then there will be a significant difference in crew wage costs which will affect the turning points.

- Design flexibility: One vessel type can be adjusted more easily than another. If a client has a special request, one vessel might be more suitable for adjustments than the other. e.g. What if more deck area is required? This is probably easier to adjust on the semi-sub than the monohull. Some extra thought should be given to their flexibility.
- Lifting direction: It is recommended to further investigate the (dis)advantages of lifting an object in other angles besides over the stern. A lift over the side has proved to have a higher workability for certain headings. Could this be taken into account and for which vessel is this the most advantageous?
- Crane position: In addition to the previous, it could prove valuable to investigate other crane positions. A crane positioned at the side of the vessel together with a higher workability at certain headings could significantly alter the turning point.
- Structural strength: In this thesis nothing has been done related to structural strength. It is, however, advisable to include this in the research. It might be that additional material is required to make the vessel feasible, due to the heavy crane loads, ballast tanks and weight distribution over the vessel. This could affect the steel-weight and the vessel cost.
- Renewable market: It would be recommended to check what the effect could be when the vessels have to be designed for the renewable market. Not only do alternative fuels influence the operational costs, it could also significantly affect the designs and increase the capital costs.

A

Heavy lift vessels

In this chapter the different types of heavy lift vessels are explained. Furthermore, a selection is made to compare the ship types.

The vessel types

In order to lift objects at sea, a heavy lift vessel can be used. There are many types of vessels that are capable of lifting heavy objects. In this section all available types are shown and their basic principles explained. A selection of the vessel types is also made.

Monohull vessels

There are multiple monohull vessel types that can be defined:

1. Monohull heavy lift crane vessels: These are the most common heavy lift vessels, and is the type that most people think of first. This type can be divided into numerous subtypes.
    - Heavy cargo vessels: These focus on the transportation of heavy cargo from one location to another. This type of vessel is often equipped with multiple cranes and has a high transit speed. A sample vessel can be seen in figure A.1.
    - Offshore installation vessels: The focus for this type lies on the ability of installing (large) offshore structures/equipment. The transit speed is often of less importance, but these ships are usually equipped with a single high capacity crane. A sample vessel can be seen in figure A.2.
    - Pipe-lay vessels: A special feature of pipe-lay vessels is its additional

A

structure at the ships aft to efficiently place pipes on the seabed. These vessels focus on the subsea infrastructure. It is therefore also equipped with a heavy lift crane. A sample vessel can be seen in figure A.3.

- Multi functional vessels: It is rare that a ship is dedicated to one of the above types. Instead, they are often designed to be able to carry out a range of operations. This makes them more flexible and are thus able to accept more contracts.

Figure A.1: The SAL type 176 vessel, fitted with 2 cranes of 700 mton capacity.

Figure A.2: The Oleg Strashnov vessel, fitted with a single 5000 mton capacity crane.

2. Jack-up vessels: A special type of monohull is the jack-up vessel. This type is similar to the previous monohull vessels, with one major difference. It is equipped with 4 or more legs that can be lowered to lift the vessel out of the water. This gives a huge advantage; the ship hull is no longer in contact with the sea surface and is therefore not disturbed by waves. Only the legs are affected by the waves, which is minimal. These ships are often used to install offshore wind turbines. It is, however, limited to a certain water depth and brings its own unique challenges. An example can be seen in figure A.4.

3. Semi-submersibles: Another unique concept is the semi-submersible mono-

Figure A.3: The Global 1200 vessel, capable of heavy lift and pipelay operations. The crane has a 1200 mton capacity.

Figure A.4: A jack-up vessel, capable of lifting itself above the water surface.

hull. This type has a large deck area which can be lowered below the sea surface by ballasting the ship. Once the cargo is floating above the ship, the ship is resurfaced and the cargo is resting on its deck area. The cargo can now be transported to another location. An example is the Black marlin, as can be seen in figure A.5.

Multihull vessels

There are also two vessel types that can be defined for the multihull:

1. Multihull heavy lift vessels: This type is less common. This type is build from two monohulls with a large crane structure in between. As the cargo is lifted ”in the middle” of the vessel, no ballasting is required to counteract

A

Figure A.5: The Black marlin, a monohull semi-submersible ship, capable of lifting cargo that floats over its deck.

heel angles. It is however limited, as (for example) it cannot have a revolving crane. It can be seen in figure A.6

2. Multihull semi-submersible vessels: These vessels are floating on two large pontoons in transit mode. Once arrived at its destination, the vessel is lowered to get into its operating mode. The pontoons are then fully submerged and the vessel only has its large columns that go through the water surface. This reduces the effect of waves and increases its operability. Two of these ships can be seen in figure A.7.

Vessel selection criteria

Criteria have to be set so that the list of reference ships is relevant to the vessels that will be compared. Some criteria have already been mentioned in previous sections, but will be listed here for a clear overview:

- Self propelled only: There will only be looked at self-propelled vessels.
- Minimum crane capacity of 500 metric tonnes: In order to have a concise overview of heavy lift vessels, the vessel's crane (combination) needs to be able to lift a minimum of 500 tonnes.
- Revolving crane: In order to carry out a wide range of offshore operations, the crane has to be of the revolving type. This means the crane can turn relative to the vessel, so that cargo can be lifted from several different angles and can be carried onto the deck.
- No jack-up vessels: These vessels are too specific compared to the multihull,

Figure A.6: A multihull heavy lift vessel, consisting of two hulls connected by a crane structure in between.

and will not be looked at. The jack-up is mostly used for installing fixed offshore wind turbines and behaves more like a structure than a vessel in operational mode.

- No semi-submersible monohulls: These vessels do not use cranes and can thus only transport objects that can float and cannot perform offshore installations.
- No multihull fixed crane vessels: Although the multihull heavy lift vessel, with a crane structure between two hulls, is an interesting concept, it will not be looked at. It has many limitations, such as no revolving crane possibilities and restricted operational flexibility.

To summarize, in this thesis there will only be looked at self propelled, >500 ton revolving crane capacity, (multifunctional) offshore installation vessels. An example of these vessels can be seen in figure A.2 and A.7.

A

Figure A.7: Two multihull semi-submersible ships carrying out an operation.

B

Heavy lift operations

Offshore market

The offshore industry primarily consists of two main groups; The oil-gas industry and the wind energy industry. The ever growing demand for more energy expands the offshore sector and demands innovative solutions for new problems.

In the past years the offshore wind industry has grown significantly, and it is expected it will continue to grow. Not only more wind turbines are installed, but the size and power per wind turbine is increased. This brings complex problems, as the larger and heavier wind turbines can be transported less efficiently and possibly not installed as a whole. It is currently the question whether jack-up vessels will become even larger or if other vessel types will install these new wind turbines.

The oil-gas industry has been around for quite some time. Due to the recent massive drop in oil prices the exploration, construction, production and support services have plummeted. The main recent interest in this industry is the decommissioning of old projects. This field of interest is emerging and is expected to grow significantly over the next decades [32]. These offshore objects, which could for example be platforms, wells or pipelines, have to be disassembled, either offshore or onshore. In this field a wide range of designs could be feasible that target these kind of operations.

An important note is the difference in cost for onshore and offshore operations. A wind turbine that can be assembled onshore and installed in a single piece by a heavy lift vessel is in general more cost efficient than a vessel that installs a wind turbine piece by piece. This is due to the complexity and higher operational costs for offshore operations. The same holds for the decommissioning market, where lifting a larger object and disassembling it onshore is more cost efficient. However, this variation in cost also depends on many other aspects. A larger vessel does mean a higher operational, depreciation and building cost, making it less cost efficient for smaller lifts. It is thus key to design an optimal vessel for a part of the market for which the availability of contracts results in the highest profit.

There are, of course, numerous more fields of interest. For example the transportation of smaller vessels, constructions for research purposes and many more. This thesis will focus on the larger industries, thus the renewable market for offshore wind energy and decommissioning projects as it shows the best promise for the future.

Operations

Heavy lift vessels can be used for a wide range of operations. Some examples are:

- Offshore wind-turbine installation: Wind turbines become more popular every day. The demand for renewable energy is high. Wind turbines grow larger for more power output and are thus also heavier. The market is clearly growing and more vessels are required that can install such wind turbines. Jack-up vessels are often used to install wind turbines, but heavy lift vessels can be used as well. It is questionable if the jack-up vessels will grow even larger, or that they will reach their limits for these heavy lifts (in term of effectiveness compared to other heavy lift vessels). The wind-turbines can be installed in parts (such as the monopile and the blades), but can also be installed as a whole. Assembling a wind turbine onshore is more cost efficient than performing it on sea. The installation of an assembled wind turbine can easily weigh up to 3000mt.
- Jacket installation: There are many types and sizes of jacket installations. Jackets are often steel structures that are attached to the seabed. On top of a jacket, a platform or object can be placed so that it is raised above the sea surface. Jackets are usually between a few hundred ton and 10,000mt.
- Topside installation: The topside structure is the platform or object that is placed on a jacket. This structure is usually heavier than the jacket and thus requires a crane with higher capacity. Most topsides are several thousand tonnes and there have been structures lifted that almost used the full capacity of the biggest heavy lift crane up to date; the Thialf (14,200mt). Because these objects are usually large in size it is often performed by a tandem lift which has several advantages. First, because the object is located between the two cranes, it is possible to lift taller (and wider) objects. The maximum crane capacity is significantly increased without the engineering effort to design a single crane that can reach the same capacity. Also, one crane can be used to perform lighter lifts without having to consume a lot of power if the vessel would be designed with a single crane.
- Other installations: There are, of course, many other structures or objects that have to be installed. Another example could be the transporting and

installation of a new crane for an offshore platform.

- Decommissioning: There are many platforms at sea that have to be removed. These are mainly old platforms that are not used any more and have to be disassembled. Instead of using several ships to disassemble the structure off-shore, a heavy lift vessel can be used to lift and transport the structure so that the time consuming disassembling can be performed onshore. This is very cost efficient in most cases. The range of crane capacities required for these type of operations vary a lot, as each platform or structure is different in weight. However, this sector often includes lift of at least 5000 mt.

C

Operational profile aspects

Function

There are many different offshore operations that can be thought of. Examples are the installation of monopiles, jackets, topsides or even complete wind turbines. Pipe-lay operations could also be a possible function of both vessel types. An elaborate description of different offshore installation functions can be found in appendix B. It is chosen to design all vessels for a single main function. Each vessel will lift an offshore object with a weight equal to its crane capacity.

Vessel speed

The vessel will spend a large portion of its operational life in transit mode. In this mode, the vessel sails to a destination. This could be an onshore location to pick up cargo, or offshore to install a structure. There are multiple choices to be discussed:

- First, the vessel speed can be selected based on the design. The vessel speed will thus be different for every concept. The advantage is that the speed is appropriate for that specific design and its capabilities. The disadvantage is that it makes the comparison for the ship types a lot more complex. After all, an increase in speed results in a higher fuel consumption. This relates in a higher fuel cost and thus OPEX. In exchange, the vessel can arrive at its destination earlier, but this raises the question; ”What is the effect of the transit duration difference on the OPEX?”. It could be, that for a certain mission, the client is more satisfied with the transit duration and is willing to pay more. This means that the profit is higher, even though the OPEX is higher. Because of these implications, it is chosen that this method falls outside the scope of the thesis.

- Another solution is to design for equal Froude numbers. This means the dimensionless vessel speed is equal. This could be done for the vessels compared, but also for all vessels together. This allows a good technical comparison between the ship types. However, as eventually there is looked at the total cost of a certain vessel, the same problem arises as in the previous method. It is therefore also chosen to not choose this strategy.
- The final solution is to pick the same real vessel speed for every design. This allows a comparison that is independent of contractual bonuses and specific OPEX related differences due to the vessel speed. The disadvantage is that the vessel speed can be unrealistic for certain designs. It is (for example), in general, more likely that a monohull has a higher transit speed than the semi-submersible. It is chosen to pick a constant speed that is the same for all designs, as it significantly decreases the complexity and still allows a clear comparison.

Operational phases duration

As described in chapter 2, an operational profile is obtained from Vuyk. Each operational phase, such as transit, DP and standby, has thus its own duration. This is assumed to be equal for each vessel.

Environment

The area that the vessel operates in is chosen to be the North-sea and the west coast of Africa. The North-sea is relatively difficult to operate in, due to its rough sea-states. However, the west coast of Africa (e.g. Angola), is relatively easy and has a different sea spectra. Because a large difference in motions and workability could be observed in these two sea-states, it is interesting to look at as the turning point could be significantly different.

Vessel lifetime

The average lifetime of a heavy lift vessel is set on 25 years. This is required to calculate the capital costs per time unit.

Cranes

The cranes are not directly part of the operational profile. It is, however, an important part of a heavy lift vessel and some decisions affect its operational profile. In this section multiple decisions involving the cranes are made. An extensive analysis and overview can be found in appendix F for more information. There are many types of cranes which could be used for offshore lift operations. There are, however, only two feasible types which are often used in the heavy lift industry; the tub crane (or A-frame crane) and mast crane. Both of these have their own unique (dis)advantages and according to the research of J. Kamp both types are, in general, equally feasible for heavy lift vessels [17]. However, there are far less mast cranes that are designed for the higher lift capacities. Due to this fact and the need for a wide range of crane capacities for the concept designs, the tub crane is chosen for all designs.

Another point of interest is the amount of cranes for each design. This is also explained in more detail in appendix F. Because a vessel with multiple cranes has a different operational profile (e.g, tandem lifts have different characteristics) it is chosen to use one crane for all designs.

Last but not least, there will only be looked at vessels that have a revolving crane. This means that sheerleg vessels are not considered. This is decided as revolving cranes can carry lifted cargo onto its deck, which is essential for many operations.

Crew and accommodation

Another important aspect of heavy lift vessels is the crew. Due to the complexity of all offshore heavy lift operations, the crew size is set the same for all designs. A difference in crew size would require an extensive analysis of the different work on-board a vessel and the requirements for each vessel related to its operation. This would widen the scope a lot, while it is expected that its impact on the results is rather limited. As the operations require a large crew, quite some accommodation units have to be on-board the vessel. The accommodation design can be quite complex as it depends on many factors. There are multiple different cabins, the kitchen, social areas, hallways and more. However, it is chosen to keep the accommodation unit as simple as possible and it is simply assumed to be a block on-board the vessel, because its influence on the result is limited. The determination of the accommodation block is explained in more detail in chapter 5.

D

Reference vessel matrix

D

Name	Lpp	Beam	Transit speed	Dead weight	Installed power	Crane cap	Accommodation cap	DP
[-]	[m]	[m]	[kts]	[mt]	[kW]	[mt]	[-]	[-]
GLOBAL INDUSTRIES								
GLOBAL 1200	151	32	15	9595	25984	1200	264	DP-2
J. RAY McDERMOTT, INC.								
DB 16	119	31	7		9200	780	184	
DB 26	119	32	6			816	293	
DB 50	142	46	6	16565	20800	4000	320	DP-2
DLV 2000	179	39	12		33000	2000	401	DP-3
OFFSHORE CONTRACTORS								
Jascon 18	148	37	12	11890	28640	1800	400	DP-3
Jascon 25	116	36	9	7867	7300	800	355	DP-2
Jascon 34	116	36	9	9378	7300	800	335	DP-3
CNOOC ENGINEERING								
Lan Jiang	154	48				3800		
Lan Jing	230	50	13	65473		7500		
DPV7500C / Hai yang shi you 201	182	39	12	25448	32640	4000	389	DP-3
SAIPEM GROUP								
Castoro Otto	183	35	8	25075	16800	2177	356	
Saipem S-3000	157	38	8		22000	2177	211	DP-3
Field development ship 2	171	32	13	6000	36000	1000	325	DP-3
SEAWAY HEAVY LIFTING ENGINEERING								
Stanislav Yudin	173	36	9	33036	24149	2500	151	
Oleg Strashnov	172	47	14	47000	28200	5000	395	DP-3
BOSKALIS								
Bokalift	212	43	14	47500	26070	3000	150	DP-2
SAPURA ACERGY								
Sapura 3000	144	38	8	17650	24000	2722	330	DP-2
LTS3000	156	38	7	15310	8800	2722	289	
MSH SHIP MANAGEMENT								
Oceanic 5000	183	48	12	39817		4400	398	DP-2
CRS COMPANY								
Hua Tian Long	168	48	5	40792	13550	4000	300	
SUBSEA 7								
Borealis	169	46	15	47000	34800	5000	400	DP-3
DEME								
Orion	204	49	14	36600	44180	3000	131	
HEEREMA								
Aegir	198	46	12	41000	49731	4000	305	DP-3
GRUPO R								
Oceanic II	165	45	11		33210	2268	504	DP-3
TOLTECA	190	36	11	25945		1814	438	

Figure D.1: Monohull reference vessels with a selection of parameters.

Name	Lpp	Beam	Transit speed	Dead weight	Installed power	Crane cap	Accommodation cap	DP
[-]	[m]	[m]	[kts]	[mt]	[kW]	[mt]	[-]	
J. RAY McDERMOTT, INC.								
DB 101	143	52	10	93400		3175	275	
SAIPEM GROUP								
Saipem 7000	175	87	9		70000	14000	725	DP-3
OOS INTERNATIONAL								
Gretha	136	81	8	22100	30880	3600	618	DP-3
Prometheus	118	70	10	29246	8600	1100	500	
Serooskerke	126	81	11		30400	4400	750	DP-3
HEEREMA MARINE CONTRACTORS								
Thialf	165	88	6	129221	58400	14200	736	DP-3
Hermod	118	86	6	59344	19355	8100	371	
Balder	118	86	7	59404	40590	6300	392	DP-3
Sleipnir	180	102	10		96000	20000	400	DP-3

Figure D.2: Semi-submersible reference vessels with a selection of parameters.

E

Detailed concept parameter selection

In this chapter the concept design process is described in more detail. The design for the monohull and semi-submersible are different, as both have significant different characteristics. As a result, both designs are described separately. First some general remarks are explained so that the design process is easier to follow. Then the monohull and semi-submersible designs are explained.

General remarks

There are a few decisions, assumptions and constraints that are the same for both the monohull and semi-submersible designs.

Design requirements

Several vessel requirements can be defined, which are listed below. These are the most important requirements.

- Crane capacity: As has been stated before, the crane capacity for the three vessel designs is 3000, 6000 and 9000 mt.
- Dead weight (DWT): A higher dead weight means the ship is able to carry more (cargo)weight. As DWT is the general term for added weight such as the fuel, crew, cargo and ballast, it is often much larger then what the vessel is capable of lifting and can carry on its deck. This is also because enough DWT has to be available for ballasting the ship in lifting mode. As will become clear during the semi-submersible design, not enough parameter relations are available for a first design. It is therefore chosen to set the DWT equal

to that of the monohull designs. This is mainly done because the DWT is an important requirement and partly determines the market/operational value of a vessel.

Block coefficient

In many steps of the design use is made of the general block coefficient equation, which is shown below in equation E.1.

$$C_B = \frac{\nabla}{L \cdot B \cdot T} \tag{E.1}$$

Graph layout

In this section, every graph is made with the same layout for a clear overview. A few notes:

- Reference ships are always indicated by blue circles.
- The concept designs are, if applicable, indicated by orange squares.
- Trend-lines are added to estimate a relation between two parameters. In most cases use is made of a linear function, which showed the best correlation for all points. In some cases the claim that the relation is indeed linear cannot be supported, but it gives a good indication to get a first estimate for that parameter.
- In the graphs an R^2 value is also indicated which is the coefficient of determination. It is the proportion of the variance in the dependent variable that is predictable from the independent variable[3]. In more simple terms it shows how good the trend-line fits the data points, with R^2=1 meaning a perfect fit and R^2=0 the worst fit.

Monohull parameter selection

In this section the monohull design process is explained. The section is explained in chronological order, starting with the determination of the vessel width by looking at the crane capacity.

Width

The first parameter that is determined is the width of the vessel, as it is the most decisive parameter for heavy lift monohull vessels for a certain crane capacity. It makes sense to linearly relate the crane capacity (P) with the width to the third power (B^3) as there is a linear relation between the inertia of the vessel and its stability. The relation can be seen in figure E.1.

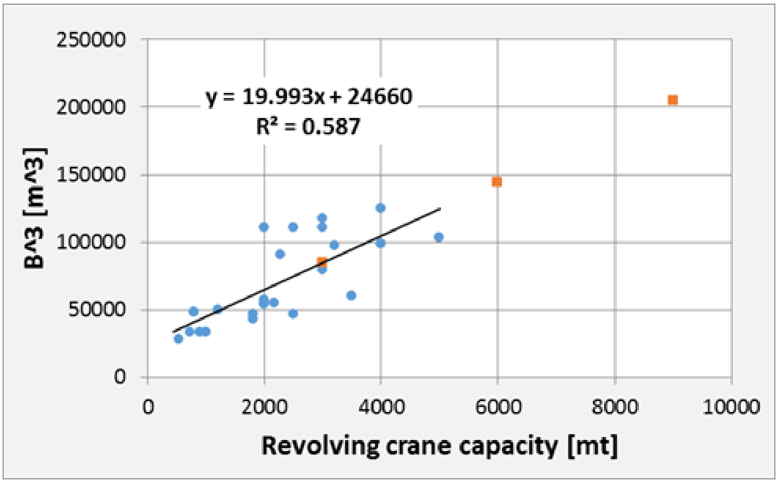

Figure E.1: Monohull: relation between the revolving crane capacity (P) and width (B^3) of the vessel.

Deadweight Tonnage

A clear relation is found between the (total) crane capacity and the deadweight tonnage. This makes sense, as vessels that can lift heavier objects, also often need to be able to carry more in weight. The relation can be seen in figure E.2.

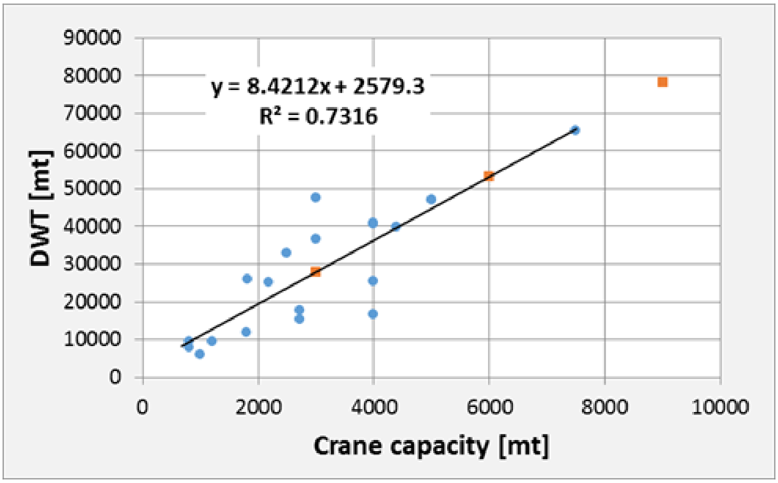

Figure E.2: Monohull: relation between the crane capacity (Q) and Deadweight tonnage (DWT) of the vessel.

Operational displacement

The relation between the operational displacement and deadweight tonnage can be seen in figure E.3. This relation is obvious, as the ship has a larger displacement when it can carry more weight.

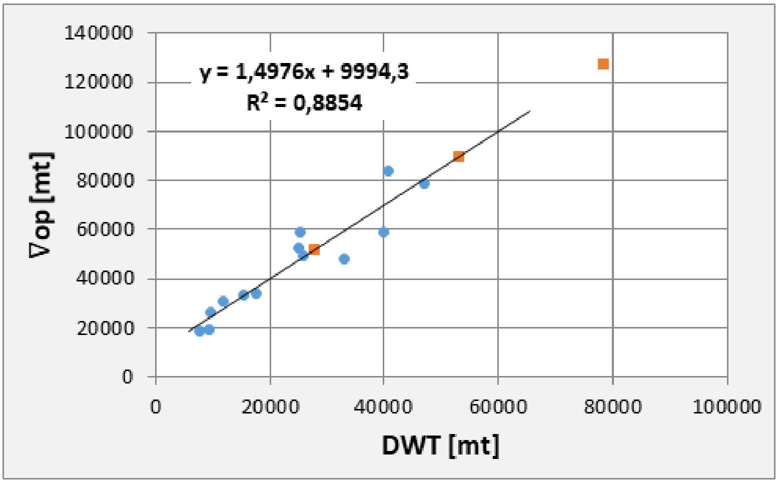

Figure E.3: Monohull: relation between the operational displacement (∇_{op}) and deadweight tonnage (DWT).

Length

The length (between perpendiculars) is determined by using the relation between L · B and the deadweight tonnage. As the width and DWT are known, the length (L_{PP}) can be determined. The relation can be seen in figure E.4.

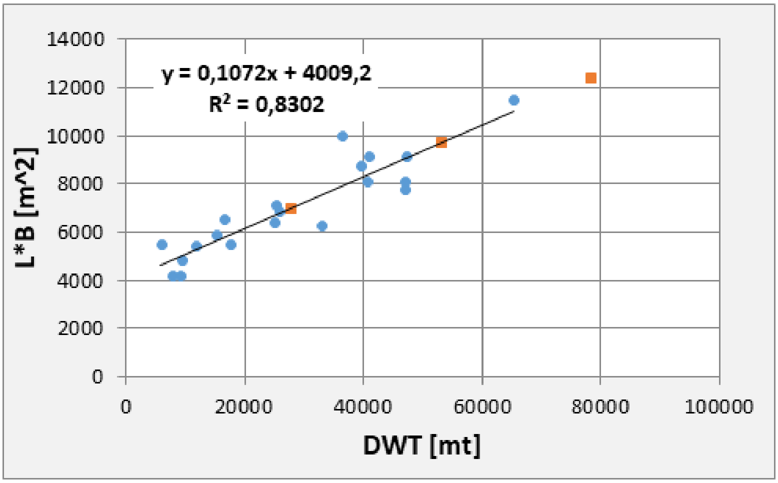

Figure E.4: Monohull: relation between the length times width ($L_{PP} \cdot B$) and deadweight tonnage (DWT).

Another parameter that can be determined is the overall length (L_{OA}). This parameter is not important for this thesis work, but as this parameter is almost always mentioned with a design, it is useful for comparison. This parameter is also used in the DP-tool in chapter 8. It is determined by taking the average L_{OA}/L_{PP} ratio of all monohull vessels.

Form coefficients

Block coefficient

The block coefficient showed no clear relation with other parameters, other than it seems to be constant. It is therefore chosen to take the average to determine this parameter. A close look at the known block coefficients, show that the block coefficient is often much higher for a barge hull type compared to a ship type. It is therefore chosen to only take the average of ship hull type vessels.

Midship coefficient

The midship coefficient is not determined, and is simply chosen as it is often close to 1 for this type of vessels. A value of 0.98 is selected. This value will prove to be an accurate estimation when the designs are finished.

Prismatic coefficient

Using the block- and midship coefficient, the prismatic coefficient can be calculated using equation E.2.

$$C_P = \frac{C_B}{C_M} \tag{E.2}$$

Draft

The operational draft can be determined now that the operational displacement, width, length and block coefficient are known. Equation E.1 is used to calculate the operational draft.

The draft in transit mode can be estimated, as data is available for both the operational and transit draft. The average ratio between these two draft values is taken to estimate the transit draft.

Transit displacement

An estimation of the displacement in transit mode can be made by making an assumption for the block coefficient. It is most likely that the block coefficient reduces at decreasing draft, but it is unknown by how much. It is assumed, as a first indication, that the block coefficient is the same for the operational and transit mode. Equation E.1 can then be used to calculate the transit displacement.

Depth

The depth of the vessel is the distance between the keel and main deck. For this parameter, the relation between the operational draft and depth is taken. The relation can be seen in figure E.5

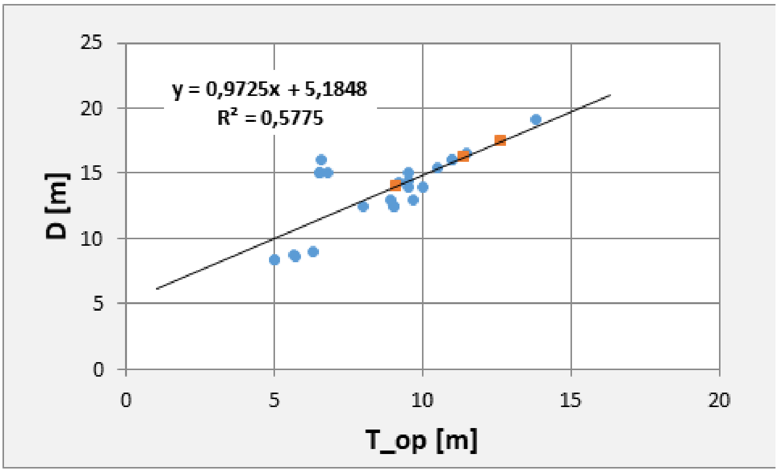

Figure E.5: Monohull: relation between the operational draft (T_{op}) and depth (D).

Semi-submersible parameter selection

The design process for the semi-submersible is different, as the ship type is significantly different. Compared to the monohull design, the width of the ship is less important and a direct derivation for this parameter shows that no clear relation can be made. It is therefore chosen to take a good look at how these vessels operate in lifting mode.

Area moment of inertia

The first clear relation is found for the area moment of inertia. The waterplane area in operational mode is determined using sketches, photos and available data. A few assumptions are made:

- The width of the floaters is assumed to be equal on both sides.
- The width of the struts is assumed to be equal to the width of the floaters.
- In general, asymmetry around the y-axis is neglected. Only for the Thialf vessel which has a significantly larger strut below the crane locations, aft of the vessel, the asymmetry is taken into account.
- Supporting structures between the struts are neglected, except for the OOS Prometheus which has two large circular pillars in between the struts (with a diameter of about 6 meter).

From the known dimensions and areas, the moment of inertia can be determined. In general equations E.3 and E.4 are used. In these equations the formula is applied for each area that goes through the water surface.

$$I_{xx} = \frac{B^3 \cdot L}{12} + A_{wp} \cdot dx^2 \tag{E.3}$$

$$I_{yy} = \frac{B \cdot L^3}{12} + A_{wp} \cdot dy^2 \tag{E.4}$$

Some vessels lift in a different way compared to others. For example, the Thialf has two cranes located aft of the vessel. In order to use its full crane capacity of 14200 ton, objects are lifted aft of the vessel. The OOS Gretha has its two cranes located at the side of the vessel and thus uses its full crane capacity at the side of the vessel. The area moment of inertia that is critical is determined. This is defined as the axis over which a lift has the least stability. For the vessels that lift aft of the vessel, I_{yy} is critical. For vessels such as the OOS Gretha, which lift from the side, I_{xx} is critical. For two vessels, the DB101 and OOS Prometheus, only one crane is used which makes them unique in the sense that the crane can be used in its revolving mode at full capacity. However, it is assumed that the DB101 lifts aft of the vessel while the OOS Prometheus is assumed to perform lifting at the side.

By comparing the critical area moment of inertia, I_{crit}, with the total combined crane capacity, the following graph in figure E.6 is made. From this relation the critical area moment of inertia is determined.

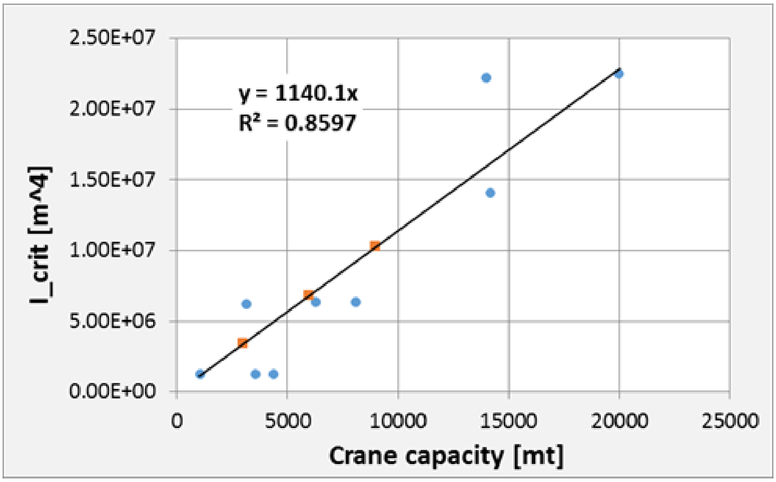

Figure E.6: Semi-submersible: Relation between the critical area moment of inertia (I_{crit}) and total crane capacity (Q).

The next step is to compare I_{crit} with the total possible area moment of inertia, I_{total}. This is as if the vessel is a monohull with a waterplane coefficient of 1. This gives a ratio that can be related to the main length and width dimensions of the vessel. The relation can be seen in figure E.7.

This relation, however, is not used directly. The results did not fit the trend of the reference vessels. This is most likely due to the sensitivity of the data, as the moment of inertia depends on parameters to the third power. A small deviation in such a parameter can give a large change in the result. The determined moments of inertia will be used at a later stage and to design the struts and floaters, as will be explained in a later section.

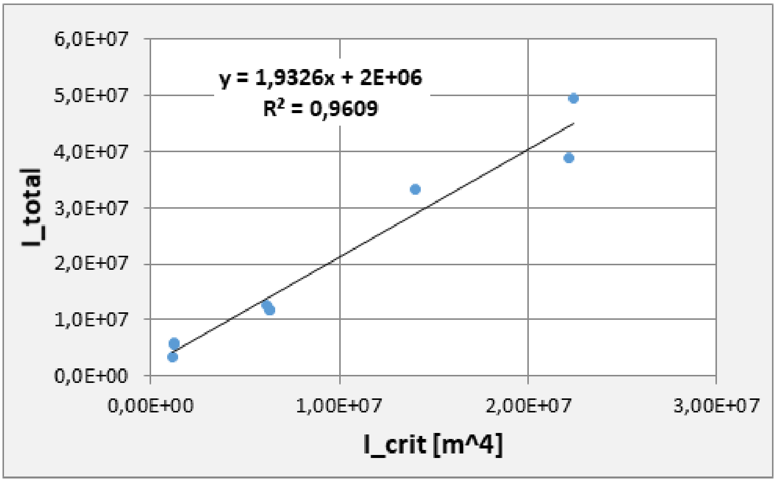

Figure E.7: Semi-submersible: Relation between the critical area moment of inertia (I_{crit}) and total area moment of inertia (I_{total}).

Waterplane area

Instead of the moment of inertia, the total waterplane area of each vessel is plotted against the total crane capacity. In figure E.8 the relation can be found. This relation is in theory not as good as the previous relation between the moment of inertia and crane capacity. It is, however, less sensitive and will prove to give better results (That is, the concept parameters follow the trend of the reference vessels).

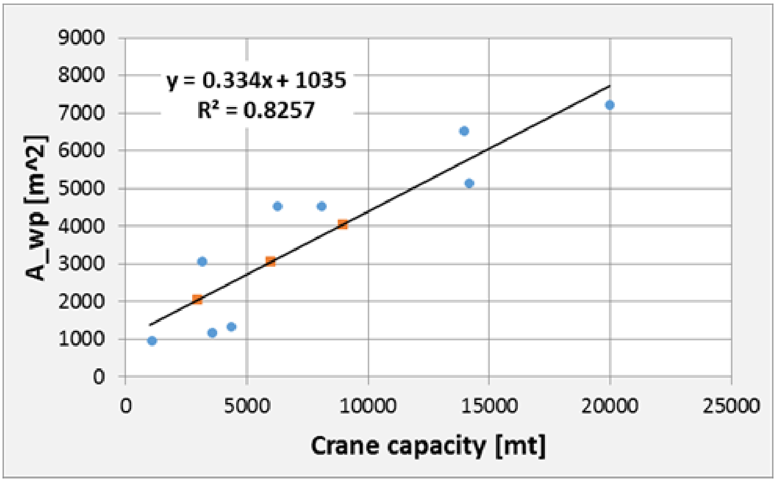

Figure E.8: Semi-submersible: Waterplane area as function of the crane capacity.

Area ratio

The ratio between the waterplane area and $L \cdot B$ area will be used so that the width can be determined. This ratio shows how large the struts are that go through the water surface in lifting mode. The relation between the area ratio and crane capacity is shown in figure E.9.

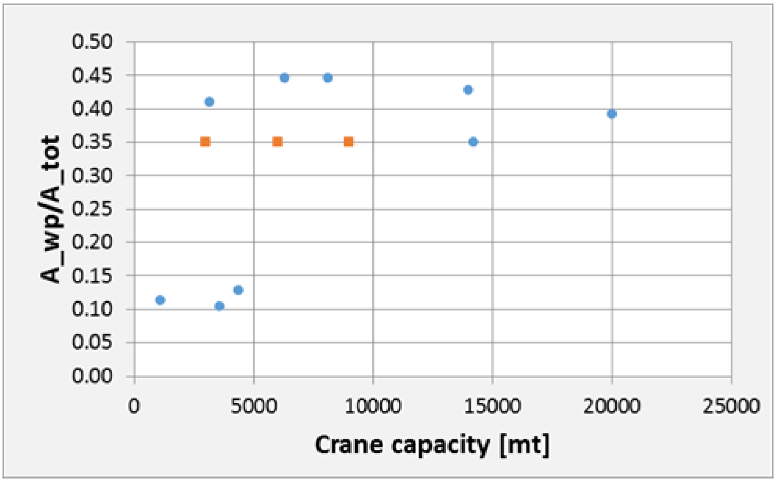

Figure E.9: Semi-submersible: Waterplane area ratio as function of the crane capacity.

At first glance, no clear relation can be found. However, the three ships that have a low ratio are the OOS Gretha, Prometheus and Serooskerke. These are ships that have relatively low crane capacity and function primarily as a working platform. The other ships focus more on the heavy lifting capabilities. It is chosen to design a ship that focuses on the heavy lifting capabilities, thus having a relatively high area ratio. The same value as for the Thialf vessel is chosen.

Length-width ratio

After numerous attempts to determine the length and width of the vessel, it showed that taking the length to width ratio to determine the main dimensions of the vessel gives the best results that fit reasonable well with the reference vessels. The relation between the length-width ratio and length can be seen in figure E.10. Together with the area ratio, waterplane area and length-width ratio, the length and width of the vessel are determined.

E

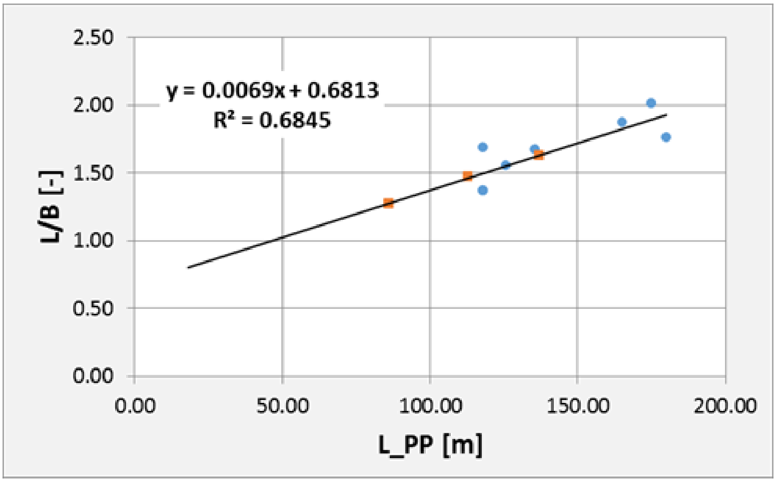

Figure E.10: Semi-submersible: Length-width ratio as function of the vessel length.

Deadweight tonnage

As explained before, the DWT is initially kept equal to the monohull designs.

Operational displacement

A clear relation is found between the DWT and operational displacement. Even though only limited data is available, it makes sense this relation exists. Also for the monohull reference vessels a clear relation was found. In figure E.11, the relation can be seen.

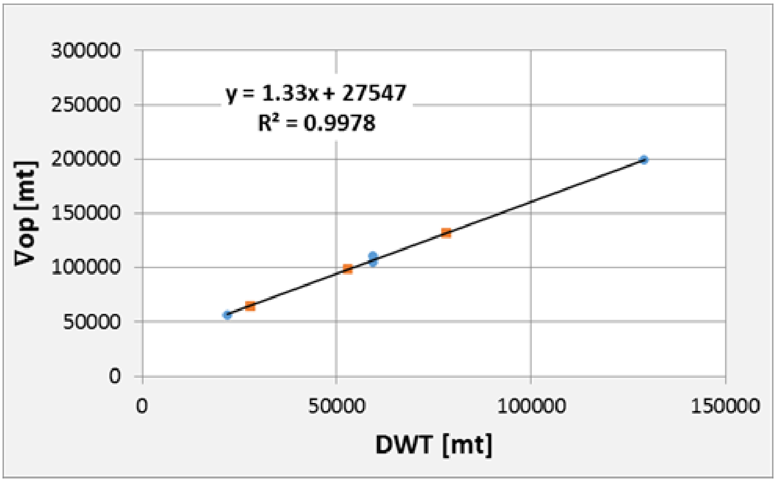

Figure E.11: Semi-submersible: Operational displacement as function of the DWT.

E

Block coefficient

Similarly to the monohull, the block coefficient showed no clear relation with other parameters, other than it seems somewhat constant. It is therefore chosen to take the average to determine this parameter. Similar to the length-to-width ratio, not all reference parameters are used.

Draft

The operational draft can be determined now that the operational displacement, width, length and block coefficient are known. Equation E.1 is used to calculate the operational draft.

The draft in transit mode can be estimated, as data is available for both the operational and transit draft. The average ratio between these two draft values is taken to estimate the transit draft.

Transit displacement

An estimation of the displacement in transit mode can be made by making an assumption for the block coefficient. It makes sense that the block coefficient increases in transit mode, as the floaters cover the length of the vessel, whereas the struts do not for better wave interaction. It is assumed that the block coefficient increases with 0.1, which is just a rough estimation, by the use of sketches. Equation E.1 can then be used to calculate the transit displacement.

Depth

For this parameter, the relation between the operational draft and depth is taken, similarly to the monohull design. The relation can be seen in figure E.12

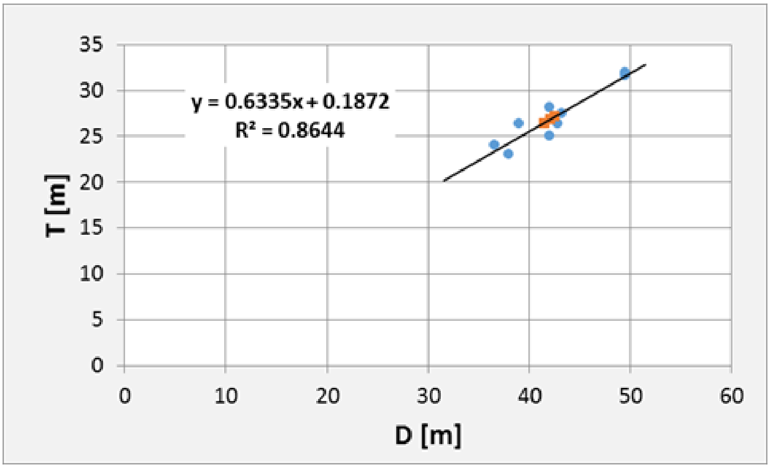

Figure E.12: Semi-submersible: relation between the operational draft (T_{op}) and depth (D).

Fine tuning

Once all parameters are known, it is checked whether the calculated area moment of inertia fits the trend of the reference vessels. It is found that, for the current geometry of the vessel, it will be hard to get a sufficient area moment of inertia. The geometry is thus adjusted (both the length and width) by still keeping certain important ratios the same (which have been discussed previously). This does mean that at least one parameter has to change, which is the ratio between the water-plane area and the $L \cdot B$ parameter. In order to increase the geometry of the vessel (and thus also the area moment of inertia) this area ratio had to be reduced. The result of the design process will be explained in the next section.

Design results

In this section the concept design results are shown and discussed. In figure E.13 the results of the monohull, semi-submersible and the differences between them are displayed.

Some important observations:

- The length is significantly less for the semi-submersible compared to the monohull. The width is, on the other hand, increased significantly. This is one of the most obvious differences between the ship types, and is as expected.
- The depth is significantly larger for the semi-submersible. This is due to the fact that the semi-submersible has two significantly different modes. In transit mode it behaves somewhat similar to a monohull, while in lifting mode the

MONOHULL	#1	#2	#3	
Q	3000	6000	9000	[mt]
LPP	159	185	211	[m]
B	42	50	59	[m]
D	14.0	16.2	17.5	[m]
T_tr	5.9	7.4	8.2	[m]
T_op	8.3	10.3	11.5	[m]
C_b	0.81	0.81	0.81	[-]
DWT	25312	46179	68148	[mt]
∇_tr	30899	53515	76131	[mt]
∇_op	46992	81386	115781	[mt]

SEMISUB	#1	#2	#3	
Q	3000	6000	9000	[mt]
LPP	110	132	151	[m]
B	76	83	88	[m]
D	31.7	36.9	39.2	[m]
T_tr	8.5	9.9	10.9	[m]
T_op	22.2	25.8	28.4	[m]
C_b	0.90	0.90	0.90	[-]
DWT	26794	48278	71245	[mt]
∇_tr	32155	48886	65617	[mt]
∇_op	58706	89252	119797	[mt]

DIFFERENCE	#1	#2	#3	
Q	0	0	0	[mt]
LPP	-49.8	-53.3	-59.7	[m]
B	34.1	32.6	28.9	[m]
D	17.7	20.7	21.7	[m]
T_tr	2.6	2.5	2.7	[m]
T_op	13.9	15.5	16.9	[m]
C_b	0.09	0.09	0.09	[-]
DWT	1482	2099	3098	[mt]
∇_tr	1256	-4629	-10514	[mt]
∇_op	11715	7866	4017	[mt]

Figure E.13: Concept design results of the monohull, semi-submersible and the differences between them, respectively.

vessel submerges, takes in ballast, and increases its displacement (and thus its operational draft).

- As the semi-submersible is a multihull in transit mode, the block coefficient is significantly lower compared to the monohull. Even though its transit displacement is also lower, the draft in transit mode is still greater compared to the monohull designs.

Crane design

The crane size is determined by looking at reference cranes. For all monohull and semi-sub vessels, an estimate of the crane size is made by using sketches, photos and available data. It makes sense that the crane size increases with increasing crane capacity. As the A-frame (or tub-crane) is chosen, only vessels with such a crane installed are used. The total number of vessels that are used for this analysis is 19 (out of the total 32).

The cranes are compared by looking at the following parameters:

- Single crane capacity
- Crane footprint radius
- Main boom length
- Auxiliary + whip boom length
- Boom hinge point height

The first clear relation is found for the crane footprint radius. In figure E.14 the radius is plotted against the crane capacity of the crane. Note: for the crane design use is made of the total single crane capacity. This is different then the crane capacities used before, as it does not have to be revolving. It also does not use the total vessel crane capacity as there is looked at single cranes.

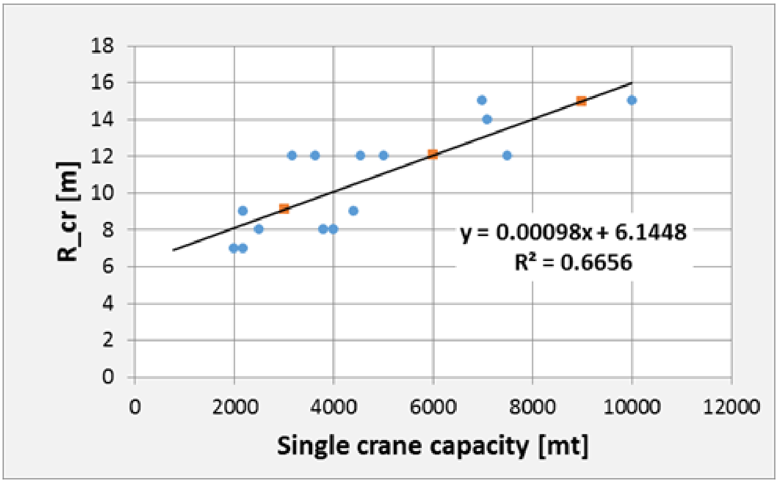

Figure E.14: Relation between the single crane capacity and footprint radius.

With the crane footprint radius known, the swing radius can be determined. The swing radius is the maximum radius of the crane pedestal above the footprint. For many cranes additional space around the crane footprint radius is required as some part can be significantly larger. This is, for example, the case when additional counterweight is added which is attached behind the crane. In figure E.15 it can be seen that a clear correlation is found.

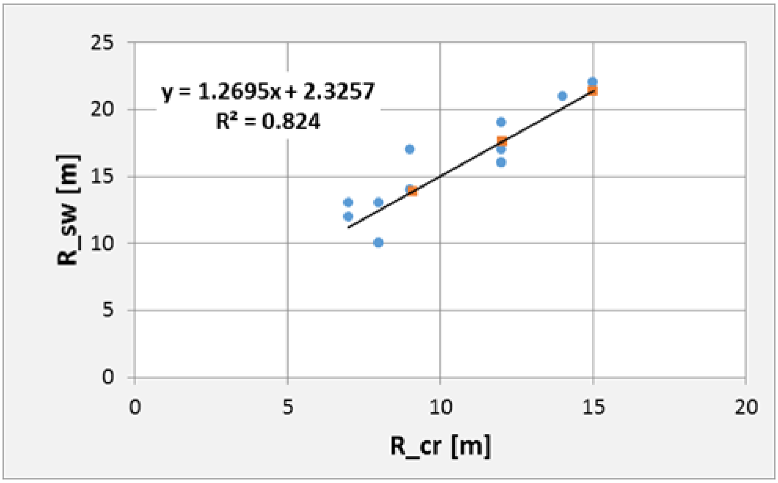

Figure E.15: Relation between the crane footprint radius and swing radius.

The boom length of the crane is divided into two parts; the main boom length and auxiliary-whip length. No clear relation was found for the total boom length, including the main, auxiliary and whip hoist. Attempts were also made to determine the moment at maximum hoist radius, but were unsuccessful. This is the reason why the main and auxiliary/whip boom length are both determined using the single crane capacity. The relation is shown in figure E.16 and E.17.

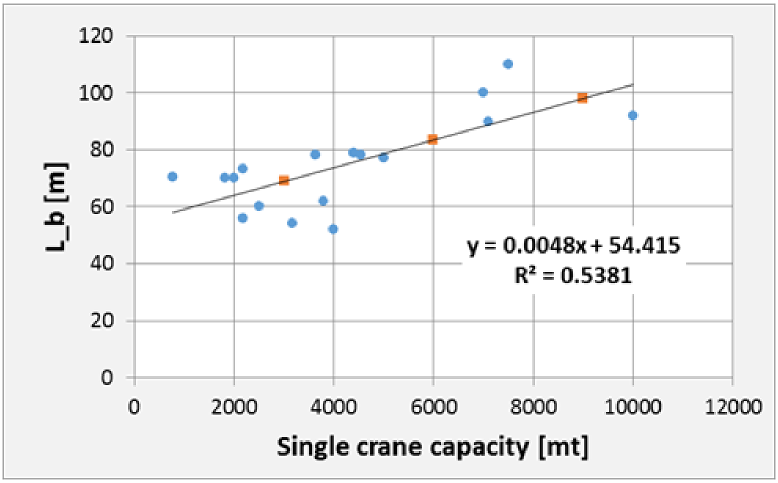

Figure E.16: Relation between the main boom length and single crane capacity.

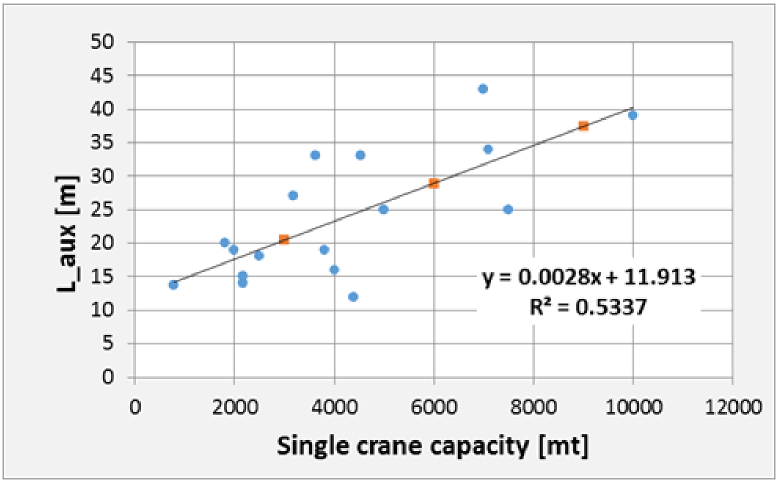

Figure E.17: Relation between the auxiliary+whip boom length and single crane capacity.

The most difficult parameter is the boom height. No clear relations are found. This is, most likely, because the boom height depends on other characteristics that are not known or are not determined. It could be that this height depends on the deck equipment and the height required to perform a certain operation. It is out of scope to determine such parameters and relations between deck equipment and crane height. It is therefore chosen to pick the height that fits well with the reference cranes. In figure E.18 the relation between the boom height and single crane capacity can be seen. Although the fit is bad, it determines three results that correspond well with the reference vessels. It also makes sense that for higher capacity cranes (and thus bigger vessels), the height increases. It is thus used in the design.

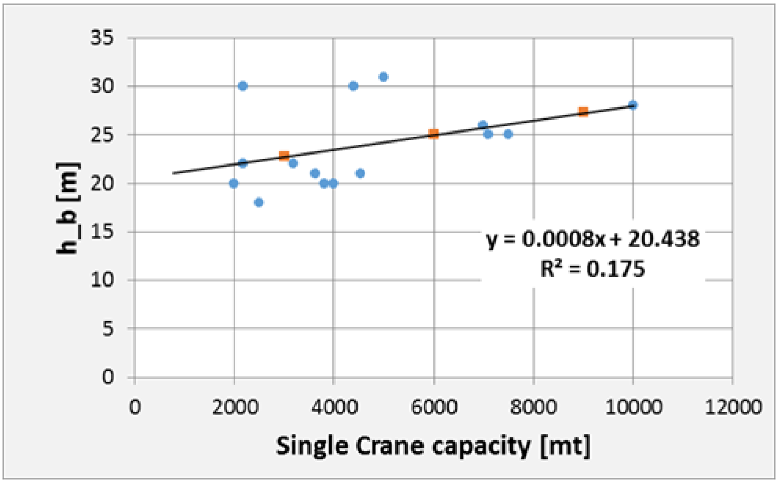

Figure E.18: Relation between the boom height and single crane capacity.

Accommodation design

The dimensions of the accommodation unit can be estimated in a similar way as for the cranes. For a few vessels, detailed sketches are available from which the dimensions of the accommodation could be estimated. A list is made of the following parameters:

- Capacity: The total amount of persons a vessel can accommodate.
- Length: The length of the accommodation unit.
- Width: The width of the accommodation unit.
- Layers: The number of layers stacked upon each other that make the accommodation unit.

The combined floor area that is used in the accommodation unit can be calculated using equation E.5. In this equation, x is the amount of layers. The total floor area is then plotted against the accommodation capacity, so that the floor area can be estimated for the designs. This can be seen in figure E.19.

$$A_{acc,T} = L_a \cdot B_a \cdot x \tag{E.5}$$

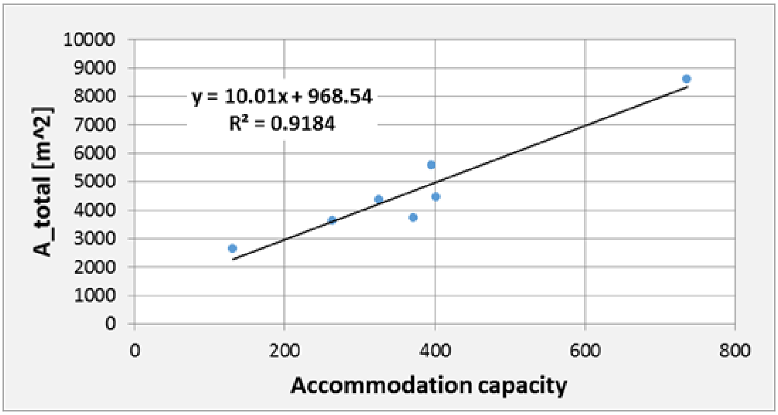

Figure E.19: The accommodation floor area as function of the accommodation capacity.

F

Crane selection and placement

In this chapter the current trends of the heavy lift cranes are explained. A heavy lift crane has a capacity of at least 250 metric tonnes according to the DNVL standards[7]. It is, however, a flexible limit and can be different for each individual.

Crane type selection

As with the ship types, the choice in different cranes is also large. There are at least two dozen different crane types that can be placed on a ship. Sometimes even mobile cranes (that are originally used onshore) are placed on a ship for simplicity, low cost and mobility. However, most of these cranes are not relevant, as the focus lies on the heavy lift sector. The different crane types that are often used onboard an offshore vessel can be seen in figure F.1. Most of these cranes are not used for the heavy lift sector. This is mainly because of the high lift capacity requirement. The following cranes are relevant for the off-shore heavy lift industry:

- A-frame: These cranes are also called pedestal or tub cranes. The boom is attached to an A-frame which is installed on a fully revolving platform. The forces are transferred to the pedestal by a slew bearing. Above the slew bearing, everything is fully revolving. Because of the load transfer mechanic it requires a relatively large footprint radius. These cranes are often equipped with counter weights which are located at the opposite side of the crane boom. While this makes heavy lifting easier, it does mean the crane has a large swing radius. As a result, more deck space around the crane has to be clear, so that the crane can rotate. This type of crane is well known for its application on the larger semi-submersible vessels such as the Thialf (7100 mt), Hermod (5000 mt) and Saipem 7000 (7000 mt) . Nevertheless, these cranes are widely used over all sort of vessels and can also be used for smaller crane capacity requirements.

- Mast: The mast crane is known for its mast, which is the tower-structure between the platform and the boom tie (which is attached to the mast head). The boom is attached to the slew platform, which can rotate around the mast. The crane footprint is relatively small, as the overturning moment is transferred by the mast rather than the slew bearing. As this crane uses no counterweight, this crane type has a larger influence on the vessel stability but has no swing radius. In general, the deck space required for this crane is relatively small. The mast cranes are usually used for smaller crane capacity requirements and are usually seen on heavy cargo vessels. They are still also used for heavy lift vessels such as on the Seven Borealis (4000 mt), Aegir (4000 mt) and Sapura 3000 (3000 mt)
- Slewing-mast: This type is similar to the mast crane, except for one key difference. The mast is attached to the platform and thus rotates as well. This means the overturning moment is transferred by the slew bearing rather than the fixed mast. As a result, the footprint radius is also relatively larger than the other mast crane. These cranes are not used often and the maximum crane capacity is for the Liebherr MTC 78000 (2000 mt).

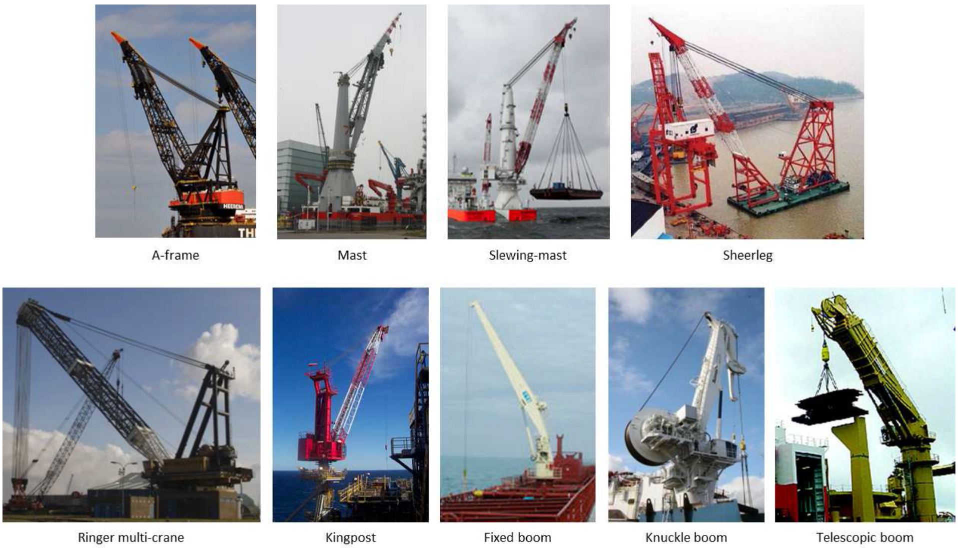

Figure F.1: The different types of cranes that can be used on-board ships. [17]

As explained before, there are multiple types of cranes that can be placed on a vessel. In particular, for the heavy lift crane vessels, usually only the A-frame and mast cranes are used. In order to keep the comparison as focused as possible (on the hull type, rather than other parameters), it is chosen to pick one crane and use it for all designs.

In the research by J. Kamp[17] a trade-off between the A-frame and mast crane is made. This is done for a 8000mt crane, but also gives a detailed analysis of the (dis)advantages. The trade-off table, which includes the weights and scores of different aspects, can be seen in figure F.2. As can be seen, there is no clear winner. Only the slewing mast crane scores significantly worse compared to the others. The two A-frame crane types are similar in size, while the mast crane is significantly smaller due to its smaller footprint area. For the further discussion it is assumed that the two A-frame crane types are similar, as it does not make a difference for this thesis study what the specific differences are. The remaining choices are thus the A-frame crane and mast crane.

		Scores				Weights*Scores			
Criteria	**Weight**	**A-frame** Roller	**A-frame** Bogie	**Mast**	**Slewing-mast**	**A-frame** Roller	**A-frame** Bogie	**Mast**	**Slewing-mast**
Lifting capacity up to 8,000mt	0,26	5	4	3	1	1,28	1,02	0,77	0,26
Lifting capacity >8,000mt	0,06	4	4	2	2	0,25	0,25	0,12	0,12
Footprint	0,17	2	2	5	3	0,34	0,34	0,84	0,51
Boom hinge point location and load curve	0,14	1	3	5	3	0,14	0,43	0,71	0,43
Tail swing	0,05	1	1	5	2	0,05	0,05	0,23	0,09
Slewing range	0,02	5	5	4	5	0,10	0,10	0,08	0,10
Air draft	0,15	3	3	2	2	0,44	0,44	0,29	0,29
Vessel stability	0,05	5	5	3	3	0,23	0,23	0,14	0,14
Deep-water lifting and lowering	0,02	4	4	3	3	0,08	0,08	0,06	0,06
Design maturity and reliability	0,09	5	4	2	1	0,47	0,37	0,19	0,09
	1,00					3,37	3,30	3,43	2,09

Figure F.2: Trade-off between the A-frame and mast crane types. [17]

In addition to the comparison in (dis)advantages, there is one major difference. In figure F.3, the amount of mast- and A-frame cranes can be seen. Most mast cranes are for lower crane capacities compared to the A-frame cranes. According to the figure, A-frame cranes are used for a wide range of crane capacities. It is therefore chosen to use the A-frame crane for all designs. This is because the engineering cost to design a new high capacity mast crane can be quite high, and might bring (new) unexpected challenges or disadvantages.

Difference in vessel type

Most monohull crane vessels have a relatively low crane capacity compared to the semi-submersibles. Monohulls usually do not have a higher crane capacity than 5000 metric tonnes. The semi-submersible vessels are often equipped with higher capacity cranes, such as the Thialf with 2x7,100 mt and there are even plans for the Sleipnir that has a total crane capacity of 2x10,000 mt.

Number of cranes

As the monohull vessels often use a single crane, while the semi-submersible designs often have two, a decision has to be made. There are multiple options pos-

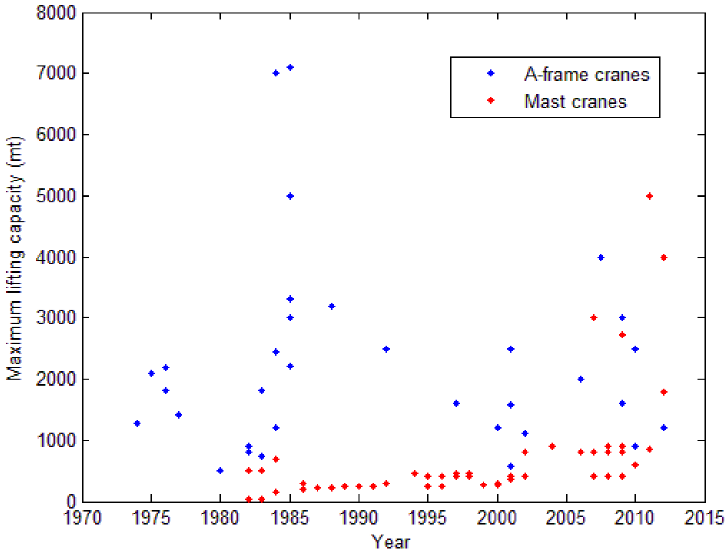

Figure F.3: Number of mast and A-frame cranes with their crane capacities over the years.

F

sible:

- All designs use two cranes. This makes sense for the semi-submersible, but for the monohull it is exceptional. The question here is also the significantly lower width of the vessel. In order to use the two cranes together, a certain distance has to be between the cranes so that an object can be lifted onto the vessel.
- All designs use a single crane. Compared to the previous option, this makes sense for the monohull designs. Although there are two semi-submersibles that use a single crane, it does not fit the average semi-sub. One of the two exceptions is the DB101, which has the highest length-width ratio of all semi-subs (thus a single crane is more favourable compared to other designs due to limited width). The other is the OOS Prometheus, which is more a hotel vessel than a vessel focused on heavy lifting.
- One crane for the monohull designs and two cranes for the semi-sub designs. This option is the best for the individual designs. It does, however, make the comparison more complicated. As now multiple load cases have to be investigated which are not identical, it widens the scope of the thesis. It also greatly influences the operational profile, as a tandem lift is significantly different than a single lift.

It is chosen to use a single crane for all designs. This is mainly because a difference in crane number significantly differs the operational profile of the vessel. This would make the comparison more complex, and might even result in unclear results.

Crane positioning

The crane position is important as it largely affects the operational profile. A crane that is located on the side of the vessel enables the possibility to lift at the side. In addition, the crane position also largely affects the ballasting requirements and thus also the general arrangement.

In order to get a good understanding of the (dis)advantages of all crane positions on both vessels, the ideal crane position is determined for both vessels. This is done by looking at various aspects such as ballasting, structural strength and the reach on deck. A trade-off is then used.

Monohull

There are a few possible crane positions for a monohull crane vessel. In figure F.4 three crane positions are shown that are investigated. Each position has its own advantages and a trade-off is made to determine the preferred crane location. It is tried to make a general comparison independent of crane capacity and its operational profile. There are, however, some important differences in these aspects which will be explained after the trade-off is made.

Figure F.4: Top view of the monohull deck with indicated possible crane locations.

Performance aspects

The crane positions will be evaluated by looking at the following aspects. It is important to note that all crane positions are evaluated by using the same size and capacity of the crane.

- Lifting operation: In order to perform an operation the crane has to be able to reach the cargo and install it outboard. This aspect is divided into these two phases:
  - Outboard reach: The crane reach outboard of the vessel is of importance, as sometimes the space around the vessel is limited and a more versatile

vessel has the advantage. Some clients even simply refuse to have a crane on the side of the vessel due to this. Position #1 has the advantage that it can carry out an operation at both sides and aft of the vessel. There is, however, a small limitation which might occur if the clearance is not large enough, as the crane is positioned at the centreline. #2 has the advantage that it has maximum reach on one side of the vessel, but it has a high limitation on the other side. Similar to #1, it has full capacity at the aft of the vessel. #3 is usually limited to operations at one side of the vessel.

– Inboard reach: It is also important to be able to reach the cargo that is positioned on deck. #1 and #2 have a similar reach, whereas #3 is capable to reach a larger deck area with its maximum crane capacity. On the other hand, the minimum crane radius is of importance here as well. #3 will lose more deck area while #1 loses less as it stands at the aft of the vessel. #2 has the least minimum crane radius disadvantage as it stands in the corner.

- Ballasting: The position of the crane has a large impact on the ballast management during an operation. Having a crane at the side of the vessel will increase the required ballast and thus requires a higher pump capacity or the operation will take longer. Compared to #1, #2 has a small advantage with installations that have to be performed aft of the vessel. As the cargo can be rotated closer to the centreline, less ballasting is required. However, if the deck is obstructed, cargo has to be rotated further away from the centreline which increases the required ballast. #3 is similar to #2 but the effect on the trim angle could be smaller, which could potentially reduce the required ballast. One additional consideration is that a vessel could lift cargo off a barge and directly install it without slewing the crane. #1 and #2 then have the advantage of requiring significantly less ballast as the operation can be performed aft of the vessel.

  For #2 and #3 not all lightship heel can be compensated by placing other weight items at the opposite site, due to the high crane weight. It is therefore necessary to use ballast to compensate this initial heeling angle. In addition, some weight items that could be used to compensate can vary during a mission (such as fresh water tanks), making a constant ballast adjustment necessary in some situations.

  A crane that has to rotate an object with a large angle has the disadvantage of time and complexity. As the cargo and boom weight shift during slewing, constant ballasting is required. A versatile working radius thus has another advantage due to this aspect.

- Structural design: This aspect can be divided into two main items. First, for a crane that is located at the aft of the vessel, the longitudinal bending moment is higher. This will thus increase the steel-weight of the hull. Secondly, since the loads have to be transferred from the crane to the hull structure, it is not desired to have the crane located at a thin part of the hull (thus, aft of the ship due to the hull shape). Finally, it is desired to have a full ship, thus a higher displacement, below the crane, which is only the case for position #3.

- Deck arrangement: This aspect can also be separated into two main items:
  - Unobstructed deck area: By positioning the crane at the side of the vessel, a larger unobstructed deck area is available for large cargo due to the location of the boom. However, for #3 the maximum deck area available for a single large cargo object is limited as the crane is more forward on deck.
  - Other equipment: Some vessels have more equipment on deck than just a heavy lift crane. For example, additional cranes can be installed or the vessel can be outfitted for pipelay operations. Even though these considerations are not required for the comparison, it is taken into account in the sense that a large unobstructed deck area is not the sole contributor to this aspect. For each crane position there is looked at consequences for additional equipment on board.

Trade-off

Now that the aspects are known, scores can be assigned for each crane location and the aspect. The method used is the multi-criteria analysis[4]. The weight of an aspect is determined by comparing each aspect with all other aspects, and assigning an importance factor. There are three different factors used as listed below.

- 1: Aspects are of (almost) equal importance,
- 2: Aspect is more important,
- 3: Aspect is much more important.

It is important to realise that these factors are estimated by own knowledge and with the help of company experts. These are thus not based on calculations. Therefore this trade-off can only be used as a rough idea rather than taking clear conclusions from it. By taking the weighted average the total weight per aspect is determined, as shown in figure F.5.

Now each crane position can be given a score for each aspect. These scores are also estimated and their meaning are listed below. The explanation for each crane position that result in these scores can be found above in the performance aspects section.

- 1: Very bad performance. Has a significant disadvantage.
- 2: Poor performance. Has a small disadvantage.
- 3: Average performance. No significant (dis)advantage.
- 4: Good performance. Has a small advantage.

- 5: Excellent performance. Has a significant advantage.

The final score for each position is then determined by multiplying each aspect score with the aspect weight and summing these scores, as shown in figure F.6. It can thus be seen that the first crane position scores the best.

	a	b	c	d		W
a	1	2	2	1	6.0	0.33
b	0.5	1	1	0.5	3.0	0.17
c	0.5	1	1	0.5	3.0	0.17
d	1	2	2	1	6.0	0.33

Figure F.5: Determination of the aspect weights by comparing each aspect with another. Lifting operation, ballasting, structural design and the deck arrangement is indicated by a, b, c and d respectively. The final weight is indicated by W.

		Crane position				Score		
		# 1	# 2	# 3	W	# 1	# 2	# 3
a	**Lifting operation**	4	3	3	0.33	1.3	1.0	1.0
b	**Ballasting**	4	4	3	0.17	0.7	0.7	0.5
c	**Structural design**	3	1	4	0.17	0.5	0.2	0.7
d	**Deck arrangement**	4	5	3	0.33	1.3	1.7	1.0
						3.8	**3.5**	**3.2**

Figure F.6: Criteria scores and result for each crane position.

Semi-sub

A similar strategy is used to determine the ideal crane position on a semi-sub. The three considerations are shown in figure F.7.

Performance aspects

The same aspects as for the monohull are used. The different crane positions are discussed shortly.

- Lifting operation: For this aspect two phases are also identified:
  - Outboard reach: The advantage of a crane in the corner is that it can perform lifting operations over the stern as well over the side. Thus #2 has a significant advantage over #1 and #3, which are both limited to one side.
  - Inboard reach: However, since #1 and #3 are more centered, their reach on deck is larger. This is especially advantageous for #1, as for #3 there is the accommodation that takes part of the deck area.

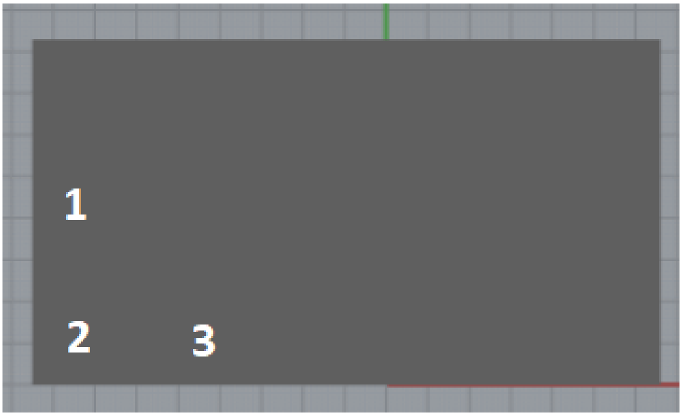

Figure F.7: Top view of the semi-submersible deck with indicated possible crane locations.

- Ballasting: During a lifting operation less ballasting is required for #1 and #2, while for #3 a relatively large amount of ballast is required due to L/B being higher than 1. For the cranes positioned on the side, there is additional disadvantage as these designs need a relatively high initial amount of ballast in order to level the vessel.
- Structural design: The main difference here is that crane #2 and #3 can be positioned above a column, providing a high initial structural strength. For #1 additional support is required, as the crane is positioned in the centre.
- Deck arrangement: In general #2 is the best for this aspect as the crane is positioned in the corner.

Trade-off

Also for the trade-off the same strategy is used. This leads to figure F.8. It can thus be seen that position #1 and #2 have a relatively high score compared to #3. A crane in the corner has the highest score mainly due to the structural and deck arrangement advantage. However, this position is not chosen, as will be described in the next sections.

		Crane position				**Score**		
		# 1	**# 2**	**# 3**	**W**	**# 1**	**# 2**	**# 3**
a	**Lifting operation**	4	4	3	0.33	1.33	1.33	1.00
b	**Ballasting**	4	2	2	0.17	0.67	0.33	0.33
c	**Structural design**	3	4	4	0.17	0.50	0.67	0.67
d	**Deck arrangement**	3	4	3	0.33	1.00	1.33	1.00
						3.5	**3.7**	**3.0**

Figure F.8: Criteria scores and result for each crane position.

Crane positioning

It can thus be concluded that by looking at the vessel types individually, a crane centered at the stern is the best for the monohull, while a crane in the corner is ideal for the semi-submersible. However, there are some important implications for both the individual designs as the comparison in this thesis.

Crane capacity implication

As the three designs have a large difference in crane capacity, it could be that the ideal crane position for a 3000mt capacity design is different than for a 9000mt design. The 3000mt design could be used for offshore wind operations, whereas the higher crane capacities are often used for e.g. decommissioning operations. For the offshore wind industry it is often the case that multiple smaller objects have to be transported and installed, giving position #3 a significant advantage due to its large deck area that the main crane can reach. Some other criteria might also be deemed less important for these operations. However, in order to compare the vessels for a generalized lifting operation, this trade-off suffices.

Multiple cranes

It would also be possible to fit two cranes on the vessels, so that tandem lifts are possible which have unique (dis)advantages. However, since the crane capacity and lifting operation is equal for both vessel types, this would have limited effect on the result. Especially since the workability will stay almost the same. This option would particularly be interesting for the largest 9000 mt design, as this crane capacity is not yet constructed. Using a single 9000mt crane would cause a larger uncertainty in the capital investment and applicability of the workability criteria. Similar to the above, this would have a rather limited effect on the comparison and is not further considered.

Comparison

Overall it can be concluded that all crane positions have their own advantages and there is no crane position that is the best for every specific operation. However, in order to make an accurate comparison within the time frame, a good choice has to be made. One might wonder how the ideal crane positions for a generalized crane operation will result in crane positions that are used for the comparison. Two options are considered:

1. It is possible to select the ideal crane position for each vessel type. This would mean that the monohull is mainly capable of lifting over the stern, while the semi-sub can perform a lift over the side as well. However, the difficulty is then to combine the results while their operational profile is actually different. It is also possible to only assume a lift over the stern, but then some of the

semi-sub crane position advantage will not be used (and might not be ideal any more).

2. An easier solution is to keep the crane position the same for both vessel types. This will largely keep the operational profile the same, as both vessels will lift over one side. This is a large advantage as it simplifies the required work as well as makes the comparison clearer. However, this means a less ideal crane position has to be used for a vessel.

Mainly due to the many implications option 1 would have, it is chosen to use the same crane position for all vessels. There has to be realised that if the crane position is not the same and the lifting operation is different, many calculations will yield results that are difficult to compare. For instance, if the workability is determined for a lifting operation over the stern, but in addition also for over the side for the semi-sub, it is questionable how these workability rates should be combined for the semi-sub design. It could be an option to only compare the lifts over the stern, or to average the workability rates. In both cases several disadvantages could be thought of, making it not ideal.

The second question is which crane position is then chosen for both vessels. Since a lifting operation over the stern is the most common, #3 is neglected. This is also logical as this location scores the worst for both vessel types. The difference between #1 and #2 is almost equal for both vessel types, making both positions viable. However, it is chosen to use #1 because a 9,000 mt crane on the side would cause additional challenges that are deemed out of scope of this thesis. A lot of initial ballast would be required and the structural design implications for placing such a heavy crane (with its load) on the corner of the deck are unknown. It would be a safer and more feasible option to place a crane on the stern which is centered on deck. This thus also implicates that the vessels will perform their lifts over the stern. However, there will still be looked at several conditions where the crane is turned to the side, which is e.g. the case when cargo is rotated over the stern. This will be further discussed in chapter 6.

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E-balances

	Nominal output	Efficiency	Power consumption	Transit			DP Normal			Stand-by			Harbour		
				Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	2800	0.9	3111	0.67	1	2084	0.2	1	591	0.1	1	296	0	0	0
2. Main thruster	2800	0.9	3111	0.67	1	2084	0.2	1	591	0.1	1	296	0	0	0
3. Retractable thruster	2200	0.9	2444	0	1	0	0.2	1	464	0.1	1	232	0	0	0
4. Retractable thruster	2200	0.9	2444	0	1	0	0.2	1	464	0.1	1	232	0	0	0
5. Retractable thruster	2200	0.9	2444	0	1	0	0.2	1	464	0.1	1	232	0	0	0
6. Retractable thruster	2200	0.9	2444	0	1	0	0.2	1	464	0.1	1	232	0	0	0
7. Tunnel thruster	1500	0.9	1667	0	1	0	0.2	1	317	0.1	1	158	0	0	0
8. Tunnel thruster	1500	0.9	1667	0	1	0	0.2	1	317	0.1	1	158	0	0	0
Crane	4295	0.9	4772	0	0	0	0.5	1	2386	0	0	0	0.2	1	954
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	500	0.9	556	0.5	1	278	0.5	1	278	0.5	1	278	0.75	1	417
Remainder	474	0.9	527	1	1	527	1	1	527	1	1	527	1	1	527
Total power			26854			6223			8114			3891			2731

Figure G.1: E-balance sheet for the 3000 mt monohull design.

	Nominal output	Efficiency	Power consumption	Transit			DP Normal			Stand-by			Harbour		
				Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	4000	0.9	4444	0.59	1	2622	0.2	1	889	0.1	1	444	0	0	0
2. Main thruster	4000	0.9	4444	0.59	1	2622	0.2	1	889	0.1	1	444	0	0	0
3. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
4. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
5. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
6. Retractable thruster	3000	0.9	3333	0	1	0	0.2	1	667	0.1	1	333	0	0	0
7. Tunnel thruster	2000	0.9	2222	0	1	0	0.2	1	444	0.1	1	222	0	0	0
8. Tunnel thruster	2000	0.9	2222	0	1	0	0.2	1	444	0.1	1	222	0	0	0
Crane	8590	0.9	9544	0	0	0	0.5	1	4772	0	0	0	0.2	1	1909
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	750	0.9	833	0.5	1	417	0.5	1	417	0.5	1	417	0.75	1	625
Remainder	697	0.9	774	1	1	774	1	1	774	1	1	774	1	1	774
Total power			39485			7685			12546			5108			4141

Figure G.2: E-balance sheet for the 6000 mt monohull design.

	Nominal output	Efficiency	Power consumption	Transit			DP Normal			Stand-by			Harbour		
				Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	4700	0.9	5222	0.59	1	3081	0.2	1	1044	0.1	1	522	0	0	0
2. Main thruster	4700	0.9	5222	0.59	1	3081	0.2	1	1044	0.1	1	522	0	0	0
3. Retractable thruster	3700	0.9	4111	0	1	0	0.2	1	822	0.1	1	411	0	0	0
4. Retractable thruster	3700	0.9	4111	0	1	0	0.2	1	822	0.1	1	411	0	0	0
5. Retractable thruster	3700	0.9	4111	0	1	0	0.2	1	822	0.1	1	411	0	0	0
6. Retractable thruster	3700	0.9	4111	0	1	0	0.2	1	822	0.1	1	411	0	0	0
7. Tunnel thruster	2600	0.9	2889	0	1	0	0.2	1	578	0.1	1	289	0	0	0
8. Tunnel thruster	2600	0.9	2889	0	1	0	0.2	1	578	0.1	1	289	0	0	0
Crane	12885	0.9	14317	0	0	0	0.5	1	7158	0	0	0	0.2	1	2863
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	900	0.9	1000	0.5	1	500	0.5	1	500	0.5	1	500	0.75	1	750
Remainder	894	0.9	993	1	1	993	1	1	993	1	1	993	1	1	993
Total power			50643			8905			16435			6010			5440

Figure G.3: E-balance sheet for the 9000 mt monohull design.

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	Nominal output	Efficiency	Power consumption	Transit			DP Normal			Stand-by			Harbour		
				Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	2700	0.9	3000	0.81	1	2435	0.2	1	600	0.1	1	300	0	0	0
2. Main thruster	2700	0.9	3000	0.81	1	2435	0.2	1	600	0.1	1	300	0	0	0
3. Retractable thruster	2700	0.9	3000	0.81	1	2435	0.2	1	600	0.1	1	300	0	0	0
4. Retractable thruster	2700	0.9	3000	0.81	1	2435	0.2	1	600	0.1	1	300	0	0	0
5. Retractable thruster	2700	0.9	3000	0	1	0	0.2	1	600	0.1	1	300	0	0	0
6. Retractable thruster	2700	0.9	3000	0	1	0	0.2	1	600	0.1	1	300	0	0	0
7. Retractable thruster	2700	0.9	3000	0	1	0	0.2	1	600	0.1	1	300	0	0	0
8. Retractable thruster	2700	0.9	3000	0	1	0	0.2	1	600	0.1	1	300	0	0	0
Crane	4295	0.9	4772	0	0	0	0.5	1	2386	0	0	0	0.2	1	954
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	500	0.9	556	0.5	1	278	0.5	1	278	0.5	1	278	0.75	1	417
Remainder	558	0.9	620	1	1	620	1	1	620	1	1	620	1	1	620
Total power			31614			11888			9334			4548			2824

Figure G.4: E-balance sheet for the 3000 mt semi-sub design.

	Nominal output	Efficiency	Power consumption	Transit			DP Normal			Stand-by			Harbour		
				Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	3500	0.9	3889	0.71	1	2749	0.2	1	778	0.1	1	389	0	0	0
2. Main thruster	3500	0.9	3889	0.71	1	2749	0.2	1	778	0.1	1	389	0	0	0
3. Retractable thruster	3500	0.9	3889	0.71	1	2749	0.2	1	778	0.1	1	389	0	0	0
4. Retractable thruster	3500	0.9	3889	0.71	1	2749	0.2	1	778	0.1	1	389	0	0	0
5. Retractable thruster	3500	0.9	3889	0	1	0	0.2	1	778	0.1	1	389	0	0	0
6. Retractable thruster	3500	0.9	3889	0	1	0	0.2	1	778	0.1	1	389	0	0	0
7. Retractable thruster	3500	0.9	3889	0	1	0	0.2	1	778	0.1	1	389	0	0	0
8. Retractable thruster	3500	0.9	3889	0	1	0	0.2	1	778	0.1	1	389	0	0	0
Crane	8590	0.9	9544	0	0	0	0.5	1	4772	0	0	0	0.2	1	1909
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	750	0.9	833	0.5	1	417	0.5	1	417	0.5	1	417	0.75	1	625
Remainder	777	0.9	863	1	1	863	1	1	863	1	1	863	1	1	863
Total power			44019			13526			13524			5641			4230

Figure G.5: E-balance sheet for the 6000 mt semi-sub design.

				Transit			DP Normal			Stand-by			Harbour		
	Nominal output	Efficiency	Power consumption	Load	in service	Power	Load	in service	Power	Load	in service	Power	Load	in service	Power
	kW	%	kW	%	-	kW	%	-	kW	%	-	kW	%	-	kW
Propulsion															
1. Main thruster	4300	0.9	4778	0.62	1	2983	0.2	1	956	0.1	1	478	0	0	0
2. Main thruster	4300	0.9	4778	0.62	1	2983	0.2	1	956	0.1	1	478	0	0	0
3. Retractable thruster	4300	0.9	4778	0.62	1	2983	0.2	1	956	0.1	1	478	0	0	0
4. Retractable thruster	4300	0.9	4778	0.62	1	2983	0.2	1	956	0.1	1	478	0	0	0
5. Retractable thruster	4300	0.9	4778	0	1	0	0.2	1	956	0.1	1	478	0	0	0
6. Retractable thruster	4300	0.9	4778	0	1	0	0.2	1	956	0.1	1	478	0	0	0
7. Retractable thruster	4300	0.9	4778	0	1	0	0.2	1	956	0.1	1	478	0	0	0
8. Retractable thruster	4300	0.9	4778	0	1	0	0.2	1	956	0.1	1	478	0	0	0
Crane	12885	0.9	14317	0	0	0	0.5	1	7158	0	0	0	0.2	1	2863
Accommodation	1500	0.9	1667	0.75	1	1250	0.75	1	1250	0.75	1	1250	0.5	1	833
Deck equipment	900	0.9	1000	0.5	1	500	0.5	1	500	0.5	1	500	0.75	1	750
Remainder	994	0.9	1104	1	1	1104	1	1	1104	1	1	1104	1	1	1104
Total power			56310			14784			17657			6676			5551

Figure G.6: E-balance sheet for the 9000 mt semi-sub design.

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