Master Thesis Offshore Engineering

A research to decrease the duration of the lifting phase for the Jumbo J-class, focussing on anti-heeling

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Abstract

For offshore installation projects Jumbo uses her J-class heavy lift crane vessels. The objects that need to be installed offshore are carried at deck or in the ship's hull. At location the object is lifted by the on-board heavy lift crane and slewed to the side of the ship. From the final overboard position the object can be lowered to the seabed. When the load is displaces by the slewing crane, the equilibrium of moments is disrupted and the ship will roll or heel. To reduce this moment change the ship pumps ballast water from one side to the other and causes a counter moment. This simple system has limited capacities and it was assumed that an improvement could reduce the duration of the lifting phase. This thesis is about a research to decrease the duration of the lifting phase for the Jumbo J-class, focussing on anti-heeling.

To understand the context of the problem, research has been done on the company and the market she is operating in. Then the phenomenon of anti-heeling, the current ballast system and the operational limits were analysed. With the information obtained from this research a list of requirements and the load cases were selected.

The analysis showed that the current system capacity is based on three manual controlled centrifugal pumps that can either been switch on or off. This system can not easily vary the moment change and was believed to be limiting for lift operations. To find a new solution design methods are used to generate a variety of concepts. For five concept calculation has been done to predict their capacity and dimensions. The analysis showed that a new solution is unlikely to be financially attractive for the market Jumbo is operating in. Therefore the decision has been made to research the effects of the most economical potential solution. Variably frequency drives were applied to the pump system to see if the variations of the rotational speed of the pumps and therefore varying the flow rate would make easier to control the roll motions. Or even shorten the lift phase.

To examine the ballast system and the sensitivities a model has been made. As this research focused on roll and heeling motions only, a 2D model is used to predict heave, sway and roll motions. The model is based on the equilibrium of moments where the ship is exposed to wave induced moments and the moments caused by the crane and ballast system. The crane and ballast system are effected by the ship motions and therefore coupled to these motions. The model allows to automate the control of the crane motions and the pump rotations. This allowed the model to test different lift scenarios under different wave conditions.

The results showed that the variable frequency drivers can result in smaller differences of the heeling angles. But the results also showed that the ballast system was not the most important limiting factor. It was the moment change caused by the crane. When the crane tip with hanging load was displaced with a certain speed, it caused an impulse making the ship oscillate around the static heeling angle. The amplitudes of these oscillations and the static heeling angle could be reduced by the ballast system. But the best results were shown when the stability properties of the vessel were improved. Increasing the effective metacentric height from 2.25 to 3.25 showed significantly reduce roll excitations.

Other conclusions are that previously very conservative roll angles of $1°$ were used, while the mast crane proved to handle roll angles of $4°$ and larger depending on the load. Also the lay out of the current pump system leaves a lot of room of improvement. These improvements can be found in the recommendations as well as suggestions how to discover potential time savings during offshore operations.

Contents

Chapter 1

Introduction

This researched is commissioned by Jumbo Maritime, a Dutch shipping and heavy lift company. This introduction will provide information about the company, the background of the problem and introduce the research questions and objectives.

1.1 Company profile

Jumbo is a privately-owned company with five decades of maritime industry experience. The company owns and operate a large fleet of specialized Heavy Lift Vessels (HLVs) with dual crane capacities ranging from 500 to 1,800 tons. Although shipping is the main business, Jumbo is becoming a well know player in the offshore installation market with their *"From-Quayside-To-Seabed"* solutions. See figure 1.1 to have this philosophy illustrated.

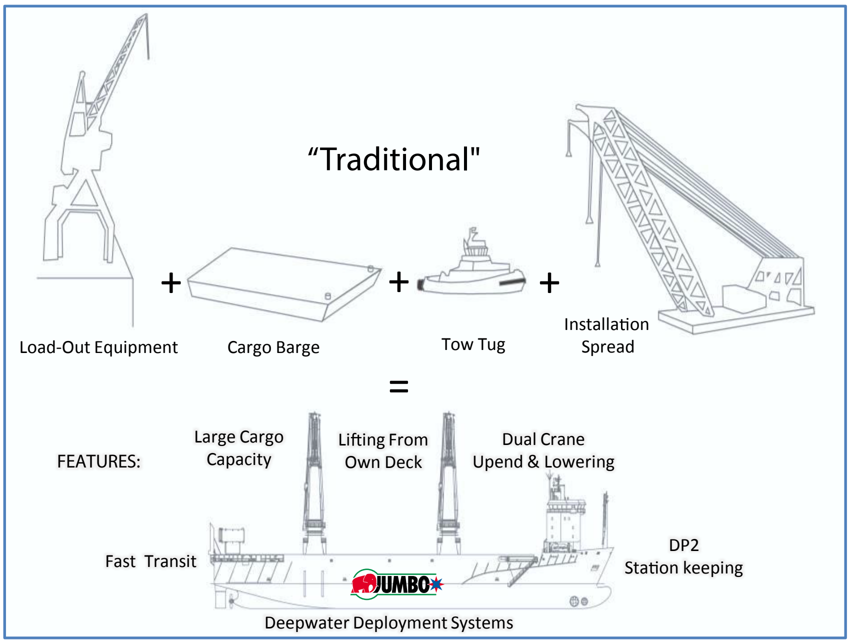


Figure 1.1: Intergrated transport and installation

The Jumbo J-class vessels are mono hull heavy lift crane vessels (144.1 x 26.7) with two mast cranes of 900 ton each. The vessel are able to pick the load from the quayside by its own crane and transport is to the offshore location with a relative high transit speed with a maximum of

17 knots. The DP2-system and a depth rating of 3000 meters makes these vessel well equipped for a great variety of offshore work. This all-in-one solution can be a great advantage over the equipment and logistic intensive traditional method.

1.1.1 Mission

Jumbo's mission is to be the partner of choice, and be acknowledged for:

- succesfull delivery of simple, robust and cost efficient offshore transportation and installation solutions,
- Strong focus on developing long-term client/partner relationships.

1.1.2 Offshore products and services

Jumbo has build long-term relationships with their customers, some of these customers trusted Jumbo to execute offshore installation work. As a result of this relationships Jumbo could work on their offshore track record.

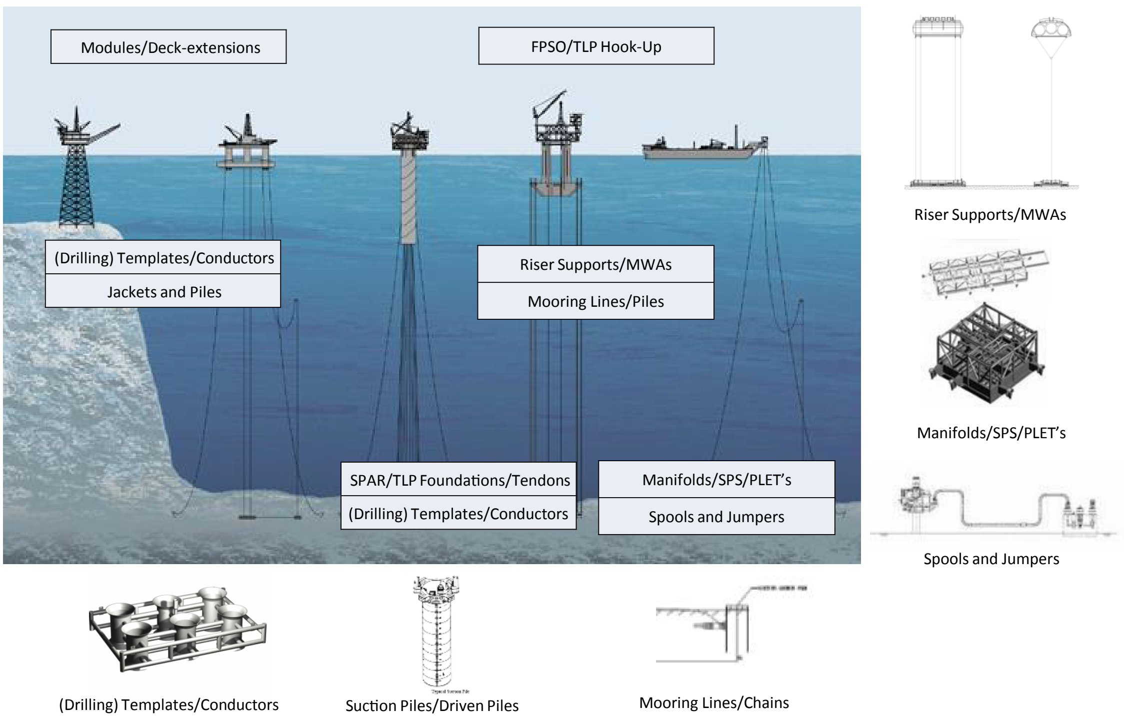


Figure 1.2: Products and services

The track record contains mostly transport and installation projects (T&I). Typical project are installation of anchor piles, moorings systems, windmill transition pieces and subsea structures as figure 1.2. Offshore lifting operation including equipment has varied from 85 up to 760 ton for Jumbo's track record. See figure 1.3 for a time-line overview. Not all lifting operations will significantly benefit from a shorter lifting phase, as some projects only have one relatively light subsea structure to be installed. Bigger numbers of repeated installation work can be found in, for instance, the offshore wind projects where 80 + transition pieces are installed.

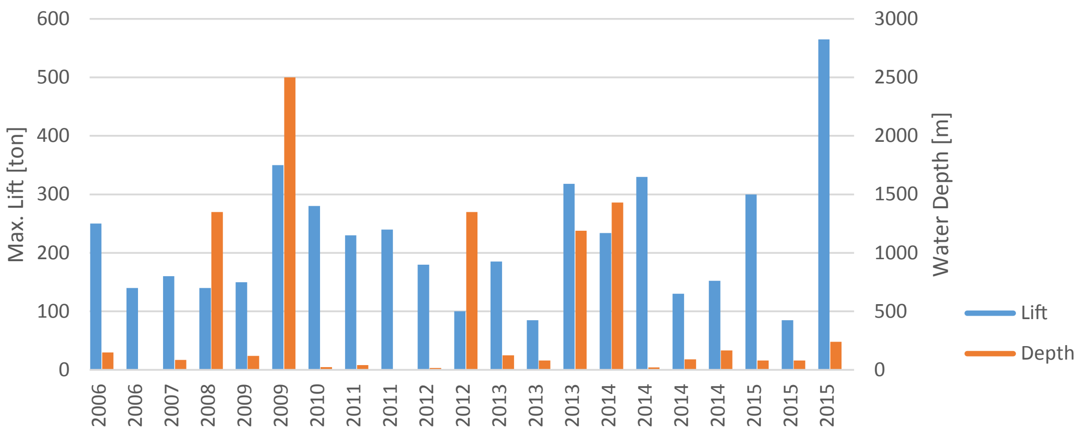


Figure 1.3: Track record time-line

The figure above shows the offshore projects for Jumbo based on the maximum offshore lift and the water depth. The biggest share of the projects done are lifts of less than 300 ton and in water depths up to 165 meters. When converting these numbers into a pie chart one can see that 80 % of the offshore lifts was lighter than 300 ton. See figure 1.4.

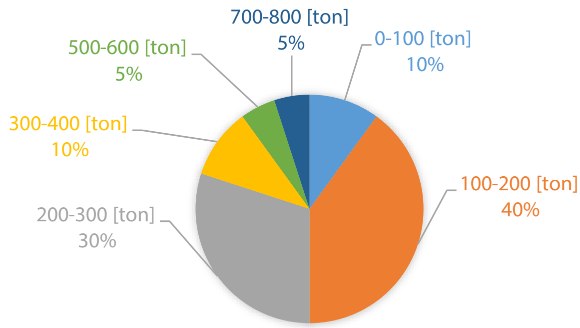


Figure 1.4: Max. lift distribution of
Jumbo's 20 offshore jobs

Future products and services

The track record history is a good indication of what Jumbo is capable of and what market they are competing in. However, offshore technology is continuously developing so the size and nature of the offshore installation may shift. This requires a market forecast and a suiting strategy. To see which kind of project are in Jumbo's interest and other future prospect will be discussed in section 2.1.

1.2 Problem definition

This section will explain where the request to do research on a faster lifting phase comes from. The moment that the object is lifted from deck until it reaches the final position overboard, is called the lifting phase. Also the scope of the research will be explained.

1.2.1 Background of the problem

The question to research the possibilities of the development of a faster anti-heeling solution comes from the Turkey-Cyprus water supply project. For this project Jumbo did the transport and installation of the subsea suspension system for the water pipeline. The scope of work comprised of the project management, engineering, load-out & seafastening, transport to site, on-site survey and installation (incl. pre-tensioning) of 126 anchor-thether-buoy assemblies of 220 ton each. The 126 subsea anchor-thether-buoy assemblies were installed in less than 75 days in up to 1400 meters water depths. The installation record was four assemblies per day.

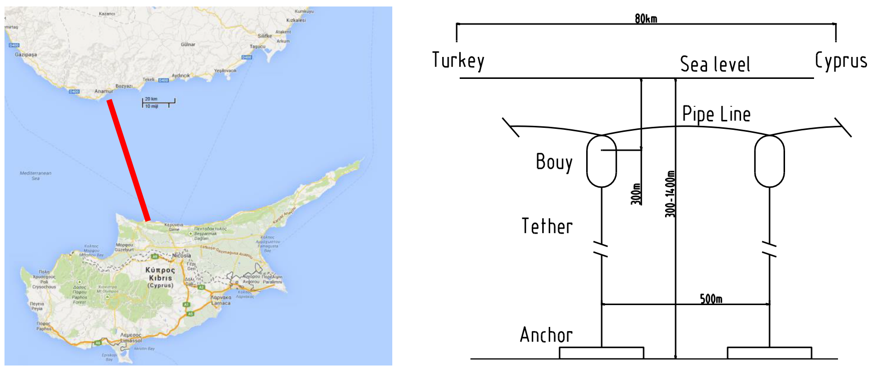


Figure 1.5: Left:Pipeline between Cyprus and Turkey
Right: Geometry subsea installation

Installation procedure

Without going into much detail, the procedure will be described briefly. First the anchor was lowered by the fore crane and the buoy followed with the aft crane. See figure 1.6. Then the ship is moved so that the anchor buoy and aft crane are in line. To remove the elongation out the tether the aft crane put a tension of 100 ton for 30 minutes on the assembly. The frame is released from the buoy and the process can start over.

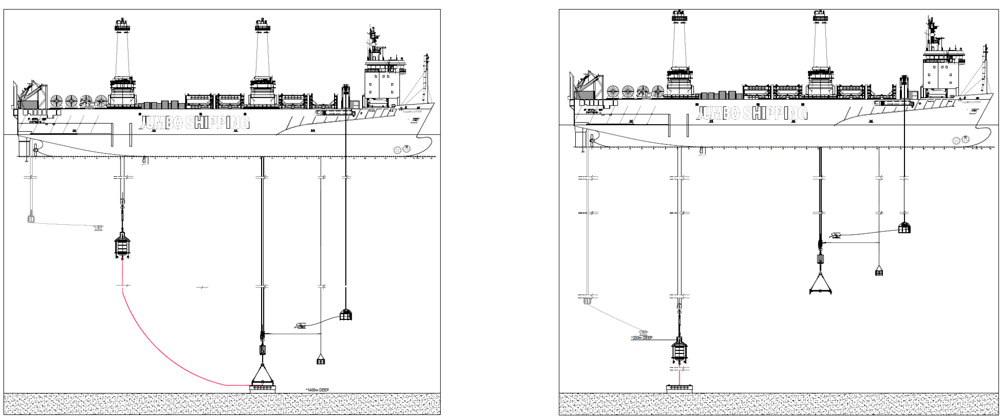


Figure 1.6: Water pipeline project installation procedure

Request for improvement

This project had 126 lifting repetitions. The lifting phase was expected to have taken a fair part of the total duration. Therefore the interest raised to research the possibilities of an improvement of the anti-heeling system. So for example if the duration of the 126 lifting operations could be reduced by thirty minutes each, the whole project could have potentially be reduced by 2,6 days.

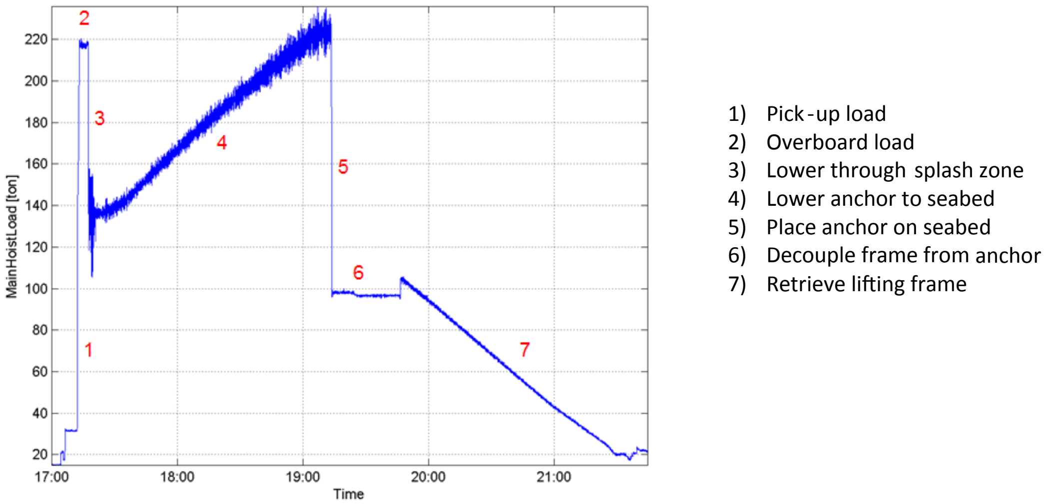


Figure 1.7: Fore crane hoist load read-out

When the read-out of the forward crane hoist load was analysed the outcome turned out somewhat different. Figure 1.7 shows a lifting operation from load pick-up to retrieving the lifting frame from the sea bed. The lifting phase, number 2 in figure 1.7, is only just over 5 minutes. If this is always the case, this project would be insignificant. The reason that in this case it only took 5 minutes is that transverse displacement of the load was relatively small and did not need much slewing motion of the crane. Figure 1.8 shows that the anchor was placed on the portside on deck and lifted overboard at portside.

Figure 1.8: Overboarding one of the anchors of the Turkey-Cyprus water pipeline

Figure 1.7 could suggest that a research to shorter hoisting phase would be a good investment for deep sea projects. But when different lifting operations with more varying crane motions and larger weights are considered, further researched showed that the current anti-heeling system and lifting procedures do leave room for improvement. This is elaborated upon in section 3.3. In consideration with the R&D department the decisioned has been made to stick to the plan and keep the focus on the anti-heeling system.

1.2.2 Scope of this thesis

This project is about the development of an anti-heeling solution for the Jumbo J-class. The current system installed on the J-class works but does not correspond with the crane motion well enough and causes delays. Developing a complete controlled anti-heeling system is not possible given the duration of this project. Therefore a realistic scope is determined. *The scope of this project is to find out the best suiting principle solution for the J-class, the required system capacities and how to realise it.* The property of "best suiting" will be determined by market conditions and technical possibilities. These will elaborated upon in the design specifications in section 4.1. The project will be lift system orientated because the variation of object to be lifted is to big to include the "load side of view".

1.2.3 Research questions

In order to evaluate the potential solution, both corporate as technical parameters should be analysed. Before technical solutions are developed, one should clarify under what conditions a shorter lifting phase is a contribution to the business. Eventually the decisions have to be made on quantitative requirements. But collection representative financial information is a study on its own. For that reason some requirements only based on general reasoning are included in the corporate part of the scope. The first research question is formulated as follows:

Under what conditions will the reduction of the lifting phase be worth the operational and capital expenditures?

When this is clear, the technical possibilities and constraints should be analysed. Therefore the second research question is:

How can the duration of the lifting phase of the Jumbo J-type be reduced?

The technical parameters for the development should researched and their sensitivity on the design should be determined. The answer of the question also include the concept generation and the technical properties of the developed system.

1.2.4 Objective

The main objectives of this thesis to do develop a beneficial solution to reduce the lifting phase are:

- Clarify the corporate design requirements
- Clarify the technical design parameters and their sensitivities
- Generate concepts
- Develop a solution
- Make a model of the most promising solution
- Draw conclusions and make recommendations

Chapter 2

Corporate Analysis

This chapter provides a summary of the research done for the thesis about the market Jumbo is operating in. To evaluate if an investment in a new or upgraded anti-heeling system is fertile, a sufficient understanding of the company and the market forecast is essential.

2.1 Offshore installation market

This section two market segments are analysed on macro economic level. The offshore wind and oil & gas are discussed separately. The civil market is not included in this research because oil, gas and wind projects are Jumbo's main drivers. The general market trends and developments are analysed and opportunities for Jumbo are discussed.

2.1.1 Offshore wind

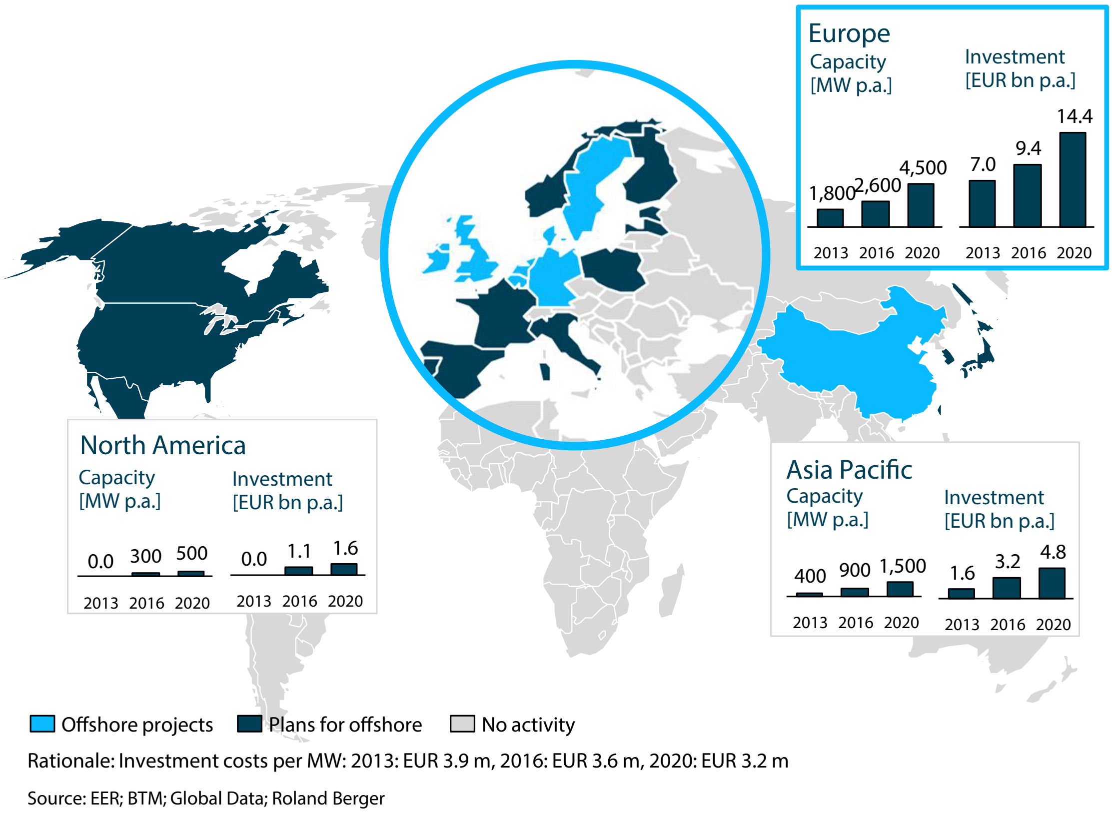


Figure 2.1: Global offshore wind market forecast of annual additions (Roland&Berger 2013)

In context of the growing environmental and social challenges, the industry of renewable energy technologies is growing fast. Offshore wind industry has shown a 5-year compound annual growth rate of 31% at the end of 2014 [10]. Europe has by far the biggest market share in terms of installed capacity of 90% by the end of 2014. But figure 2.1 shows that Asia Pacific will catch up and North America follows with lower levels.

Europe

Europe leads in the development of the offshore wind industry. Its capacity could triple from 8GW today to 23.5GW in 2020 [10]. The UK and Germany will have the biggest share to the main growth. France and the Netherlands are expected to have a fast deployment. See figure 2.2. These number are expected under the condition of political stability and the background of still recovering regulatory and financial stability.

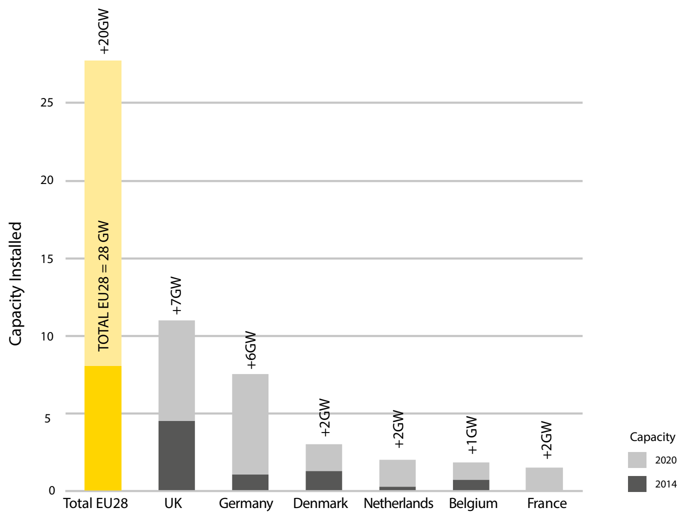


Figure 2.2: A breakdown of potential growth in 2020 (EY 2015 [10]).

Asia Pacific and North America

The opportunities for Jumbo in the offshore wind industry outside of Europe are relatively marginal. The plans of South Korea and Taiwan are lagging behind and India is still taken her first steps. China already realised offshore wind projects and there are more to come, but China is a protected market. The Jones act in the USA decreases the possibilities for Jumbo in North America. A potential market for Jumbo could be Japan. Japan has the ambition to install 6 GW of offshore wind capacity by the year 2030. But water depth close to the coastline increase rapidly to 200 meter. Therefore Japan is interested in floating wind turbines. Floating wind turbines also need mooring system, which installation Jumbo also has experience in.

Levelized cost of electricity and cost reduction

As a competitive energy source on the market, offshore wind is still dependant on governmental subsidies. Even compared to other renewable energy technologies offshore wind is expensive. See figure 2.3 where the cost of electricity in Euro's per kWh (LCoE) per energy source is shown for the German market. To be a viable option in the long run, the energy production cost of offshore wind electricity must be reduced to be competitive with other sources.

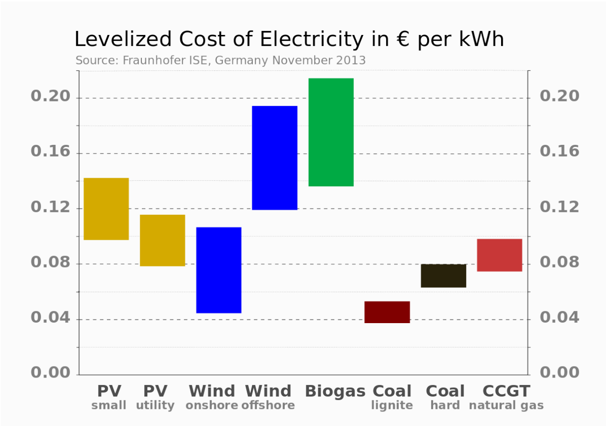


Figure 2.3: LCOE of renewable energy technologies and conventional power plants at locations in Germany in 2013 [8].

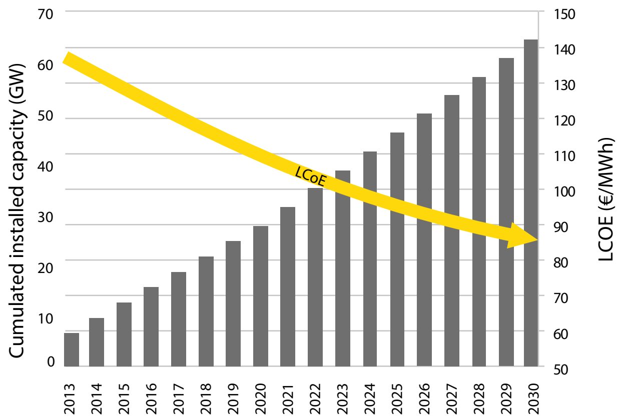


Figure 2.4: Evolution of the LCoE according to the cumulated offshore wind capacity installed. EY 2015 [10].

Large-scale deployment is beneficial for the installation market. This is not only dependant on cost reduction but also how fast these cost can be reduced. Against a backdrop of continuous growth and with the market realizing its technological, supply chain and investment goals, a evolution of the LCoE as in figure 2.4 can be realised. In that trend the LCoE could go down to €90/MWh by 2030.

Installation

Generally speaking the installation of a offshore wind turbine is devided in three stages:

1. Installation of foundation or substructure
2. Installation of transition piece
3. installation of mast, nacelle and blades

The possible opportunities for Jumbo in the offshore wind industry are in the installation phase. The floating Jumbo vessels are suitable to installation of the monopiles but these operations have small tolerances. For installing the mast, nacelle and blades the cranes of Jumbo do not reach high enough. For water depths up to 60m, Jack-up vessels are often preferred here. Installation of transition pieces is no problem for Jumbo.

The installation cost is accountable for only 11% of the total costs. See figure 2.5. Jumbo's potential involvement in the projects is relatively small, because other segments are already professionalized by other companies. Opportunities can ly in being a valuable partner in optimizing logistics and developing new installation concepts. To see where Jumbo fits in, the market trends are analysed.

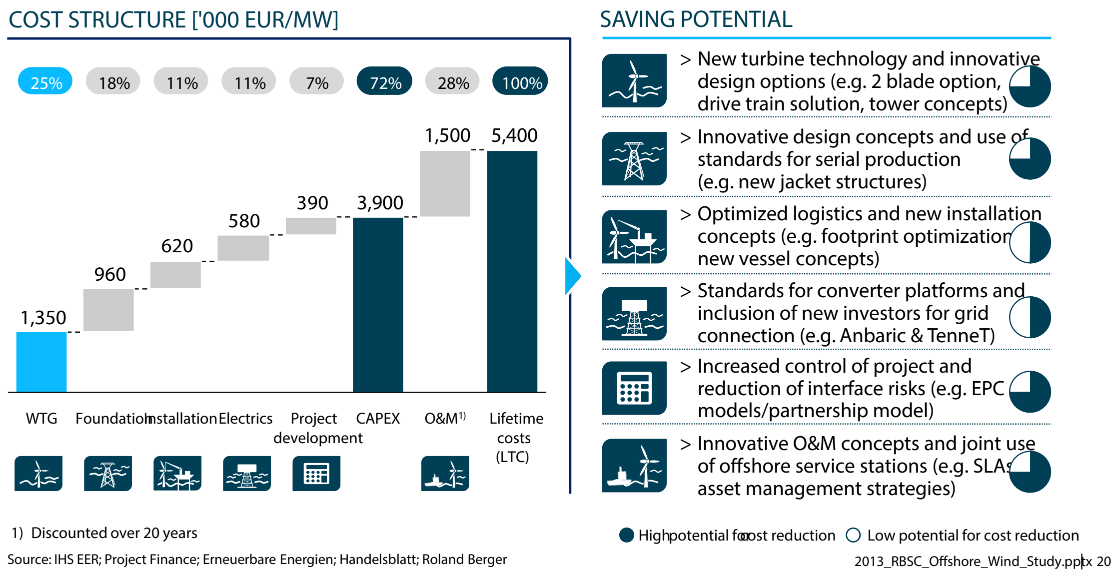


Figure 2.5: Cost & saving potential (Roland Berger 2013)

Trends

The future of offshore wind is offering great opportunities and various players in the industry develop new generations of vessels specifically designed for offshore wind. The trend goes towards increasing turbine size and water depths. See figure 2.6. For the Westermost Rough offshore wind farm transition pieces of the 6MW turbines are 367 ton [6]. Both turbines and TP's will increase in size in the future. Larger cranes and deeper operational capabilities will be required. As an example, the new type of jack-up wind turbine installation vessels can work in water depths up to 60 meter, have a 1200 ton crane and can carry twelve 3.6MT wind turbine generators. See figure 2.7 for an impression.

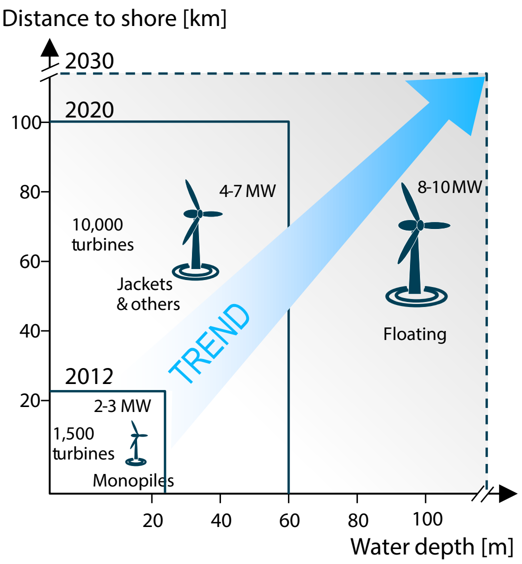


Figure 2.6: Trends

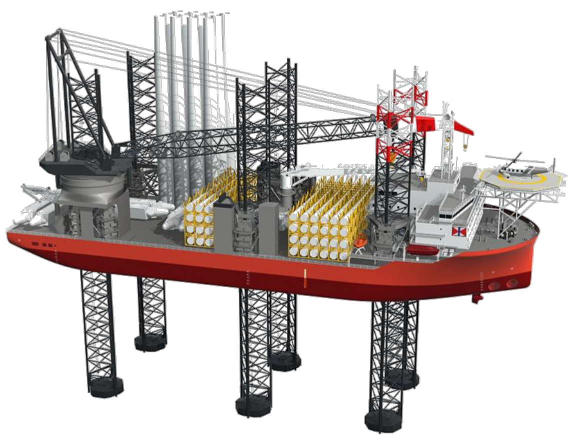


Figure 2.7: Samsung SPO WTIV

As the supply of WTIV's increases, the transition pieces typically installed by the same vessels as the turbines itself. The different types of substructures and their market share is given in figures 2.8 and 2.9. The market share of monopiles is expected to be 75 % toward 2020. But the designs are evolving to new installation methods are developed [5]. The combination of the supply of TWIV's and changing designs will intensify the competition for Jumbo to play a role in the monopile wind turbine installation market.

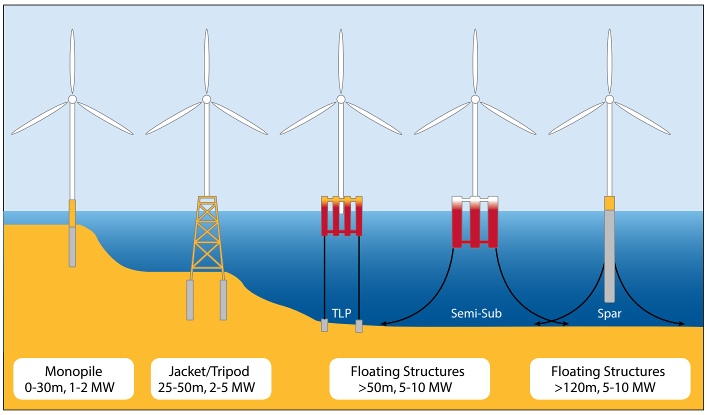


Figure 2.8: Offshore wind foundation

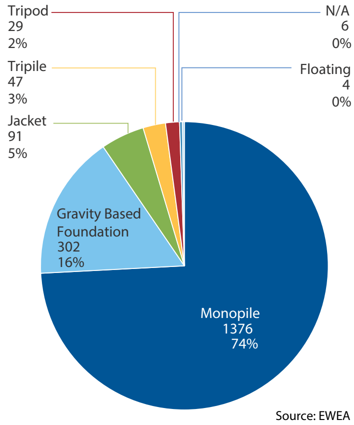


Figure 2.9: Share of substructure types for online wind farms , end 2012 (UNITS)

Jackets will increase their share due to their flexibility and low weight, comparing with the amount of steel is used for monopiles. Jackets become commercially worthwhile from water depths of 35 meter and deeper [20]. Jumbo's vessel are suitable for this if the crane height is increased. In the past fly-jibs were temporary attached to obtain extra height. But these also have their limits.

An other trend is that construction companies are, as their experience increases, expanding their scopes of work and offering turn key solutions or EPCI contracts. Jumbo is not one of these companies so this will reduce the potential opportunities to be involved.

2.1.2 Offshore oil and gas

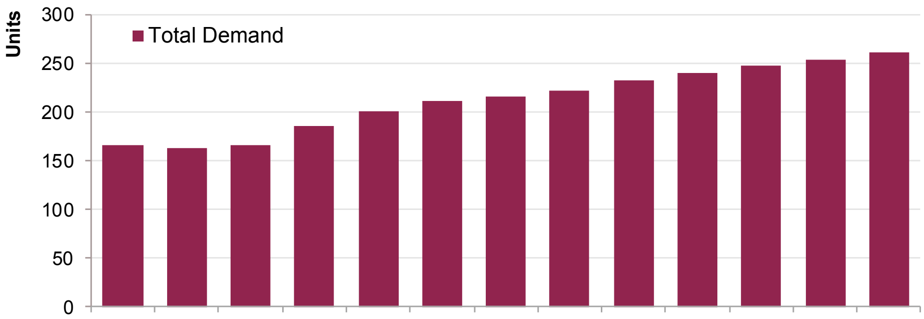
As mentioned in 1.1.2, Jumbo's track record contains mostly oil and related installations. This section will discuss the potential market in the oil and gas segment for Jumbo. The demand of offshore installations strongly depends on the oil price. As the oil price dropped tremendously in 2015, this section will discuss the expected development on the offshore oil and gas market related to Jumbo's interest.

Oil price

The drop of the oil price is driven by the inclining demand from China, North America and Europe and the global overproduction of oil. The surges of shale oil production in the USA and the decision of the OPEC to protect its market share and keep the production up are the major causes. But as global population and GDP will continue to increase, the consumption of hydrocarbons will also increase. The supply and demand will level again and the oil prices will go up in the long term. Exploration & production companies take a long term view and will merely delay rather then cancel projects. As more reserves in deeper waters are unlocked, the demand for floating platforms and subsea infrastructure will increase.

FPSO

The growth of the FPSO market is a good indicator of mooring and subsea related potential installation jobs ahead. The FPSO market is predominantly project driven and according to Laurent Daniel (Organisation for Economic Co-operation and Development 2014) the global demand of FPSO is expected to increase by 129 units over the 2014-2025 period [4].



Units	2013	2014	2015	2016	2017	2018	2019	2020	2021	2022	2023	2024	2025
Africa	44	42	41	44	48	53	56	59	61	62	63	65	67
Asia	41	43	43	47	48	48	47	47	51	52	53	55	56
Australasia	12	12	12	13	13	12	13	11	11	12	12	12	13
Eastern Europe & FSU	0	0	0	0	0	0	0	0	0	0	0	0	0
Latin America	43	42	43	53	58	62	64	72	72	74	76	78	80
Middle East	0	0	0	0	2	2	2	2	2	2	2	2	2
North America	4	4	4	5	5	5	5	4	5	5	6	7	7
Western Europe	23	20	23	24	27	30	29	28	32	34	35	36	37
Total	167	163	166	186	201	212	216	223	233	241	248	254	262

Figure 2.10: FPSO market growth expectadtions (Douglas Westwood 2014)

As figure 2.10 shows, Latin America and (West) Africa will account the largest demand of FPSO's. Projects in Latin America are mostly in Brazil, where project execution for Petrobras demands a 70 % local content.

TLP

Installation work related to TLP's is comparable with that of FPSO's. Douglas Westwood expect an increase of 15 TLP units over the 2014-2025 period. North America (52%), Africa (14%) and Asia (20%) account for the majority of demand over the forecast period [4]. Projects typically involve EPC contractors as Technip and McDermott and Hyundai Heavy Industries. Therefore Jumbo's potential lies in partnership with these contractors.

Subsea

The period of high oil prices enabled investments in deep water technology. Also previous marginal fields could be developed. But the oil price dropped and many project are being deffered. Still a growth of subsea infrastructure expenditure is expected by Douglas-Westwood (August 2015). Main drivers are the move to deep water, marginal and remote field development and fields in harsh environments . And due to revisions of field developments man projects are developed as tiebacks rather than with their own platform [26].

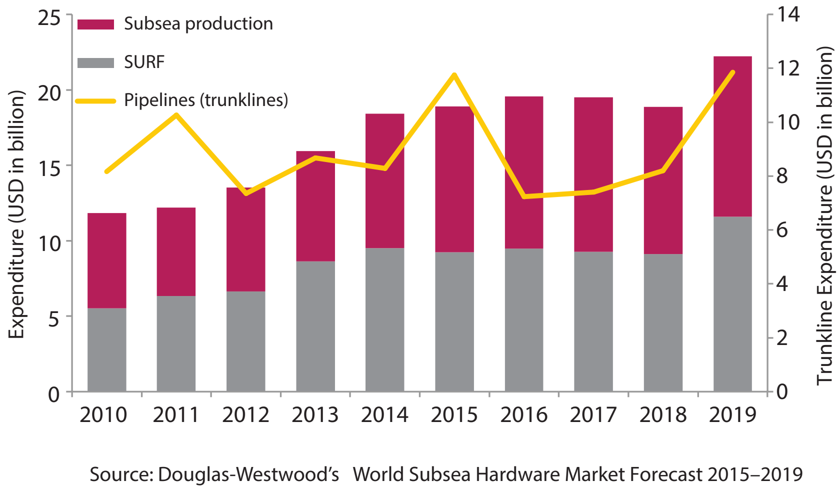


Figure 2.11: Subsea production hardware; subsea umbilical, riser, and flowline (SURF); and pipeline expenditure, 2010 - 2019

Subsea vessel oversupply

Over the past 5 years there was a high demand of high-specification vessels, driven by the high oil prices. This caused a corresponding increase of day rates. As a result the most recent build cycle has been characterised by typically larger and high-specification vessels ready for the deepwater challenges. Now the oil companies have smaller budgets, day rates are decreasing. And under the current market conditions this trend is likely to continue. Although this resulted in a decline of vessels ordered, the oversupply will continue until the current order book is worked off [1]. Despite the current conditions the offshore investment will remain strong as the global demand of energy increases. The day rates however are unlikely to meet the levels of 5 years ago and could decline even more before the rates stabilize again.

Trends

The increasing volume of subsea developments and installations is coupled with a high volume of inspection repair and maintenance activities. Due to increasing depths and increasing size and weight of subsea infrastructures, there is a trend towards larger crane capacity and deck space.

2.2 Business Development

This section will discuss the business development process to foresee the possible consequences of an improvement of capabilities of Jumbo's fleet. As detailed information is for internal use only, the process is briefly explained.

2.2.1 Business development process of Jumbo Offshore

The process from analysis to tender goes through four stages. See figure 2.12.

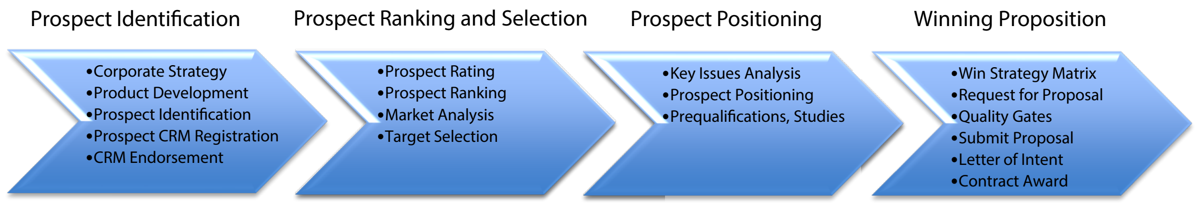


Figure 2.12: Sales and business development process

Phase 1: Prospect identification

The objective of the prospect identification stage is to obtain a potential probable & prospect list. This list is based on information available from various prospects (fields) and from costumer relations. The prospects on the list should be in line with Jumbo's strategy and product development.

Phase 2: Prospect ranking and selection

Before the selection process starts, the prospects are ranked first. First the market is segmented and the volume/revenue is estimated. Simultaneously the trends are analysed. Then, a target list with a back-up list is formed. The sorting of this target list is based on the probability the project will be executed, the probability Jumbo will get the prospect, the estimated margin and project timing. Also the synergy between projects is taken into account. And then the selection takes place. This selection is based on market intelligences, area condition, local issues, need for partnering and corporate governance.

Phase 3: Prospect positioning

When the targets are selected, Jumbo's capacities are measured against the competitors and a partner plan is formed if needed. Then a key issue analysis is performed. Pre-qualifications are studied and legal matters are checked. If all the opportunities and issues are worked out, a action plan for the winning strategy is created.

Phase 4: Winning proposition

First the invitation to tender is obtained and reviewed. Then decision to bid or not to bid is made. If Jumbo is going to bid, the tender team en budget is determined. A technical proposal is prepared by the engineering department. And the commerce department will take it to the commercial deliverables to the potential client.

Effect of a shorter lifting phase on the business development process

A shorter lifting phase can effect on the prospect ranking and positioning in phase 2 and 3. This is only the case if it significant reduces the duration on the execution of the project. So no major differences in the business development are expected.

2.2.2 Vessel adjustment

There is no standardised development protocol for vessel improvements or investments. Suggestions from the field or opportunities that are picked-up by the R&D department are evaluated if they might be in Jumbo's interest. A selection of these is continuously in development by the R&D or engineering department. When the results are promising the board will eventually decide whether a budget will be available or not. Jumbo prefers to earn a vessel investment back within the same project. Or if a vessel needs a special adjustment for a projects, Jumbo will look for opportunities involve the client in the investment.

For complete new vessels it works different. First the market position is determined and then a vessel is designed corresponding to that market position.

2.2.3 Strategy

Jumbo can be divided in a shipping unit and an offshore unit. The shipping unit focuses on transport of extraordinary loads and the offshore unit on the services mentioned in 1.1.2. For most offshore project Jumbo was involved as subcontractor. Over time Jumbo gained more experience in deep water projects and now offers fully integrated EPIC packages for mooring system installations. This growth is all plan of the strategy to become a main contractor for projects corresponding to Jumbo's abilities as figure 2.13 illustrates.

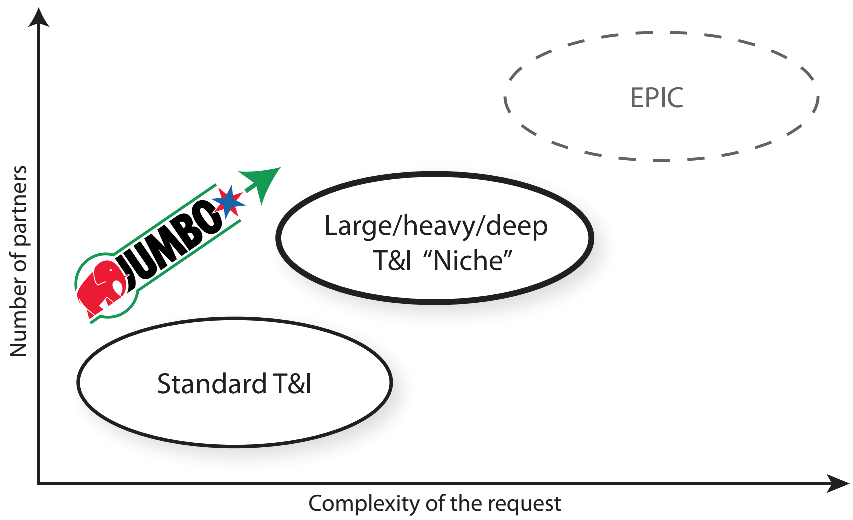


Figure 2.13: Project category development

2.3 Financial analysis

A new solution for the heeling problem becomes interesting when it is financially beneficial. To express the saved by a shorter lifting phase, the time saving is related to the vessel day rate.

$$Financial\ saving = \frac{time\ saving}{project\ time} \cdot day\ rate \tag{2.1}$$

The time saving is not directly related as equation 2.1 suggests, but is an indication of the order of savings to deal with.

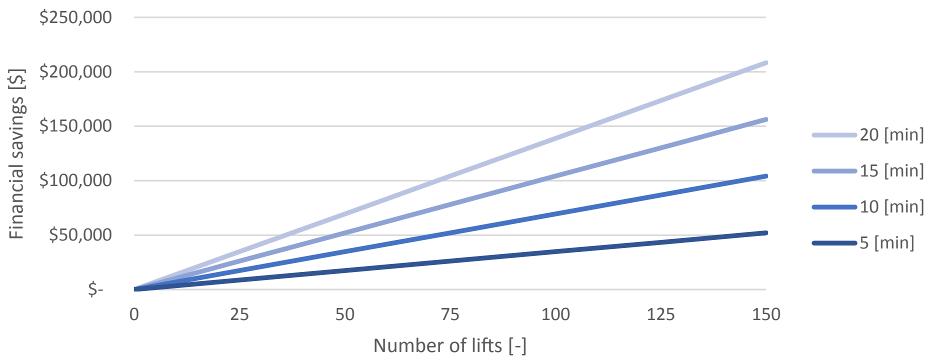


Figure 2.14: Potential financial savings related to the number of lifts and time saving per lift

In graph 2.14 the potential financial savings are shown when a single lifting phase is shorted with 5, 10, 15 or 20 minutes. This is related to the number of lifts involved in the project. The day rates of Jumbo are not public, so an estimation of a day rate of $ 100,000 has been taken. This is low for a 1800t offshore heavy lift vessel, but realistic under current market conditions.

When the lifting phase can be reduces by 5 minutes, the saving will be around $ 350 per lifting operation. If you directly link time to day rate, which is not realistic. But $ 350 is a small saving. Even when the project involves 100 repetitions the total savings will just be $35,000.

2.3.1 Cost of energy

To evaluate the energy consumption, the energy can be expressed in dollars. The fuel consumption of the J-class is about 200 grams per 1 kWh or 3.6 MJ. The current price of 1 ton of fuel dropped to $ 125 on the 6th of January 2016, see figure 2.15. The price of energy can is expressed as $125 \cdot 0.2 \,/\, 3.6 = 0.007$ $/MJ.

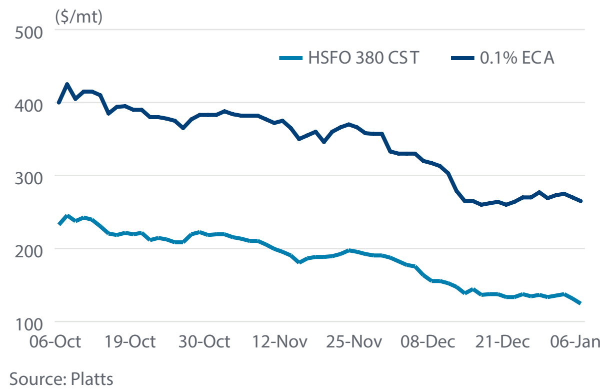


Figure 2.15: Bunker prices Rotterdam [19]

2.3.2 Contracts

A reduction of the project duration does not have to be a necessarily financial beneficial to Jumbo Maritime. This strongly depends on the type of contract between Jumbo Maritime and the client.

Mobilisation / demobilisation fee

In case of a mobilisation / demobilisation fee and a fixed day-rate, the risk of delay is completely carried by the client. When the total offshore time is reduced, the total cost for the client are reduced. This results in a lower revenue for Jumbo Maritime, because Jumbo's services are hired for a shorter period. An potential benefit for Jumbo is this case of shorter project duration, is a positive contribution to their reputation. This can have a positive effect on bagging new projects.

Lump Sum

In case of a Lump Sum contract, the total price of the execution of the project is agreed between Jumbo and the client. When the duration of the project is reduced, the cost for Jumbo can potentially be reduced. A potential reduction by means that the operational costs, like personnel and energy consumption do not exceed the benefits of saved offshore time.

2.4 Conclusions corporate analysis

The most important findings of the corporate analysis will be given in bullet points below

- 80% of the offshore lifts executed by Jumbo is below 300 ton.
- Offshore wind market is still growing. Mainly in Europe.
- Due to investments in specialized installation vessel of the competition, Jumbo's involvement in this market is limited.
- Opportunities in the offshore wind market for Jumbo could be installation of TP's. In the future Jumbo can be involved in the installation of floating offshore wind structures.
- TP's for 6MW turbines weigh up to 400 ton. Larger turbines require heavier TP's
- Oil price is low but the energy demand will increase on the long run and therefore the expenditures in offshore oil and gas will also increase.
- Opportunities for Jumbo in the offshore oil and gas market are installation of (deep sea) mooring systems and subsea components.
- The potential impact of a faster lifting phase will hardly have an effect on Jumbo's business development and prospect positioning.
- Due to small potential time savings, the allowable expenditures on a new solution are very limited.

Chapter 3

Technical Analysis

This chapter provides a summary of the technical analysis of the project. The analysis includes ship hydrostatics, working condition for the J-class, properties of the current system and a brief analysis of anti-heeling systems available on the market.

3.1 Ship hydrostatics

To have a better understanding of the phenomenon heeling this section will give a basic explanation about ship hydrostatic. The movements of the vessel that disturb the equilibrium state were assumed to be relatively slow and the dynamic effect are not included in the concept development calculations. The system dynamics can have a significant effect on the roll angle and are therefore included in the model of the selected solution. In this section, the relevant principles, definitions, assumptions and calculations concerning ship hydrostatics will be explained.

3.1.1 General assumptions

For the ship hydrostatics the textbooks Basic Ship Hydrostatics [2] and Offshore Hydromechanics [14] are used as reference material. Concerning problem related to basic ship hydrostatics we use the following assumptions:

1. the water is incompressible;
2. viscosity plays no role;
3. surface tension plays no role;
4. the water surface is plane;
5. the floating bodies are perfectly rigid.

For the first assumption, water is practically incompressible for the depths concerning this thesis. As the heeling motion is a relatively slow movement and only hydrostatics are considerer, the viscosity does not play a role. The vessel is a large structure, so surface tension does not have a influence. Although the water surface will never be plane, the fourth assumption is very helpful to derive general results. The last assumption allows to work with concentrated forces.

Definition of motions

Ship motions can be build up from six basic motions. See figure 3.1.

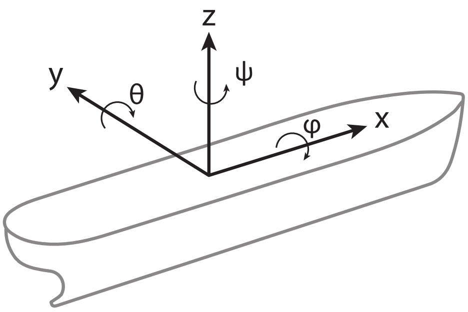


Figure 3.1: Definition of ship motion in six degrees of freedom

These motions are defined by translations of the ship's centre of gravity:

- Surge in x-direction (X)
- Sway in y-direction (Y)
- Heave in z-direction (Z)

And by rotations about the axes:

- Roll about the x-axis (ϕ)
- Pitch about the y-axis (ψ)
- Yaw about the z-axis (Θ)

3.1.2 Conditions of equilibrium

Heeling is a combination of the forces and moments acting on the floating body. It is a rotation around the x-axis so only the equilibria in the y,z-plane are considered.

Forces

For a structure to float, the upward force needs to be equal to the weight of the structure. In case of Figure 3.2 this means $B = G$. This upward force is called buoyancy force and act on the centre of the displaced volume.

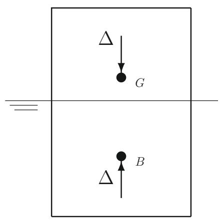


Figure 3.2: Archimedus

According to Archimedes, the weight of the volume of water displaced by the structure is equal to the weight of the structure.

$$\Delta = W \tag{3.1}$$

And to express the buoyancy force:

$$F_{\nabla} = \nabla . \rho_{water} . g \tag{3.2}$$

Moments

For a structure to float at rest, the sum of the moments of all forces must be zero. Considering only buoyancy and gravity forces, the centre of buoyancy and the centre of gravity must be on the same vertical line. This is called Stevin's law.

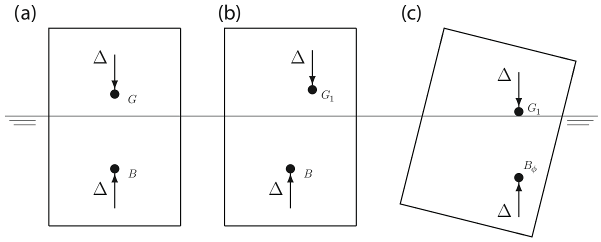


Figure 3.3: Stevin's Law

If the centre of gravity moves, the centre of buoyancy moves as well un till the two centres are on the same vertical line again.

3.1.3 Stability

A floating structure is called stable if it returns to its initial position after a small change of position is cause by some force or moment. If a symmetric ship is heeled to the starboard with a small angle ϕ, a volume submerges at starboard and a equal volume emerges at port. The centre of buoyancy moves to starboard, see figure 3.4. When the weights are fixed, the centre of gravity stays on the same position. As the forces B and G are perpendicular to the water line, they are not in a vertical line any more and therefore create a moment. This restoring moment as it is called, rotated the ship towards port and returns the ship back to her initial upright position.

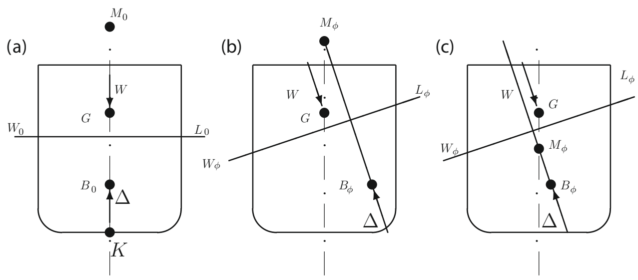


Figure 3.4: Initial stability

In some situations when a ship is heeled, G and B do not form a restoring moment but a additional heeling moment, see figure 3.4. This will create an unstable situation and the ship will capsize.

Metacentre

To determine if a ship is stable the concept of metacentre is used. The metacentre is a imaginary point on the centreline of the vessel. This centreline is a trace of the port-starboard symmetry plane. This centreline is the vertical line where the buoyancy and gravity forces are act along when the ship's moments are in equilibrium. When the ship heels, the centre of buoyancy moves and a new line of action of the buoyancy force perpendicular to the waterline intersects with the centreline of the ship. This point is the the metacentre M_ϕ. With this method the equilibrium of the ship can be tested on stability. If the metacentre is situated above the centre of gravity, the ship is stable.

Metacentric height

The ship is stable when the metacentre is above the centre of gravity. The distance between these points is called metacentric height and can also be written as $\overline{GM}$ and expressed as:

$$\overline{GM} = \overline{KB} + \overline{BM} - \overline{KG} \tag{3.3}$$

Where B is the centre of buoyancy and K is the origin of the z-coordinate. Conventionally K is chosen as the keel of the ship because this is the lowest point of the ship. Condition of stability is expressed as:

$$\overline{GM} > 0 \tag{3.4}$$

Metacentric radius

The metacentric radius is the distance between the metacentre and the centre of buoyancy. Given equation 3.3 one can see that $\overline{BM}$, the metacentric radius, plays an important role. $\overline{BM}$ is the ratio of the waterplane moment of inertia, about the x-axis, to the volume of displacement. There are three 'methods' to determine the $\overline{BM}$.

1. Initial stability: wall sided ship and sideways shift of B
2. Scribanti: wall sided ship and B shifts sideways and up
3. 'Real ship': determine GZ curve for actual shape

Small heeling angles (smaller than 5 degrees) and a wall sided ship are considered in this thesis. Therefore the method of initial stability is used and the for $\overline{BM}$ is:

$$\overline{BM} = \frac{I}{\nabla} \tag{3.5}$$

If the ship is relatively wide for its displacement it will have a large metacentric radius. This can results in a larger metacentric height.

Restoring moment

When the metacentre is positive, the ship will return in its initial position when the heeling occurs. This happens under condition that the centre of gravity does not change and the deck will not submerge. As mention earlier this section, the centre of buoyancy moves when the ships heels. This movement ofB creates a distance $\overline{GZ}$ between the lines of action of G and B_ϕ. See figure 3.5. This distance $\overline{GZ}$ is called the *righting arm*

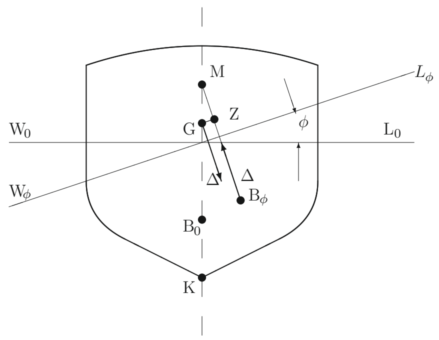


Figure 3.5: Righting arm

As G and B_ϕ are both forces, a moment is created that will return the ship in its initial position. This is called the *restoring moment*. The magnitude of this force is given by :

$$M_{Restoring} = \Delta \cdot \overline{GZ} \tag{3.6}$$

Stability curve

The static relation of the heeling angle and the righting arm of a ship is usually given in a static-stability curve. This relation changes when the loading condition of the ship changes. For small angles the relation between the angles of heel and the righting arm $\overline{GZ}$ can be considered linear. As can be seen in figure 3.6, up to eight degrees the GM line follows the stability curve.

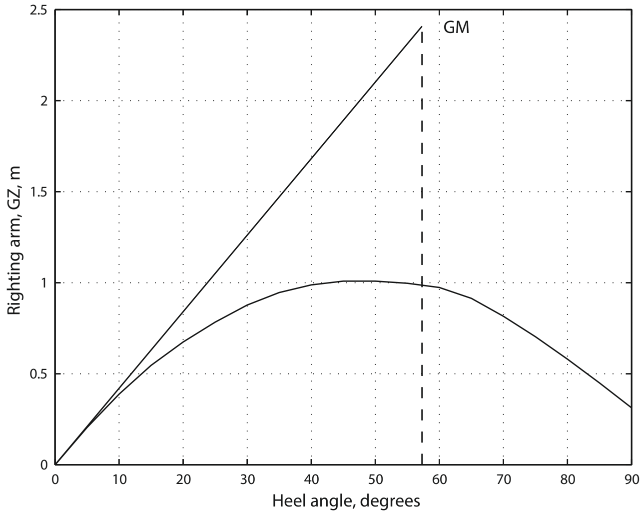


Figure 3.6: Example of a static stability curve

The expression for the righting arm $\overline{GZ}$ for small angles can be assumed to be:

$$\overline{GZ} = \overline{GM} \cdot sin(\phi) \tag{3.7}$$

This relation (3.7) can also be used to calculate small heeling angles if the heeling moment is known. Assume that the ship is stable and in equilibrium of moments so:

$$M_{Heeling} + M_{Restoring} = 0 \tag{3.8}$$

When the equations 3.6, 3.7 and 3.8 are combined, an estimation of the heeling angle can be found as given in equation 3.9.

$$\phi = \arcsin\left(-\frac{M_{Heeling}}{\Delta \cdot \overline{GM}}\right) \tag{3.9}$$

Where $M_{Heeling}$ is result of the summation of all the moments acting on the ship causing a rotation around the x-axis: $M_{Heeling} = \sum_{i=1}^{n} M_i$.

3.1.4 Effects causing heeling

This subsection will discuss the effects that can significant influence the heeling angle in relation with the offshore installation activities of the J-class. Lifting the object from deck and move it overboard is basically transversely displacing a hanging load. Although environmental loads are not considered for this analysis, wind loads are briefly discussed. Also the possible free surface effect of the water in the ballast tanks is introduced.

Hanging loads

Before installation the ship is at her initial stable upright position. The object rests at deck and the gravitational load of the object acts in its centre of gravity. When the crane lift the object, the gravitational force of the object jumps to the crane tip. This means that the centre of gravity of the whole system jumps as well and the $\overline{GM}$ reduces. Therefore the initial-stability calculations must be corrected. The corrected metacentric height is called the effective metacentric height and is expressed as:

$$\overline{GM}_{eff} = \overline{GM} - \frac{h \cdot m}{\Delta} \tag{3.10}$$

Where h is the distance the load is raised, m the mass of the load and Δ is the mass of the displaced water volume. Ones the load is lifted and a moment causes the ship to heel, the hanging mass moves transversely by $r.tan(\phi)$.

To maintain the initial moment equilibrium it is important that the tip of the crane is in vertical line with the centre of gravity of the object. If not, the instant increase of the heeling moment and the reduction of the $\overline{GM}$ can load to a uncontrollable situation.

Transversely displaced loads

When the load is hanging on the crane, it can be moved transversely. Moving the load result in of new position of the centre of gravity. The distance between the initial G_0 and the new G_1 is equal to:

$$\overline{G_0 G_1} = \frac{d \cdot m}{\Delta} \tag{3.11}$$

Where d is the transverse distance the load is displaced.

This shift of centre of gravity creates a heeling moment and the ship will heel. Due to the shift of the centre of gravity the righting arm is reduced to an effective value:

$$\overline{GZ_{eff}} = \overline{GZ} - \frac{d \cdot m}{\Delta} cos(\phi) \tag{3.12}$$

This effect is similar to a vertical shift of the centre of gravity to a higher position.

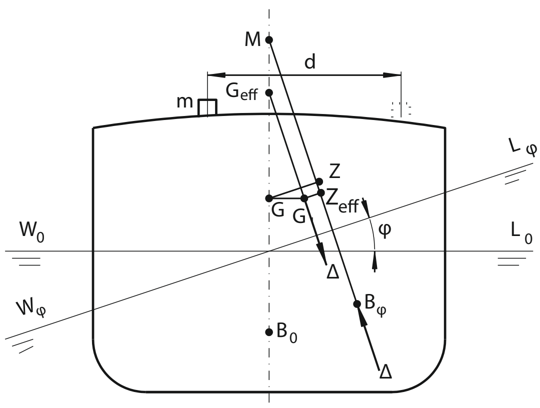


Figure 3.7: Destabilized by a transversely moved load

Wind loads

When the ship has a large above water ship surface and/or very large cargo surface, beam wind can cause a large heeling moment.

$$M_{wind} = F_{wind} \cdot l_{wind}(\phi) \tag{3.13}$$

Free surfaces effect of water in ballast tanks

The vessel has a great number of tanks with liquids. Think of fuel, ballast water, fresh water, lubricating oils, etc. All together this make a significant amount of free surface. The moving liquids in the tanks cause a reduction of stability. This reduction is caused by a inclination of the effective metacentric height and effective righting arm. The has multiple tanks. These has to be summed up for the calculations of the effective metacentric height

$$\overline{GM_{eff}} = \overline{GM} - \frac{\sum_{k=1}^{n} \cdot \rho_k \cdot i_{Bk}}{\Delta} \tag{3.14}$$

and the effective righting arm

$$\overline{GZ_{eff}} = \overline{GZ} - \frac{\sum_{k=1}^{n} \cdot \rho_k \cdot i_{Bk}}{\Delta} \cdot sin(\phi) \tag{3.15}$$

Where n is the number of tanks, i_B the moment of inertia of the liquid surface and ρ_k the liquid density.

The heeling angle produced by the inclination of the liquid surface is

$$M_l = \sum_{k=1}^{n} \cdot \rho_k \cdot i_{Bk} \cdot tan(\phi) \tag{3.16}$$

3.1.5 Limitations of the calculations

The calculations used are rough approximation of reality. Heeling is not just a static problem. The ship is always exposed to environmental loads. Water opposes the ship motions with forces that depend on amplitude, speed and acceleration of the ship motions.

3.2 Workability

The workability of a heavy lift vessel is the percentage of time that offshore operations can be executed in certain areas. The workability depends on three components:

1. Seakeeping behaviour of the vessel under a certain loading condition
2. Environmental conditions of the working area
3. Criteria to operate safely

A complete workability study is essential but not included in the scope of this project. However, it is important to know what possible effects the anti-heeling solution could have on the workability. Therefore the three components mentioned above are briefly discussed.

3.2.1 Seakeeping behaviour

The seakeeping behaviour of a ship can be assessed by its amplitudes of motion in certain sea states. The natural period, damping level and RAO's are the most important parameters. When the ship is oscillated with periods close to its natural period, relatively large motions are likely to occur. Figures 3.8 and 3.9 show the significant double amplitudes for the J-class for 165 degree headings for two different load cases. The load cases are elaborated in appendix A.

Safetrans 165 LC1							SDA Roll unstab [deg]									
		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
HS		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
0	0.25	0.00	0.01	0.02	0.02	0.02	0.02	0.04	0.06	0.11	0.16	0.23	0.31	0.41	0.45	0.40
0.25	0.5	0.01	0.02	0.03	0.05	0.05	0.05	0.07	0.13	0.22	0.33	0.45	0.62	0.82	0.89	0.79
0.5	0.75	0.01	0.03	0.05	0.07	0.07	0.07	0.11	0.19	0.33	0.49	0.68	0.93	1.23	1.34	1.19
0.75	1	0.02	0.04	0.07	0.09	0.10	0.10	0.14	0.26	0.44	0.66	0.91	1.24	1.64	1.79	1.59
1	1.25	0.02	0.05	0.08	0.12	0.12	0.12	0.18	0.32	0.55	0.82	1.13	1.56	2.05	2.23	1.99
1.25	1.5	0.02	0.05	0.10	0.14	0.15	0.15	0.21	0.38	0.66	0.99	1.36	1.87	2.45	2.68	2.38
1.5	1.75	0.03	0.06	0.12	0.16	0.17	0.17	0.25	0.45	0.76	1.15	1.59	2.16	2.78	3.02	2.71
1.75	2	0.03	0.07	0.13	0.19	0.19	0.20	0.29	0.51	0.87	1.31	1.81	2.45	3.11	3.35	3.04
2	2.25	0.03	0.08	0.15	0.21	0.22	0.22	0.32	0.57	0.98	1.48	2.04	2.74	3.44	3.69	3.36
2.25	2.5	0.04	0.09	0.17	0.23	0.24	0.25	0.36	0.64	1.09	1.64	2.27	3.03	3.77	4.03	3.69

Figure 3.8: SDA values for Roll for a $\overline{GM}_{eff}$ of 2.25 meter with a heading of 165 degrees [24]

Safetrans 165 LC2							SDA Roll unstab [deg]									
		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
HS		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
0	0.25	0.00	0.01	0.02	0.03	0.03	0.06	0.12	0.20	0.30	0.44	0.52	0.47	0.38	0.32	0.29
0.25	0.5	0.01	0.02	0.04	0.05	0.06	0.12	0.24	0.40	0.61	0.88	1.04	0.94	0.75	0.64	0.58
0.5	0.75	0.01	0.03	0.06	0.08	0.10	0.18	0.36	0.61	0.91	1.31	1.55	1.40	1.13	0.96	0.87
0.75	1	0.02	0.04	0.07	0.10	0.13	0.24	0.48	0.81	1.21	1.75	2.05	1.86	1.51	1.28	1.16
1	1.25	0.02	0.05	0.09	0.13	0.16	0.30	0.61	1.01	1.52	2.18	2.56	2.32	1.88	1.60	1.46
1.25	1.5	0.02	0.06	0.11	0.16	0.19	0.36	0.73	1.21	1.82	2.61	3.06	2.78	2.26	1.92	1.75
1.5	1.75	0.03	0.07	0.13	0.18	0.23	0.42	0.85	1.41	2.10	2.96	3.44	3.14	2.59	2.23	2.03
1.75	2	0.03	0.08	0.15	0.21	0.26	0.47	0.97	1.62	2.39	3.31	3.81	3.50	2.92	2.53	2.32
2	2.25	0.04	0.09	0.17	0.24	0.29	0.53	1.09	1.82	2.67	3.65	4.18	3.86	3.25	2.83	2.61
2.25	2.5	0.04	0.10	0.18	0.26	0.32	0.59	1.21	2.02	2.96	4.00	4.55	4.21	3.57	3.14	2.89

Figure 3.9: SDA values for Roll for a $\overline{GM}_{eff}$ of 3.25 meter with a heading of 165 degrees [24]

The red areas are the SDA's larger than 1.5 degree, which is a commonly use criteria by Jumbo. Figure 3.8 show the highest roll motions around the 17 seconds and figure 3.9 around the 14 seconds. This corresponds with the natural periods of those load cases. The figures also show that a higher $\overline{GM}_{eff}$ result in a lower natural period. This relation is given as:

$$T_{0\o} = 2\pi\sqrt{\frac{I}{g \cdot \Delta \cdot \overline{GM}_{eff}}} \tag{3.17}$$

where the $\overline{GM}_{eff}$ is explained in 3.1.4 and I is given as:

$$I = k_{xx}^2 \cdot \Delta + m_{øø} + I_{additional} \tag{3.18}$$

Here I is in $kg \cdot m^2$, k_{xx} is the radius of gyration in meters, ∇ is the displacement of the ship in kg, $m_{øø}$ is the added mass of the ship.

During a lifting operation the mass moment of inertia changes due to movement of masses. As the radii of gyration are hard to calculate exact, the masses are considered as a point masses. Therefore the additional inertia will be determined by

$$I_{additional} = \sum_{i=1}^{N} m_i \cdot r_i^2 \tag{3.19}$$

where m_i is the mass of point mass i in kg and r is the distance in meters to the point mass i.

Equation 3.17 is particularly sensitive to variations in the $\overline{GM}_{eff}$. For the mass moment of inertia, the additional masses on the ship are relatively small compared to the weight of the ship it self.

3.2.2 Environmental conditions

The figures 3.8 and 3.9 show the SDA's for certain combinations significant wave heights (H_s) and periods (T_p). How often certain combinations occur depend on the geographical location and the time of the year. This information is given in scatter plots of the specific geographical location. The workability is calculated by summing up all the scatter diagram percentages of the allowable H_s - T_p combinations.

A workability study is not included in the scope of this project, but areas of interest are the North Sea, Campos Basin and West Africa as explained in 2.1. For both North Sea as Campos Basin 90 % of the wave periods are typically below 10 seconds during their geographical summer. During winter at the North Sea 80 % of the wave periods are still below 10 seconds but the waves are typically higher, resulting in a lower workability. At the west coast of Africa wave periods above 10 seconds are more common, but with relatively low wave heights of 1.5 meters and lower [24].

3.2.3 Criteria to operate safely set by the company

Criteria to operate safely can be set by the company, the captain and/or a classification societies. Jumbo's uses its own criteria based on experienced. At location the captain can adapt these criteria based on the local conditions. For this project the criteria will be limited by the ship stability properties and crane limitations. For the stability only the natural periods and GM (in the Y,Z-plane) are analysed. This is insufficient for offshore conditions, but it gives a fairly good first impression of the probable sea keeping behaviour of the vessel when interested in roll motions.

Natural period

As mentioned in 3.2.2 about 90% of the wave periods are below 10 second. Therefore the natural period of the ship should be larger than 10 second. In this way rolling is not a problem in moderate sea state.

$$T_{0ø} > 10\, sec \tag{3.20}$$

GM

An in-house guideline of Jumbo is that if a load is transversely displaced the static inclination angle is not allow to exceed 1°.

$$\overline{GM}_{min} = \frac{1 \cdot m_{load}}{\nabla \cdot \sin(1^o)} \tag{3.21}$$

In practise a $\overline{GM}$ between 2.25 and 3.25 meter is used for the J-class. This is for both $\overline{GM}_{initial}$ as for $\overline{GM}_{eff}$. To keep the natural period above 10 seconds $\overline{GM}$ should be smaller than 6 meter.

$$2.25 \leq \overline{GM} \leq 6\, meters \tag{3.22}$$

limitations of the crane

The crane is not only limited by the maximum crane load and the usual crane curve. When a crane is on a floating structure it is also exposed increased moments cause by inclination angles of the vessel and the swinging load. These be divided in side- , to- and off lead which are the summations of the crane base inclination and the load swing. The cranes on the J-class are originally not designed for offshore, but can be used offshore under certain conditions. The limitations of the cranes with a crane load of 900 and 250 ton are given in the appendix B. The curves show that when the crane radius is kept between the 10 and 20 meter the SDA's are allowed to go up to 4° for a 900 ton crane load. For 450 ton this is 5° for all radii.

Roll and heeling angles

For most offshore project on the track record Jumbo used a significant double amplitude criteria of 1.5°. See appendix A. The static crane inclinations criteria has been 1° for offshore lifts and heavy lift in port. These low static angles proved to be safe but unnecessary. In port an inclination angle of 1° or 2° are still used for optical reasons. It looks remarkable when a ship is inclined by 5° while the rest of the buildings and object in port are perfectly straight. At sea the captain sometimes allows heeling angles up to 4° or 5°. Based on Jumbo's experience and the crane load curves the inclination criteria become:

$$|\theta_{roll}| \leq 1.5^\circ \tag{3.23}$$

$$\left|\theta_{heeling} + \theta_{roll}\right| \leq 4^\circ \tag{3.24}$$

3.3 Current anti-heeling solution of Jumbo's J-class

To counter act heeling, the J-class uses wing tanks and ballast pumps. This section will elaborate on the main components of the system and how it is used in practise.

3.3.1 General arrangement of the anti-heeling equipment

The ballast system of the J-class is an integrated system of 22 ballast tanks 3 ballast pumps and many valves and piping routes. 12 of these tanks are wing tanks and are used for anti-heeling. Each side has 6 wing tanks of varying volumes. The positions of the wing tanks are illustrated in figure 3.10. The tanks are smaller on starboard because the cranes are installed here.

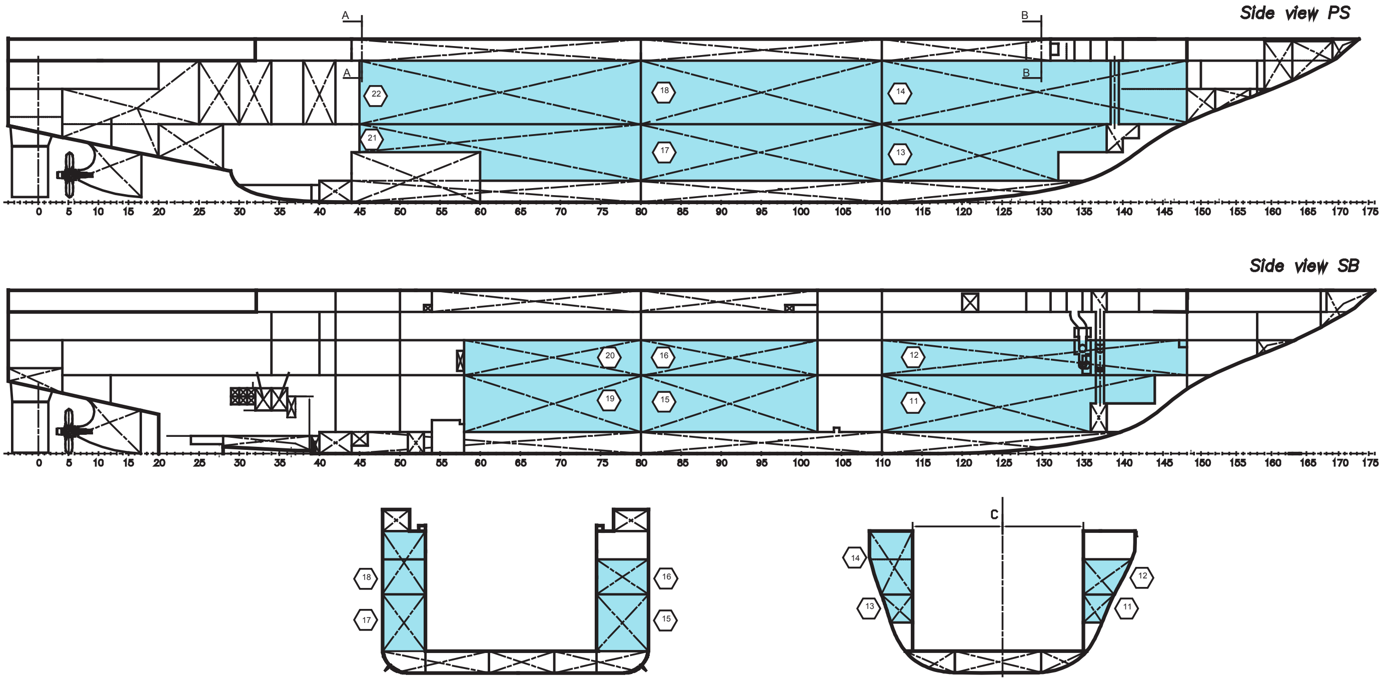


Figure 3.10: Water ballast tank arrangement with anti-heeling tanks marked blue

In order to be able to displace the ballast water as preferred, the ballast tanks are connected with a 12 inch glass fiber reinforced plastic (GRP) piping. These are all guided through the pump room in the front of the ship. See figure 3.11

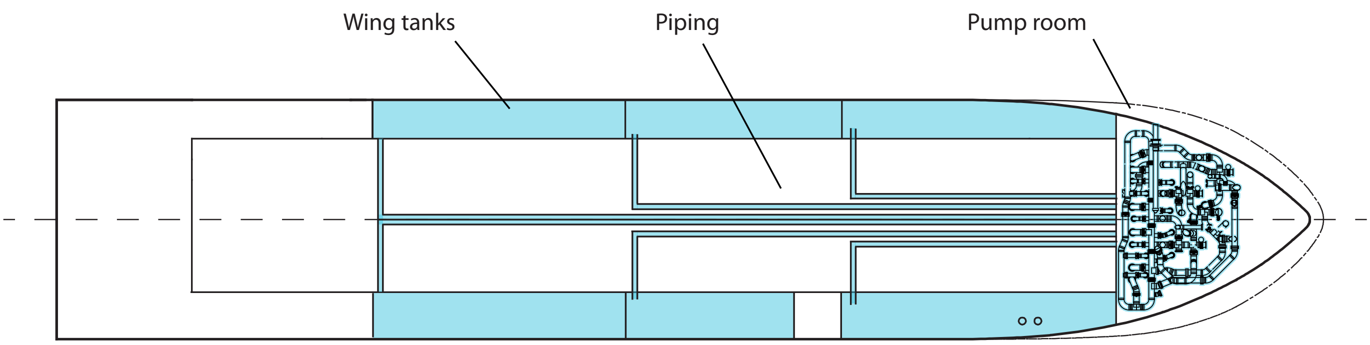


Figure 3.11: Piping from wing tanks to pump room

Here the three 700 m^3/hr centrifugal pumps supply the pump capacity. All the tanks and pumps are to a manifold as shown in figure 3.12. When varying the valve configuration, all tanks can be filled or emptied by a pump of choice. In practise one pump is used to operate the opposite wing tanks. The centrifugal pump flows are one directional. With altering the flow route around the pump, the ballast water can be pumped from both starboard to port-side as the other way around. Detailed information and original drawings can be found in appendix C.

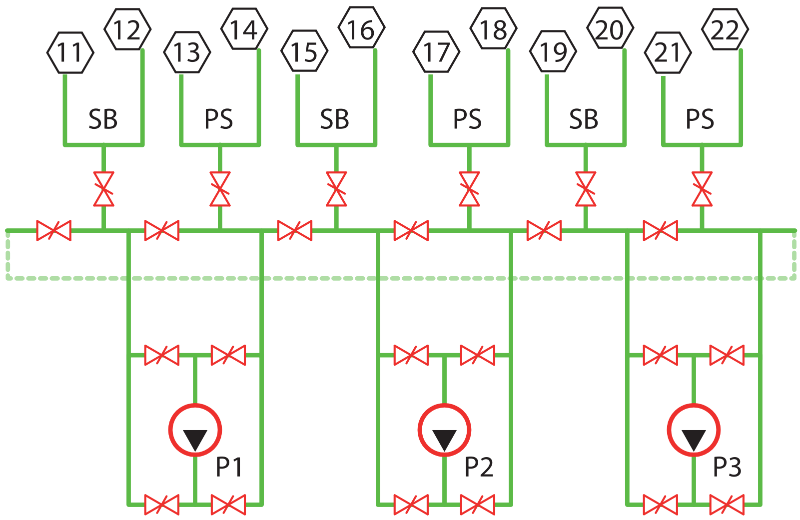


Figure 3.12: Simplified scheme of anti-heeling ballast system (Excluding 9 double bottom tanks)

3.3.2 Procedure

The heeling pumps pump the ballast water from one side to another to compensate the heeling moment. The general description of the procedure is given below. The detailed description can be found in the short report of observations of the operational practise during the field trip on the Jumbo Javelin in appendix D.

1. The tanks are filled with a sufficient amount water to be able to compensate for the final heeling moment.
2. The crane is positioned, the rigging is connected and the sea fastening is disconnected
3. The object is carefully lifted from the deck till the preferred height
4. The ballast pumps are started
5. The object is slowly moved overboard due to slewing of the crane. Meanwhile the heeling angle is monitored by the officer on the bridge on a inclinometer.
6. When the heeling-angle is nearing the 1 degree the cranes are stopped from slewing
7. Now wait un till enough ballast water is pumped to the tank on the opposite side till the heeling angle nears -1 degree and start slewing
8. In case the water ballast goes fasten then the slewing, the pumps are stopped and the preferred heeling angle is obtained by slewing the crane, or reversing the ballast water
9. From here step 5 to 9 are repeated till the object reaches its final position overboard.

Both the crane driving and operating the anti-heeling system is controlled manually. The crane is driven from deck and the ballast system from the bridge. On the bridge the crane heeling and roll angels are checked on an inclinometer. Real-time crane and ballast data is given on the screens. All the operational aspects are coordinated and communicated with walkietalkies or handheld transceivers.

3.3.3 Ballast capacity current solution indication

Currently the J-class has three ballast pump installed with a design capacity of 700 m^3/hr each. The slewing speed of the crane is 0.2 rpm when fully loaded. This means that the crane can slew 360 degrees in five minutes. To have an idea to what the current system is capable of, the static moments of a few scenarios are calculated. For these scenarios the assumptions are made that the vessel is restricted from rotating around the x-axis and Q is constant. So $\phi_{heeling} = 0$ and $M_{Ballast} + M_{Load} = 0$. Then the heeling moment caused by the load and the water ballast at a certain time is calculated. The ballast moment is expressed in the percentage of the total moment caused by the crane load and is called the theoretical coverage. This coverage is given as

$$Coverage = \left|\frac{M_{Ballast}}{M_{Load}}\right| \cdot 100\% = \left|\frac{t \cdot Q_{total} \cdot \rho_{water} \cdot \left(r_{wb_{PS}} - r_{wb_{SB}}\right)}{m_{Load} \cdot r_{Load}}\right| \cdot 100\% \tag{3.25}$$

This gives an simplified indication of the relation between the ballast capacity, load and time. The installed design capacity is 3 x 700 = 2100 m^3/hr but the system proved to be capable to produce 3000 m^3/hr and over at certain static heads. In figure 3.13 the coverage is shown for various loads and lifting phase durations and both capacities.

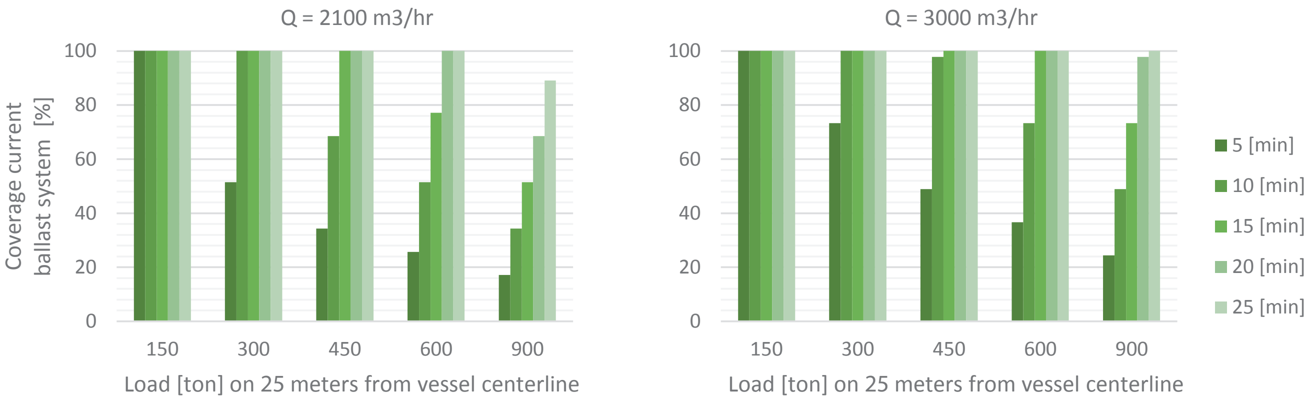


Figure 3.13: Theoretical coverage of the current ballast capacity

The figures show that the system is capable to cover the total moment caused by a 300 ton lift within 10 minutes for both volumetric flow rates. This corresponds with 80% of the offshore lifts in Jumbo's track record. At higher loads the coverage obviously decreases. But to overboard 900 ton a lifting phase of 25 minutes is still double according to the figure. However these high level calculation dos not take the moment changes over time into account and the system sensitivities cannot be determined. Therefore a dynamic model of the system should be written.

3.3.4 Operational time losses

The ship motions of the J-class are sensitive to hook loads. The heavier the load, the larger the overturning moments. This sensitivity can be illustrated by checking the heeling angle if a load is displaced one meter transversely. The static relation between the heeling angle and the hook load is given as

$$\sin(\phi) = \frac{1m \cdot F_{hook}}{\overline{GM}_{eff} \cdot \Delta} \tag{3.26}$$

Where m_{load} is the transversely displaced load in kg, $\overline{GM}_{eff}$ the effective GM after lift off in meters and Δ the water displacement in kg. This relation is a based on equations 3.9 and 3.10 which are explained in 3.1.3.

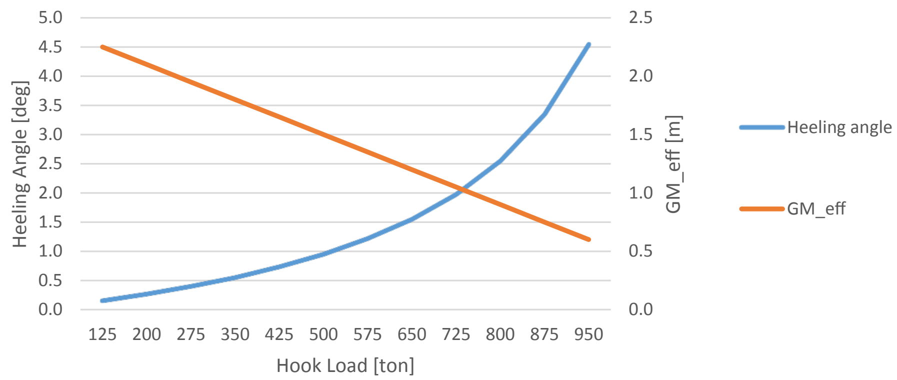


Figure 3.14: Static heeling angles after one meter transversely displaced loads. $\overline{GM}_{initial} = 2.5$ meter.

When this relation 3.26 is drawn into graph 3.14 it shows that the heeling angle increases exponentially with larger hook loads when the $\overline{GM}_{initial}$ is equal for all loads. This is a result of an increasing heeling moment and a decreasing $\overline{GM}_{eff}$. Which means that at larger hook loads, the heeling angle becomes more sensitive to differences between the heeling moment caused by the hook load and the moment caused by the ballast water. In case of the manual operated system of Jumbo, it can result is numerous "start and stops" of the pumps when lifting heavy loads to control the ship motions. This is also shown when looking at the operational log books of the Jumbo Javelin.

Lift job examples

At the 14th of November 2015 a reel of 150 ton had to be lifted from the deck to the quay. The reel was lifted from the deck the crane could slew to the final outboard position in one go. No starts and stops during slewing.

Earlier that year, on the 25th of January, a 815 ton lift was executed. This slewing operation took 76 minutes and each pump had to be stopped at least 10 times. See figure 3.15 were the total run time stand for the sum of minutes the pump was switched on. The inability to properly control the flow rate could potentially be accounted for the main share of the pump downtime of 36 minutes. So not the total volumetric flow rate, but the lack of variation in the flow rate increases the downtime. As figure D.3 in appendix D shows.

	Total run time [min]	Start/stop count [-]
Pump 1	28	10
Pump 2	28	10
Pump 3	35	14

Total duration	76	[min]
Down time	36	[min]

Figure 3.15: Summary of the 815 ton lift operation extracted from appendix D.2

Other operational time losses

When looking at the lifting operations the ballast system is not one of the most important causes of the total time loss. Various human factors are. Think of communication errors, rigging going wrong or heavy weather. As this is not included in the analysis of the anti-heeling system. If interested, appendix D.2 elaborates on time losses other than equipment capacity related.

3.3.5 Time loss due to equipment

Other than the limited pump capacity, the reaction time of the pump system plays an important role.

Reaction time of the pumps

Every time the operator starts the pump, the valves need te be opened. When they are fully open the pump can start. Then it the pumps need some time to reach the maximum rpm. The process of stopping the pump is the other way around. First the electricity supply to he pump it stopped. When the pump stopped, the valves can be closed. Both the start and stop phase cost 30 seconds each.

Losses in the piping system

The extensive piping system has many corners and butterfly valves. This results in mechanical energy losses. To determine the effect on the capacity the losses should be calculated. This is done later on in the project. Additionally, no examples of wear, cavitation or other typical damage cases in the piping of the J-class are known by the chief engineer of the Jumbo Javelin.

3.3.6 Summary of limitations of the current system

The operational limitations of the anti-heeling becomes noticeable at higher hook loads. The slew motion of the crane will have a larger impact on the heeling angle. To compensate this , the J-class has only a limited total pump capacity divided over three pumps. So basically three different volumetric flow rates to work with. And activating and deactivating the pumps goes with delays.

The lifting operation is supervised from the bridge were the inclinometer is checked and the pumps are controlled manually. Through walkietalkies orders are given to the officer of the deck and crane driver. This manual controlling of the crane and ballast system results can result in large differences of both overturning moments and can cause delays when reacting to the changes.

To sum up, the limitations of the current system are:

- Limited pump capacity for higher crane loads
- Limited variation of volumetric flow rate
- Reaction time of system components
- Inaccurate (manual) matching of the overturning moments
- Human influences

3.3.7 Discussion

All the information in this section is based on limited interviews, drawings, field experience, log books and simplified calculations. Every lifting operation has its own particularities and sensitivities. With the information and results of this section, an attempt has been made to give a general overview of the capabilities of the current system at own interpretation. But as the results are comparable with reality, one can take the information in the section as simplified but realistic interpretation of the characteristics of the J-class anti-heeling system.

3.4 Anti-heeling systems

In this section a selection of anti-heeling-solutions used in practice is described. First stand-alone solutions available on the market are described. Then an example from the industry is analysed to learn from an application in practise.

3.4.1 Systems available on the market

There are a number of different solutions developed to counter act the inclination of a ship. Most of them are anti-roll solutions for vessels in sailing motion and just a few of them are zero speed solutions. This subsection will focus on zero speed solution. So when the ship is not in sailing mode. Non-zero speed solutions have also been looked at but since it is not applicable for heavy lift operations, they are not incorporated in this thesis.

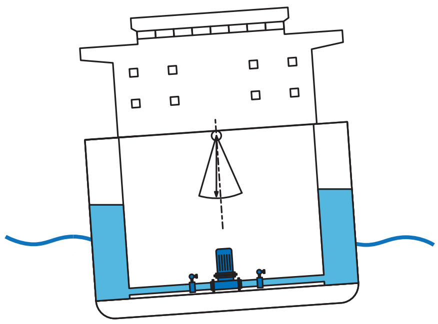


Figure 3.16: Anti-heeling system [11]

Conventional anti-heeling systems

This system reduces the heeling moment by pumping water to the wing tanks of the ship to act as counter ballast. The system is basically a combination of ballast tanks, pumps, valves, piping and a control system with sensors. Pumping water from one side of the ship to the other creates a moment in the opposite direction of the heeling moment. The ballast tanks are positioned at the sides of the ship and create a larger lever angle. The ballast tanks are usually connected through different piping routes, so the ballast plan can be adjusted relatively easily.

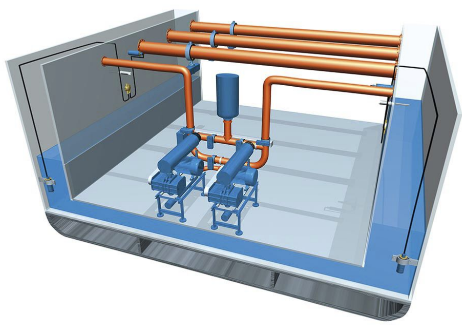


Figure 3.17: Hoppe Marine Blower Anti-Heeling System

Wing tanks with air compression

This system also makes use of wing tanks to compensate the overturning moment. The difference is that air pressure in the wing tanks is used in stead of in-line pumps. The system is a closed system and the compressed air from the blower system shifts the water between the side tanks due to pressure differences. The side tanks can either be connected by piping or be connected at the bottom. This last option is called a U-tank, see figure 3.17. The advantages of this principle is that it has a quick response. The pressures used for operation can be generated outside the wing tanks and can directly be applied whenever needed. Therefore this system is also used as a stabilizer to reduce roll motions. Hoppe Marine developed such system [12] and applied it for example on the EMAS' Lewek Constallation (2014). A 178 meter ultra deep water flexible pipelay, heavy lift and construction vessel.

Magnus Stabaliser

The Magnus stabliser is a rotating cylinder. When this cylinder is exposed to a flow and the cylinder is circulating, it will create a pressure difference. The reason is that the circulation will increase a tangential flow velocity on one side of the cylinder and decrease flow velocity on the opposite side. See figure 3.18. This is better know as the Magnus effect, named after the physicist Gustav Magnus. In 1853 the German professor published his finding about the effect in about the deflection of projectiles and peculiar phenomenon noticed in rotary bodies".

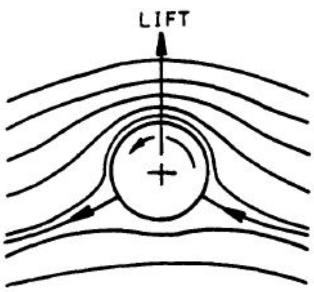


Figure 3.18: Magnus effect

Figure 3.19: Quantum Magnus

This effect only occurs if there is a flow around the cylinder. So to use this principle of Magnus effect as a zero-speed solution the cylinders must be dragged through the water wile rotating. This system is normally used as a anti-roll solution and typically used on luxury yachts. It is less effective as a anti-heel solution because the force generated is not constant due to its sinusoidal movement around the z-axis. As an example, the Quantas Magnum is shown in figure 3.19.

3.4.2 Anti-heeling in the high-end heavy lift industry

To see how anti-heeling on heavy lift vessels is applied in practise Heerema's Aegir is analysed. See figure 3.20. The Aegir is a high-end mono hull 210x46.2 meter construction vessel and relatively new (christened in 2013). The Aegir is designed for heavy lifting, ultra-deep water pipe-laying and installation of subsea structures. The crane has a capacity of 4000 ton up to a 40 meter radius and 1500 ton up to a 78 meter radius. This makes it a high-end big brother of the Jumbo J-class and therefore a good example to analyse. The analysis is based on information from an article in the June 2015 edition of SWZ Maritime [27] and the websites of Hoppe Marine and Heerema.

Figure 3.20: Heerema Aegir

Ballast capacity

The ballast system of the Aegir is based on 5 pairs of wing tanks. Each tank pair is operated by three reversible variable frequency driven propeller pumps. The nominal pump capacity of these 15 pumps combined is 15000 to 25000 m^3/hr. The maximum heeling moment caused by a 4000 ton crane load on a radius of 40 meters is around 215000 $ton.m$. This is the most extreme case. To compensate this moment, around 2500 m^3 of water needs to be pumped from the starboard wing tanks to the port side wing tanks. Every cubic meter of water pumped counts double for the overturning moment. This is because a cubic meter on starboard is lost and a cubic meter on port side is won. If assumed that the crane moment is only compensated by the water ballast the ballast water displacement can be expressed as

$$V_{Displacement} = \frac{m_{Load} \cdot r_{Load}}{\left(r_{wingtank_{PS}} - r_{wingtank_{SB}}\right) \cdot \rho_{water}} \tag{3.27}$$

V is the displaced ballast water in m^3, m the mass of the object in kg, r the lever arms from the ship's centre line in meters and ρ the density in kg/m^3.

Ballast follows the crane

When the ballast capacity is known, the slewing motion corresponding to the ballast displacement can be calculated. However, working with such big loads makes the ship highly sensitive for heeling if the moments are not matched properly. Additionally the other effects causing heeling described in 3.1.4, make it impossible to predict the heeling angle accurate enough before hand. Therefore Hoppe Marine developed an anti-heeling system with Load Moment Control (LMC), as they call it. The LMC simulates the impact the crane slewing process under a given crane load and floating condition. Then it computes the appropriate ballast operations. The LMC combined with a real-time inertial measurement system enables a reference to control the variable frequency drives of the pumps. By varying the frequencies of the pumps, numerous starts and stops of the pumps and opening and closing of valves can be avoid. A smoother and faster crane operation can be realised. Full scale crane slewing tests with a load of 4000 ton kept the vessel within 0.5 degrees of heeling.

3.5 Conclusion technical analysis

The most important conclusion of the technical analysis is that not the maximum volumetric flow rate of the current system is limiting, but the range of volumetric flow rate. The system should be able to correspond to the actual crane motion by varying the moment rate of the system. The other findings are included in the design criteria 4.1. As they are already mentioned in this chapter and included in the design criteria, there is no need to name them more than twice.

Chapter 4

Design parameters

This chapter will elaborate the design criteria and load cases. In order to develop a solution, the selection of the load cases and design criteria are important decisions. These need to representative to present and future operations. The design criteria limit the freedom of design possibilities and the load cases govern the evaluation. The considerations related to the selection and the sensitivities are discussed in this chapter.

4.1 Design criteria

The design criteria consist of a number of requirements. A requirement is an objective that every alternative must meet. The programme of requirements is a list of these objectives, or goals [22].

The main goal is to develop a solution for the J-class that is able to keep the ship's heeling angle within its operational limits at various crane motions and loads. In order to develop possible solutions and be able to asses them, design criteria are formulated. The criteria are clustered in four categories listed below

- System capacity
- Effects on ship motion
- Effects on (deck) operations
- Investments, installation and maintenance

4.1.1 System capacity

For the system capacity only the anti-heeling potentials of the principle solutions are evaluated.

Total moment capacity

The system should be able to compensate for the maximum moment the crane is able to create. The maximum crane capacity is 900t at 25 meter from the crane base. The crane base is 10 meters from the centre line. So the total moment capacity should be at least (25+10) x 900 = 31500 ton.m.

Moment rate capacity range

The crane motion is governing the heeling motion. This motion can have all the variations possible. To follow these crane motions, the system should be able to vary its moment rate. This will smooth the lifting operation. The lower and upper boundaries of this range can be determined by running different load cases in a crane model.

Start/stop reaction time of the system

If varying the moment rate is not the sufficient enough and start/stops are necessary, these should take as less time as possible. The reaction time on the current system is 20 seconds.

Redundancy

In case of an component failure the system should be able to remain stable.

Sensitivity to ship motions

The ship will move in all the 6 degrees of freedom. These motions should not significantly effect the capacity (as described above) of the system.

Ability to alter an increase or decrease of the moment quickly

When the moment cause by the system is significantly larger than the crane moment, it could be preferable to decrease the counter moment rather than increase the crane moment.

4.1.2 Effects on ship motion

All additions to the ship will have an effect of the ship's stability. This depends on the weight, location and ability to move of the system. Also the waves can have an effect on the anti-heeling system. To be able to predict what effect the solution on the ship motions could have, the solution is evaluated on hydrostatic criteria discussed in 3.2.3.

Increase of KG

An increased KG generally means larger GM possibilities. This results in a larger restoring arm. For reasons given in section 3.2.3 the GM should between the 2.25 and 6 meters.

Natural period

To decrease the change of resonating roll motions the natural period should be larger than 10 seconds as explained in section 3.2.3.

Mass moment of inertia

A larger mass moment of inertia increases the natural period and makes the vessel less sensitive to roll excitations. A down side could be that with a larger mass moment of inertia the roll motion become less easy to control.

Sensitive to effects causing heeling

The new system should not signifactly increase the potential effects causing heeling as discussed in section 3.1.4.

Effect of waves

Wave effects can cause force and moment oscillations. It is preferred to keep the counter moment as stable and predictable as possible. Therefore the wave effects should be kept to a minimum.

Integrity of stability after component failure

Obviously the ship is not allowed to have a sudden loss of stability after component failure. When a load is hanging in the crane the maximum inclination angle should not be bigger than 4° for the reasons given in section 3.2.3.

4.1.3 Effects on (deck) operations

All operations require different equipment, crew, space and time. Also the size, location and potential hazards of the system have their influence. To minimize potential risks the interference with the deck operation should be kept at a minimum. The equipment used for the anti-heeling system might also need preparation. With less influence on the deck operations, the system is less sensitive to time lost caused by human interaction. The criteria considering these are explained below.

Deck or hold space required

Deck and/or hold space is a valuable property. The ability to handle larger object sizes or amounts gives compatible advantages. In the case of a repetitive quayside to seabed project, this means that the ship need less transits.

Safety risks for crew

Interaction of the crew and system can cause safety risks. So moving parts, flammable component and other potential hazards should be avoided or kept a safe distance from the crew.

Weight of the system

The weight of the system can potentially decrease the KG or increase the mass moment of inertia. Heavy systems can also result in higher expenses.

Mobility of the system

When the system is taking space or is heavy, it might be preferable to let it be able to be displaced. Or taken off board when it is not used.

Extra procedures to prepare the system

When the solution required extra procedures resulting in a longer lifting phases, the solution does not reach it goal.

Ability to use in port

For the quayside-to-seabed solution the ship lifts both in port and offshore. Therefore the solution should be suitable for both.

4.1.4 Investments, installation and maintenance

The last cluster covers the financial criteria. For conceptual designs it is hard to evaluate on qualitative criteria. But the financial criteria are the most because they will tell if the solution will pay itself back. Therefore the cost will be roughly estimated based on the complexity and size of the system.

Ability to integrate with current system

Generally speaking the less amount of adaptation of the ship, the more economic. And for 80% the current system is fast enough.

Complexity of the ship conversion

Like with the ability to integrate with current system, when the conversion is less complex the investment is potentially more economic.

Initial investment

Section 2.3 concluded that potential savings are relatively small. This means that the amount of money available to invest is also small.

Operational expenses

In line with small potential savings and the investment, the operational expenses are not left much room to increase.

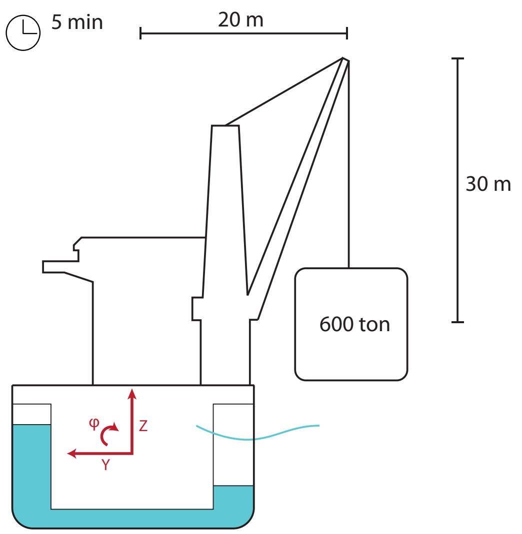
4.2 Initial load case

In the next phase candidate concepts are generated. To be able to evaluate the concepts (partly) based on quantitative results, they have to be exposed to the same load case. The selected load case and the considerations are given in figure 4.1.

Equilibrium of moments

For the candidate concepts the static equilibrium of moment is calculated. The sum of the moments is of the same magnitude as the restoring moment with a corresponding heeling angle as given in equation 3.9. The equilibrium of moments of the concepts is

$$M_{Ballast} + M_{Crane} + M_{Concept} = -M_{restoring} \tag{4.1}$$



- 600 ton lift
- 5 minutes slewing time
- 25 meters from midship
- 30 meters crane line
- $\overline{GM}_{initial} = 3.5$ meter
- $\overline{GM}_{min} > 2.25$ meter

Figure 4.1: Initial load case

4.2.1 Chosen parameters

The effect of the parameters given this load case are already discussed in sections 3.1.4, 3.2 and 3.3.4. In this section the chosen values of the parameters will be discussed.

Weight of the load

A typical offshore lift for Jumbo, about 80% is lighter than 300 ton. The heaviest manifold lifted by jumbo is 574 ton, so 600 ton is in the higher segment.

Duration

A slewing time of 5 minutes is the capacity when the crane is fully loaded. Figure 3.13 shows that the slewing time of a load of 600 ton could potentially save 10 minutes. 5 minutes is short for the current ballast system to compensate for the 600 ton lift and will only be able to compensate for 30% of the heeling moment. Doubling the $t_{slewing}$ will double the $M_{Ballast}$. But the $t_{slewing}$ is preferred to be as short as possible.

Load displacement

A lever arm of 25 meters from midship leaves a space of 12 meters between the crane hook and the side of the ship. When the object in lifted from the deck, 30 meters is a realistic distance between the centre of gravity of the load and the crane tip.

4.2.2 Discussion of the load case

The static load case does take the moment changes over time and ship motions into account. It only calculates the eventual equilibrium of moments. So the roll angle at any given time is unknown. Also the load case is sensitive to its chosen values. When changing the lever arm of the load from 20 to 25 meters results in a moment increase of 25% for example. To try different load cases for the concepts an Excel spreadsheet has been written. But to compare the concepts the load case mentioned above has been chosen. For the selected concept a more realistic model has to be written.

Chapter 5

Development process

This chapter will explain the steps taken and the models used to manage the process.

5.1 Systematic approach

Smart solutions or designs are made by small, or sometimes large inventive steps. In some cases smart solutions are the result of a spontaneous bright idea. However this is not something one can count upon in the development of a solution. Systematic approaches can lead to a smart solution or even stimulate bright ideas. But the most important advantage of a systematic approach is the development of insight into the problem. A systematic approach is especially beneficial in more complex engineering problems and sub-problems.

5.1.1 Analysis versus synthesis

When searching a solution to an engineering problem, one can use of a "classical analytical thought process" or a "synthesis-orientated thought process". The difference between the two is described by Prof. Amaresh Chakrabarti in his book *Engineering Design Synthesis- Understanding, Approaches and Tools* 2002 [3].

The classical analytical thought process is to extract facts from observations of a phenomena and to try to give best explanations of these facts from a set of axioms or hypotheses. This is commonly used in academic research. However the main focus of this thesis is not about an in-depth observation of the phenomenon of heeling during heavy lifting only. The main objective of this thesis is to develop and compare alternative solutions that will shorten the lifting phase. The solutions will be a considerations of various viewpoints based on requirements. Therefore a classical analytical thought process alone is not sufficient for this thesis.

Synthesis-orientated thought process in the context of design works different from the classical. Here, the analysis is to derive descriptions of the designed object's behaviour and function. The synthesis is to derive descriptions about the design object from functional requirements to the design object. The synthesis involves multiple viewpoints as axioms. Therefore many hypothesis are acceptable.

What it basically comes down to is that the analysis-oriented thought process is to examine "something". The synthesis-orientated thought process is to create or find "something". But the synthesis also involves examining processes. In this sense, analysis and synthesis are complementary to each other in the synthesis-orientated thought process. The methods used to generate solution principles are chosen with the synthesis orientation in mind, because it potentially creates more room for new ideas.

5.1.2 Framework

As time is relatively short to develop the solution, one should not waste time on elaborating procedures that do not serve the goal of the project. The eventual solution will be sought in a (partly) physical system or product. To make progress, Michael J. French [9] suggest that the problem analysis, the conceptual, embodiment and detail phases, and a simple table of functions or requirements and alternative means are enough of a systematic framework.

5.2 Phase model of the design process

To structure the engineering design process a phase model of the process, shown in figure 5.1, is developed. The phase model is a schematic overview of the whole process.

As suggested by MJ French in 5.1.2, Pahl & Beitz also used the same four phases in their model. The model of Pahl & Beitz is problem-orientated approach to systematically generate solutions. Later in time, the Verein Deutscher Ingenieure generalised the model of Pahl & Beitz and divided the phases into stages with corresponding results to make the model more applicable to a wide variety of engineering tasks [21]. The phase model used for this thesis is based on these models and can be found in appendix E. Using a phase model will help to manage the process.

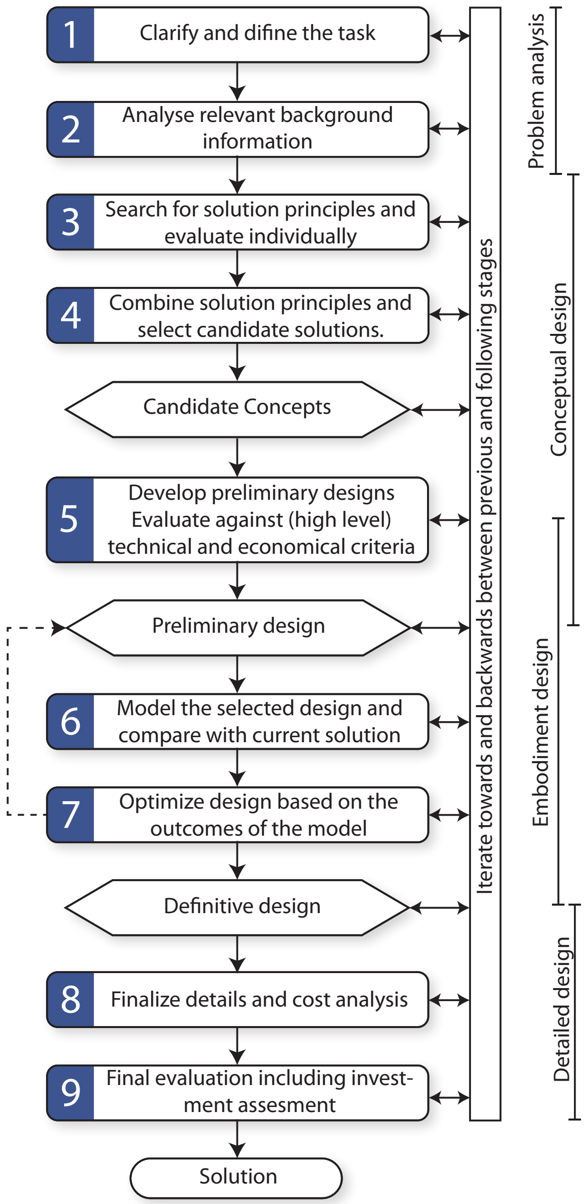


Figure 5.1: Phase Model of the Process

5.2.1 First phase: Problem analysis

This phase is the start of the whole process. The purpose of this phase is to get familiar with the subject, determine clear aims and lay the foundation of all the following phases.

Stage 1: Clarify and define the task

This stage is basically the definition of the problem as given in 1.2. The problem is analysed to determine the objectives and formulate the research question.

Stage 2: Analyse relevant background information

Information on the problem is collected and analysed. Based on the analysis a program of requirements is drawn up.

5.2.2 Second phase: Conceptual design

The second phase is to generate different solution principles, combine these to principle solutions and translate them into concepts. Different creativity techniques and selection methods are used for the conceptual generation.

Stage 3: Search for solution principles

The purpose of this stage is to create a divergence of ideas. Techniques used to reformulate the problem and reflect different points of view towards the problem, are the "How-To's" problem statements. The implementation of this technique is described in 6.1.1. Then a variety of solutions principles to these problem statements form a basis for the next stage.

Stage 4: Combine solution principles and select candidate solutions

In this stage the solution principles are structured and further elaborated. The method used here is a morphological analysis and can be found in 6.1.2. With the morphological analysis a variety of combinations is created, called principle solutions. From these principle solutions the most potential are selected. These are the candidate concepts.

Stage 5: Develop preliminary designs and select most potential designs

This stage is the transition between the concept design phase and the embodiment design phase. The candidate concepts are translated into preliminary designs and evaluated against technical and economical criteria. The selection method can be found in 6.3

5.2.3 Third Phase: Embodiment design

The embodiment design phase is where the research question will be answered and verified by calculation. The most promising preliminary design will be detailed to a point, that it can be modelled and tested.

Stage 6: Model the selected design and compare with current solution

When the results are evaluated and compared with the current situation, conclusions can be drawn.

Stage 7: Optimize design based on the outcome of stage six

From stage six one can choose to start a iterative process. Changes are implemented in the preliminary design. Now the model has to be adapted to this re-design. This process can be repeated. When satisfied with the optimization, the definitive design is obtained.

5.2.4 Forth phase: Detailed design

The forth phase is the final stage. This phase will not be included in the scope of the project.

Stage 8: Finalize details and cost evaluation

Now the definitive design can be detailed till a final solution is created. Also the corresponding cost of the final design need te be estimated.

Stage 9: Final evaluation and investment assessment

The final design is evaluated against the list of requirements. Also the expected investment is evaluated. After evaluation one can decide to do an extra iteration to improve the final solution.

Chapter 6

Concept Generation

The purpose of the analysis phase is to gain relevant knowledge and lay a foundation to develop different solution principles. In the concept generation phase a combination of design methods is used to generate these different solution principles.

6.1 Candidate solutions generation

To start the design process the research question formulated in 1.2.3, can be answered on a high level and categorise the different solution principles. So the installation time, focussing on the lifting phase, can be shortened by the following principles:

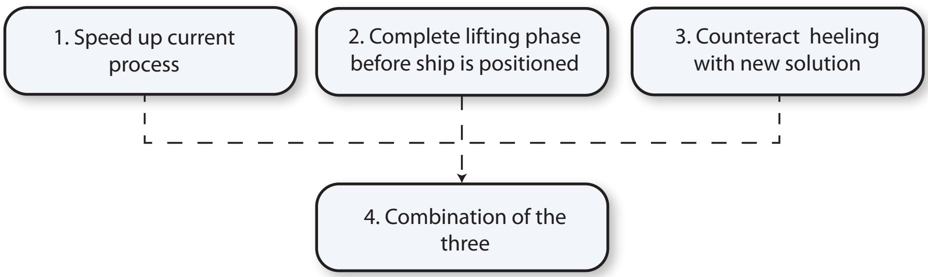


Figure 6.1: Solutions principles

6.1.1 "How to" analysis

In figure 6.1 the solutions principles to the research questions are shown. These solutions principles can be elaborated into sub-solutions principles. For the elaboration, the "How to" creativity technique is used (Delft Design Guide, 2010 [22]). With this method the problem statements are rephrased in the form of "How to.." Examples are: How to reduce a moment or how to create a force? The solutions, on their turn, are rewritten in the "how to" form. This process can be repeated un till a significant level of detail is reached. This process is a strive for quantity rather than quality in order to generate alternative solutions. An schematic overview of this process is shown in figure 6.2.

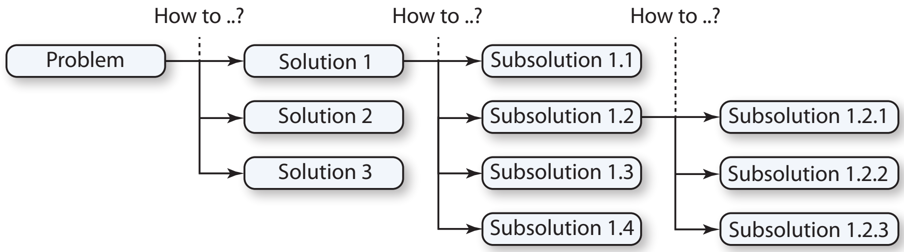


Figure 6.2: "How to's" problem description scheme

The purpose of this method is that problem reformulations can reflect different points of view toward the problem. To give an example, this project started with the assignment to develop a quick anti-heeling system. Rewriting the problem in to the 'how to' form could give: How to reduce installation time? An answer to this could be: complete the lifting phase before the ship is positioned for installation. This solution is fundamentally different from a fast anti-heeling system solution.

Below the solution principles from figure 6.1 are rewritten in the 'how to' form. The solutions form the basis for the next step where the "how to's" are combined with a morphological analysis and solution principles are synthesised.

1. How to speed the current process?

1. Faster shifting of ballast water
2. Accept larger heeling angles
3. Calculate the operation in advance so the cranes and the ballast pumps can work simultaneously as a continuous system.

2. How to complete the lift phase before the ship is positioned at location?

1. Complete lifting phase when in transit
2. Complete lifting phase when positioning takes place

3. How to counteract the heeling?

1. Reduce the varying heeling moment
2. Increase the restoring moment

4. How to develop a combination?

To develop combinations of sub solution the morphological analysis is used in the next subsection.

6.1.2 Morphological analysis

A morphological analysis is a method to generate ideas and principle solutions in a systematic manner. Typically a chart or matrix of functions and components is used to create an overview. This parameters are independent, abstract and preferably not described in words, but visually displayed. Through selection and combination of components, conceptual solutions come out. The morphological chart used for this thesis is not the classic function-component matrix, but it is used to structure the 'How To' analysis in the style of the visual morphological chart. As an example the 'how to counteract heeling' question is visualised in figure 6.3. The whole analysis can be found in appendix F.

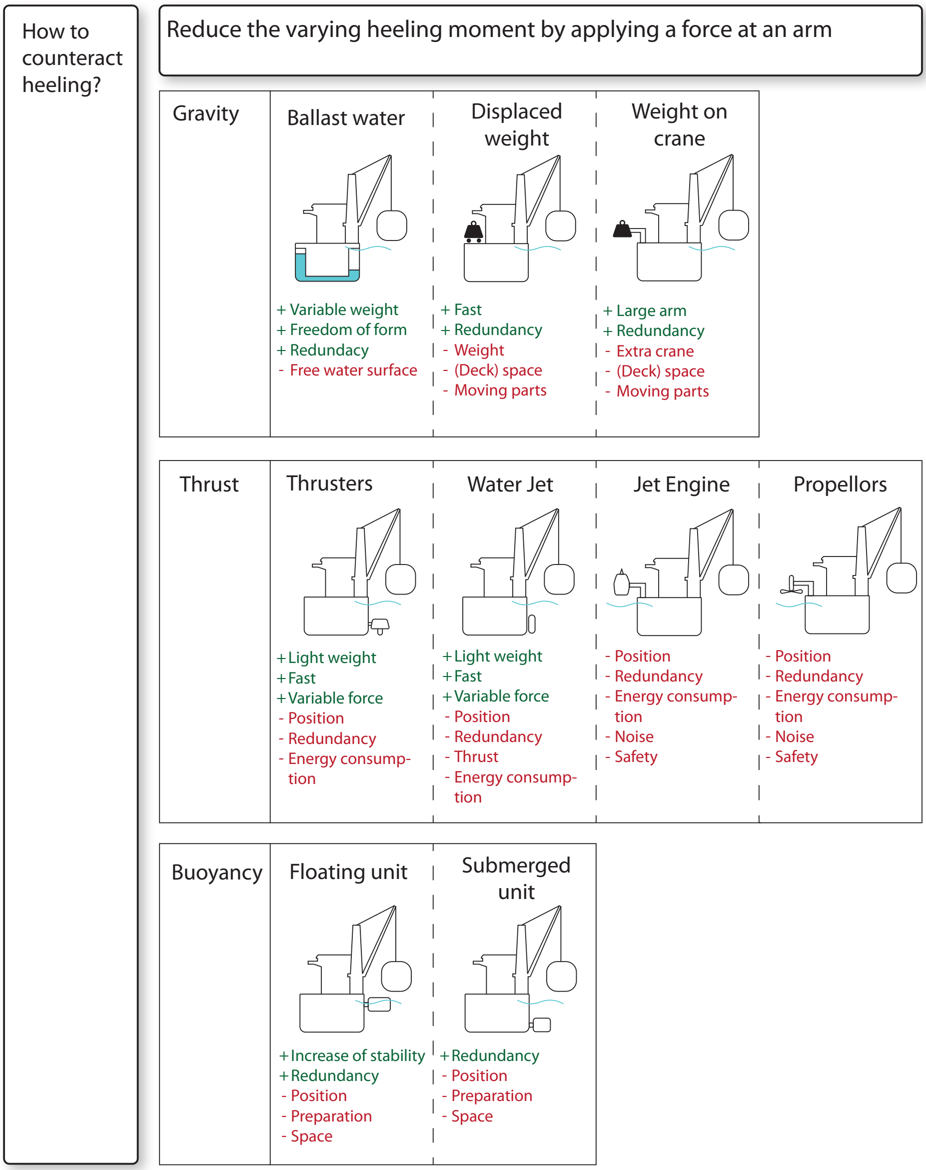


Figure 6.3: Reduce heeling moment

In order to limit the number of options the evaluation method of analysis of the rows is applied [22]. Here the individual options are evaluated against (a part of) the design criteria. In addition, the solution principles are provided with a so called itemized responses. Which are the intuitive negative and positive aspects. Using only the first and second preferences per row reduces the number of solution principles to use for the synthesis. The next section will introduce the candidate solution, based on the result of the analysis.

6.2 Candidate concepts

In this section the selected principle solutions, called candidate concepts, are discussed based on basic hand calculations. The calculations are based on static stability and equilibrium of moments $\sum_{i=1}^{N} M_i = 0$. All the masses are considered to be point masses. Environmental loads and horizontal shifts of centres of gravity and buoyancy along the y-axis and other dynamic effects are neglected. So the calculations are just an indication of which order of weights and moments can be expected. But as mentioned in 4.2.2 just looking at the total moment capacity is not sufficient. Therefore the potential of each concepts will be briefly discussed.

Selected candidate concepts

From the morphological analysis 5 concepts are selected to be taken a closer look at. They have been selected because they are likely to allow varying moments and will not loose its moment when its drive system suddenly fails.

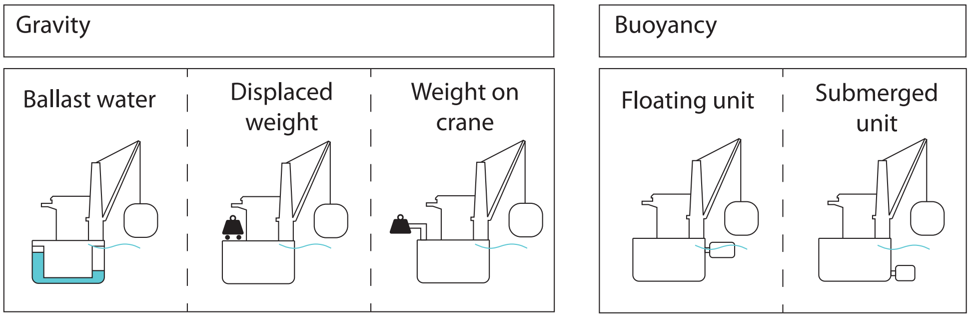
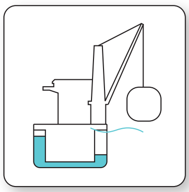


Figure 6.4: Reduce the varying heeling moment by applying a force at an arm

All the candidate concepts are based upon reducing the resulting moment by applying a counter moment. The force causing the moment is applied by gravity or buoyancy, see figure 6.4. In the following section all 5 concepts will be exposed to the load case explained in section 4.2.

6.2.1 Concept 1: Ballast tanks



Using ballast tanks is currently been done following the procedure described in 3.3.2. The maximum capacity for this system is the limited volumetric flow rate these three pumps can generate. To increase the volumetric flow rate one can consider more powerful pumps, pressurise the system and/or enlarge the cross-sectional area. When replacing the pumps, the available space might be limited. When pressurised tanks are considered, the whole tank system must be converted into a closed system. If in any case an extra connection between the two sides of the ship must be realised, the available space might be limiting. At first sight the conversion of the current system, seems to be a practical solution. But before this can be concluded the effect on the lifting phase and the complexity of the conversion needs to be analysed.

Capacity

The required pump capacity is governed by the heeling moment created by the crane load and the slewing time of the crane. To calculate the required pump capacity, equation 6.1 is used under the load condition described in 4.2.

$$Q_{pump_{total}} \cdot t \cdot \left(r_{wb_{SB}} - r_{wb_{PS}}\right) \cdot \rho_{water} + m_{Load} \cdot r_{Load} = 0 \tag{6.1}$$

Q is the volumetric flow rate in m^3/hr, m the mass of the object in kg, r the lever arms in meters, t the slewing time in hours and ρ the density in kg/m^3.

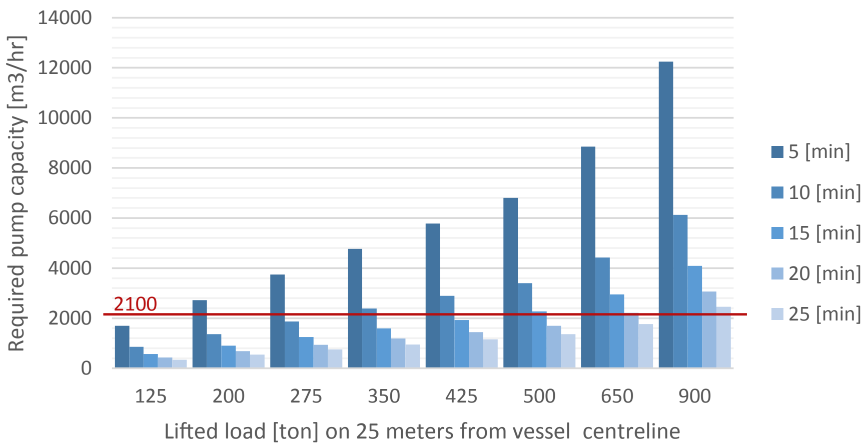


Figure 6.5: Required ballast pump capacity to per load [ton] at 25 meters from centre line

Figure 6.5 shows the relation between slewing time and required pump capacity. The shorter the slewing time, the larger the required pump capacity. The currently installed capacity is 2100 m^3/hr and is displayed as the red line. If the blue bar stays under the red line, the system is capable of generating the required moment within the given time. So for example, if a 650 ton load have to be displaced displacement a slewing time of 20 minutes or increase the capacity by 400% is required.

Results concept 1 at initial load case

The heeling moment following the lift scenario described in 4.2, almost fully compensated by the ballast water. When the total capacity of the pumps is increased to 7700 m^3/hr, a relatively small resulting moment will be left to compensate. The ship will heel 0.9 degrees and creates the restoring moment. See figure 6.6.

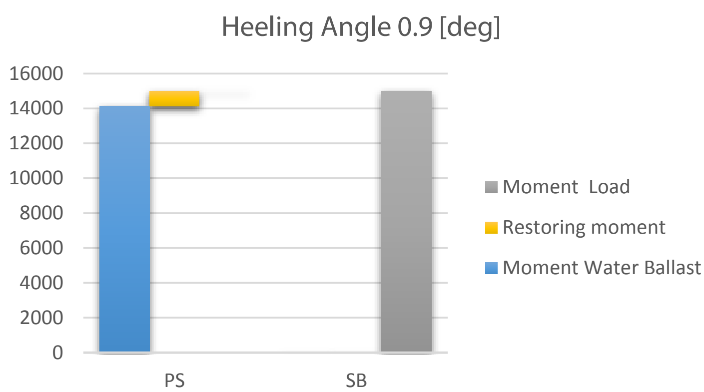


Figure 6.6: Static moments concept 1 in $ton.m$ [600ton, 25m, 5min]

Before Lift		
GM_initial	3.5	[m]
T_0ø	15.1	[s]

After Lift		
GM_eff	2.6	[m]
T_0ø_eff	17.7	[s]

Q heeling pumps	7700	[m3/hr]
Displaced ballast water	642	[m3]

Figure 6.7: Result summary concept 1 [600ton,25m,5min]

Figure 6.7 shows that $\overline{GM}$ drops after lift.Also the mass moment of inertia increases as the masses of the load and the water ballast move away from the rotation point of the ship. The additional inertia in this case is calculated as:

$$I_{additional} = m_{Load} \cdot d_{Load}^2 \tag{6.2}$$

Here I is the mass moment of inertia in $kg \cdot m^2$, m_{Load} the mass of the lifted object in kg d_{load} the distance to the crane tip in meters, $m_{waterballast}$ mass of the displaced water in kg and $d_{waterballast}$ is the distance to the centre of gravity of the water filled wing tanks in meters.

A decreased $\overline{GM}$ and a increased I will result in a increase of the natural period ($T_{0ø}$), as explained in 3.2.1. The increase of $T_{0ø}$ after lifting is governed by this drop of the $\overline{GM}$ in this case. The larger mass moment of inertia makes is harder to reduce the roll motion.

The required volumetric flow rate is 3.6 times higher than the $Q_{initial}$ in this case.

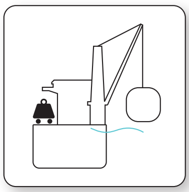
Ability of varying the moment

When the moment of the water ballast is not corresponding with the moment of the crane load, the total volumetric flow rate should be adjusted. Even with a delay it can be used to smooth out the process rather than start and stop it. The effects of varying the flow rate need to be checked if they really makes a difference. Also the possibilities of reversing the flow direction and its effect could be researched.

Discussion

At first sight this could be realised by a conversion of the current system by adding frequency controllers on the pumps. But the required system properties should first be determined by researching various crane motions and loads.

6.2.2 Concept 2: Displacing weight on board



Moving a weight on board can be considered if the size and weight are reasonable. Having a large extra weight on board takes space and can be expensive. On the other hand, this option has the potential to be developed into a movable product. The figures 6.8 and 6.9 show the required counterweights excluded and included water ballasting. From 13 meters the weight will be overboard. The overboard option is discussed as concept 3 in 6.2.3. Therefore the graphs are divided by the dotted line into concept 2 and 3.

Capacity

To calculate the required counterweight, equations 6.3 and 6.4 are used at the static load case described in 4.2. Where equation used for the case excluding the ballast system is

$$m_{load} \cdot r_{load} + m_{counterweight} \cdot r_{counterweight} = 0 \tag{6.3}$$

For the case including the current ballast system the equation is

$$m_{load} \cdot r_{load} + m_{counterweight} \cdot r_{counterweight} + 5/60 \cdot Q_{initial} \cdot \rho_{water} \cdot r_{waterballast} = 0 \quad (6.4)$$

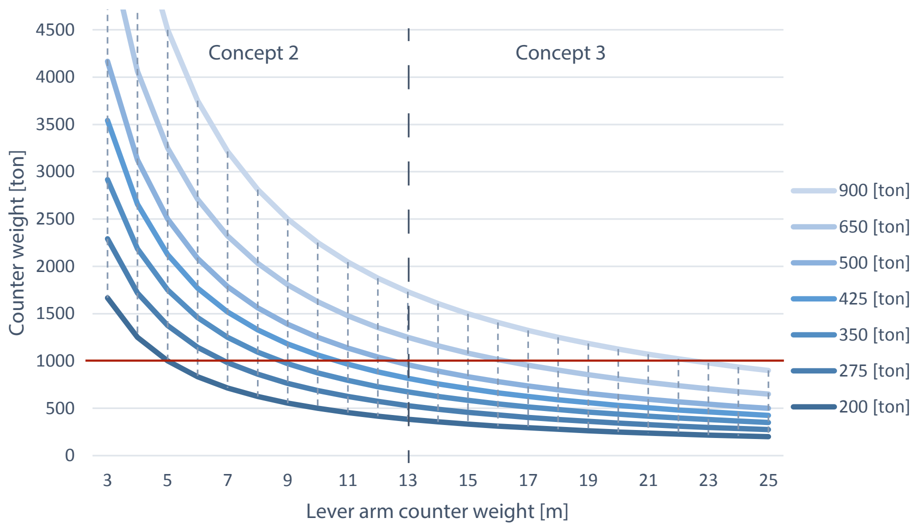


Figure 6.8: Counterweight - lever arm relation **excluding** water ballast

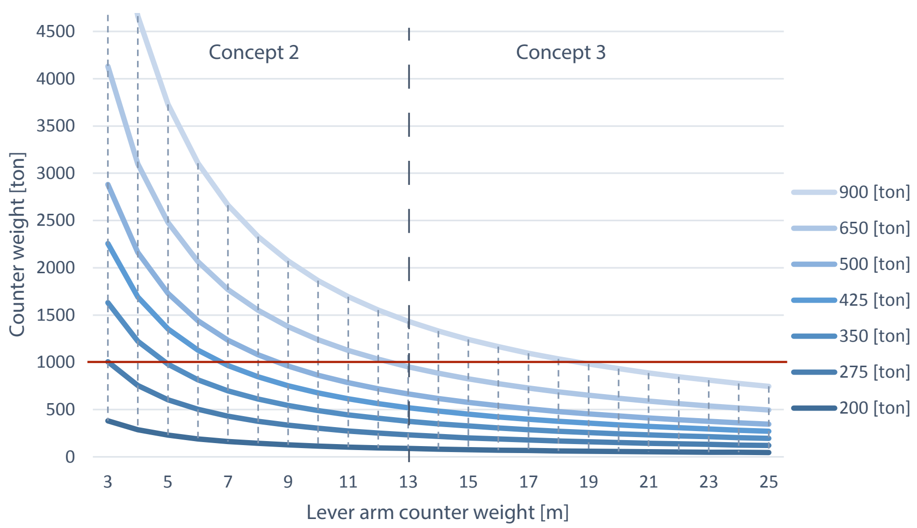


Figure 6.9: Counterweight - lever arm relation **including** water ballast [5min]

There is a significant difference in required counterweight between the systems including and excluding water ballast. But for both cases the counterweights are pretty heavy and go easily above 500 ton. Moving around 500+ ton objects on board, is a heavy operation. If a non-adjustable counterweight is used, this weight is not suitable to cover all crane loads properly without making use of water ballast.

Results concept 2 at initial load case

The heeling moment following the lift scenario described in 4.2, is for 67% compensated by the counterweight. When the point mass of the counterweight acts on 10 meters from the centre line, the required counterweight is 1000 ton.

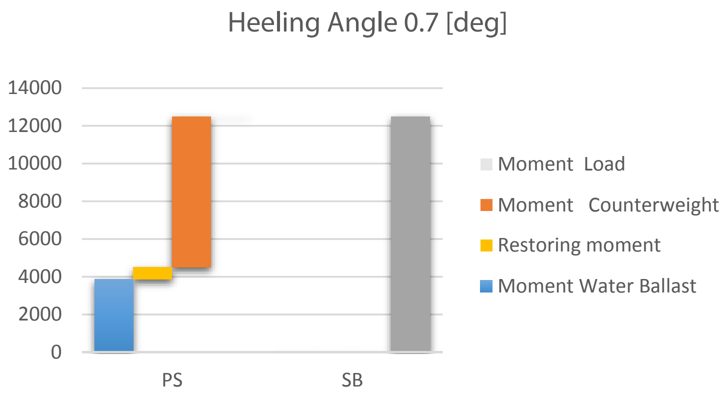


Figure 6.10: Static moments concept 2 in $ton.m$ [500ton,25m,5min]

Before Lift		
GM_initial	3.50	[m]
T_0φ	14.90	[s]

After Lift incl water ballast		
GM_eff	2.43	[m]
T_0φ_eff	18.05	[s]

Q heeling pumps	2100	[m3/hr]
Displaced ballast water	175	[m3]

Position counterweight	-10	[m]
Mass counter weight	800	[ton]

Figure 6.11: Result summary concept 2 in $ton.m$ [500ton,25m,5min]

Figure 6.7 shows that $\overline{GM}$ drops after lift. Also the mass moment of inertia increases as the mass of the load moves away from the rotation point of the ship. The additional mass moment of inertia in this case is calculated as

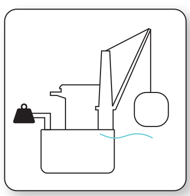
$$I_{additional} = m_{Load} \cdot d_{Load}^2 + m_{Counterweight} \cdot d_{Counterweight}^2 \quad (6.5)$$

A decreased $\overline{GM}$ and a increased I will result in a increase of the natural period .

Discussion

A moving weight could work complementary to the ballast system to vary the moment rate. But to design an expensive heavy weight system on deck to win 10 minutes on a unusual heavy single lift, will not pay itself back easily.

6.2.3 Concept 3: Displacing weight over board



Displacing weight over board creates a better moment-weight rate than concept 3 and requires minimal deck space. A special device or tool needs to be installed which might lead to conversion of the ship for a sufficient structural support. Combining a movable weight with the ballast tanks is considered.

Capacity

Figure 6.12 shows the required overboard counterweights for various crane loads. The figure shows that if a non-adjustable counterweight is used, this weight is not suitable to cover all crane loads properly without making use of water ballast. Looking at figure 6.8 a point mass of 600 ton could cover all heeling moments with a lever arm range from 5 till 22 meters.

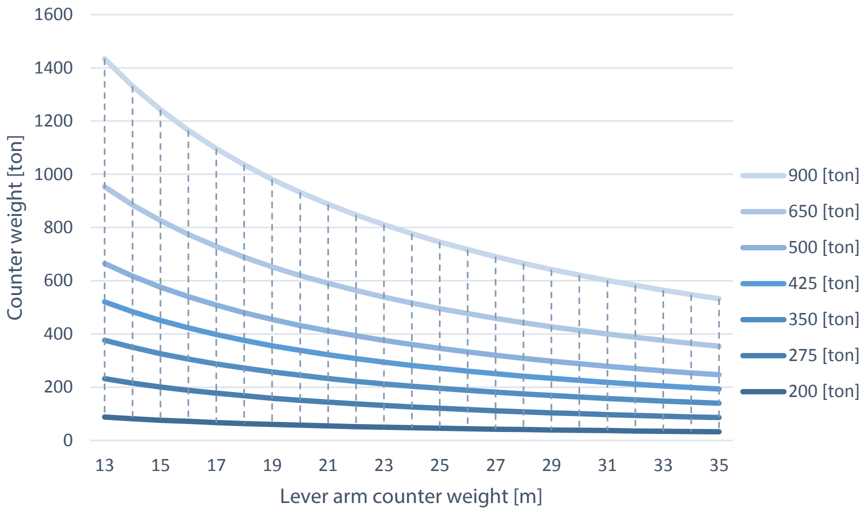


Figure 6.12: Counterweight - lever arm relation including water ballast

Results concept 3 at initial load case

The result of concept 3 show similar effects on the $\overline{GM}$ and $T_{0ø}$, as the they are basically the same but with different weight and lever arm. See tables 6.13.

Before Lift		
GM_initial	3.50	[m]
T_0ø	14.90	[s]

After Lift incl water ballast		
GM_eff	2.48	[m]
T_0ø_eff	18.1	[s]

Q heeling pumps	13	[m3/hr]
Displaced ballast water	175	[m3]

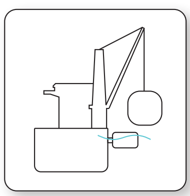
Position counterweight	-20	[m]
Mass counter weight	450	[ton]

Figure 6.13: Result summary concept 3 in $ton.m$ [500ton,25m,5min]

Discussion

This concept has the same disadvantages as concept 2. The only advantages over concept 2 are already mentioned.

6.2.4 Concept 4: Use partly submerged stabiliser



Stabilisers are already being used to increase the lifting stability in sheltered waters. The stabilisers only take limited deck space in transit mode. When varying the submerged volume of the pontoon, various moments can be created. When offshore, the waves can have a significant effect on the stabiliser and ship movements. Installation of the stabilisers also takes time.

Capacity

The capacity of concept 4 is governed by the dimensions of the pontoon. A larger the water plane moment of inertia $I_{pontoon}$ created by the pontoon results in a higher $\overline{GM}$ value and therefore a larger restoring capacity of the ship. And a larger water displacement of the pontoon results in a larger moment. The most significant effect of the pontoon on the $\overline{GM}$ is the increase of the $\overline{BM}$ which can be calculated as

$$\overline{BM} = \frac{I_{waterplane}}{\nabla_{total}} = \frac{I_{initial} + I_{pontoon}}{\nabla_{ship} + \nabla_{pontoon}} \tag{6.6}$$

With I water plane moment of inertia and ∇ displaced volume. The water plane moment of a single pontoon according to Steiner's theorem, if assumed to be rectangular, is determined as

$$I_{pontoon} = \frac{L \cdot B^3}{12} + L \cdot B \cdot d^2 \tag{6.7}$$

Where L is the length of the pontoon along the ship in meters, B is the breadth of the pontoon in meters and d is the distance in meters between the centre line of the ship and the centreline of the pontoon.

Although a larger $I_{waterplane}$ results in a larger $\overline{GM}$, there are limitations. When the $\overline{GM}$ increases the $T_{0ø}$ decreases. As a design criteria the $T_{0ø}$ should be larger than 10 seconds. As $\overline{GM}$ drops after lift, the $T_{0ø}$ before lift is governing. The relation between $\overline{GM}$, $T_{0ø}$ and $I_{pontoon}$ is shown in figure 6.14. The graph shows that $I_{pontoon_{max}} = 70000 m^4$.

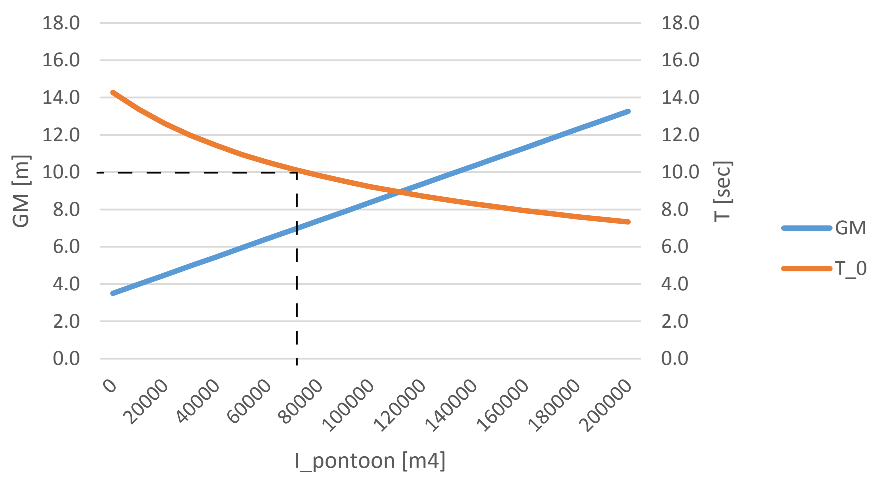


Figure 6.14: Relation between $\overline{GM}$, $T_{0ø}$ and $I_{pontoon}$ for concept 4 before lift

A larger restoring moment can be realised with the pontoon, but it is still relatively small compared to the heeling moment caused by the load. Therefore the water ballast and the moment created by the buoyancy force of the pontoon must take up the rest of heeling moment. This moment is calculated as

$$M_{pontoon} = (m_{pontoon} - \nabla_{pontoon}) \cdot r_{pontoon} \tag{6.8}$$

where $m_{pontoon}$ is the mass of the pontoon in air in kg, $\nabla_{pontoon}$ is the displaced volume of water by the pontoon in m^3 and r the lever arm of the pontoon. The centre of buoyancy and mass of the pontoon are assumed to be on the same horizontal position. As mentioned in 6.2 horizontal shift of centres of buoyancy are not taken into account for the heeling calculations. Normally the shift of centre of buoyancy would be

$$\overline{B_0B_1} = \frac{d \cdot \nabla_{Pontoon}}{\nabla_{Total}} \tag{6.9}$$

where $\overline{B_0B_1}$ and d are in meters and ∇ is in kg.

The vertical shifts of centres of buoyancy and gravity can be neglected. The $m_{pontoon}$ and $\nabla_{pontoon}$ are just between 0 and 2 % of the total displacement and the vertical distances from the original centres they act upon are relatively small. Therefore the vertical shift is in the order of centimetres.

Results concept 4 at initial load case

The heeling moment following the lift scenario described in 4.2, is for 65% compensated by the pontoon. The ship will heel 1 degree. See figure 6.15.

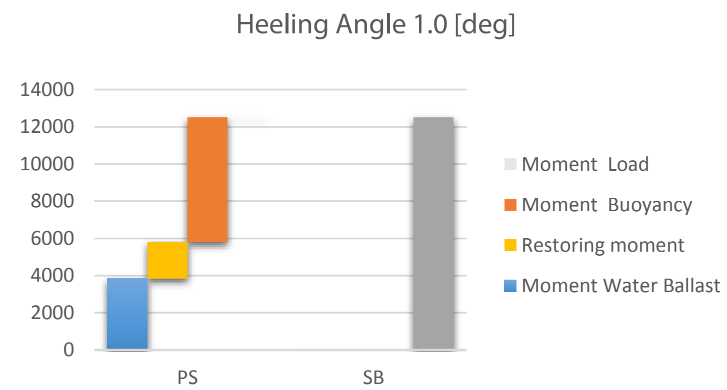


Figure 6.15: Static moments concept 4 in $ton.m$ [500ton,25m,5min]

Before Lift		
GM_initial	6.39	[m]
T_0ø	11.03	[s]

After Lift incl water ballast		
GM_eff	5.43	[m]
T_0ø_eff	12.0	[s]

Q heeling pumps	2100	[m3/hr]
Displaced ballast water	175	[m3]

L allong vessel	11	[m]
B	12.5	[m]
h	5	[m]
T	3	[m]

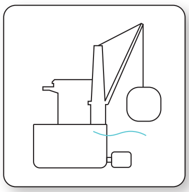
Figure 6.16: Result summary concept 4 in $ton.m$ [500ton,25m,5min]

Different than from the other concepts, concept 4 has a significantly higher $\overline{GM}$ and lower $T_{0ø}$. Both before and after lift, as can been seen above in figure 6.16. Natural periods of 10 seconds and lower can influence the workability.The largest moment to counter act the heeling moments comes from the buoyancy of the pontoon. Therefore it must be have a draught of 3.5 meters with a resulting force of 480 ton.

discussion

With the installation time involved with the pontoon it is unlikely that this concept will result in a time saving solution. Even if this could be automated it will be to expensive. And from the load side of view, being at the water surface will result in moment osculations caused by the waves.

6.2.5 Concept 5: Submerged buoyancy tank



This concept is similar to concept 4. The difference is that this buoyancy tank is not at the waterline. So the waves hardly have influence on the tanks. A submerged tank is hard to install. An external system has to be designed to pump the air into the buoyancy tank to make the buoyancy force variable. An advantage of the system could be that the buoyancy force of the tank can quickly be reduced by draining the tank.

Capacity

The capacity is determined by the water displacement and the weight of the pontoon. Similar as for concept 4, the equation 6.8 is used to calculate the moment created by the pontoon.

Results concept 4 at compare scenario

The heeling moment following the lift scenario described in 4.2, is for 54% compensated by the pontoon. See figure 6.17.

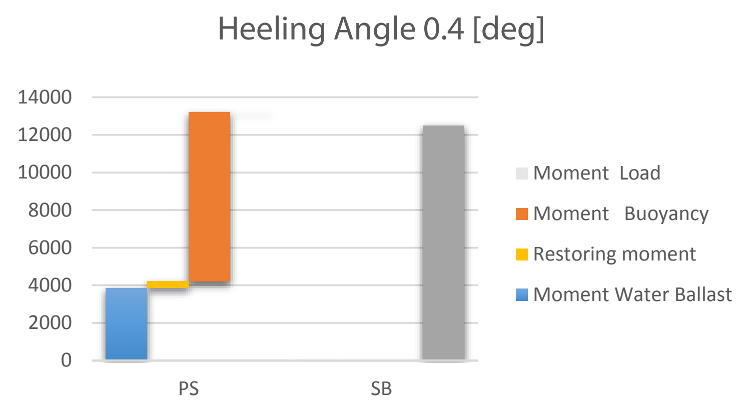


Figure 6.17: Static moments concept 5 in $ton.m$ [500ton,25m,5min]

Before Lift		
GM_initial	3.50	[m]
T_0ɷ	14.90	[s]

After Lift incl water ballast		
GM_eff	2.55	[m]
T_0ɷ_eff	17.5	[s]

Q heeling pumps	2100	[m3/hr]
Displaced ballast water	175	[m3]

L allong vessel	10	[m]
B	11	[m]
h	5	[m]

Figure 6.18: Result summary concept 5 in $ton.m$ [500ton,25m,5min]

Discussion

This concept has the same disadvantages as concept 4. This concept is even more complex and therefore expensive.

6.3 Design selection

In this section the method and considerations used for the concept evaluation and selection are shortly discussed. The information discussed here is a summery of the more detailed evaluation given in appendix G.4.

6.3.1 Weighted objectives method

For the concept evaluation the weighted objectives method [22] is used. With this method the various concepts are evaluated on the same criteria and given a certain score for each criteria. Then the different criteria are assigned a weight factor to be able to rank the most important criteria. The concept with the highest score is theoretically the best concept, according to this method. The objectives of the method are divided into four categories following from the design specifications given in 4.1. This creates a better overview of the strengths and weaknesses of the concepts. Also, some clustered objectives per category allow a less accurate evaluation than the other. The complete evaluation can be found in appendix G.4. The total scores of the concepts per category is shown in figure 6.19. A 100% score is the best score per category.

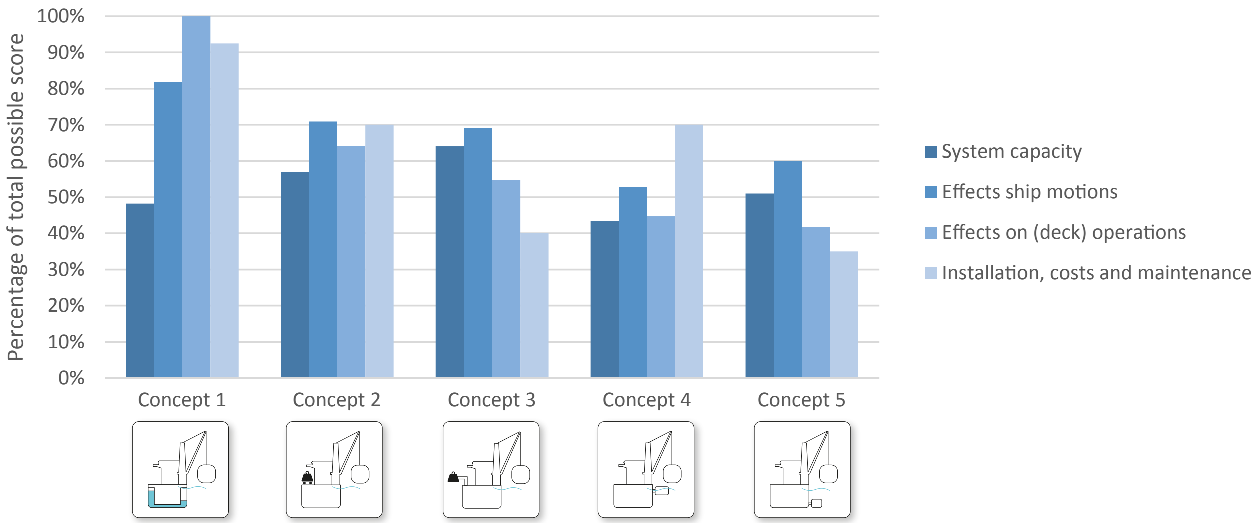


Figure 6.19: Scores of the concepts per category

Capacity

The capacity can not be evaluated by the principle solution only. Only the properties of a completed design can tell the true capacity. So the capacity depends how powerful the solution is designed. Concept 2, 3, 4, and 5 can potentially quickly start and stop its actions and therefore have and higher score than concept 1. Also the capacity range of concept 1 is smaller because of the pump limitations.

Effects on ship motion

The effect on ship motions are for all concept according to the design criteria so this category is not significantly influence the outcome.

Effects on (deck) operations

Concept 1 has a perfect score on the operations effect category because the system is concealed in the ship hull and does not interfere with the crew and no space is lost. Concept 4 and 5 have a low score here because of the extra handling of equipment to install the buoyancy tanks.

Installation, maintenance and repair

Concept 1 is an adaptation of the current system on board. The other concept requires custom designed equipment and is therefore expected to be more capital intensive.

6.3.2 Conclusion of the design selection

Because of the limited potential time savings for the lifting phase, it is hard to make a solution profitable. As the project with repetitive lifts above 300 ton are uncommon, the concept with best capacity does not have to be the winner. Therefore the concept with the best combination of moment rate range and low investment will be the best for Jumbo's case. Concept 1 is expected to be the most economic investment with no influences on deck operation and with a sufficient capacity. So concept 1 is going to be investigated further and will be tested with varying crane motions and volumetric flow rates.

Chapter 7

System model

The scope of the model is to calculate the moment changes over time. These moment changes govern the heeling angle. The resulting moment is expected to change relatively slow over time and the heeling angle are small so that the system is operating in the linear part of the stability curve. Therefore the heeling angle be found as given in equation 3.9. From the effects causing heeling mentioned in 3.1.4 the model focusses on the moment caused by the ballast tanks and the crane. The equilibrium of moments used is

$$M_{restoring} + M_{Ballast} + M_{Crane} = 0 \tag{7.1}$$

The static heeling angle on a certain moment is time is than given as

$$\phi_{heeling_{static}}(t) = sin^{-1}\left(\frac{M_{Ballast}(t) + M_{crane}(t)}{\Delta \cdot \overline{GM}_{eff}}\right) \tag{7.2}$$

This equation give a good estimation of the static heeling angle. But the roll motion of a ship experiences minimum damping. Therefore the ship dynamics must also be taken into account. To calculate the roll angle over time the wave, ballast and crane moments need te be determined. These moments influence the ship motions. The ship motions on their turn influence the ballast and crane moments again. This principle is illustrated in figure 7.1. The actuators of the pumps and crane are scripted to simulate the lifting operations.

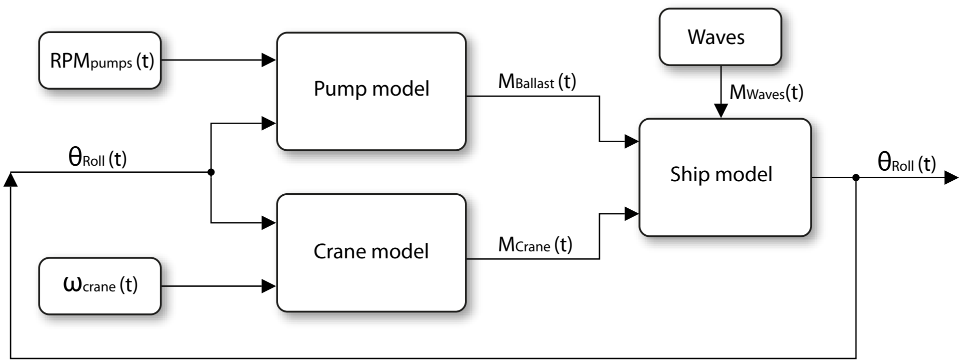


Figure 7.1: Model set-up

This chapter will elaborate on the calculation on the models of the ballast system, the crane and the capabilities of the system when variable frequency drivers are added.

7.1 The ballast system model

As discussed in 3.3 the tank arrangement consists of 12 wing tanks and 3 ballast pumps. All interconnected. In anti-heeling mode however, the model is divided in to three different anti-heeling combinations of four wing tanks and a pump. See figure 7.2. Per anti-heeling combination, the flow goes from one starboard wing tank to one portside wing tank, or the other way around. The cranes are on starboard, so generally the flow goes from starboard to portside.

The ballast moment is determined by the 12 water volumes V_i of the tanks and their lever arms r_i. The total moment is given as the sum of these 12 individual moments

$$M_{Ballast}(t) = \rho g \sum_{i}^{12} r_i \cdot V_i(t) \tag{7.3}$$

To determine the volumes of the ballast tanks over time a model has been made to determine the volumetric flow rates of the three pumps. For the model the system is split into three tank pairs because the one pump pumps from one ballast tank to one other ballast tank. The pipe flow calculations are based on Bernoulli and is elaborated upon below and in appendix H.

7.1.1 The Bernoulli equation

The amount of mechanical energy available in a pipeline flow of fluid can be quantified in the Bernoulli equation [16]. Named after Daniel Bernoulli. When a flow is steady, frictionless and incompressible the relation at any location is

$$h + \frac{P}{\rho g} + \frac{v^2}{2g} = constant \tag{7.4}$$

Where h is the elevation above datum in meters, P the pressure in Pascal, ρ the density of the fluid in kg/m^3, g the gravitational acceleration in m/s^2 and v the velocity of the fluid in m/s.

Each term in equation 7.4 represent a head in meters. Head can be considered as energy per unit gravity force. The equation is also commonly expressed in pressure form. The three term can be considered as the potential energy, flow energy and kinetic energy. Along the flow the sum of these energy term is constant. However, due to mechanical energy dissipation of the flow between two locations, energy is lost. So the relation becomes

$$h_1 + \frac{P_1}{\rho g} + \frac{v_1^2}{2g} = h_2 + \frac{P_2}{\rho g} + \frac{v_2^2}{2g} + \sum_{i=1}^{n} h_{loss_i} \tag{7.5}$$

Where $\sum_{i=1}^{n} h_i$ represents the sum of al the losses due to mechanical energy dissipation and i an element with head loss (piping, fitting, valves, bends, etcetera).

7.1.2 Assumptions

Although the Bernoulli equation is widely used for streamline flows, it contains important restrictions. A more general relation for a flow with one inlet and one outlet is the steady-flow energy equation [25]. This relation takes friction, heat transfer, shaft work and viscous work into account. These effects are explained in appendix H. To summarize, the following assumptions are made:

- Steady flow

- Incompressible flow
- Flow along a single streamline
- No heat transfer

7.1.3 Pump system equation

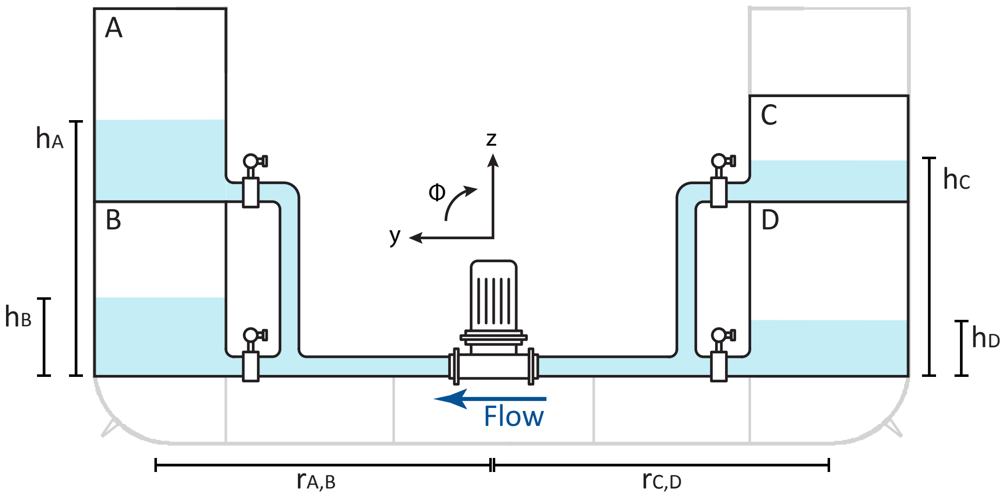


Figure 7.2: Anti-heeling model of one pump system

Figure 7.2 shows a simplification of pump and ballast tanks combination. When adding a pump to equation 7.5 the expression of the flow from starboard to portside becomes

$$\frac{P_{C,D}}{\rho g} + \frac{v_{C,D}^2}{2g} + h_{pump} + \Delta h = \frac{P_{A,B}}{\rho g} + \frac{v_{A,B}^2}{2g} + \sum_{i=1}^{n} h_{loss_i} \tag{7.6}$$

Where h_{pump} is the manometric head that is delivered by the pump in meters and Δh the geometric head between the water surfaces of the connected tanks.

The geometric head is both depending on the volumetric flow as on the rolling angle of the ship.

$$\Delta h = h_{C,D} - h_{A,B} - (r_{A,B} + r_{C,D})\sin(\phi) \tag{7.7}$$

where the height of the water level in a reservoir can be calculated by integrating the velocity of the water surface

$$h(t) = h_{initial} \pm \int_0^t \frac{Q(t)}{A} \cdot dt \tag{7.8}$$

When the current tank levels h_i are known, the volumes can easily been found with $V_i(t) = h_i(t) \cdot A_i$. Considering relatively small roll angles $r_i \cdot cos(0) \approx r_i \cdot cos(\phi)$ the moment of one ballast tank i is given as

$$M_{ballast_i}(t) = \rho\ g \cdot r_i \left(V_{initial} \pm \int_0^t Q(t) \cdot dt \right)_i \tag{7.9}$$

with

$$Q_i(t) = v_{pipe_i}(t) \cdot A_{pipe} \tag{7.10}$$

Taken the sum of all the moments will give the resulting moment caused by the water ballast.

$$M_{Ballast}(t) = \sum_i^{12} M_{ballast_i}(t) \tag{7.11}$$

7.1.4 Head loss

Mechanical energy is dissipated due to friction in the flow through the pipeline. The total head loss is the sum of the pipeline resistance and flow obstructions like bends and valves.

Straight pipelines

The frictional head loss of a water flow through a straight pipeline is determined using the Darcy-Weisbach equation [16].

$$h_{loss,pipe} = f \frac{L}{D} \frac{v_{pipe}^2}{2 \cdot g} \tag{7.12}$$

Where f is the flow friction coefficient, L and D the length and diameter of the pipe in meters, v_{pipe} the flow velocity in m/s and g the gravitational acceleration in m/s.

The flow friction coefficient f, or Darcy friction factor, of a duct flow is depending on the surface roughness ϵ and the present Reynolds number [25]. The Reynolds number of a duct flow is given as

$$Re_d = \frac{\rho \cdot v \cdot D}{\mu} \tag{7.13}$$

Where μ is the viscosity of the fluid in kg/ms depending on the temperature. When the Reynolds number and the surface roughness are known, the friction factor can be extracted from the Moody chart given in appendix J figure J.2 or calculated the formula given by Haaland [25]

$$\frac{1}{f^{1/2}} \approx -1.8 log \left[\frac{6.9}{Re_d} + \left(\frac{\epsilon/D}{3.7} \right)^{1.11} \right] \tag{7.14}$$

Obstructions

Obstructions cause flow separation and dissipate energy. The obstructions in the system of the J-class are butterfly valves and 90 and 120 degrees bends. It is hard te determine these losses. Common practise is to use a friction coefficient. These factors are empirically determined and discussed in appendix J. The head losses due to valves and bends are given as

$$h_{loss,valves} = K_{valve} \frac{v_{pipe}^2}{2 \cdot g} \tag{7.15}$$

$$h_{loss,90^\circ bend} = K_{90^\circ bend} \frac{v_{pipe}^2}{2 \cdot g} \tag{7.16}$$

$$h_{loss,120^\circ bend} = K_{120^\circ bend} \frac{v_{pipe}^2}{2 \cdot g} \tag{7.17}$$

$$h_{loss,inlet} = K_{inlet} \frac{v_{pipe}^2}{2 \cdot g} \tag{7.18}$$

Where K are the loss coefficients depending on the obstructions.

Total head loss

All the named head losses are depending on $v_{pipe_i}^2$. De pipe diameter does not change over the whole length, so the coefficients can be added together. The head loss of one tank pair becomes

$$h_{loss_i} = \frac{v_{pipe_i}^2}{2 \cdot g} \left(f \frac{L}{D} + N_{valves} K_{valve} + N_{90bends} K_{90bend} + N_{120bends} K_{120bend} \right)_i \tag{7.19}$$

were N is the number of that specific object in the piping route.

7.1.5 Pump curve

The energy added to the system is generated by a centrifugal pump. How a centrifugal pump works is explained in appendix K. This energy added can be extracted from the pump curve provided by the manufacturer. This curve shows the relation between the delivered head with a corresponding volumetric flow rate at a certain rotational speed of the pump shaft. See figure 7.3.

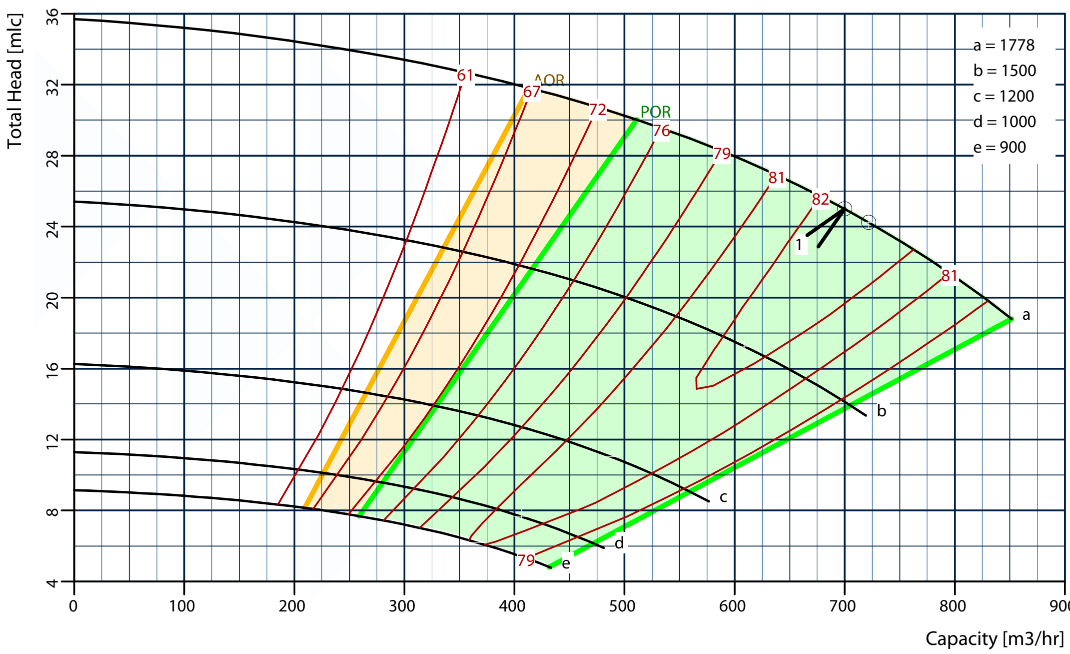


Figure 7.3: Pump curve 700 m^3/hr centrifugal pump

Using the simplified equation L.7 discussed in appendix L the following equation is used to find the so called working point on the pump curve corresponding to a certain ω_{shaft}.

$$h_{pump}(t, \omega_{shaft}) = h_{loss}\left(Q(t, \omega_{shaft})\right) + h_{static}\left(\int_0^t Q(t, \omega_{shaft})dt, \phi_{roll}(t)\right) \tag{7.20}$$

To make the fluid flow the pump head should be larger than the sum of head loss and the static head. The head loss depends on v_{pipe}^2 and so on the $Q(t)$. The static head depends on ϕ_{roll} and the water levels in the tank and so on the $\int_0^t Q(t)$.

Affinity law

To produce pump curves for different than the curves provided, affinity laws can be used [16].

$$\frac{h_{pump,\omega_1}}{h_{pump,\omega_2}} = \left(\frac{\omega_1}{\omega_2}\right)^2 \tag{7.21}$$

This relation is used to determine the $h_{pump} - Q$ relation for the varying ω_{shaft}.

7.1.6 Effect of roll motion on the ballast system

The roll motion of the ship causes an oscillating variation of the hydrostatic head over the roll period of the ship as given in equation 7.7. To demonstrate the effect on the moment rate of the ballast system, two examples of the moment rates including and excluding the roll motion are displayed below.

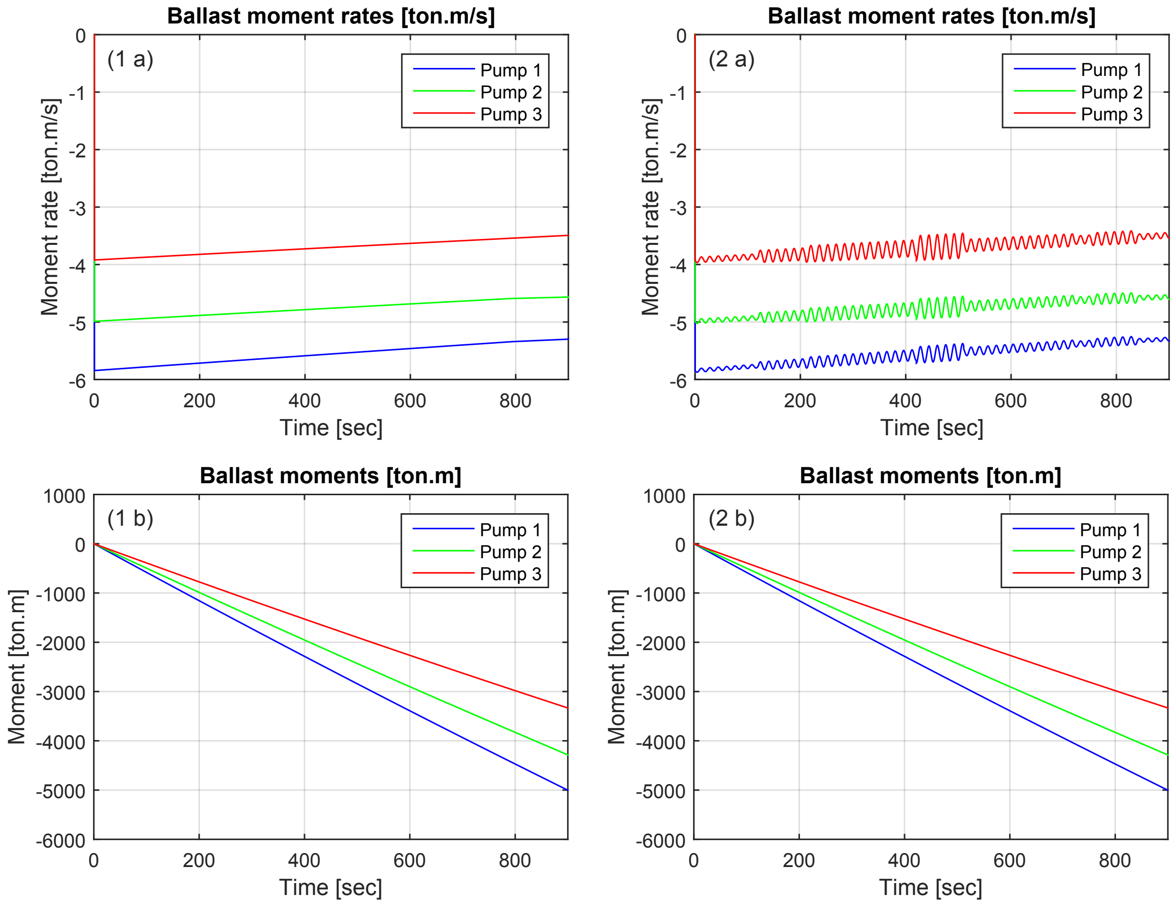


Figure 7.4: (1) Ballast system without roll motions. (2) Ballast system with roll motions.

In figure 7.4 examples of the moment rates of the three pumps are shown. The rpm's are constant over time. The moment rate decreases slightly because of the water transport the hydrostatic head decreases. The only difference is the oscilating variation in figure (2 a), but the average moment rate is similar to the rate of figure (1 a). To see what the effect on the total Ballast moment is, both rates are integrated over time.

$$M_{ballast}(t) = \int_0^t \dot{M}_{ballast}(t)dt \tag{7.22}$$

The results are shown in (1 b) and (2 b) and show that the oscillations do not have an significant effect on the total ballast moment. The fluctuations equal each other out as their time domain is small compared with the time domain of the ballast system. $T_{roll} << t_{antiheeling}$. From the result one can conclude that the roll motion does not significantly effect the total ballast moment if the roll motions have relatively constant amplitudes and periods over their own time domain of T_{roll}.

Effect of the heeling angle on the ballast system

As with the rolling angle the heeling angle also has an effect on the hydrostatic head. But $\phi_{heeling}$ does not oscillate, so the heeling angle will effect the average moment rate and therefore the total moment. The additional static head of a tank pair will be given as

$$\Delta h_{heeling} = (r_i + r_j) \cdot sin(\phi_{heeling}) \tag{7.23}$$

where r_i and r_j are the lever arms of the centres of gravity of the ballast tanks at starboard and port side.

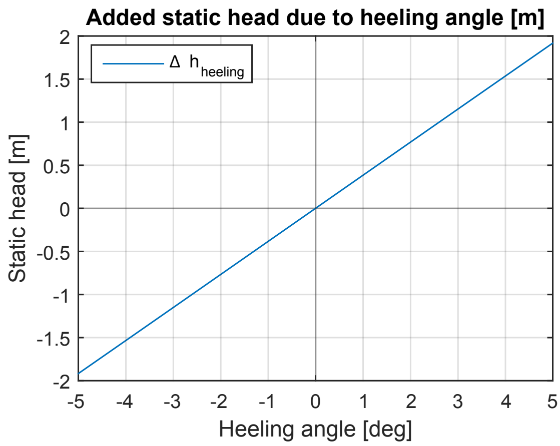


Figure 7.5: Additional static head due to heeling when pumping from SB to PS.

Figure 7.5 shows the additional static head due to heeling. When pumping from PS to SB the graph will change sign. The centres of gravity of the tanks are all between 10.6 and 11.1 meters, so $(r_i + r_j)$ has been chosen to be 22 to illustrate the effect in figure 7.5. A variation of one meter in the static head can result in a total moment rate change in the range of 0.3 and 0.6 ton.m/s at 1785 and 900 rpm respectively. These rates correspond to a rate loss between the 1.5 and the 7 %. Its significance is covered in the results of chapter 8.

Free surface effect

When a ship is in rolling motion the liquids in the tanks cause a free surface effect, as mentioned in 3.1.4. Due to the moving liquid a moment is created.

$$M_l = \rho \cdot i_B \cdot tan(\phi) \tag{7.24}$$

where i_B is the moment of inertia of the liquid surface [2].

According to the captain of the Jumbo Javelin the free-surface effect is neglectable during lift operations. The widths of wing tanks are relatively small and so are the roll angles. Therefore the effect is not taken into the scope.

7.1.7 Ballast moment ranges

To be able to anticipate on the varying crane motions, the rotational speeds of the pumps are varied. The current pumps installed have a range of 900 till 1785 rpm. To find the upper and the lower boundaries of the capacity, the total moment rates of the three pumps combined are calculated. The moment rate and corresponding power required of each pump depends on the ω_{shaft} and the static head.

$$\dot{M}_{Ballast}(t) = \sum_{i=1}^{3} \dot{M}_{pump_i}(rpm_i, \Delta h_i, t_i) \tag{7.25}$$

$$P_{total}(t) = \sum_{i=1}^{3} P_{pump_i}(rpm_i, \Delta h_i, t_i) \tag{7.26}$$

The results are shown in figure 7.6. The green line is the upper limit when all three pump are running at 1785 rpm and the red line the lower limit when all three pumps are running at 900 rpm. The horizontal axis show the vertical difference between the starboad and the portside water surfaces. All three tanks are set equal on each side.

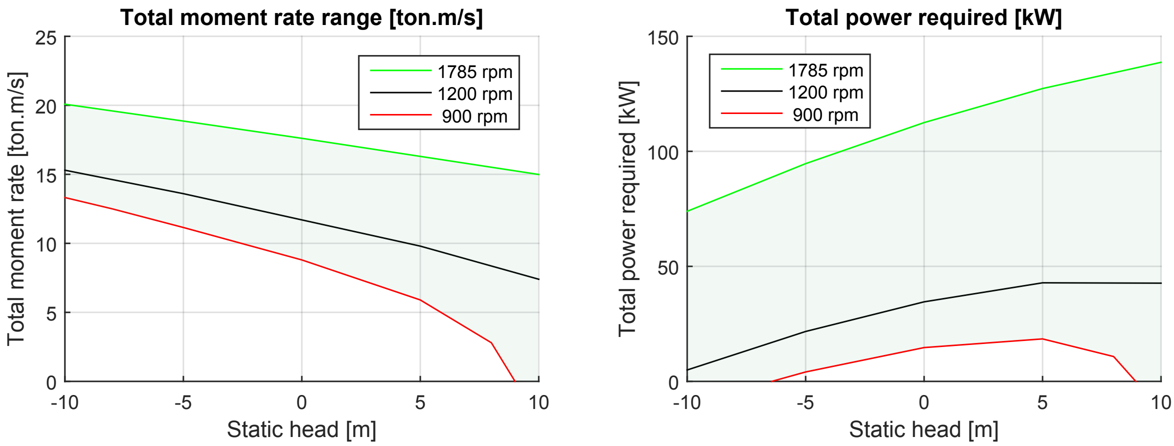


Figure 7.6: Working ranges of the ballast system

Without comparing the ballast moment rates with the crane moment rates, figure 7.6 does not say much. Figure 7.7 shows the moment rates at a certain time of the crane when the slewing speed is kept constant. In this figure the roll motions are not taken into account.

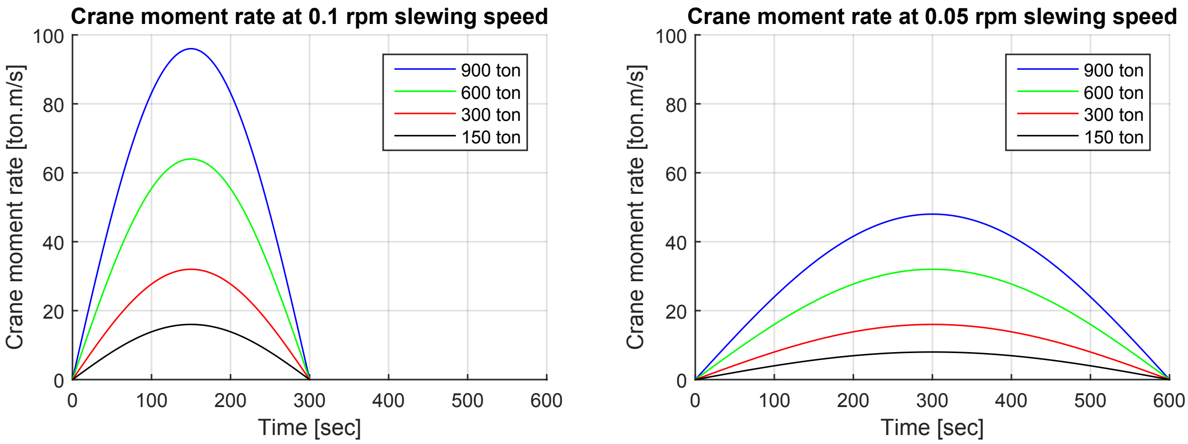


Figure 7.7: Crane moment rates over a 20 meter transverse displacement

In the given time frames the peak moment rates easily exceed the maximum moment rate of the ballast system. However, this does not mean that the ballast system in not capable to generate a large enough moment to compensate for the crane moment. Therefore the total moment difference should be checked at a certain time. If the relation 7.1 is written in the form of equation 7.22 the relation becomes

$$M_{heeling}(t) = \int_0^t \dot{M}_{ballast}(t)dt + \int_0^t \dot{M}_{crane}(t)dt \tag{7.27}$$

In operational practice the crane moment rate graphs will never look like the perfect sinus as in figure 7.7. Depending on the lifting job, crane can be moved in any way possible and started and stopped at all times. In the next section the crane moment model is elaborated upon which is used to simulate crane operations.

7.2 Crane model

The slewing crane with the load is considered as a transversely displaced hanging load as only moments in the y-z plane are considered, as explained in 3.1.4. The heeling moment caused by the shifting load on the crane is governed by the y coordinate of the crane tip. Additionally the lever arm of the load can increase or decrease depending on the roll angle of the ship. The equation of the crane moment is given as

$$M_{crane} = m_{load} \cdot \left(y_{cranetip} + sin(\phi) \cdot l_{craneline}\right) \tag{7.28}$$

The distance $y_{cranetip}$ is determined by two angles. The first is the horizontal slewing angel of the crane and the second is the vertical boom angle. With y_{cor} as the distance from the centre line of the ship to the centre of rotation of the crane, the expression becomes:

$$y_{cranetip} = y_{cor} - cos\left(\theta_{slewing}(t)\right) \cdot cos\left(\theta_{boom}(t)\right) \cdot l_{boom} \tag{7.29}$$

where $\theta_{slewing}$ is the horizontal slewing angle (see figure 7.8) of the crane and θ_{boom} the vertical angle of the boom.

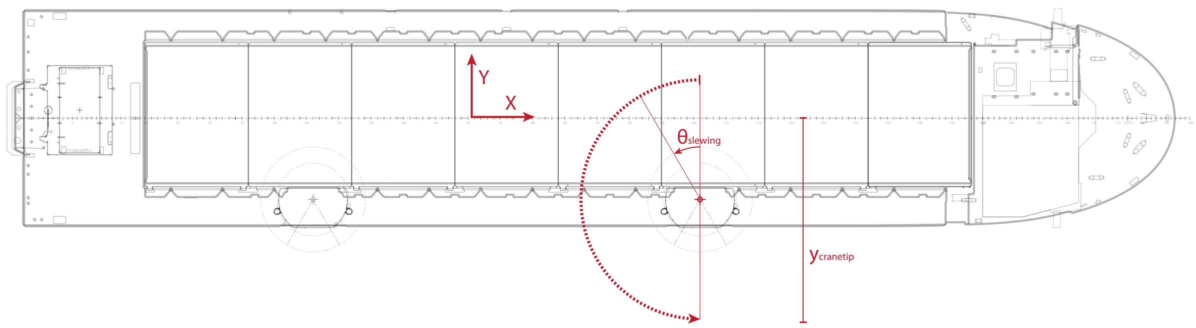


Figure 7.8: Transversal distance of the cranetip relative to the centreline

During operation the slewing speed and the boom angle are manually controlled and can vary in speed. To determine the angle the integrals of the rotational speeds are taken.

$$\theta_{slewing}(t) = \int_0^t \omega_{slewing}(t)dt \tag{7.30}$$

To see the effect of varying the slewing speed, it is compared with a constant slewing speed in the following subsection.

7.2.1 Constant vs varying slewing motion

Preferably the slewing speed $\omega_{slewing}(t)$ is constant. The moment rate will be a smooth sinus function. See figure 7.9. Depending on the magnitude of constant slewing speed, this crane moment rate is relatively easy to compensate with the ballast system. However, this is hardly ever the case. To mimic the slewing speed variations, starts and stops, a piecewise function has been used to make $\omega_{slewing}(t)$ time depending.

$$\omega_{slewing}(t) = \begin{cases} \omega_1 & t \leq t_{ref1} \\ \omega_2 & t_{ref1} \leq t \leq t_{ref2} \\ \omega_3 & t_{ref2} \leq t \leq t_{ref3} \\ \omega_4 & t_{ref3} \leq t \\ etc. & \end{cases}$$

Figure 7.9 shows the difference between a constant and a varying slewing speeds. To compare the two the total slewing time and transverse displacement are kept equal. For the constant slewing speed the development of the moment rate 1a and total moment 1b the development is smooth for both. When the $\omega_{slewing}(t)$ is varying over time and the crane is stopped a few times, a moment rate pattern as shown in 2a is obtain fo example. Due to the discontinuities 2a looks very different than 1a. When looking at the total moment the figures 1b and 2b have more similarities despite the slope differences. There are no uncontrollable extreme developments in the moment curve 2b caused by the extreme moment rate pattern of 2a.

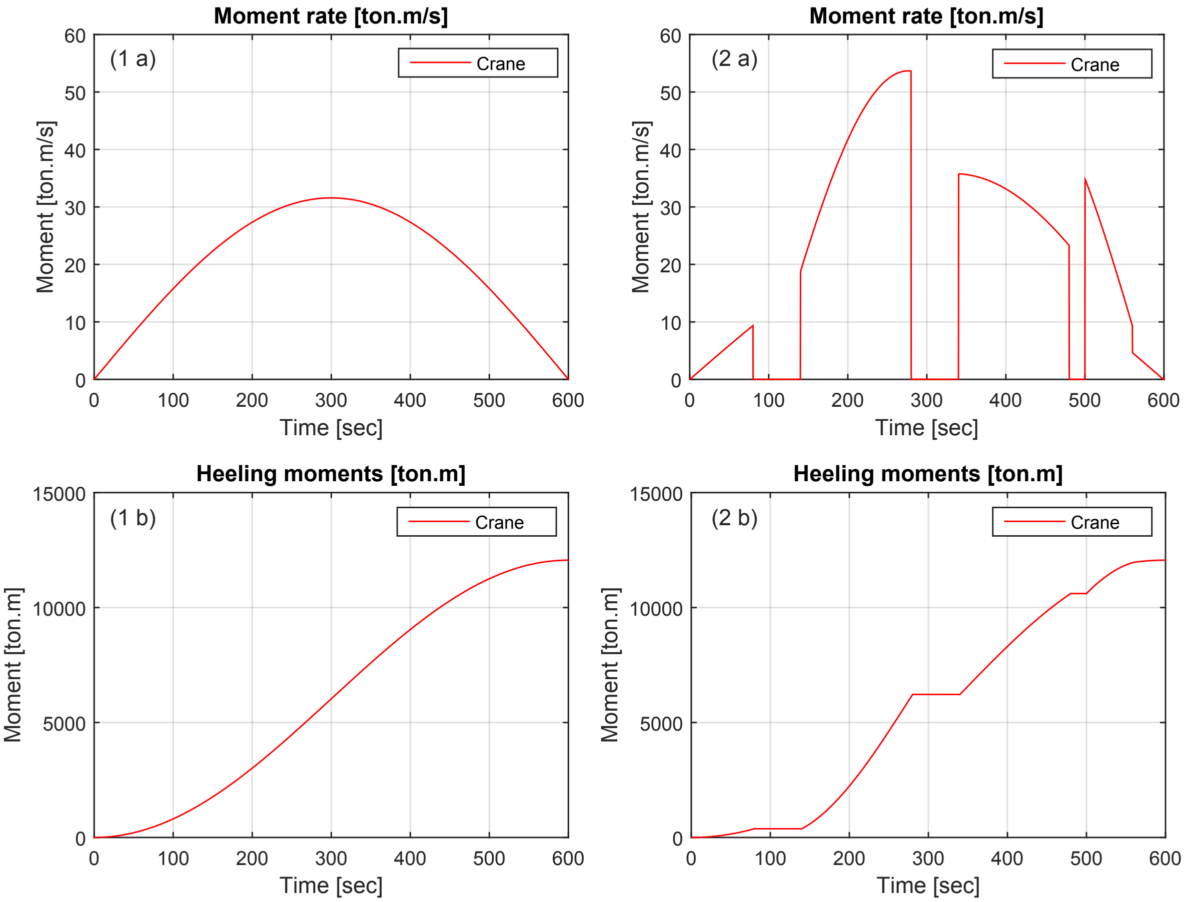


Figure 7.9: 600 ton 20m 180 deg

7.2.2 Effect of roll motion on the crane moment

The centre of gravity of the hanging load on the crane is at a vertical difference $l_{craneline}$ from the crane tip. When the ship rolls the crane line will stay vertical and results is an sinusoidal variation $sin(\phi_{roll}(t)) \cdot l_{craneline}$ of the horizontal distance of the load and the centre of rotation of the ship. The expression of the crane moment becomes

$$M_{crane}(t) = m_{load} \cdot \left(y_{cranetip}(t) + sin(\phi_{roll}(t)) \cdot l_{craneline}\right) \tag{7.31}$$

Figure 7.10 shows the same situation as in figure 7.9 but with the ship exposed to 1.5° to 4.5° roll angles. The load is 600 ton and the crane line is 30 meters. This results in a quite extreme situation but it give an good illustration of the effect of roll motion on the moment rates.

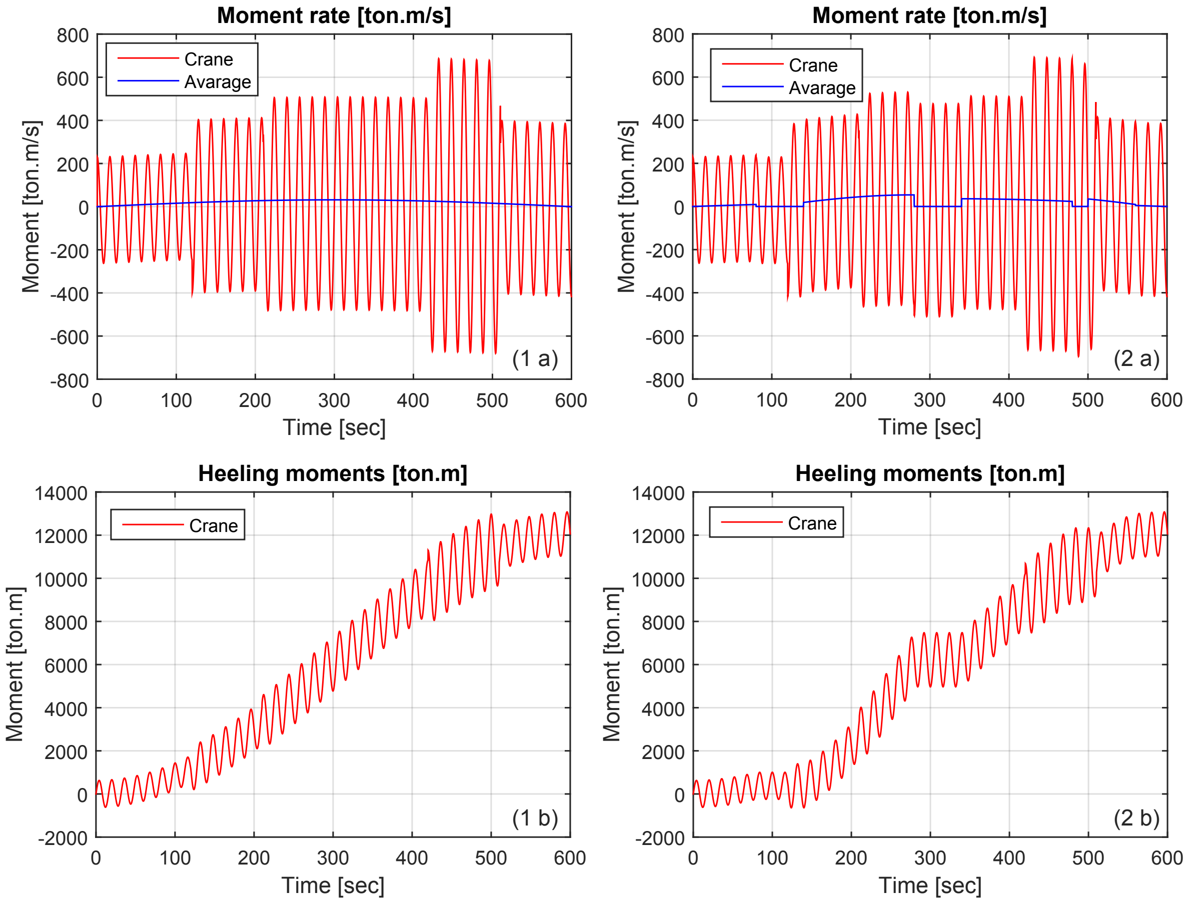


Figure 7.10: 600 ton 20 meters 180 deg

To see the effect roll motion to the total crane moment the moment rate is integrated over time

$$M_{Crane}(t) = \int_0^t \dot{M}_{Crane}(t)dt \tag{7.32}$$

The transverse displacement of the load by the slewing motion of the crane is relatively slow comparing with the transverse oscillation of the hanging load caused by the roll motion. Comparable with the effect of roll motion on the ballast system, the oscillations of the hanging load do not have an significant effect on the average total moment causing heeling. But different from the case with the total moment of the ballast system, the oscillations do have a notable effect on the total crane moment $M_{Crane}(t)$.

7.3 Ship model

An in-house developed ship model of the Jumbo J-class is used to analyse the dynamic response of the ship with the crane and ballast moments. This model is written by Jasper van Heijst in Matlab and validated with Orcaflex and model test results from MARIN [23]. The equation of motion is based on Newtons first law and given as

$$\sum_{k=1}^{6} M_{jk} \cdot \ddot{q}_k(t) = F_j^{HD} + F_j^{HS} + F_j^{D} + F_j^{A} + F_j^{E} \qquad j = 1...6 \tag{7.33}$$

where the right hand system gives a summation of forces and moments. The forces F_j^{HD}, F_j^{HS}, F_j^{D}, F_j^{A}, F_j^{E} stand for hydrodynamic, hydrostatic, diffraction, actuator and environmental forces. To analyse the ship motion for this project the ballast and crane moments are added at the right hand side of the equation as illustrated in figure 7.1.

7.3.1 Effect of crane moment on roll motion

When the crane is moved slow enough the system the maximum heeling angle can be determined with the static relation given as equation 7.34.

$$\phi_{heeling} = sin^{-1}\left(\frac{m_{load} \cdot (dy + l_{craneline} \cdot sin(\phi_{heeling}))}{\nabla \cdot \overline{GM}_{eff}}\right) \tag{7.34}$$

The equation shows that when the heeling angle increases the lever arm also increases and therefore the heeling angle is increased again until an equilibrium is found. To illustrate the effect of the length of the crane line the difference between moving an object on deck or with a crane is shown in figure 7.11.

	Weight	150	300	600	900	1200	1400	[ton]
Fixed load	Angle	0.2	0.4	0.8	1.1	1.5	1.8	[deg]
Hanging load (30 m)	Angle	0.2	0.5	1.3	2.8	7.1	13	[deg]

Figure 7.11: Static heeling angle at $\overline{GM}_{eff} = 2.25$ m

For the fixed load $y_{craneline} = 0$ m and for the hanging load $y_{craneline} = 30$ m. In both cases the crane tip is given a traverse displacement of 1 meter but the hanging load will cause a larger heeling angle because the lever arm increases. For high loads the system becomes more sensitive. To see the effect on the ship dynamics the 1 meter transverse displacement is applied at different speeds.

600 ton in 30 seconds

Figures M.4 shows the roll angles when the displacement is done in 30 seconds. The ship will roll to its static heeling angle of 1.4° and has a very small oscillation. The static heeling angle corresponds with figure 7.11.

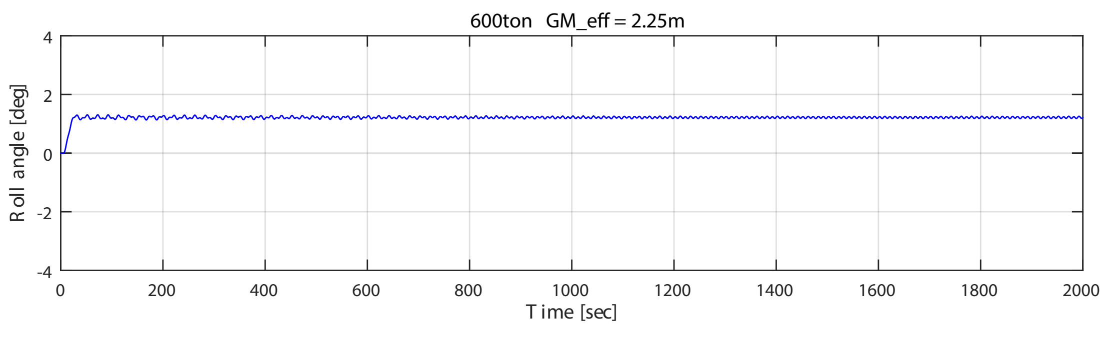


Figure 7.12: 1 meter transverse cranetip displacement in 30 seconds

600 ton in 5 seconds

If the load is moved fast enough, an impulse is created and will make the ship oscillate around the static equilibrium. The amplitude of these motion depends on the moment rate caused by the crane. Figure 7.13 shows the ship response when the 600 ton hanging load is moved in 5 sec. The ship will roll up to a angle of 2.4° and then will damp out very slowly until the static heeling angle of 1.4°.

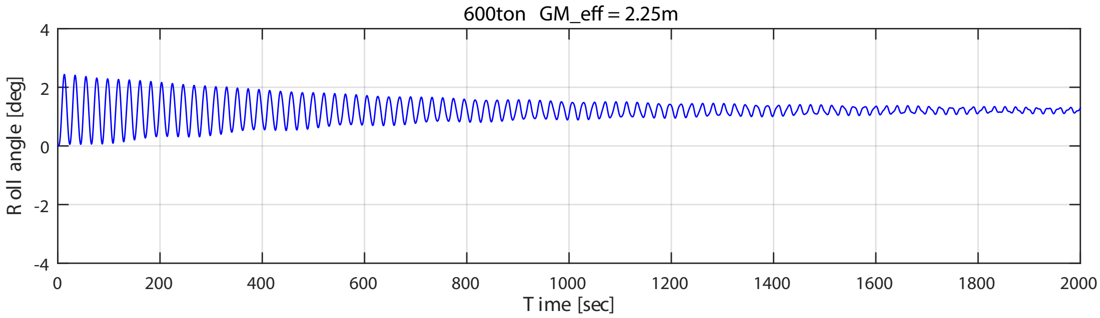


Figure 7.13: 1 meter transverse cranetip displacement in 5 seconds

600 ton in 5 seconds in waves

In offshore conditions the ship is exposed environmental forces from which waves forces are governing. The ships natural period is 16.2 seconds but, as mentioned in 3.2, 90% of the waves periods is below 10 seconds. Conditions with a significant wave height above 2.5 meter is considered as not workable by Jumbo. Therefore the ship is exposed to JONSWAP waves spectrum with a T_p of 10 seconds and a H_s of 2.5 meters.

For most heavy lifts a heading of 165° is preferred as this heading result in the lowest roll motions. With these sea conditions the influence on the roll motion of the J-class appear to be minimal. Figure 7.14 shows the same 5 seconds transverse displacement as in figure 7.13 but with waves. The roll motion develops similar but is slightly distorted by the waves.

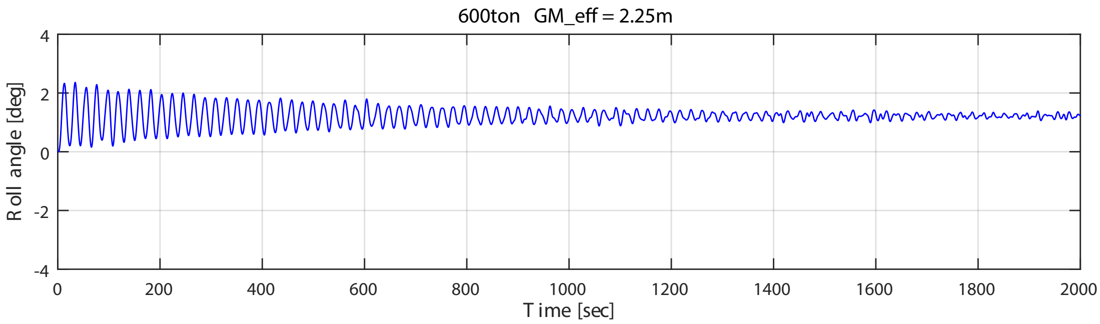


Figure 7.14: 1 meter transverse cranetip displacement in 5 seconds with $H_s = 2.5m, T_p = 10s$ and 165° heading

A 90° heading result in the highest roll motions. Figure 7.15 shows again the same 5 seconds transverse displacement but with 90° heading waves. In this case the equilibrium is still at the static heeling angle but the oscillating motions will not damp.

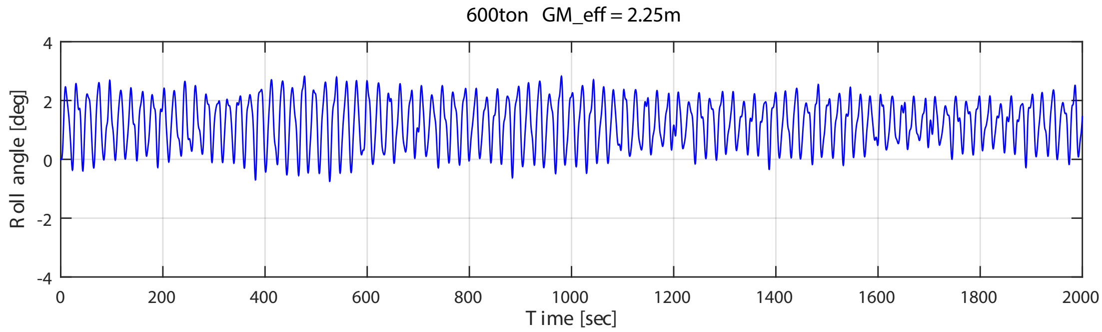


Figure 7.15: 1 meter transverse cranetip displacement in 5 seconds with $H_s = 2.5m, T_p = 10s$ and 90° heading

7.3.2 Effect of ballast moment on roll motion

The moment rates of the ballast system are generally lower and variate less than the crane moment rates. To see if the ballast system also creates an impulse causing the system to oscillate, all three pumps are actuated and stopped at the same time. Figure 7.16 show that the ballast system at full capacity does not cause the system to oscillate.

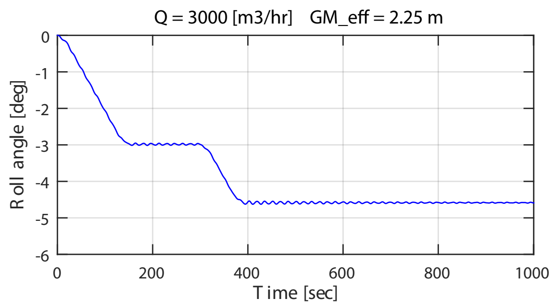


Figure 7.16: Water ballast induced roll angle with 2 start/stops

So the system is the most sensitive to the impulse created by the crane. In higher sea states with peak periods closer to the natural period, the system will experience larger heeling angels resulting in increased crane moments and impulses.

7.3.3 Coupled motions of ship and crane load

In this chapter the effects of rotational motions around the x-axis on the ballast system and the crane moment has been covered. The effect on the systems has been researched when the ship is exposed to different moments. In reality the wave induced motion of a crane ship are made up of a complex interaction of the body motions, elastic deformations of the hull and the crane, together with the vertical stretching of the crane line and swinging pendulum oscillations of the load [17]. However, all these interaction can be reduced to a system of 9 degrees of freedom and, as explained in appendix N, the system can be written as

$$M'\ddot{Z}' + B'\dot{Z}' + \left(K' + K_c\right) Z' = F' e^{i\omega t} \tag{7.35}$$

where Z' is containing surge, sway, heave, roll, pitch and yaw motions of the ship and surge, sway and heave motions of the crane load.

$$Z' = \left[\begin{array}{ccccccccc} X & Y & Z & \theta_x & \theta_y & \theta_z & X_c & Y_c & Z_c \end{array} \right]^T$$

According to Patel and Witz the motion coupling of the ship hull and the hook load can be considered through the vertical and lateral elasticities of the crane housing and jib and of the lift wires. This results in, considering the equation of motion 7.35, that the only coupling of the crane load and the ship motions is in the stiffness matrix K_c. This is a 9x9 matrix, but as this report focusses on the moments around the x-axis the fourth row is given below to display the coupled roll moments.

$$M_{c\theta_x} = \theta_x \begin{array}{c} \begin{array}{ccccccccc} X & Y & Z & \theta_x & \theta_y & \theta_z & X_c & Y_c & Z_c \end{array} \\ \left[\begin{array}{ccccccccc} 0 & z'k_t & -y'k_c & k_{\theta_x} + y'^2 k_c + z'^2 k_t & -x'y'k_c & -x'z'k_t & 0 & -z'k_t & y'k_c \end{array} \right] \end{array} \cdot Z'$$

where x', y' and z' are the coordinates of the crane load suspension point relative to the centre of gravity of the ship[1]. k_c is the vertical stiffness of the crane structure and wire. k_t is the transverse stiffness of the swinging load.

This teaches that sway, heave, roll, pitch and yaw motions of the ship and sway and heave motions of the crane load all contribute to the motion around the x-axis. Throughout the project all calculations are limited to motions in the y,z-plane. So the equation of motion will be rewritten considering only the motions in the x,y-plane.

Equation of motion for Y,Z plane

As figure 7.17 shows, the degrees of freedom reduces to five when surge, pitch and yaw motions of the ship and the surge motion of the crane load are left out of the equation. Equation 7.35 remains valid but with a new Z'.

$$Z' = \begin{bmatrix} Y & Z & \theta_x & Y_c & Z_c \end{bmatrix}^T \tag{7.36}$$

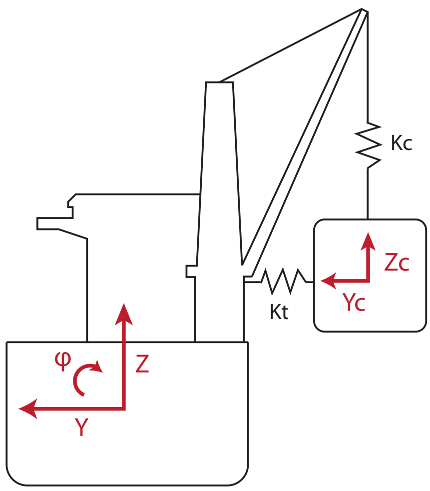


Figure 7.17: 5 DOF system in the X,Y-plane

Determine the M', B', K' and K_c matrices

Now the system in simplified the mass, damping and stiffness matrices are left te determine. Not all the five degrees of freedom are coupled. According to T. Perez and T.I. Fossen for slender

[1]These coordinates should actually be the relative to the centre of rotation, but to be able to work with equation 7.35 the centre of gravity is used. This is common practise as the ship's the centre of rotation is usually relatively close above the centre of gravity to from a stable system.

lateral symmetric ships the surge-heave-pitch and sway-roll-yaw motions are uncoupled when relative small angles are considered [18]. For the vessel motions given the system of figure 7.17 only sway and roll are coupled. This will result in a sway-roll added mass coupling and because of the lateral symmetry assumption the matrix is symmetric as well.

$$M' = \begin{bmatrix} m + a_{yy} & 0 & a_{y\theta_x} & 0 & 0 \\ 0 & m + a_{zz} & 0 & 0 & 0 \\ a_{y\theta_x} & 0 & I_{\theta_x\theta_x} + a_{\theta_x\theta_x} & 0 & 0 \\ 0 & 0 & 0 & m_L & 0 \\ 0 & 0 & 0 & 0 & m_L \end{bmatrix} \tag{7.37}$$

m and m_L are the mass of the structure and crane load, I the mass moment of inertia and a the additional mass of fluid entrained with the acceleration of the hull form.

The damping of the system can be divided by the hydrodynamic damping b and the structural damping b' associated with crane dynamics [17]. Similar to the added mass the hydrodynamic damping also has a sway-roll coupling and a symmetric matrix.

$$B' = \begin{bmatrix} b_{yy} & 0 & b_{y\theta_x} & 0 & 0 \\ 0 & b_{zz} & 0 & 0 & 0 \\ b_{y\theta_x} & 0 & b_{\theta_x\theta_x} & 0 & 0 \\ 0 & 0 & 0 & b'_{y_c} & 0 \\ 0 & 0 & 0 & 0 & b'_{z_c} \end{bmatrix} \tag{7.38}$$

The hydrodynamic damping factors are made up from radiation losses and linearised drag damping. The damping terms b'_{y_c} and b'_{z_c} can roughly be estimated as a small percentages, 1 and 5 percent respectively, of the critical damping according to Patel and Witz [17].

The stiffness matrix is divided in a hydrostatic part K' and crane coupling part K_c. For the hull motions only the heave and roll motion have restoring capabilities. So the K' is given as

$$K' = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ 0 & k_{zz} & 0 & 0 & 0 \\ 0 & 0 & k_{\theta_x\theta_x} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \end{bmatrix} \tag{7.39}$$

were the restoring "spring" terms are $k_{zz} = \rho g \cdot A_{wl}$ for the heave motion of the hull and $k_{\theta_x\theta_x} = \rho g \cdot \nabla \cdot \overline{GM}_{eff}$ for the roll motion [14].

The matrix K_c provides the coupling between the ship motion and the crane load motions. This is also a symmetric matrix but with many couplings.

$$K_c = \begin{bmatrix} k_t & 0 & z'k_t & -k_t & 0 \\ 0 & k_c & -y'k_c & 0 & -k_c \\ z'k_t & -y'k_c & y'^2k_c + z'^2k_t & -z'k_t & y'k_c \\ -k_t & 0 & -z'k_t & k_t & 0 \\ 0 & -k_c & y'k_c & 0 & k_c \end{bmatrix} \tag{7.40}$$

where transverse stiffness can be written as $k_t = m_L g/L$ and k_c depends on the crane and wire. The Y_c, Z_c, y' and z' are depending on the crane operation ($\theta_{slewing}, \theta_{boom}, l_{craneline}$) and the roll angle of the ship.

Excluding coupled motions from the model

Including the coupling crane load motions and ship motions into a dynamic model is a complex problem which goes beyond the scope of the project. Due to the oscillating nature of the dynamic motions and the short time domain relative to the heeling time domain, the decision has been made to focus on the moment balance described in this chapter because the oscillations do not significantly effect the average resulting heeling moment considering this simplified linear system.

Chapter 8

Results

In this chapter the results of the model are summarized and discussed. The concept of a water ballast system with variable frequency drivers (VFD) is compared with the current system. This is done for 5 different crane loads. For each crane load the effect of the ballast system on the development of the heeling angle and required power is analysed. The ballast system controls the starts, stops and frequency changes of the 3 pumps. This will be explain in 8.1.

8.1 Lift scenarios

The lift scenarios used for the results are based on the load case of section 4.2. But the system capacities are now evaluated under given varying slewing motions of the crane as explained in section 7.2.1. The lift scenarios differ in crane loads, duration and the number of interruptions of the crane motion. Crane loads of 150 and 300 ton represent the 80% of Jumbo's offshore project portfolio. Crane loads of 600 and 900 the represent the exceptional heavy offshore lifts. And the 1800 ton crane load represent the very extreme case. All loads are 20 meters transversely displaced by the slewing crane.

Ballast system control

To compensate for the crane motion the reference pump shaft rotations need to be chosen to generate enough water ballast. These shaft rotations correspond to certain water displacements of the pump as explained in the pump curve section 7.1.5. In the left graph of figure 8.1 the current system is shown. The pumps can either have 0 rpm (off) of 1785 rpm (on). On the right side the rotations of the system with VFD's are shown. This in this system the pump rotate between the 900 and 1785 RMP or been switched off. In the model the control of the pumps have been automated by a script as a reaction to the crane motion.

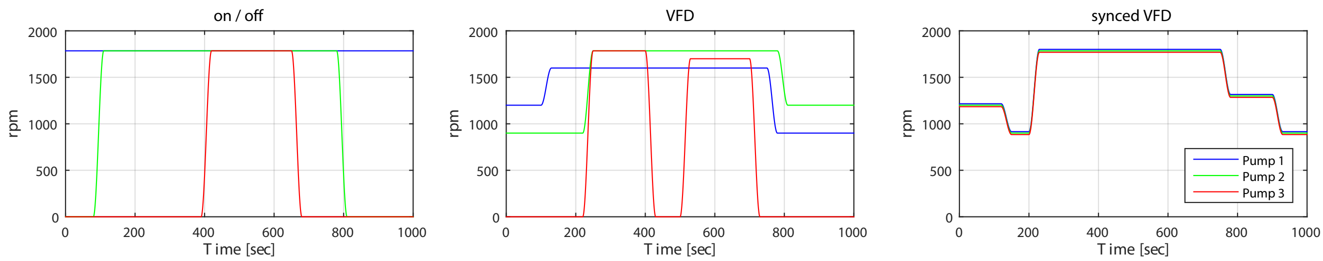


Figure 8.1: Examples of reference pump shaft rotations

8.2 Results of the lift scenarios

The model is provide with many signal scopes to keep track of the developments in the system. The most informative graphs are plotted on one data summary sheet every time the model is run. Figures 8.3 and L.1 in appendix L shows examples of this.

To compare the current system with the set-up including the VFD's the model results of the different lift scenarios are summarized in tables. The tables contain information about

- The duration of the lifting phase
- How often the crane was interrupted
- The maximum excitation angle
- The amount of power the system required
- Fuel consumption
- The rotational speed of the crane when in motion

For the *control* column the "on/off" indicates the use of the current system, the "VFD" use of individually frequency controlled pumps and the "synced VFD" means that the shaft rotation are equal for all pumps.

8.2.1 150 ton

For the 150 ton scenario a lift duration of 300 seconds is chosen with 3 crane interruptions. Because of the relatively small crane load, the total volume of displaced water is also relatively small. Therefore the use of less pumps is examined. The results for scenarios for the 150 ton lift are shown in figure 8.2.

Crane load [ton]	Time [sec]	Control [-]	no. pumps [-]	no. crane interuptions [-]	Largest angle [deg]	Power [MJ]	Fuel [kg]	RPM [rpm]
150	300	on/off	1	5	2.2	10.3	0.57	0.13
150	300	on/off	2	5	0.9	16.7	0.93	0.13
150	300	VFD	2	5	1.1	12.1	0.67	0.13
150	300	synced VFD	3	5	0.7	4.8	0.27	0.13

Figure 8.2: Model result summary of a 150 ton lift phase with $\overline{GM}_{eff} = 2.25m$

The table shows that the largest heeling angle is 2.2° . This is much lower than the 4° based on the criteria given in section 3.2.3. So all systems have sufficient capacity. The most remarkable result is that 3 pumps with synced VFD use 53% less energy than 1 pump on full speed. But as the energy is \$ 0.007 per MJ this saving is 4 dollar cents and can therefore be considered neglectable.

Figure 8.4 show the roll angle (1a), wave height (1b), heeling moments (2a) and moment rates (2b) when all 3 pumps run at constant 1000 rpm for 300 seconds. The roll angle graph 1a shows that the roll angle will hardly move away from the 0°. Due to roll motions and the 5 starts and stops of the crane, the moment rate of the crane fluctuates while moment rate of the ballast system is relatively constant. However these excitations are not big enough to give the system significant roll angles.

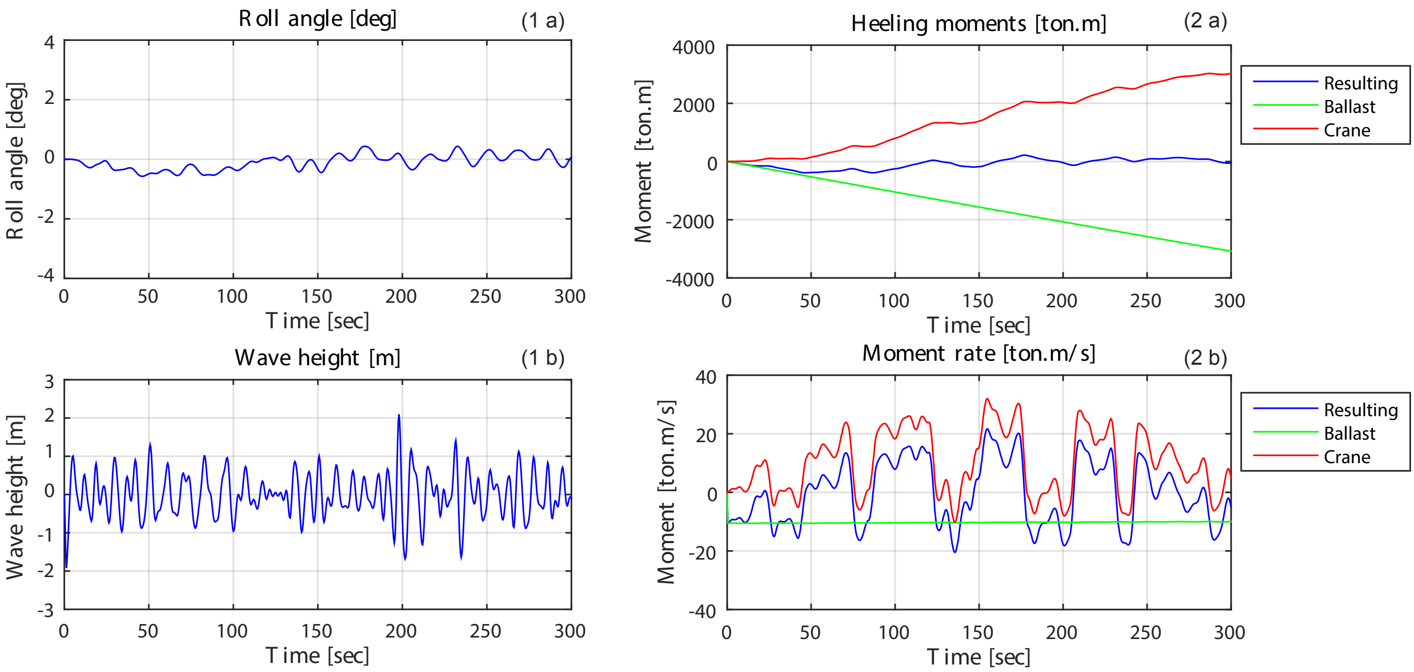


Figure 8.3: 150 ton lift in synced VFD mode
$H_s = 2.5m, T_p = 10s$ and 165° heading

8.2.2 300 ton

For the 300 ton scenario a lift duration of 300 seconds is chosen with 3 crane interruptions and a lift duration of 600 seconds with 5 crane interruptions. The results are shown in figure 8.4.

Crane load [ton]	Time [sec]	Control [-]	no. pumps [-]	no. crane interuptions [-]	Largest angle [deg]	Power [MJ]	Fuel [kg]	RPM [rpm]
300	300	on/off	3	3	1.9	31	1.72	0.13
300	600	on/off	2	5	1.8	36	2.00	0.08
300	600	synced VFD	3	5	1.3	5.9	0.33	0.08

Figure 8.4: Model result summary of a 300 ton lift phase with $\overline{GM}_{eff} = 2.25m$

The current system is capable to have a lift phase of 300 seconds. All 3 pumps have rotate at full speed to compensate the crane moment. When a lift duration of 600 seconds with only 2 pumps is analysed the heeling angle is similar but the power required becomes slightly more. Again, the use of the synced VFD's show the lowest heeling angle and power use. So when allowing the lifting phase to taken 600 seconds in stead of 300, the power use can be reduced by 70% with a neglectable $ 0.16 savings.

A 300 ton lift of 600 seconds show similar results as for 150 ton in 300 seconds. Moment rate peak of 40 ton.m/s are reached, causing roll angles up to 2°. The current system is capable to safely move a 300 ton hanging load within 300 to 600 seconds.

8.2.3 600 ton

For the 600 ton scenario lift durations of 700 and 1000 seconds are chosen with 5 crane interruptions. The results are shown in figure 8.4.

Crane load [ton]	Time [sec]	Control [-]	no. pumps [-]	no. crane interuptions [-]	Largest angle [deg]	Power [MJ]	Fuel [kg]	RPM [rpm]
600	700	on/off	3	5	2.8	71.3	3.96	0.07
600	1000	on/off	3	5	3	74.4	4.13	0.05
600	1000	VFD	3	5	2.3	52	2.89	0.05
600	1000	synced VFD	3	5	2.1	48.1	2.67	0.05

Figure 8.5: Model result summary of a 600 ton lift phase with $\overline{GM}_{eff} = 2.25m$

The current system is capable to have a lift phase of 700 seconds when all 3 pumps rotate at full speed to compensate the crane moment. The power used with the current system is similar for the 700 and 1000 second scenario. This is because in both cases the total amount of running time of the 3 pumps together is similar to move the required amount of water ballast. Also for the 600 ton case the synced VFD setting uses the least power and has the lowest heeling angle.

An other development noticed is that the heeling angle become more sensitive to the crane motions, as mentioned in 7.3.1. This is of course because a higher crane load can result in larger moment rates. But also the maximum moment rate of the ballast system is limited. So the ballast system is relatively less powerful to the crane moment rate. This is illustrated in figure 8.6

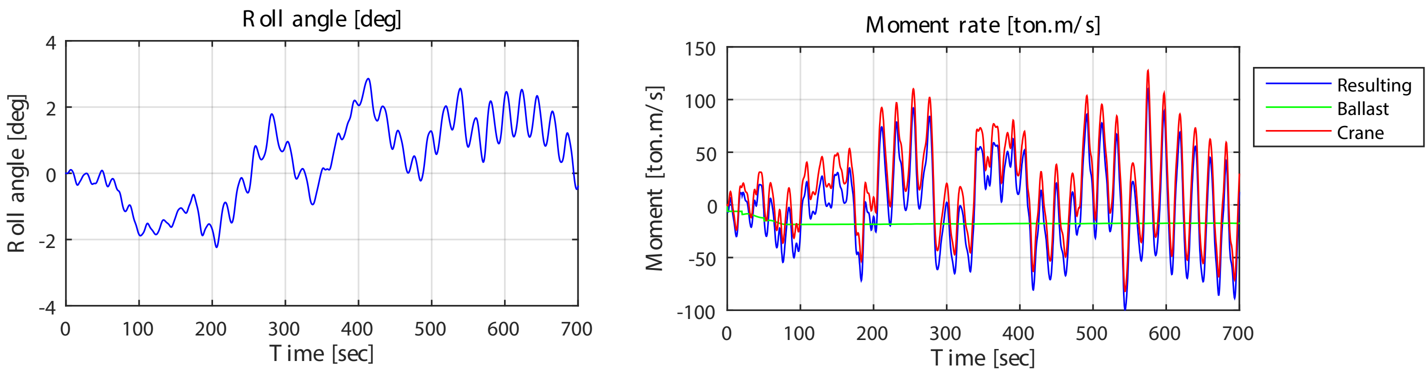


Figure 8.6: 600 ton lift with current ballast system
$H_s = 2.5m, T_p = 10s$ and 165° heading

Even with very slow crane rotation of 0.05 rpm, the moment rates of the crane can easily outsize the moment rate of the ballast system. However, this does not have to be a problem as the heeling angle is still within the limits and the load can still be displaced in 700 to 1000 seconds. Repetitive lifts of 600 ton are uncommon. So there is no lucrative advantage of a faster lifting phase yet.

8.2.4 900 ton

For the 900 ton lift durations of 1500 and 1800 are examined. The results are shown in figure 8.7.

Crane load [ton]	Time [sec]	Control [-]	no. pumps [-]	no. crane interuptions [-]	Largest angle [deg]	Power [MJ]	Fuel [kg]	RPM [rpm]
900	1500	on/off	3	5	4.5	115	6.39	0.032
900	1500	synced VFD	3	5	4.1	82	4.56	0.032
900	1800	on/off	3	5	3.4	130	7.22	0.025

Figure 8.7: Model result summary of a 900 ton lift phase with $\overline{GM}_{eff} = 2.25m$

The results show that both systems do not comply to the heeling angle criteria when the lifting phase is taken 1500 second. When the duration of the lifting phase is increased the slewing speed can be decreases. This can result in smaller heeling angles. To see of the angles can be decreased, the lift scenarios are repeated with an increased water ballast capacity, an increased $\overline{GM}_{eff}$ and a combination of the two.

Effect of extra capacity

To analyse the effect of larger volumetric flow rate of the ballast system, a higher pump capacity has been put into the model. Figure 8.8 shows the current system on the left and an upgraded system on the right. In the upgraded system the 3 pumps are displace by pump capable to displace 1600 m^3/hr at a pump head of 35 meters each.

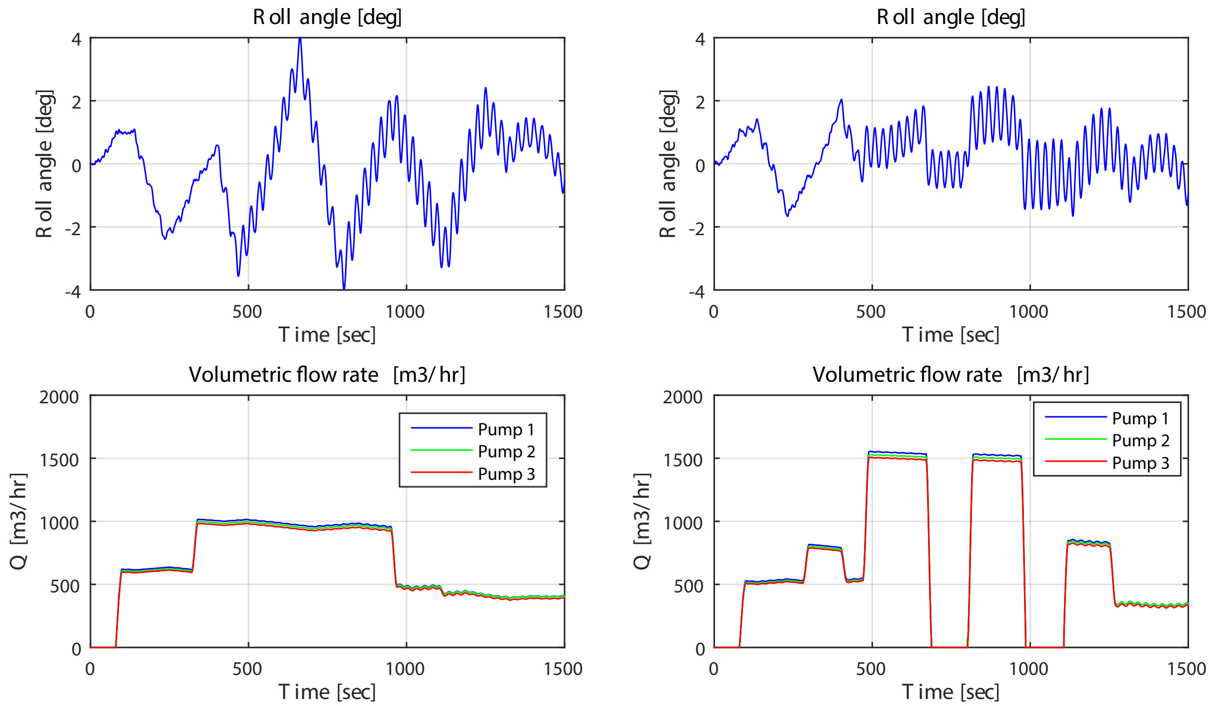


Figure 8.8: Comparison of the current system and a system (left) with a larger variable volumetric flow rate (right).

At the left the ballast system runs at full capacity and first creates a larger moment than the crane, making the ship heel 3.6° to port side. Then the crane moment is increased faster than the ballast moment until the crane motion is stopped at a heeling angle of 4.1° starboard. The ballast pumps are still running and the heeling angle decreases again. The crane is moved another 2 times until it reached its position.

At the right the capacity of the maximum volumetric flow rate is 50% larger. When the crane moment rate is relatively low, the volumetric flow rate is minimized. As the moment rate of the crane increases the flow rate is increased. In this scenario the heeling angle is kept closer to 0° and the peak is at 1.6°.

In both graph the system starts to have larger roll excitations around the heeling angle from 500 seconds and further. This is because the large 900 ton hanging load causes an extra roll induced moment. And as the damping in the system is very small, this excitation does not damp within the lifting phase. Therefore the effect of an increased $\overline{GM}_{eff}$ is analysed.

Effect of increasing the GM

To see the effect on the roll angles the GM is increased by 1 meter and tested for the same lift scenarios of figure 8.8. The graphs of figure 8.9 show that the 1 meter increase of the GM has a large effect on the roll angles. The shape of the graph remains similar as the crane motions and the pump automation are kept the same. But the excitations are largely reduced.

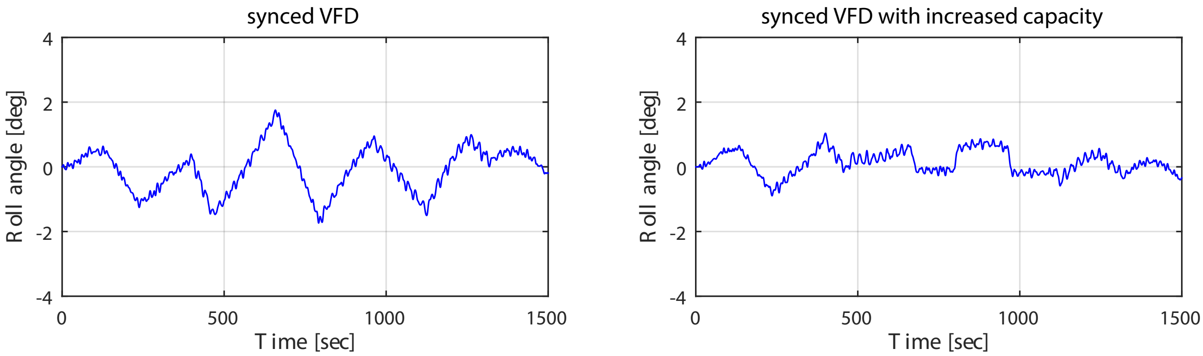


Figure 8.9: Scenarios of figure 8.8 but with $\overline{GM}_{eff} = 3.25$ m

An increased $\overline{GM}_{eff}$ results in a increase of restoring capacity. To illustrate the improvement the static heeling angle are calculated for 1 meter transverse displacements, as in 7.3.1. The results are shown in figure 8.10. The static heeling angle of a 1 meter displacement of 900 ton is 1.3° with a $\overline{GM}_{eff}$ of 2.25 meter where is was 2.8° with a $\overline{GM}_{eff}$ of 3.25 meter. This is a decrease of 55%.

	Weight	150	300	600	900	1200	1400	[ton]
Fixed load	Angle	0.1	0.3	0.5	0.8	1	1.2	[deg]
Hanging load (30 m)	Angle	0.1	0.3	0.7	1.3	2.3	3.4	[deg]

Figure 8.10: Static heeling angle at $\overline{GM}_{eff} = 3.25$ m
for a 1 meter transverse displacement

The increase of the $\overline{GM}_{eff}$ decreases the natural period of the system. A GM of 3.25 meters results in a natural period of 14 seconds. This is still within the criteria of 4.1.

8.2.5 1800 ton

The maximum lift capacity of the cranes is 900 ton per crane. The J-class has two of them on board. With a tandem lift 1800 ton can be reached. Jumbo does not lift these heavy loads offshore. In port pontoons are used to increase the water plane area and therefore its restoring capacity. When the 1800 modelled as a hanging load in the current model with a $\overline{GM}_{eff}$ of 2.25 meter, it shows that the system is very unstable. Within half a minute the situation becomes uncontrollable. See figure 8.11.

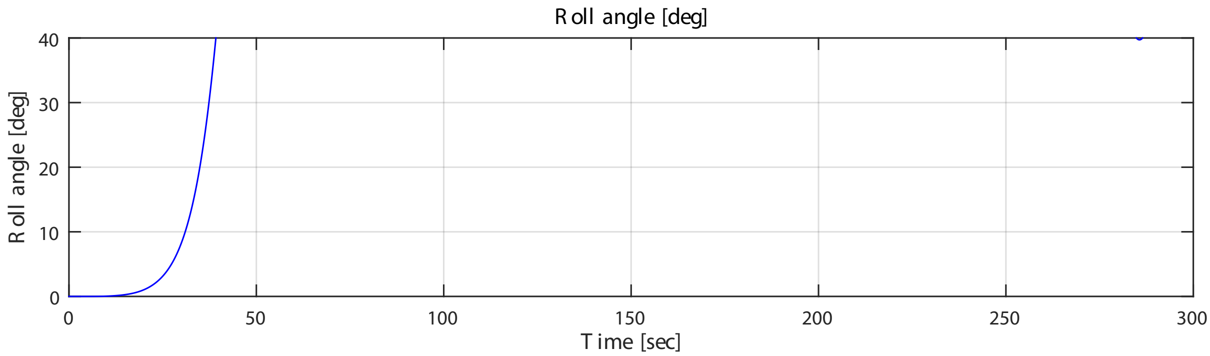


Figure 8.11: Unstable behaviour of the 1800 ton lift

$$\overline{GM}_{eff} = \frac{m_{load} \cdot \left(1m + l_{craneline} \cdot sin(\phi_{heeling})\right)}{\nabla \cdot sin(\phi_{heeling})} \tag{8.1}$$

To ensure sufficient stability the GM can be increased. If $\phi_{heeling} \leq 1°$ is required the $\overline{GM}_{eff}$ should be at least 7.5 meters. This will make the natural period of the vessel drop below 10 seconds and the system will not meet the criteria of 4.1. Making the J-class suitable for 1800 ton offshore lifts is not within the scope of this project.

8.2.6 Discussion

Both the crane as the ballast system are modelled as a simple moment based on a point mass at a horizontal distance from the ship's point of rotation. The change of mass moment of inertia caused by the crane and ballast is not included in the model. By moving the masses away from the centre of rotation the mass moment of inertia is increased and can result in a larger natural period. In the case of a 900 ton load and corresponding ballast water the mass moment of inertia can be increased by 15%. However for both the mass moment of ienrtia as for the restoring capacity of the ship the GM is governing. See equations 8.1 and 8.2

$$T_{0ø} = 2\pi\sqrt{\frac{k_{xx}^2 \cdot \Delta + m_{øø} + I_{additional}}{g \cdot \Delta \cdot \overline{GM}_{eff}}} \tag{8.2}$$

As the system is based on equation of moments and the roll angles are relatively small it is assumed that the model results are accurate enough to be representative for the prediction of the systems behaviour.

Dynamic positioning

Not included in the model but an essential part of the offshore installation operations is the DP-system. The system creates an extra moment around the point of rotation. In normal operational conditions this moment is assumed to be relative constant and is therefore not included in the scope for this thesis.

Chapter 9

Conclusion and recommendations

The purpose of this project was to develop of an anti-heeling solution for the Jumbo J-class. From the corporate and technical analysis a list of design requirements has been selected. Several concepts has been developed to create a variety of ideas. With the list of requirements the most suitable concept for the J-class has been selected to develop into a model. This chapter will summarize the most important conclusions and give recommendations for improvement and further research.

9.1 Conclusion

For the corporate analysis, technical analysis and concept generation the conclusions are already given in the report. For the general conclusion the researched questions will be answered including the model results.

9.1.1 Research questions

In the first phase of the project two researched question were formulated to be able to evaluate the solution. The first one is:

Under what conditions will the reduction of the lifting phase be worth the operational and capital expenditures?

Complex equipment investments in the ship are generally in the order of hundred thousands up to millions of dollars. In section 2.3 the potential time saving has been expressed in the ship's day rate. This showed that if 5 or 10 minutes could be saved per lifting phase 150+ repetitive lifts needed to be done to be able to save between the \$50.000 and \$ 100.000. Additionally 80% of the offshore project track record did not include lift heavier than 300 ton. The current system is capable to overboard up to 300 ton within 10 minutes. So every dollar that is put into this project will most likely not be returned. Unless large projects with 150+ lift repetitions are contracted or if the new solution will significantly increase the workability. But in the near future there are no such projects for Jumbo.

To understand the anti-heeling problem a technical analysis has been done and a model has been made. With the knowledged obtained the second research question can be answered.

How can the duration of the lifting phase of the Jumbo J-type be reduced?

The current manual controlled system has limited pump capacity and the pump can either pump at full capacity or been switched off. The reaction time of the system is between 20 and 30 seconds. Pumps with variable volumetric flow rates has been added to the system to analyse the effects. The results show that for load up to 600 ton the roll angles and lift duration are not significantly improved. For the 900 ton load the lift te roll angles did benefit from variable flow rates, but the pump capacity had to be increased by 50%.

The best result however were seen when the $\overline{GM}$ was increased from 2.25 meter to 3.25 meter. The higher $\overline{GM}$ resulted not only a smaller static heeling angle but also smaller excitations of the ship roll motions.

The system also showed a dynamic response when to load was displaced with a certain speed. The impulse created by the crane movement can cause large roll motion and the hanging load enlarges these moments. These moments can not be compensated accurately with the current system. These roll motions can be decreased by a higher $\overline{GM}$ or a slower crane movement. An increase of the $\overline{GM}$ increases the restoring arm and makes the ship less sensitive to moment changes.

So the current system can be improved but will not be worth an investment under Jumbo's current market conditions. The most economical option to reduce roll and heeling angles is to slowly move the crane to decrease the impulse and increase the $\overline{GM}$ to reduce the ship response.

9.1.2 Sources causing lost time

Based on the simple nature of the anti-heeling system, limited capacity and the manual control it was assumed that a new anti-heeling solution could prove advantages and safe time. It turned out that for the projects the J-class executes the current system capacities are sufficient enough.

Apart from lifting phase there are many other aspects were time savings could be realised. In case of the Cyprus-Turkey water pipeline project, the hoisting of the loads to and from the seabed was very time consuming for example. And according to the captain and deckcrew interviews on the field trip (Appendix D) al lot of time is lost due to meetings, communication errors, rigging going wrong, cultural differences, crowded ships, negative working spirit and other human factors.

If Jumbo wants to save time on offshore projects it could consider to research all aspects causing lost time. This research should contain the technical capacities and limitations of the equipment but also involve the crew and the interaction between the two.

9.2 Recommendations

Investing in a new anti-heeling solution is not advised according to this research. However, there is room for improvement. This section will introduce recommendations to improve the anti-heeling solution and how to avoid time lost caused by human involvement.

9.2.1 Stability

The results of chapter 8 showed that the system is sensitive to roll motions. For the J-class load up to 900 ton can still be executed with roll angles below the 2° when the $\overline{GM}_{eff}$ is increased to 3.25 meter. Under the condition that a lifting phase of 1500 seconds is acceptable.

If Jumbo is moving toward heavier offshore lifts, sufficient restoring capacity is required to guarantee safe operation. When the load is prevented from swinging, smaller moment excitations caused by the load are experienced. Taglines are commonly used for this, but the applicability for Jumbo should be researched first. Roll reduction systems could also be considered, as they can damp the roll excitations. Studies on roll reduction systems has been done at the R&D department of Jumbo but did only cover conceptual roll reduction systems to compensate for wave induced roll motions. Including the crane forces makes the problem even more complex.

9.2.2 Recommended pump system

The current anti-heeling solution is elaborated in section 3.3. To remind, the system consist of 12 wingtanks, 9 double bottom tanks and a fore tank. All tanks are connected by pipeline trough the double bottom to the manifold in the pump room at the front of the ship. This is illustrated in figure 9.1. This section will elaborate the recommendations concerning the design of the ballast system.

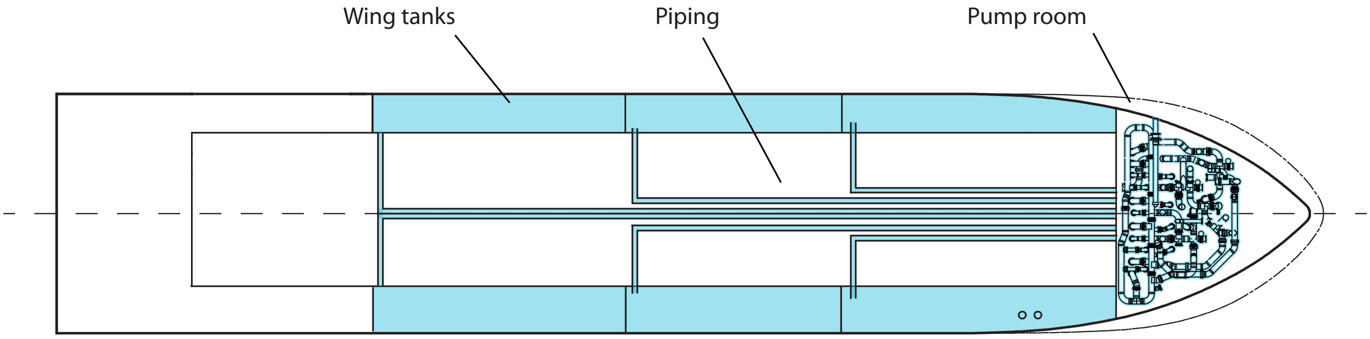


Figure 9.1: Impression of current piping route through the double bottom

Pump selection

The centrifugal pumps are one directional. Different routing of the piping is required if the flow should change direction, as illustrated in figure 3.12. If bidirectional pumps are used, a more elegant solution can be designed. A bidirectional pump is an in-line pump with a symmetrical configuration consisting of two counter-rotating impellers with variable speed. See figure 9.2 for an impression.

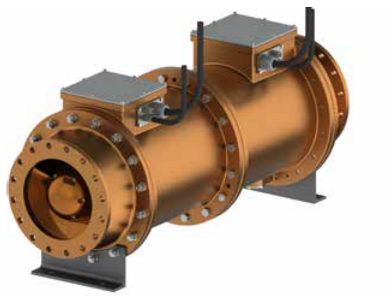


Figure 9.2: Nijhuis BiFlow Pump [7]

When taking the 900 ton lift with the increased capacity as representative for exceptional lift projects, the pumps should be able to produce a variable volumetric flow rate up to 1500 m^3/hr at a pump head of 30 meters. According to Fairbanks Nijhuis, the manufacturer of these pumps, that is what the pumps are made for.

System lay-out

With the bidirectional pump the current extensive system lay out can be redesigned in more simple and compact system. Figure 9.3 shows that the pumps can be included in the manifold. This is significantly different from the current system illustrated in figure 3.12 on page 33. Advantages are that not only less piping and valves are used, but also the tank pairs can be directly be connected to each other. The new lay out also allows every single pumps to be connected to all tank pair combinations.

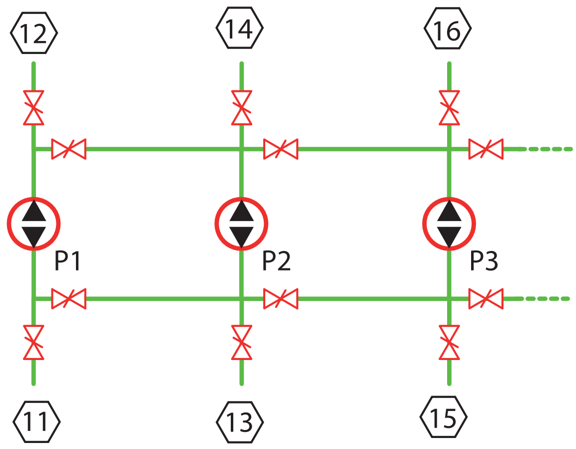


Figure 9.3: Recommended system layout

An other difference is that the wing tanks are not separated in a upper and lower wing tank but as one wing tank. For lifts above 600 ton the showed that tank heights of 3.5 meters were not enough and the system has to switch tank during operation.

Physical positioning of compartments

Due to the relatively compact dimensions the pumps can be installed in the double bottom of the ship. This means that the recommended layout of figure 9.3 can be build in the double bottom. The pump room can therefore be significantly be decreased in size to a small service pump room where the system is connected to the sea water. The wing tanks can also be connected by straight piping and reduce pipe length and head loss. See figure 9.4.

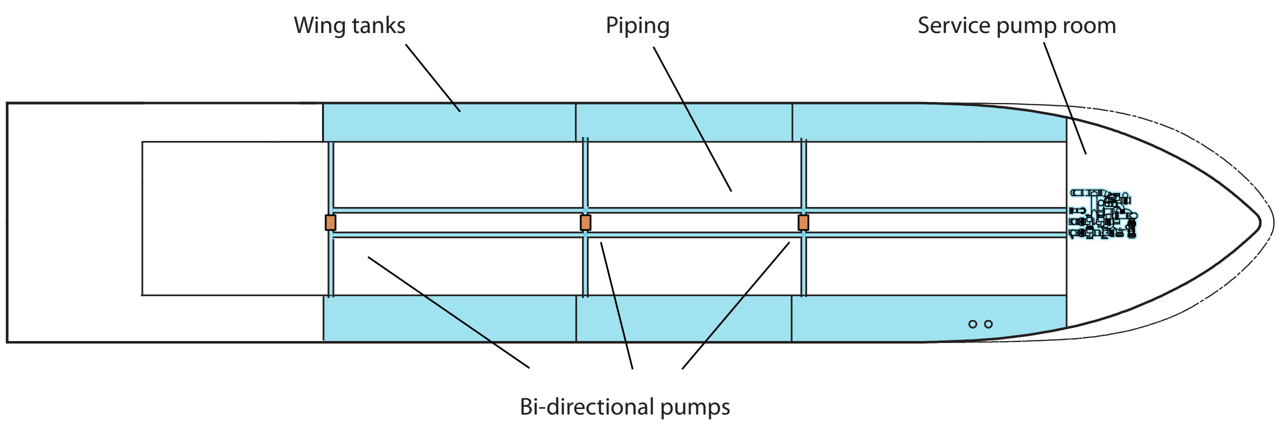


Figure 9.4: Impression of the recommended ballast system.

Excluded from the figures are the double bottom tanks. All 9 of them can be added to the manifold in the double bottom. This is illustrated in figure 9.5. The advantage here is again that pipe length is reduced as the pipes do not go to the front of the ship. In the current solution these pipes are also brought to the front of the ship and are there connected to the manifold.

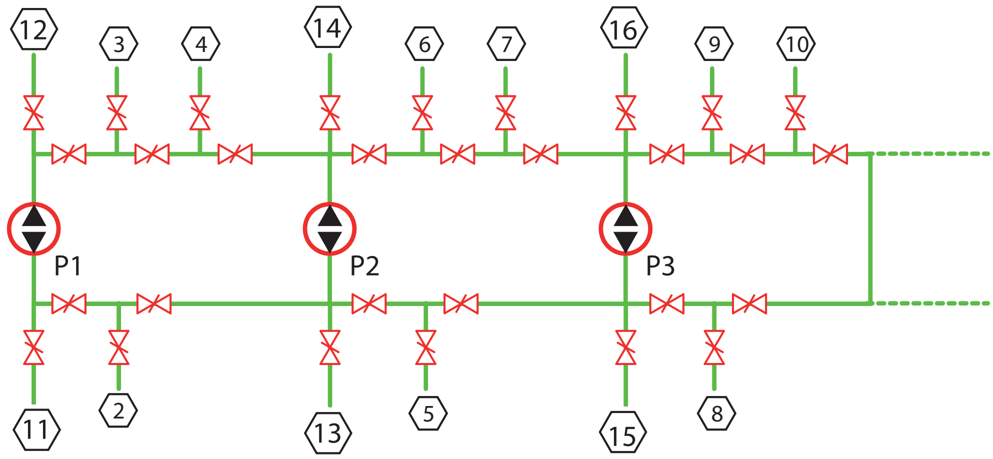


Figure 9.5

Manual control vs (semi) automated system

According to the captain a manual controlled ballast system is preferred over an automated system. The reason is that the captain would like to keep full control over the ship and adjust the water ballast the way he likes. The ballast system is also used to carefully pick up or put down loads if the moving the crane result in relatively rough displacements of the object.

Adjusting the water ballast is also possible with an (semi) automated system. The difference is that instead of selecting the ballast tanks and run the pumps, the captain has to select a new set point. The advantage of the manual system is that it is more simple and the investment is smaller. To consider a semi automated system a research should be done whether the captain or a computer is a better controller for the lifting jobs Jumbo does.

In order to allow automation of the ballast system an accurate ship model (including crane) is required to be included in the control system. This model should include all effects on the roll angle and will therefore become complex. But lets assume an accurate control system. The system will control the water flow according to the measured changes of the crane moment and ship motions. The captain will control the volumetric water flow by what he sees and what his plans are. So the automated system is better in anticipating quickly on small differences and the captain is better in anticipating on the lift operation itself but will react slower. Due to the stability capacities of the ship, the control of the ballast system does not require the lowest reaction time. So smart but slow system is preferred over a fast but stupid system. A manual controlled system is therefore recommended for the J-class because it is smart, economical and is expected to have sufficient capacities given the results of chapter 8.

9.2.3 Recommendations to avoid lost time

As mentioned in 9.1.2 there are many other aspects were time savings could be realised. Therefore it is recommended to survey projects to find bottlenecks, show stoppers and other causes of lost time. It is also recommended to be more involved in the aspect concerning the daily business on the ship. This will be briefly explained below.

Survey and improve

To be able to save time on offshore projects in a constructive and measurable way, Jumbo needs to know why every task does take as long as it takes. All the delays, extra meetings and their causes should also be listed. With the obtained information an overview can be created where the time consumptions are shown. When this is clear Jumbo can decide which aspects have the most potential to save time on. In the case of this project, a faster hoisting speed had much more potential to safe time for example.

Get involved

That the mentality on board of the ship and at the office are two different worlds is no surprise. Common complaints from the crew on board is that the support from office does not really understand how things work at sea. At the office complaints are that the crew on board is not flexible enough at some occasions. There is no right or wrong in this case, but in the end of the day it is the crew on board that executes the work offshore. As the project plans and preparations are done at the office, Jumbo could consider a stronger involvement of the crew or a crew representative to create a stronger partnering. This is not so much for the technical execution of the project, but more to get the most out the crew and create a good working spirit. Lets take an example.

As explained in appendix D the Jumbo J-class is not a dedicated offshore ship. The ship is designed for a crew of 20 and putting an extra accommodation block on deck with 60 extra beds, puts an extra load on the existing facilities. So the ship is four times more crowded and with a largely unfamiliar crew. When crew members are getting in the way of each other and experience discomfort when there shift is over, the captain has to spend a significant amount of time on extra meetings and on keeping everyone happy. When involving the crew in the creation of a good working environment they can have a better focus on the work while at the job.

9.3 Summary

Looking at the result the conclusion can be drawn that for the offshore project Jumbo does, the current system is sufficient. The potential amount of time a new solution could save is not large enough to return an investment. The ship stability properties are the most limiting for the operations, not the ballast system. An increase of the GM up to 3.25 meter showed that loads up 900 ton can be lifted safely under normal conditions.

The current lay-out of the system is a cumbersome way of displacing the ballast water. The recommended lay-out showed a more simple and compact solution.

When Jumbo want the see where they can safe offshore time on projects, a detailed overview of the project tasks and their time consumption should be created. Time can be saved in many ways and the solution does not necessarily has to be a technical one. Get the right information and the right people involved to get the most out of the crew and equipment. This could lead to smarter project planning and smarter investments.

Bibliography

[1] Mark Adeosun. Oversupply of Subsea Vessels To Continue Until Gradual Recovery in Operators Spending. 2016.

[2] Adrian Biran and Rubén López-Pulido. *Second Edition Ship Hydrostatics and Stability Second Edition*. 2014.

[3] Amaresh Chakrabarti. *Engineering Design Synthesis- Understanding, Approaches and Tools*. 2002.

[4] Laurent Daniel. Offshore vessel mobile offshore drilling unit & floating production unit market review. 2014.

[5] Douglas-Westwood. Assessment of vessel requirements for the U.S. offshore wind sector. 2013.

[6] Dredging International. Westermost Rough Offshore Wind Farm. 2015.

[7] Fairbanks Nijhuis. Product overview.

[8] Fraunhofer ISE. Levelized Cost of Electricity Renewable Energy Technologies. 2013.

[9] M.J. French. Conceptual design for engineers. 1998.

[10] Andrew R. Henderson. Offshore wind in Europe. 2002.

[11] Hopper Marine. Anti-Heeling System Brochure.

[12] Hopper Marine. Blower Anti-Heeling System Brochure.

[13] Huisman Special Lifting Equipment BV. Expanded load curves for side and off lead. 2008.

[14] J.M.J. Journée and W .W. Massie. *Offshore hydromechanics*. 2001.

[15] Mathworks. SimHydraulics User's Guide. 2015.

[16] V Matousek. Dredge pumps and slurry transport: 7. Pump and pipeline characteristics. 2009.

[17] M. H. Patel and J. A. Witz. *Compliant Offshore Structures*.

[18] T Perez and T I Fossen. *Kinematics of Ship Motion*.

[19] Platts McGraw Hill Financial. Bunkerwire. 40, 2016.

[20] Roland Berger. Offshore Wind Toward 2020. 2013.

[21] N.F.M. Roozenburg and J. Eekels. Product Design, Fundamentals and Methods, 1995.

[22] Annemiek van Boeijen and Jaap Daalhuizen. Delft Design Guide: Design Strategies and Methods, 2010.

[23] Jasper van Heijst. Improvement of the workability of monohull heavy lift vessels during operations by reducing roll motion. 2015.

[24] Tom van Schalkwijk. Increasing the workability for offshore lifting operations by reducing the roll motion. 2015.

[25] Frank. M. White. *Fluid Mechanics*. 2010.

[26] Ben Wilby and Douglas Westwood. Subsea Infrastructure Growth Tempered by Oil Price Collapse. 2015.

[27] S Winkel, H Amtsberg, and W Schulz. Keeping Ships Upright. *SWZ Maritime June 2015*, 2015.

Appendix A

Seakeeping behaviour J-class

This appendix provides the significant double amplitudes of the Jumbo- J-class related to the JONSWAP wave spectrum for two load cases. The information given is extracted from the master thesis Tom van Schalkwijk wrote for Jumbo [24].

Criteria for maximum responses

Assuming that the DP-system can compensate the surge, sway and yaw motions and a (passive) heave compensator for the heave mootion, the workability is governed by the pitch and roll motion. For significant wave heights above 2.5 meter these assumptions might not be valid any more. Therefore all conditions with a significant wave height above 2.5 meter will be considered as not workable.

Looking at the history of Jumbo's offshore lifting operations the following criteria has been used:

- Roll angle Significant Double Amplitude (SDA): 1.5°
- Pitch angle Significant Double Amplitude : 1.5°

For this thesis only the motions in the Y,Z-plane are considered. So the pitch is excluded for the workability. The roll angle is sensitive to the ships stability and therefore its $\overline{GM}_{eff}$ and T_0. By ballasting the centre of gravity can be shifted vertically and cause an increase or decrease of the $\overline{GM}_{eff}$. For offshore lifting operation the $\overline{GM}_{eff}$ of the J-class is usually between 2.25 and 3.25 meters. This gives the following load cases:

	LC1	LC2
Displacement ∇ [m3]	17845	17845
Metacentric height GM [m]	2.25	3.25
Vertical position centre of gravity KG [m]	10.66	9.66
Natural period of roll T_0 [s]	16.8	14.0

Figure A.1: Loading condition LC1 and LC2

SDA values

For the calculations the computer program Safetrans (Marin, 1997) was used. This software has been used by Jumbo for over 10 years and proved to correspond to the actual ship motions relatively well. The SDA's are calculated for 90 and 165 degrees headings. A heading of 90 degrees results in the highest roll motions. A heading of 165 degrees is preferred for most heavy lift operations because it results in low roll motions.

Safetrans 90 LC1								SDA Roll unstab [deg]								
		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
HS		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
0	0.25	0.04	0.03	0.02	0.03	0.04	0.07	0.12	0.23	0.39	0.59	0.82	1.13	1.46	1.58	1.43
0.25	0.5	0.07	0.06	0.05	0.05	0.09	0.15	0.25	0.46	0.79	1.19	1.64	2.26	2.93	3.17	2.86
0.5	0.75	0.11	0.08	0.07	0.08	0.13	0.22	0.37	0.68	1.18	1.74	2.33	3.06	3.81	4.06	3.73
0.75	1	0.14	0.11	0.09	0.11	0.17	0.29	0.50	0.91	1.57	2.30	3.02	3.86	4.69	4.96	4.61
1	1.25	0.18	0.14	0.12	0.14	0.22	0.37	0.62	1.14	1.96	2.85	3.70	4.67	5.56	5.86	5.48
1.25	1.5	0.21	0.17	0.14	0.16	0.26	0.44	0.75	1.37	2.35	3.40	4.39	5.47	6.44	6.76	6.35
1.5	1.75	0.25	0.20	0.16	0.19	0.31	0.51	0.87	1.59	2.68	3.82	4.87	6.04	7.11	7.46	7.01
1.75	2	0.28	0.23	0.19	0.22	0.35	0.58	1.00	1.82	3.02	4.23	5.36	6.62	7.78	8.17	7.68
2	2.25	0.32	0.25	0.21	0.25	0.39	0.66	1.12	2.05	3.35	4.65	5.84	7.19	8.45	8.87	8.34
2.25	2.5	0.36	0.28	0.23	0.27	0.44	0.73	1.25	2.27	3.68	5.06	6.33	7.77	9.12	9.57	9.00

Safetrans 165 LC1								SDA Roll unstab [deg]								
		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
HS		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
0	0.25	0.00	0.01	0.02	0.02	0.02	0.02	0.04	0.06	0.11	0.16	0.23	0.31	0.41	0.45	0.40
0.25	0.5	0.01	0.02	0.03	0.05	0.05	0.05	0.07	0.13	0.22	0.33	0.45	0.62	0.82	0.89	0.79
0.5	0.75	0.01	0.03	0.05	0.07	0.07	0.07	0.11	0.19	0.33	0.49	0.68	0.93	1.23	1.34	1.19
0.75	1	0.02	0.04	0.07	0.09	0.10	0.10	0.14	0.26	0.44	0.66	0.91	1.24	1.64	1.79	1.59
1	1.25	0.02	0.05	0.08	0.12	0.12	0.12	0.18	0.32	0.55	0.82	1.13	1.56	2.05	2.23	1.99
1.25	1.5	0.02	0.05	0.10	0.14	0.15	0.15	0.21	0.38	0.66	0.99	1.36	1.87	2.45	2.68	2.38
1.5	1.75	0.03	0.06	0.12	0.16	0.17	0.17	0.25	0.45	0.76	1.15	1.59	2.16	2.78	3.02	2.71
1.75	2	0.03	0.07	0.13	0.19	0.19	0.20	0.29	0.51	0.87	1.31	1.81	2.45	3.11	3.35	3.04
2	2.25	0.03	0.08	0.15	0.21	0.22	0.22	0.32	0.57	0.98	1.48	2.04	2.74	3.44	3.69	3.36
2.25	2.5	0.04	0.09	0.17	0.23	0.24	0.25	0.36	0.64	1.09	1.64	2.27	3.03	3.77	4.03	3.69

Figure A.2: SDA values for Roll for LC1 with a heading of 90 deg. (top) and 165 deg. (bottom) relative to the waves. JONSWAP wave spectrum with 3<Tp<18 and 0<Hs<2.5

Safetrans 90 LC2								SDA Roll unstab [deg]								
		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
HS		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
0	0.25	0.04	0.04	0.05	0.09	0.15	0.29	0.56	0.90	1.31	1.79	2.03	1.87	1.57	1.37	1.24
0.25	0.5	0.08	0.08	0.10	0.17	0.30	0.59	1.12	1.80	2.62	3.58	4.07	3.74	3.15	2.73	2.49
0.5	0.75	0.12	0.12	0.16	0.26	0.45	0.88	1.64	2.53	3.53	4.66	5.24	4.85	4.14	3.65	3.36
0.75	1	0.16	0.16	0.21	0.34	0.59	1.18	2.16	3.27	4.44	5.74	6.41	5.97	5.14	4.57	4.24
1	1.25	0.20	0.20	0.26	0.43	0.74	1.47	2.69	4.01	5.35	6.82	7.58	7.08	6.14	5.49	5.12
1.25	1.5	0.23	0.24	0.31	0.51	0.89	1.77	3.21	4.75	6.27	7.90	8.75	8.19	7.14	6.41	5.99
1.5	1.75	0.27	0.28	0.37	0.60	1.04	2.04	3.63	5.31	6.98	8.73	9.62	9.04	7.94	7.14	6.66
1.75	2	0.31	0.32	0.42	0.68	1.19	2.31	4.06	5.87	7.69	9.56	10.48	9.89	8.74	7.86	7.32
2	2.25	0.35	0.36	0.47	0.77	1.34	2.59	4.48	6.42	8.41	10.39	11.34	10.75	9.54	8.59	7.99
2.25	2.5	0.39	0.40	0.52	0.85	1.49	2.86	4.90	6.98	9.12	11.23	12.20	11.60	10.34	9.31	8.66

Safetrans 165 LC2								SDA Roll unstab [deg]								
		3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
HS		4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
0	0.25	0.00	0.01	0.02	0.03	0.03	0.06	0.12	0.20	0.30	0.44	0.52	0.47	0.38	0.32	0.29
0.25	0.5	0.01	0.02	0.04	0.05	0.06	0.12	0.24	0.40	0.61	0.88	1.04	0.94	0.75	0.64	0.58
0.5	0.75	0.01	0.03	0.06	0.08	0.10	0.18	0.36	0.61	0.91	1.31	1.55	1.40	1.13	0.96	0.87
0.75	1	0.02	0.04	0.07	0.10	0.13	0.24	0.48	0.81	1.21	1.75	2.05	1.86	1.51	1.28	1.16
1	1.25	0.02	0.05	0.09	0.13	0.16	0.30	0.61	1.01	1.52	2.18	2.56	2.32	1.88	1.60	1.46
1.25	1.5	0.02	0.06	0.11	0.16	0.19	0.36	0.73	1.21	1.82	2.61	3.06	2.78	2.26	1.92	1.75
1.5	1.75	0.03	0.07	0.13	0.18	0.23	0.42	0.85	1.41	2.10	2.96	3.44	3.14	2.59	2.23	2.03
1.75	2	0.03	0.08	0.15	0.21	0.26	0.47	0.97	1.62	2.39	3.31	3.81	3.50	2.92	2.53	2.32
2	2.25	0.04	0.09	0.17	0.24	0.29	0.53	1.09	1.82	2.67	3.65	4.18	3.86	3.25	2.83	2.61
2.25	2.5	0.04	0.10	0.18	0.26	0.32	0.59	1.21	2.02	2.96	4.00	4.55	4.21	3.57	3.14	2.89

Figure A.3: SDA values for Roll for LC2 with a heading of 90 deg. (top) and 165 deg. (bottom) relative to the waves. JONSWAP wave spectrum with 3<Tp<18 and 0<Hs<2.5

Appendix B

Load curves of the mast crane

Previously, the load curves of the 900 ton heavy lift marine cranes of the Jumbo J-class were calculated for an off lead of 1° and side leads up to 3°. This appendix will show updated load curves for 450 and 900 ton. Figure B.1 shows the definitions that are used for calculating the load curves. The cranes are manufactured and tested by Huisman Special Lifting Equipment BV [13].

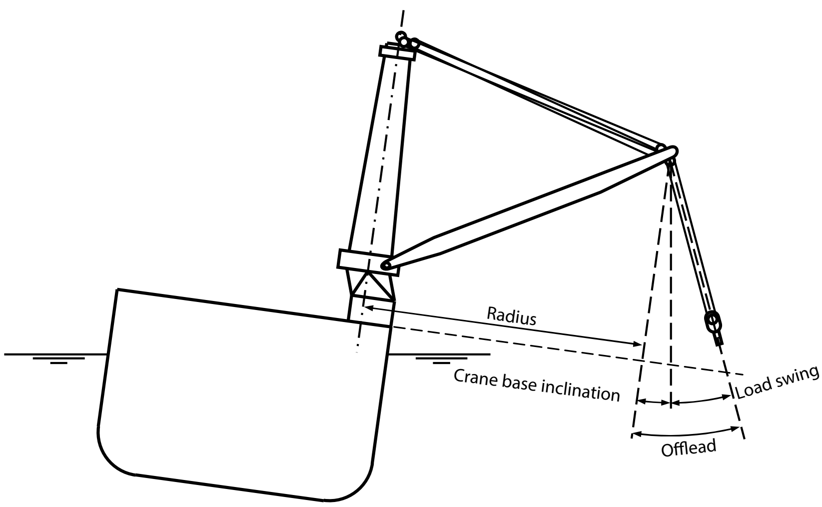


Figure B.1: Definition of radius and off lead

Side and off lead

The side- and off lead are both the summation of the crane base inclination and the load swing. Off lead is in the direction of the boom as shown in figure B.1. Side lead is perpendicular to the off lead. To lead is the lead in the opposite direction of off load.

Radius

The radius is taken as the horizontal distance between the crane base line the jib as if the ship is in its upright position, see figure B.1.

Dynamic load factors

When working at the crane limits one should consider using dynamic load factors to represent the highest peak load during operation. For these curves a vertical acceleration 0.5 m/s^2 is assumed due to vessel motions. And therefore the duty factor is set to 1.05. When larger accelerations are expected, new factors should be determined.

450 ton

Figure B.2 shows the maximum lead angles for a 450 ton lift. For side and off leads up to 5° the crane are fully capable to perform a 450 ton lift. The drop of load capacity at lower radii and larger leads is because a small radius corresponds with a small boom angle. With small boom angles the boom is at a relatively vertical position and the boom can fall back at larger lead angles.

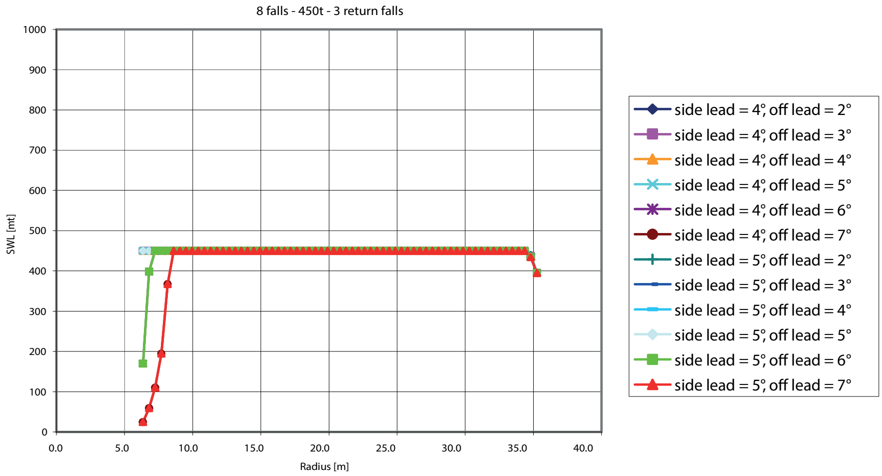


Figure B.2: To lead

900 ton

Figure B.3 shows the maximum lead angles for a 900 ton lift. The workable radii are limited for this lift because of the limited allowable mast bending moment of 273000 kNm. For leads up to 4° the crane can use its full 900 ton capacity when the radius is kept between 13 and 21 meter.

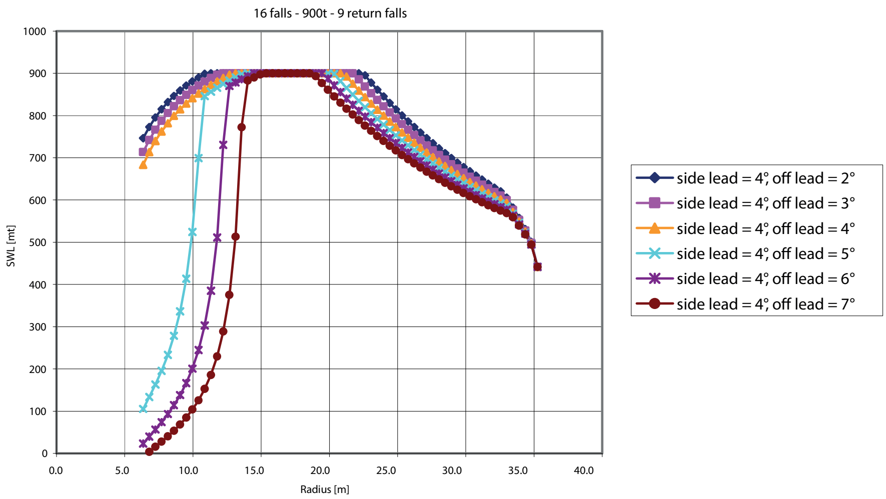


Figure B.3: Off lead

Appendix C

Pump arrangement

This appendix contains the pump scheme considering the connections of the wing tanks and the three ballast pumps. The scheme is extracted from the total ballast scheme.

nr.	Identification	Tank type			Volume [m3]	Weight [ton]	TCG [m]	Heeling Moment [kNm]
13	LW 1 PS WB	Anti-heeling	Lower	PS	359.072	368.049	-10.698	-38625.8
17	LW 2 PS WB	Anti-heeling	Lower	PS	561.11	575.138	-11.099	-62621.7
21	LW 3 PS WB	Anti-heeling	Lower	PS	519.15	532.129	-11.104	-57964.9
Subtotal					1439.332	1475.316		-159212
14	UW 1 PS WB	Anti-heeling	Upper	PS	703.936	721.535	-10.952	-77521.1
18	UW 2 PS WB	Anti-heeling	Upper	PS	627.068	642.744	-11.109	-70045.8
22	UW 3 PS WB	Anti-heeling	Upper	PS	729.446	747.682	-11.105	-81452.5
Subtotal					2060.45	2111.961		-229019
Total PS					3499.782	3587.277		-388232
nr.	Identification	Tank type			Volume [m3]	Weight [ton]	TCG [m]	Heeling Moment [kNm]
11	LW 1 SB WB	Anti-heeling	Lower	SB	492.505	504.817	10.132	50176.25
15	LW 2 SB WB	Anti-heeling	Lower	SB	499.103	511.581	10.654	53468.27
19	LW 3 SB WB	Anti-heeling	Lower	SB	503.682	516.274	10.662	53999.28
Subtotal					1495.29	1532.672		157643.8
12	UW 1 SB WB	Anti-heeling	Upper	SB	447.488	458.676	10.365	46638.47
16	UW 2 SB WB	Anti-heeling	Upper	SB	308.556	316.27	10.662	33080.01
20	UW 3 SB WB	Anti-heeling	Upper	SB	311.639	319.43	10.641	33344.73
Subtotal					1067.683	1094.376		113063.2
Total SB					2562.973	2627.048		270707

Figure C.1

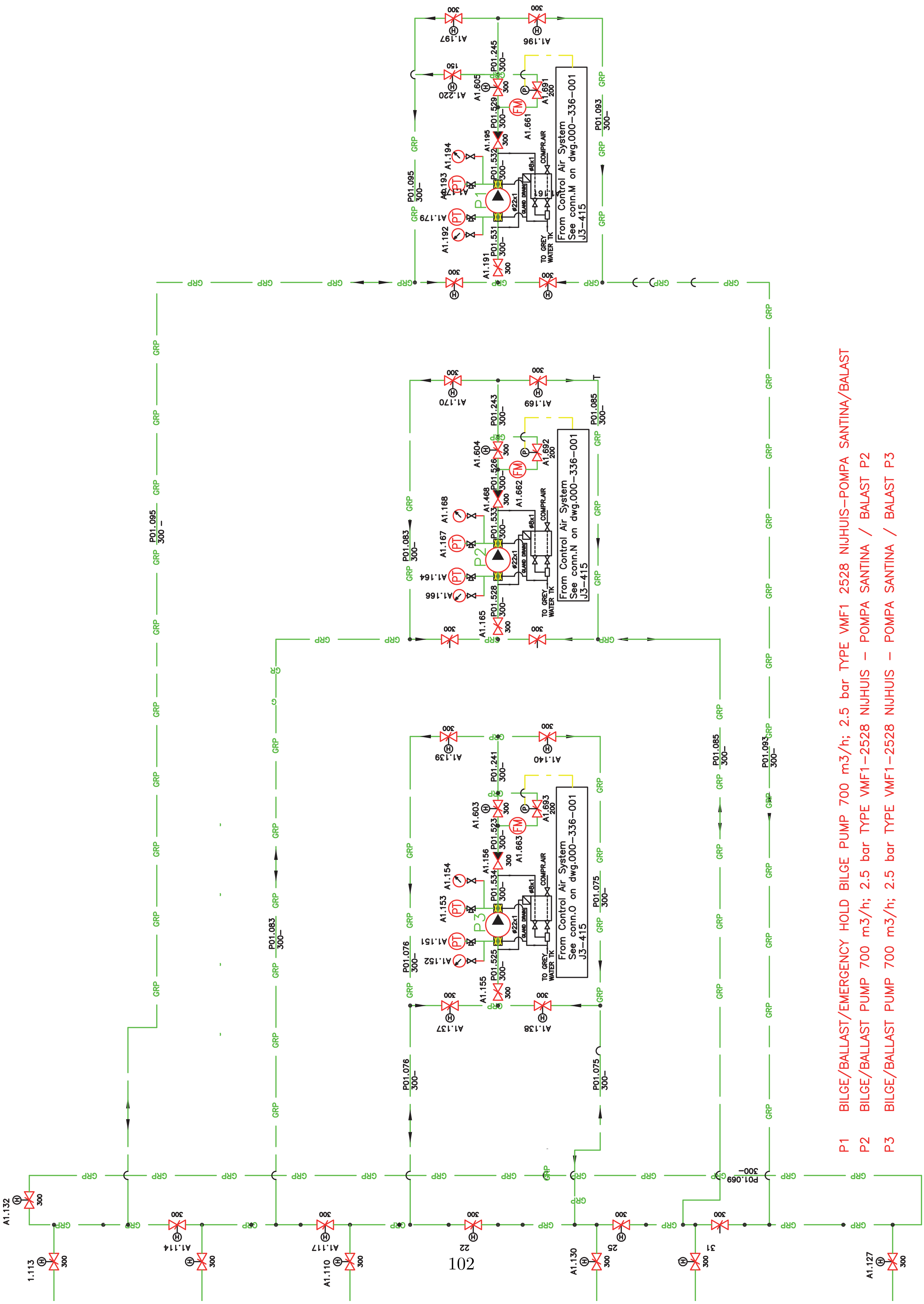


Figure C.2: Manifold and pumps

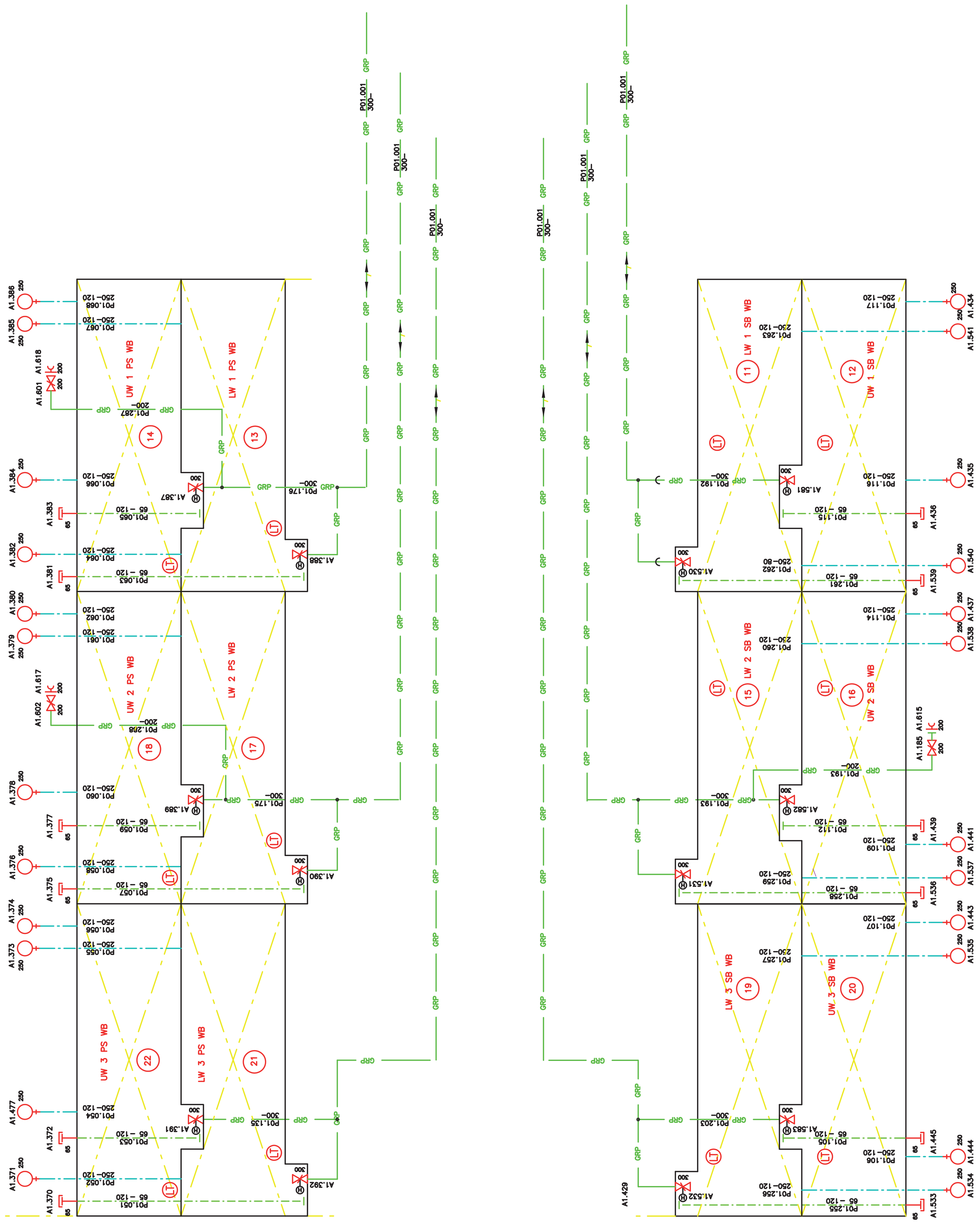


Figure C.3: Wing tanks

Appendix D

Field Trip Jumbo Javelin

As my graduation thesis progressed I familiarised the relevant aspects of lifting operations and ship stability. After basic calculations and high-level concept development, it was time to experience the operational reality on a J-class. Therefore I joined the Jumbo Javelin from Le Trait in France to Leith in Schotland the third week of November 2015. This appendix summarizes the observations and interviews on board.

In 2012 Technip was awarded a large subsea installation contract for the Chevron-operated Wheatstone Project in Australia. As part of the contract, Technip had to supply and install 41 kilometres of umbilicals. As the project is running to an end, it was up to Jumbo to return the empty umbilical reels (150 metric ton each) to Europe. On board of the Jumbo Javelin were three reels and supporting equipment. Two reels were offloaded at Technip in Le Trait and one reel in Leith.

D.1 Equipment, crew and procedure

The equipment used for lifting operations, apart from communication and control devices, can be divided into three separate systems:

- Heavy lift crane and rigging equipment
- Ballasting system
- Dynamic positioning

These are handled by different crew members. See figure D.1. The whole operation is supervised on the bridge where the captain or chief mate is the centre of communication between the DP-operator and the officer of the deck. The officer of the deck, who is physically on deck, supervises the crew on deck. The deck crew consist of the crane driver and a few able seamen.

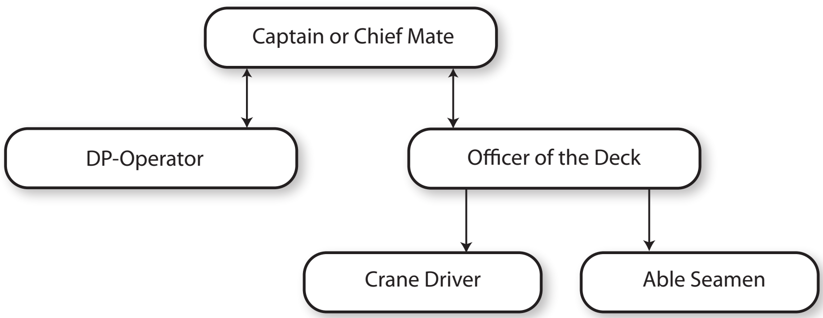


Figure D.1: Organisational overview of the crew members during lifting operations

Procedure

The lifting procedure is highly dependent on the lift plan. This explanation of the procedure of lifting the reel from deck to the quarry. This is a standard routine job and can be assumed representative for regular offshore lifting operations, apart from the absence of a DP-operator.

The procedure in the port of Leith:

1. Unfasten the sea fastening
2. Position the crane hook with slings and shackles above rigging slots of the reel
3. Rig the reel one shackle at the time
4. Ensure that the crane hook is in line with the centre of gravity of the reel
5. Check if the reel is loosened from all its sea fastening
6. Carefully put some load on the crane hook
7. Check again if the crane hook is still in line with the centre of gravity of the reel
8. Lift the reel from the deck
9. Start al three anti-heeling pumps
10. Start slewing the crane
11. check upon the heeling angle. If to large, stop slewing and let the ballast catch up
12. When the reel is at the preferred outboard position, stop both slewing and pumping
13. Lower the reel on the ground un till the crane hook has lost a part of its load
14. Reverse the route of the ballast water to loose load from the crane hook and bring the ship eventually in upright position
15. Speed up the process by lowering the crane hook
16. Unfasten rigging

At offshore jobs, this procedure is governed by the position keeping ability of the DP-system. Before positioning can start, the reference system first needs to be lowered to the seabed. Then it takes between one or two hours to calibrate the systems.

D.2 Loss of time during operation

When looking at the operational reality of lifting operations one can divide the loss of time potential into technical capabilities and human factors.

Loss of time due to technical capabilities

Considering lost time during slewing only, one can observe a relation between load and lost time. The heavier the load, the more delicate the operation. The heeling angle and stability are sensitive to the hook load. As the crane, DP and ballast pumps are all handled manual, the operation is paused numerous times to await the ship (reactive) motions and to maintain control.

Loss of time due to human behaviour

In general the procedures where manual handling and human interaction is involved, are the most time consuming and the most unpredictable. People are sensitive for tiredness, communication, mistakes, weather conditions, spirit, etc. Before lift, the unfastening of the sea-fastening and especially handling the heavy rigging equipment are accountable for lost time. At the port of Leith, for example, one of the slots on the reel was slightly to tight for the shackle. To handle that one heavy shackle and sling and manoeuvre them in the right way, took more then half an hour. Additionally throughout all the processes, taking time for safety checks is necessary.

For offshore operations the weather conditions and DP have an obvious effect on lost time. But when the Jumbo Javelin is used for offshore work, she has her own particularities according to the captains. The J-class is a shipping vessel with DP. This make it suitable for offshore work but it does not make it a dedicated offshore ship. The ship is designed for a crew of twenty persons. An additional accommodation block can be put on deck during offshore operations to house an extra 60 beds. The extra accommodation does not come with an extra kitchen and mess room. So although working on 24/7 schedule with 12-hour shifts, the rooms are tight for this amount of people. Also the water supply and disposal are not fully capable and the wifi does not work properly on regular basis. This may sound like minor short comings, but they cause serious irritations among the crew members. Keeping everyone happy is key for a vital workforce. During offshore work people from different companies with different interests are on board. Therefore significant time loss can easily be caused by mitigations and many meetings. So again, having all system up en running and having everyone satisfied everyone's work and living conditions could potentially saves the largest bit of lost time.

Advantage faster anti heel

Only for transition piece installation. When transition piece connected with the monopile, while de de-ballasting of the ship. The ship can float away and pull the transition piece sidewards.

D.3 Pump log book

	Total run time [min]	Start/stop count [-]
Pump 1	28	10
Pump 2	28	10
Pump 3	35	14

Total duration	76	[min]
Down time	36	[min]

Figure D.2: Log book of 815 ton slewing operation

time [min]	dif [min]	P1 [on/off]	P2 [on/off]	P3 [on/off]	pump time [min]
0	0	0	0	1	1
1	1	0	0	0	0
4	3	1	0	1	6
7	3	0	0	0	0
10	3	1	1	1	6
12	2	0	0	0	0
13	1	1	0	1	2
14	1	0	0	0	0
18	4	1	1	1	24
26	8	1	0	1	2
27	1	1	1	1	12
31	4	1	1	1	12
35	4	1	0	0	3
38	3	1	1	1	6
40	2	0	0	0	0
41	1	1	1	1	0
41	0	0	0	0	0
46	5	0	1	0	1
47	1	0	0	0	0
48	1	0	0	1	2
50	2	0	0	0	0
54	4	0	1	0	1
55	1	0	0	0	0
59	4	0	1	1	4
61	2	0	0	0	0
68	7	0	1	1	8
72	4	0	0	0	0
75	3	0	0	1	1
76	1	0	0	0	
Total	76	10	10	14	91

Figure D.3: Log book of 815 ton slewing operation

Appendix E

Engineering design model

Figure E.1 shows the Pahl & Beits example of a consensus model. Their model has four phases: clarification of the task, conceptual design, embodiment design and detail design.

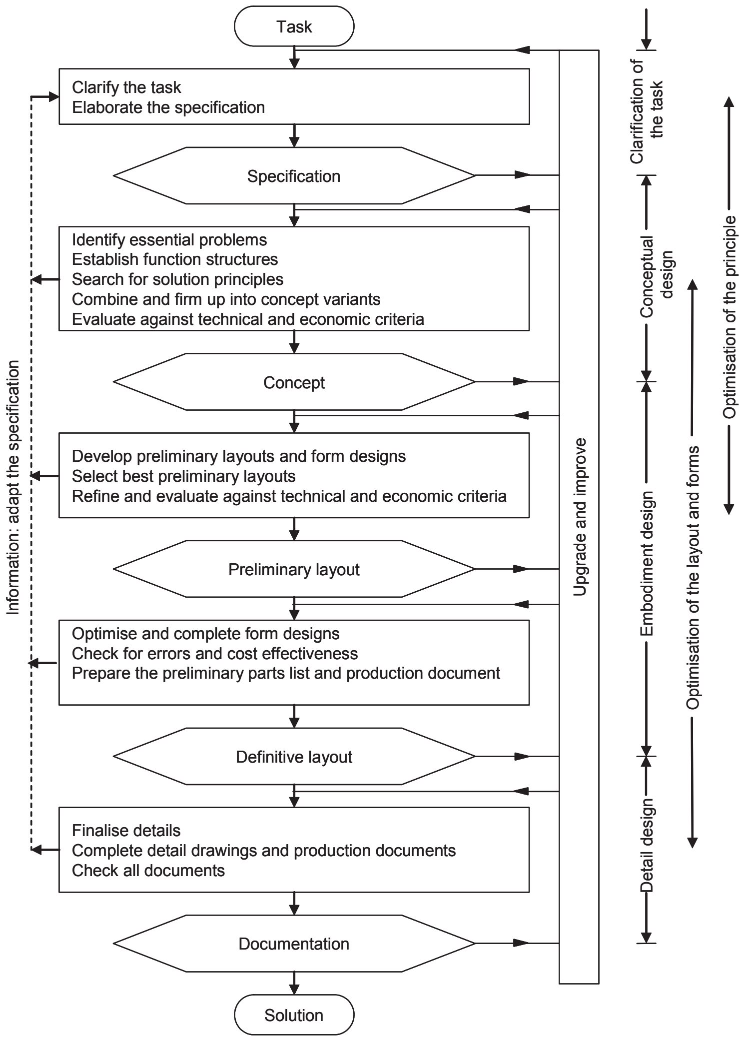


Figure E.1: Phase model of design process by Pahl and Beits

Appendix F

Morphological analysis

The purpose of the concept generation was to generate candidate concepts to solve to the second research question: How can the duration of the lifting phase of the Jumbo J-type be reduced? In section 6.1 the design methods are briefly explained and the four potential solutions principles were presented. These are repeated again in figure F.1. A morphological analysis has been done to create an overview of solution principles physically could be designed.

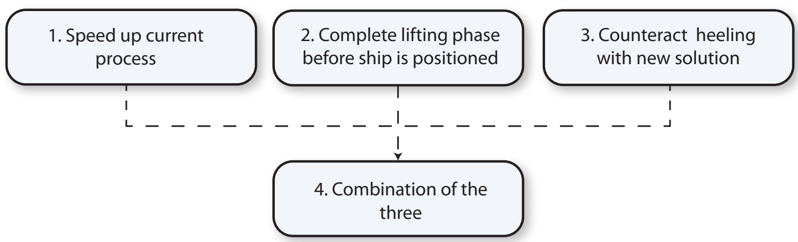


Figure F.1: Solutions principles

F.1 Speed up the current process

To speed up the current process can be done in 3 different ways:

- Faster shifting of ballast water
- Accept larger heeling angles
- Calculate the operation in advance so the cranes and the ballast pumps can work simultaneously as a continuous system.

The sub-solutions are visualized in figure F.2.

F.1.1 Faster shifting of ballast water

To generate a larger moment with water ballast in shorter time, one could consider to increase the volumetric flow rate. The could be done by increasing the rpm of the pump, install bigger pumps and, if required, increase the cross section of the piping.

Pump faster

With increasing the rpm of the pump, a larger pump head can be realised. However this depends on the pump design and and is therefore limited. Pump manufacturers specify the minimum and maximum rpm of their pump.

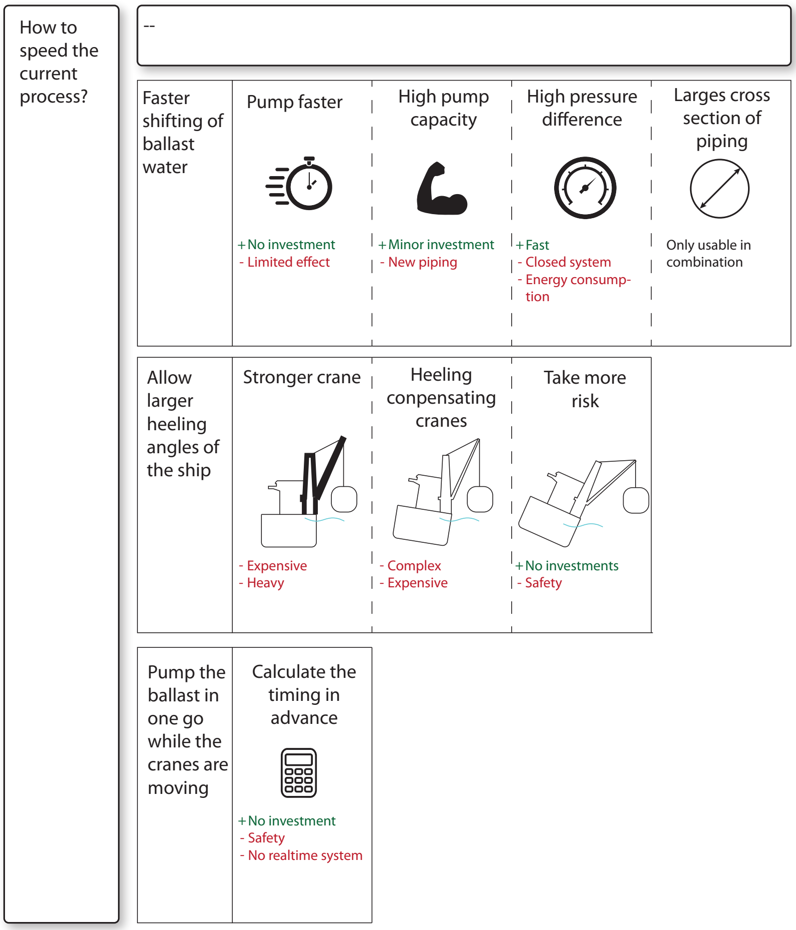


Figure F.2: Speed up current process

Larger pumps

If increasing the RPM is not possible or does not have the preferred effect, one can consider larger pumps. The space available in the pump room and the current installed piping must allow to install larger pumps.

Pump with pressure difference between the tanks

When the wing tanks are pressurised a large pressure difference between the tanks can be applied instantly. This principle can also be used for a roll reduction system. The disadvantage is that the tank system must be transformed into a closed system and an advanced control system should be developed.

Enlarge cross section of the piping

Enlarging the current cross sectional is preferably avoid. There are hundreds of meters of piping with valves in narrow spaces. Replacing the piping is a labour-intensive activity.

F.1.2 Accept larger heeling angles

The workability of the system is limited by the allowable heeling and roll angles. The maximum working angles are currently limited by the structural properties of the cranes. Larger angles can be realised when the crane allows it. So stronger cranes or motion compensate crane could be considered. An other option is to stretch the current limitations.

Install stronger cranes

The current cranes are originally not made for offshore conditions. Stronger cranes could allow large lead angles. Replacing crane can also require system and structural changes. Installing new crane is potentially an expensive option.

Install heeling compensating cranes

Keeping the cranes in a upright position will ensure the best crane capacity. However this is not necessary for crane loads up to 450 ton. Developing a heeling compensating structure for the heavy lift crane is a technical challenge which is preferably avoid.

Take more risk

The maximum allowable angles are based on the structural capabilities of the crane. So for each project different allowable angles apply. As the lifting phase is mainly limited by the roll angle, research in stretching the boundaries of allowable angles may be an beneficial solution.

F.1.3 Lift following calculated plan

Calculate different lifting scenarios in advance might reveal significantly beneficial scenarios.

F.2 Complete lifting phase before ship is positioned

One could consider the cancel out the waiting time of the lifting phase by lift the load before the ship is at position. This can be done during transit to the site or during positioning with the DP-system. The sub-solutions are visualized in figure F.3.

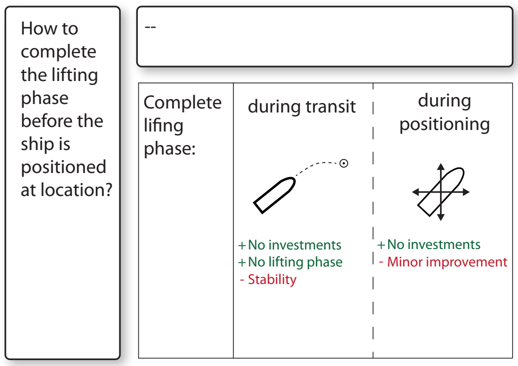


Figure F.3: Before positioning

Lift during transit

If the load is already at the preferred position at arrival, no times is lost due to moving the load. Sailing with a swinging heavy load on the cranes does not sound like a good idea. Even with tag lines large peak load can occur. The ship stability is also effected.

Lift during positioning

Here the same problems apply as during transit. The difference is that the system is not effected by ship motions during sailing. But when the DP-system is calibrating it can cause roll motions. With a load overboard the system can be sensitive to these moments.

F.3 Counter act heeling by reducing the varying heeling moment with applying a force at an arm

Applying a force on the ship can be done by gravity, thrust or buoyancy. The sub-solutions are visualized in figure F.4.

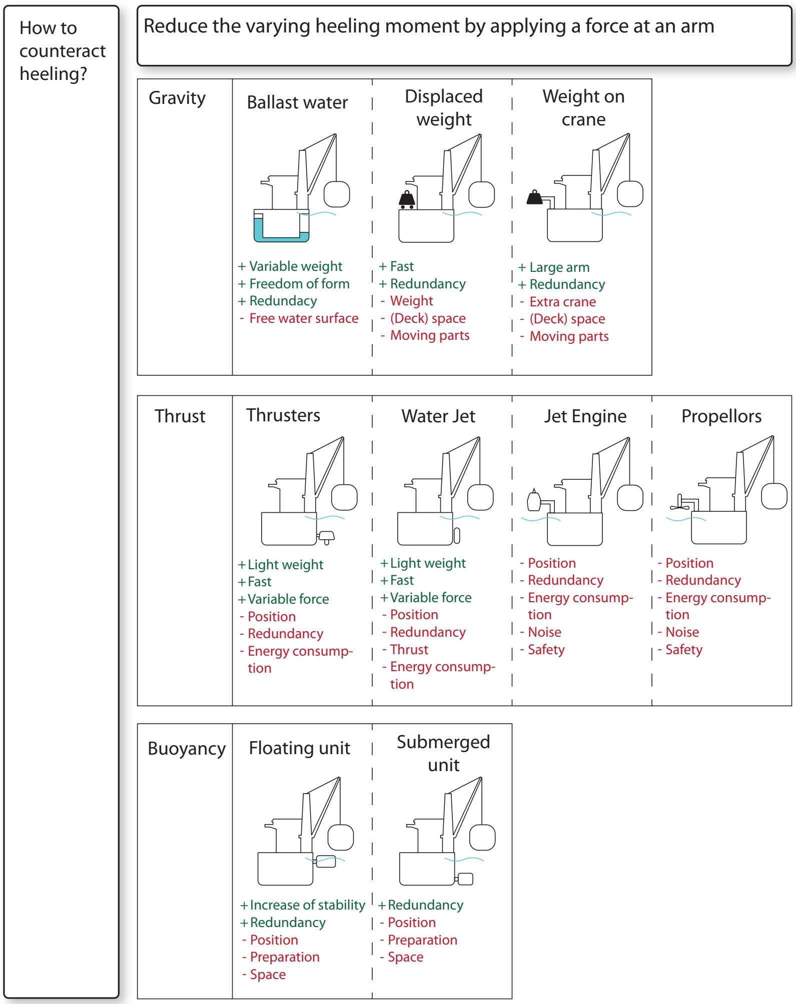


Figure F.4: Reduce heeling moment

F.3.1 Generate force by gravity

To apply a gravity force a large mass is required. This can be done by the water in the ballast system or to have a movable mass on board.

Ballast water

Ballast water is currently used. The advantages are the variable volume and the ability to transport. The moment rates are however limited by the volumetric flow rate. The system also does not loose its moment when a component fails during operation.

Displaced weight on deck

A movable loan on board can generate an accurate predictable moment. This option has the potential to be developed into a movable product. A disadvantage could be that having a large extra weight on board takes space and can be expensive.

Displaced weight on a crane

This option is similar to the previous one. However the lever arm can be larger and a smaller weight is required. To be able to move the weight out board does require a special crane structure and can be expensive.

F.3.2 Generate force by thrust

To generate a force by thrust mechanical system are required. Thrust can be applied with water or air.

Thrusters

Thrusters are widely used for propulsion and DP. In they could also be used to apply a force to compensate for the crane moment. The advantage is their weight an ability to create a variable force. However their consistency can be questioned when operating relatively close to the water line. This option also loses its complete moment when the component fails. Also making a retractable thruster in the comes with a high price.

Water jets

Similar to the thruster.

Jet engine or propellers

Possible in theory but an unrealistic option.

F.3.3 Generate force by buoyancy

An obvious way to generate a force on the water is with buoyancy. A buoyancy force can be applied by a body in the waterline or below.

Floating unit

A floating unit is already used by Jumbo. It is called a flipper or pontoon and is used to increase the moment of inertia of the water plane and increase the GM. When a larger volume is displaced a larger force can be generated. A disadvatnage of the prinicple could be its installation. Also a system to cause a variable water displacement needs tp be developed. Being at the water surface wave could influence the force applied to the unit.

Submerged unit

This option is similar to the above but is not influenced by the waves.

F.4 Counter act heeling by increasing the restoring moment

An increase of the GM can result in smaller heeling angles. The GM can be increase by enlarging the water plane area or lower the centre of gravity. The sub-solutions are visualized in figure F.5.

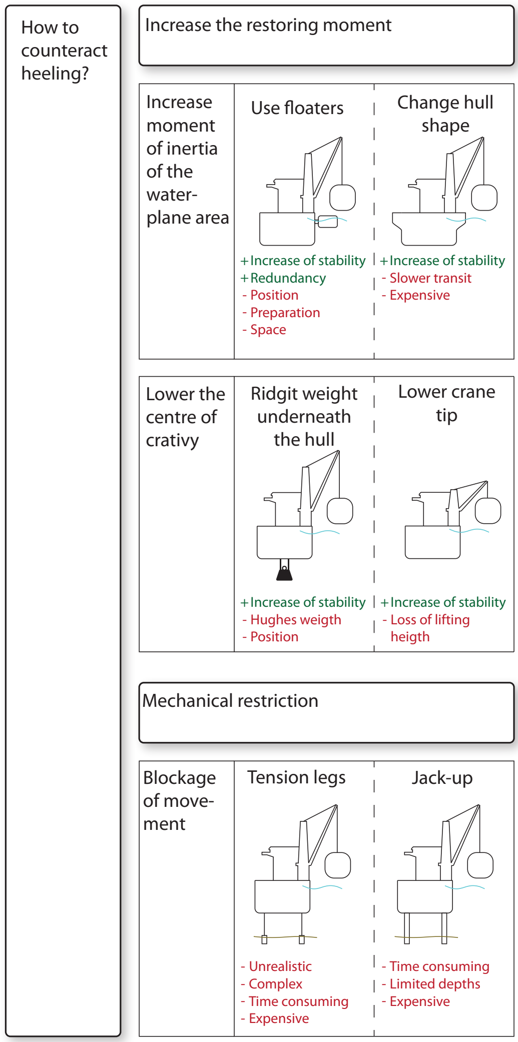


Figure F.5: Increase restoring moment

F.4.1 Increase moment of inertia of the water plane area

Increasing the water plane area results in an increase of the GM independent of the side of the ship.

Floaters

This method is relatively easy but requires the installation of a body at the side of the ship.

Change hull shape

Changing the hull shape is expensive and might result in disadvantages during transit mode.

F.4.2 Lower centre of gravity

Lowering the centre of gravity also results is a smaller lever arm when the ship rolls.

Apply weight underneath the hull

Applying a weight underneath the hull not only lowers the centre of gravity vu also results in a counter moment when rolling. However the weight is water is much less then in air so a very large weight in needed. The hull shape is also effected by an object outside the hull and causes friction.

The principle of lowering the centre of gravity however, is currently used. The double bottom water tanks are filled with water and lowering the centre of gravity and increasing the draught.

Lower crane tip

Lowering the crane tip results in a smaller different in $\overline{GM}_{initial}$ and $\overline{GM}_{eff}$. A decrease crane height capacity is a huge operational disadvantage and the effect on the stability is minor.

F.5 Block roll motions with mechanical restrictions

The ship can also be restricted from rolling by blocking the movement. This can be realised by a tensioned anchor system or by lifting the ship above the water level.

Tension legs

With tension legs the ship get pulled toward the sea floor by the anchor system. The legs or cables are tensioned and the ship does not roll. This option in unrealistic when the ship has to relocated often.

Jack-up

Jack-ups are widely used for drilling, mono pile and wind turbine installation. jacking up the ship above the water line is time consuming. Rebuilding the ship with such a system expensive.

F.6 Concept selection

Important criteria for candidate selection are the potential of varying the moments, moment redundancy at component failure and the space required. Also minor adjustments to the ship are preferred. The 5 candidate concepts selected are shown in figure F.6. The candidate concepts are elaborated in section 6.2.

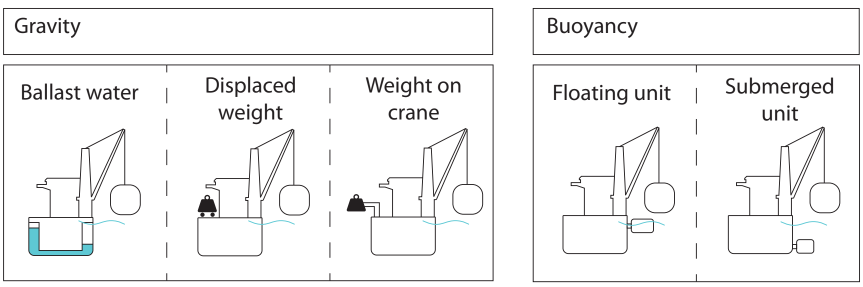


Figure F.6: Reduce the varying heeling moment by applying a force at an arm

Appendix G

Weighted objectives method

In secton 6.3 the weighted objectives method and the chosen categories are briefly explained. This appendix contains the scoring tables and its parameters. The scores are on a scale of 1 to 5 where 5 is the best score. The parameters and results are discussed here. As a reminder the concepts are illustrated below.

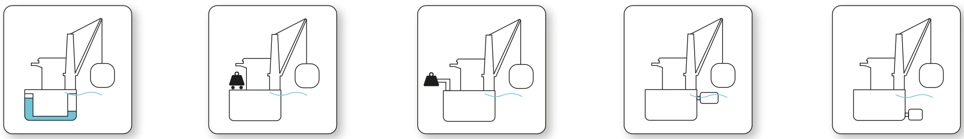


Figure G.1: Candidate concepts

G.1 System capacity

In the system capacity category all parameters considering the potential technical capacities of the system are rated. The scores per parameter are given in figure G.2.

System capacity	Weight	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5
Total moment capacity	1	3	3	3	3	3
Speed of moment change	1	3	5	5	3	3
Moment rate capacity range	1.5	3	4	5	2	3
Start/stop reaction time of the system	0.8	1	4	5	3	3
Ability to alter an increase or a decrease of the moment quickly	1	1	5	4	4	4
Redundancy	1	5	3	3	3	3
Sensitivity to ship motions	1.5	4	2	4	1	3
Total score	39	18.8	22.2	25	16.9	19.9
%Total possible score		48%	57%	64%	43%	51%

Figure G.2: System capacity

Total moment capacity

The system should be able to handle al allowable crane moments. The scores of this parameter are all equal because this depends on how large the tanks and weight are designed.

Speed of moment change

To be able to follow the crane moments, a wide range of moment rates is preferred. Concept 2 and 3 have a movable solid weight and are therefore expected to be the fastest system. Moving a rigid mass in air is generally faster than moving large water volumes.

Start/stop reaction time of the system

When the operation requires the system to stop and start, a quick reaction time is an advantage. Preferable the systems capacity allow to follow the crane moment without (numerous) starts and stops. Therefore this parameter is considered less important and is the weight factor slightly decreased.

Ability to alter an increase or decrease of the moment quickly

Movement of the crane might require a ballast system that not only increases the moment but also decreases. In case of the water ballast the flow first must be stopped and that changed in direction.

Redundancy

Preferably the system is (partly) redundant. In the case of the water ballast, if one tank pair is out of order there are still two others.

Sensitivity to ship motions

To be able to accurately control the moment generated by the system, it should not be significantly effected by the ship motion. Therefore this parameter is given a weight factor of 1.5. Concept 4 has a pontoon in the water line and is effected by waves and ship motions.

G.2 Effects on ship motion

The ship motions can have effect on the anti-heeling system, but the system will also effect the ship motions and stability. This is crucial is relation to the workability. The scores per parameter are given in figure G.3.

Effects ship motions	Weight	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5
Increase of GM	1	4	2	3	5	3
Mass moment of inertia	0.5	3	3	2	3	3
Natural period	1	3	3	3	1	3
Sensitivity to effects causing heeling	1	4	5	4	3	3
Effect of waves	1	5	5	5	1	3
Effect on heeling angle after component failure	1	5	3	3	3	3
Total score	27.5	22.5	19.5	19	14.5	16.5
%Total possible score		82%	71%	69%	53%	60%

Figure G.3: Effects on ship motion

Increase of GM

A larger GM results in a larger restoring arm. Concept 2 and 3 will decrease the GM by having the vertical displacement of the centre of gravity by their heavy weights on deck. Concept 4 has an increase moment of inertia of the water plane area.

Mass moment of inertia

A larger mass moment of inertia results in a larger natural period but also makes it harder to control the roll motions. The addition mass moment of inertia of the system is expected to be relatively small to the one on the ship. Therefore the weight factor is lowered to 0.5.

Natural period of the ship

Concept 4 is the only concept that significantly influences the natural period. With GM larger that 6 meters the natural period becomes smaller than 10 seconds.

Sensitivity to effects causing heeling

The moment caused by a hanging load becomes even larger when the ship will roll. The ballast system should not be significantly effect the ship motions.

Effect of waves

Concept 4 and 5 are the only concepts directly influenced by the waves. As concept 5 is underneath the water surface and will therefore be less effected than concept 4.

Effect on heeling angles after component failure

When a component fails the system should not loose it moment and preferably has some capacity left after component failure.

G.3 Effects on (deck) operations

An decreased lifting phase should not be at expense of other properties of the ship. Also influencing the deck operations could potentially result in delays. The scores per parameter are given in figure G.4.

Effects on (deck) operations	Weight	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5
Deck or hold space required	1	5	1	3	2	2
Safety risks of moving parts	1	5	3	2	5	5
Weight of the device	0.8	5	1	2	4	4
Mobility of the system	1	5	4	1	1	1
Extra procedures for preparation system	2	5	4	4	1	1
Ability to use in port	1	5	5	3	2	1
Total score	34	34	21.8	18.6	15.2	14.2
%Total possible score		100%	64%	55%	45%	42%

Figure G.4: Effects on (deck) operations

Deck or hold space required

Deck and hold space, together with the crane capacity, are the most valuable properties of the ship. The less space taken by the system. Concept 1 has the best score here because the wing tanks has the least influence on the space. Concept 2 has the worst because the whole system is on deck.

Safety risks of moving parts

Moving part and heavy weight are potential hazards. When on deck the crew might be seriously hurt by the object.

Weight of the device

The weight can influence the stability and the draught. Heavy devices also are potentially expensive. The advantage of concept 1 is that is can pump the ballast water out when not needed. Concepts 4 and 5 are large objects but are filled with air.

Mobility of the system

Depending on the objects on deck, it might be preferred to make the anti-heeling solution mobile on the ship. For concepts 3,4 and 5 there position on the ship are determined. Concept 1 does not influence the deck operations. Concept 2 can potentially be on various positions on deck.

Extra procedures for preparation of the system

Extra procedures can be time consuming. The purpose of the solution is to save time. Concept 4 and 5 include installation on the side of the ship and will have therefore the lowest score.

Ability to use in port

The Ships are also used for shipping projects. Therefore it is preferable that a developed solution can also be used in port. Concept 4 and 5 make use of buoyancy forces and there use is limited by the quayside.

G.4 Installation, maintenance and repair

The last cluster is "financial" cluster. These parameters are harder to evaluate based on general concepts.

Installation, costs and maintenance	Weight	Concept 1	Concept 2	Concept 3	Concept 4	Concept 5
Ability to intergrade with current system	1	4	5	2	3	1
Energy consumption	1	4	4	4	3	3
Complexity of ship conversion	1.5	5	4	1	4	2
Initial investment	2	5	1	1	4	1
Maintenance	1	5	4	4	3	3
Total score	32.5	30.5	21	13.5	23	12
%Total possible score		94%	65%	42%	71%	37%

Figure G.5: Installation, maintenance and repair

Ability to integrate with current system

Integrating the solution with the current system potentially saves on newly build systems.

Energy consumption

Hard to say something about the energy consumption. Running a few pump does consume less energy than moving cranes. But relatively small amount energy use for a anti-heeling system is not really expensive, so the difference between the concept will not be determined by this parameter.

Complexity of the ship conversion

Complex ship conversions are expensive. Therefore the weight factor is 1.5. Concept 3 requires a new structure to support a weight out board and concept 5 requires a system to bring the buoyancy tank underneath the water. Concept 1 can make use of the current system.

Initial investment

Concept 2 and 3 are heavy systems and need supportive devices to move the weights. Concept 5 requires a system to submerge the buoyancy tank and vary the buoyancy forces. This are expensive solutions. Concept 4 might make use of the existing buoyancy tanks. The initial investment is the lowest for concept 1. As the purpose of the shorter lifting phase is to save money so the weight factor of this parameter is 2.

Maintenance

The equipment used offshore can be expensive in maintenance. The more simple and heavy duty the solution the better.

Appendix H

Flow-analysis technique

Moving ballast water with a hydraulic transport system can be considered as a fluid-flow problem. Three basic method Frank M. White [25] suggest to solve this are problem are:

1. Control-volume, or *integral* analysis
2. Infinitesimal system, or *differential* analysis
3. Experimental study, or *dimensional* analysis

As for this model the total amount of water volume transported from one reservoir to another over time is the most important, the integral analysis with a control volume is the most suitable. This appendix will give a brief explanation of this method and the application to the model.

H.1 Control volume flow-analysis technique

A control volume is a finite region with open boundaries through which mass, momentum, and energy are allowed to cross. This control volume can contain a single component or a complete system. See figure H.1. When considering the flow properties over the cross sections of the inlet or exit nearly uniform, they can be considered one-dimensional for calculations. A balance can be made between the incoming and outgoing fluid. Although the details of the flow are ignored, the quantitative information of the system, like flow rate, friction loss and pump head can be determined with this method.

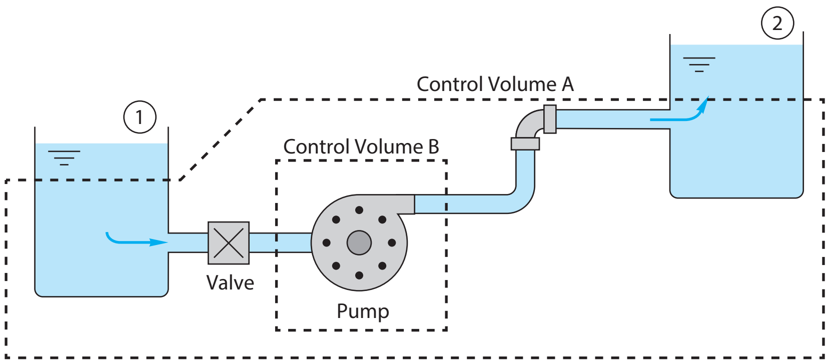


Figure H.1: Example of control volumes

Although this is a widely used method, it is a crude one. In order to us this method to model the flow, it has to satisfy four basic laws of mechanics plus a thermodynamic state relation and associated boundary conditions:

1. Conservation of mass (continuity)
2. Conservation of species (incompressible)
3. Linear momentum (Newtons second law)
4. Conservation of energy (first law of thermodynamics)
5. A state relation like $\rho = \rho(p, T)$
6. Appropriate boundary conditions at solid surfaces, interfaces, inlets, and exits

H.1.1 Conservation of mass

The mass of the system is conserved and does not change. So mass can neither be created nor destroyed. Mass can be moved or relocated in space. Conservation of mass is simply expressed as:

$$\left(\frac{dm}{dt}\right) = 0 \tag{H.1}$$

Considering a number one-dimensional inlets and exits of the control volume, the conservation of mass can be written as

$$\int_{CV} \frac{\delta\rho}{\delta t} dV + \sum_i (\rho_i A_i v_i)_{out} - \sum_i (\rho_i A_i v_i)_{in} = 0 \tag{H.2}$$

H.1.2 Conservation of species

The ballast system contain sea water, which is a variable mixtures containing components as salt. This could cause density variations. However this variations are negligible considering the flow rate of the system. So $\delta\rho/\delta t = 0$ (incompressible fluid) and in combination with a steady flow, the equation H.2 becomes:

$$\sum_i Q_{out} - \sum_i Q_{in} = 0 \tag{H.3}$$

The mass flows entering and leaving the control volume are equal.

H.1.3 Linear momentum

Newton's second law states that if the surroundings exert a force on the system, the mass will begin to accelerate.

$$\sum F = \frac{d}{dt}(mv)_{syst} \tag{H.4}$$

When mass is flowing in and out the control volume it can be expressed as:

$$\sum F = \frac{d}{dt}\left(\int_{CV} v \cdot \rho \cdot dV\right) + \sum (\dot{m}_i v_i)_{out} - \sum (\dot{m}_i v_i)_{in} \tag{H.5}$$

H.1.4 Conservation of energy

The first law of thermodynamics states that if heat is added to the system or work is done by the system, the system energy must change.

$$\frac{dQ}{dt} - \frac{dW}{dt} = \frac{dE}{dt} \tag{H.6}$$

Fluid transport systems typically contain internal, kinetic and potential energy. For steady flow in a system with one inlet and one outlet and containing machinery like a pump, the equation can be written as

$$\frac{\dot{Q}}{\dot{m}} - \frac{\dot{W}_{shaft}}{\dot{m}} - \frac{\dot{W}_{viscous}}{\dot{m}} = (\hat{h} + 1/2v^2 + gz)_2 - (\hat{h} + 1/2v^2 + gz)_1 \tag{H.7}$$

Where W_{shaft} is the work done by a machine and $W_{viscous}$ the shear work due to viscous stresses at the control surface.This energy relation is widely used for many engineering analysis.

When considering no heat transfer $dQ/dt = 0$ en divide equation H.7 by g, each term has become a length or *head*.

$$h_{pump} - h_{friction} = \left(\frac{p}{\rho g} + \frac{v^2}{2g} + z\right)_2 - \left(\frac{p}{\rho g} + \frac{v^2}{2g} + z\right)_1 \tag{H.8}$$

Where h_{pump} is the head added by the pump, $h_{friction}$ the energy lost due to friction within the system and the terms on the right hand side can be considered as the flow energy, kinetic energy and the potential energy.

H.1.5 State relation

As mentioned in H.1.2 the variations in density is assumed to be negligible. additionally the possible temperature variations of a few degrees are not expected to have a significant effect on pressure differences. Therefore $\rho = \rho(p, T) = 1025 kg/m^3$ and considered constant.

H.1.6 Boundary conditions

All walls are assumed solid and impermeable

Appendix I

SimHydraulics

This appendix provides information about the SimHydraulics software. The information is based on the SimHydraulics User's Guide [15].

I.0.7 Program description

SimHydraulics provides component libraries for modelling and simulating hydraulic systems is the Simulink environment. Simulink is a graphical customizable block diagramming tool developed by MathWorks. SimHydraulics is specially developed to cover modelling scenarios with hydraulic actuators as a part of a control system. Typical SimHydraulics components are pumps, valves, tanks, pipes and actuators.

I.0.8 Model Simulation

Mathematical representation are the foundation for physical simulation. The mathematical representation of the physical system contains a set of equations of ordinary differential equations and algebraic equations. When using Simulink, this set of equations must be constructed manually to create a block diagram representing the physical system. Other than Simulink, where the signals are essentially unitless, SimHydraulics uses "physical signals". This physical signals can represent information of the flow like volumetric flow rate, pressure and elevation from the reference plane. Using physical connections between the components the SimHydraulic model can match the physical structure of the real system.

Modelling low-pressure fluid transport systems

Fluid transport systems usually operate at low pressures about 2 - 4 bar. SimHydraulic provides library components for low pressure fluid transportation systems. A component with one entrance, one exit and a steady uniform flow is characterized by the following energy equation

$$-\frac{\dot{W}_s}{\dot{m}g} = \frac{v_2^2 - v_1^2}{2g} + \frac{p_2 - p_1}{\rho g} + z_2 - z_1 + h_{loss} \tag{I.1}$$

Where $\dot{W}_s$ is the work rate performed by the fluid and $\dot{m}$ the mass flow rate and is essentially the same as equation H.7 based on the steady-flow energy equation.

I.0.9 Simulation phases

When running the model, SimHydraulic automatically goes through the program's simulation phases to simulate the model over time. The phases are show in the flow chart I.1.

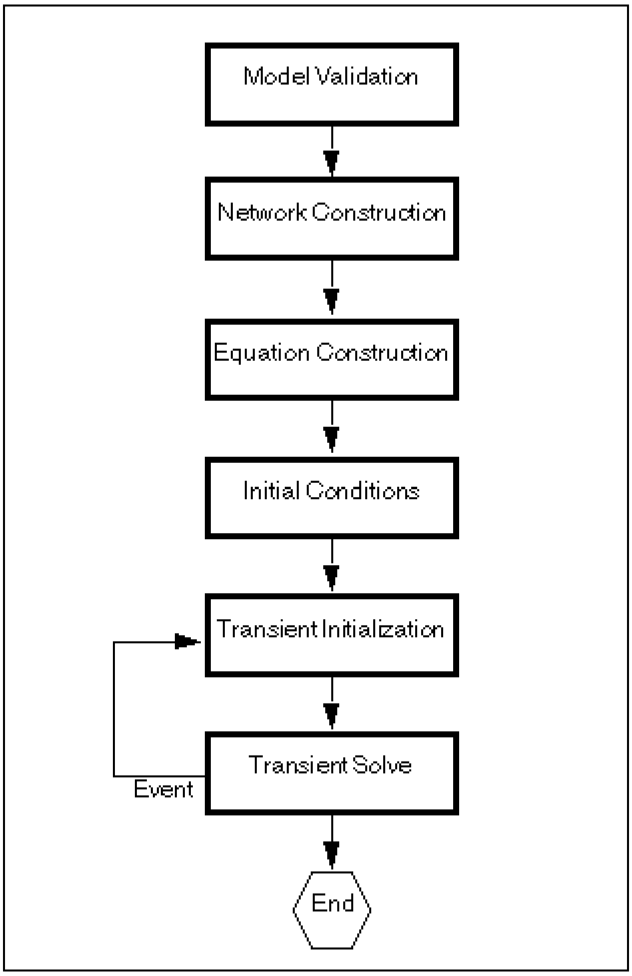


Figure I.1: Flow chart of the simulation phases of SimHydraulics

Model validation

Checks the data entries of the block diagram boxes, the connections and if the signal types between the blocks are matching.

Network construction

The solver constructs the physical network based on the block diagram.

Equation construction

Based on the network configuration, the parameter values in the block dialog boxes, and the global parameters defined by the fluid properties the solver constructs the system of equations for the model.

Initial conditions computation

The solver computes the initial conditions by finding initial values for all the system variables that exactly satisfy all the model equations.

Transient initialization

Transient initialization fixes all dynamic variables and solves for algebraic variables and derivatives of dynamic variables. The goal of transient initialization is to provide a consistent set of initial conditions for the next phase, transient solve.

Transient solve

Finally, the solver performs transient solve of the system of equations. In transient solve, continuous differential equations are integrated in time to compute all the variables as a function of time.

Appendix J

Frictional head loss

This appendix will elaborate on the used friction factors and discuses the sensitivities of the parameters effecting the head loss in the system.

J.1 Straight pipe sections

The frictional head loss in straight pipe sections is given as the Darcy-Weisbach equation

$$h_{loss,pipe} = f(v_{pipe}, D) \cdot \frac{L}{D} \cdot \frac{v_{pipe}^2}{2 \cdot g} \tag{J.1}$$

J.1.1 Friction factor

For the calculation of the friction factor the reformulated equation of Haaland is used

$$f \approx \left[-1.8 log \left[\frac{6.9}{Re_D(Q)} + \left(\frac{\epsilon/D}{3.7} \right)^{1.11} \right] \right]^{-2} \tag{J.2}$$

with

$$Re_D = \frac{\rho \cdot Q \cdot D}{\mu \cdot A_{pipe}} \tag{J.3}$$

With an increasing volumetric flow rate the term $6.9/Re_D$ goes to zero. therefore the friction factor stagnates and the relative roughness becomes governing. See figure J.1.

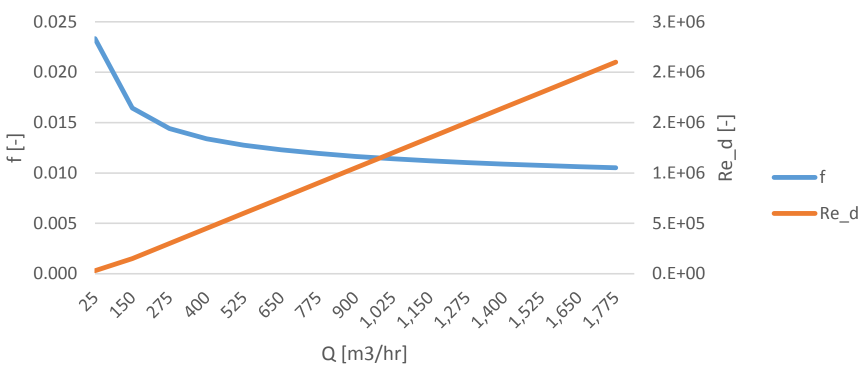


Figure J.1: Realtion between the friction factor and the volumetric flow rate.
$D = 0.3m$ and $\epsilon = 0.002mm$

GRS has an absolute roughness of $0.002mm$ and a relative roughness of $6{\cdot}10^{-6}$, which corresponds to a relatively smooth pipe. To validate the calculated friction factor, it is compared with the moody chart shown in figure J.2. This shows that the friction factors are approximately correct.

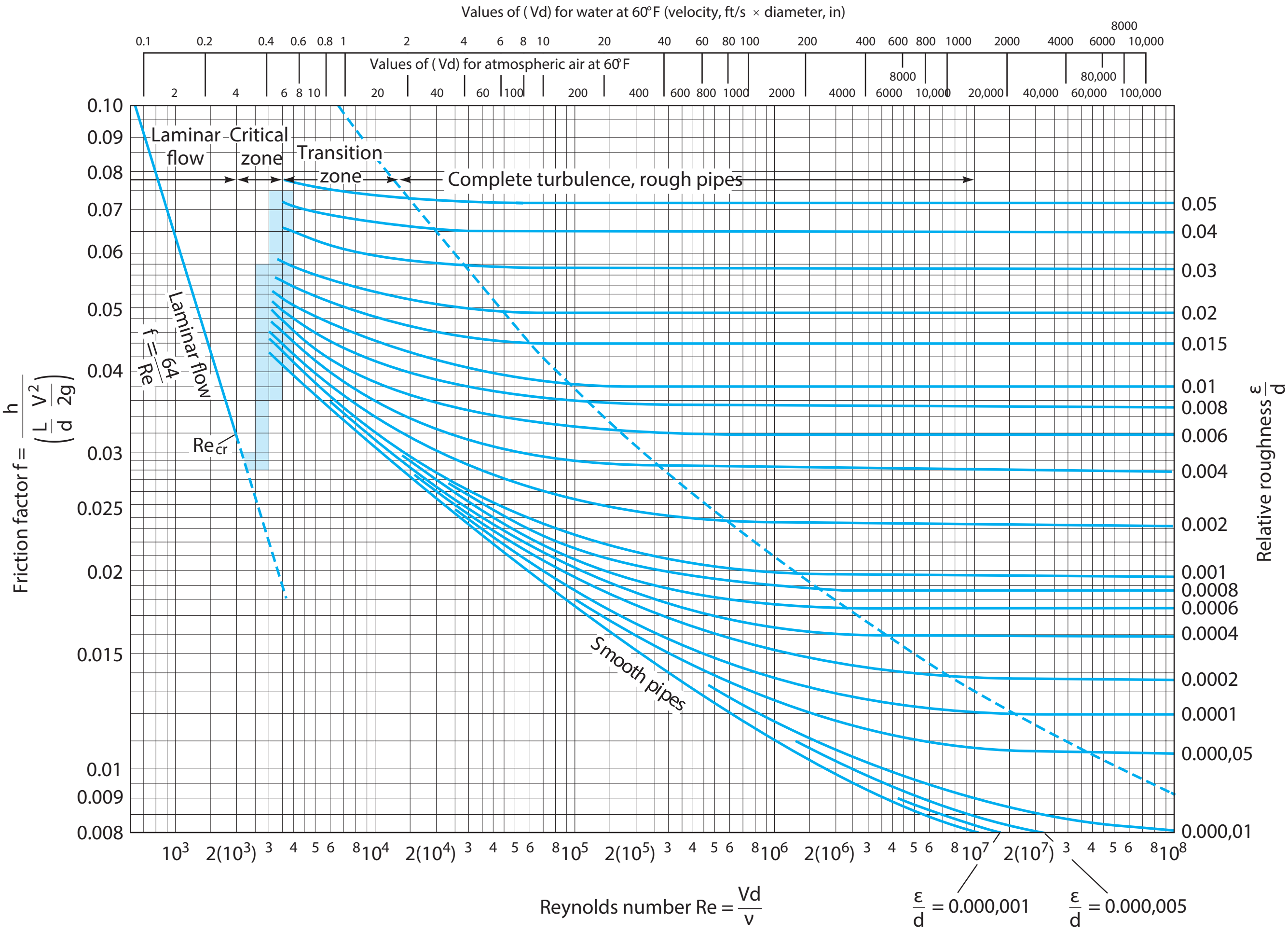


Figure J.2: The Moody chart for pipe friction with smooth and rough walls [25].

J.1.2 Sensitivities of the Darcy-Weisbach equation

The frictional head loss of a pipe section with a steady flow depends on:

1. fluid velocity
2. Pipe dimension
3. Roughness of the pipe material
4. Fluid properties

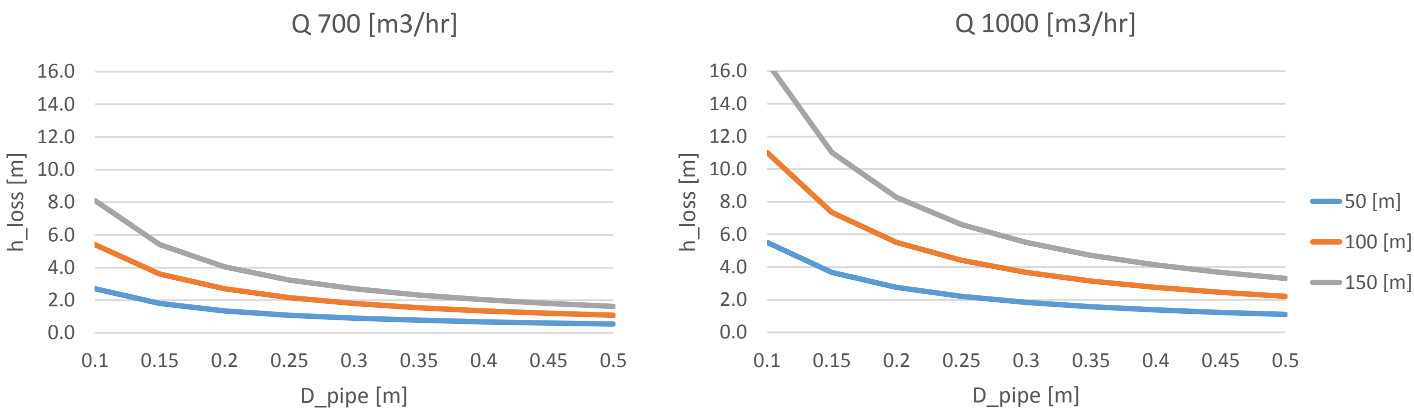


Figure J.3: Head loss of straight pipe lines over various pipe diameters and lengths

In figure J.3 the correlation between the pipe diameter and the head loss is shown. Three different pipe lengths are calculated over two corresponding flow rates. To be more consistent the flow rate should be replaced by the velocity, as the velocity changes over varying pipe diameters. However, the ballast system is all about transporting water volumes so keeping the flow rate constant gives a better view of the varying pipe dimensions on the head loss of the system.

J.2 Minor losses

Additionally to the Moody-type friction loss caused by the length of the pipe, there are also additional losses in the system caused by fittings, valves, inlets, outlets and tees. These are called minor losses. All the minor losses summed up can cause a significant loss head.

$$h_{minor} = \sum (N_i K_i) \frac{v_{pipe}^2}{2g} \tag{J.4}$$

with

$$\sum (N_i K_i) = N_{bends} K_{bend} + N_{valves} K_{valve} + N_{tees} K_{tee} + N_{inlets} K_{inlet} \tag{J.5}$$

Where K is the friction factor of the component causing friction loss and N is the number of components in the system.

J.2.1 Bends

The resistance coefficient K_{bend} is determined by the dimensions of the bend and the roughness of the material. K_{bend} remains constant over the flow velocity. Figure J.4 shown the resistance coefficient K for 90 degrees bends.

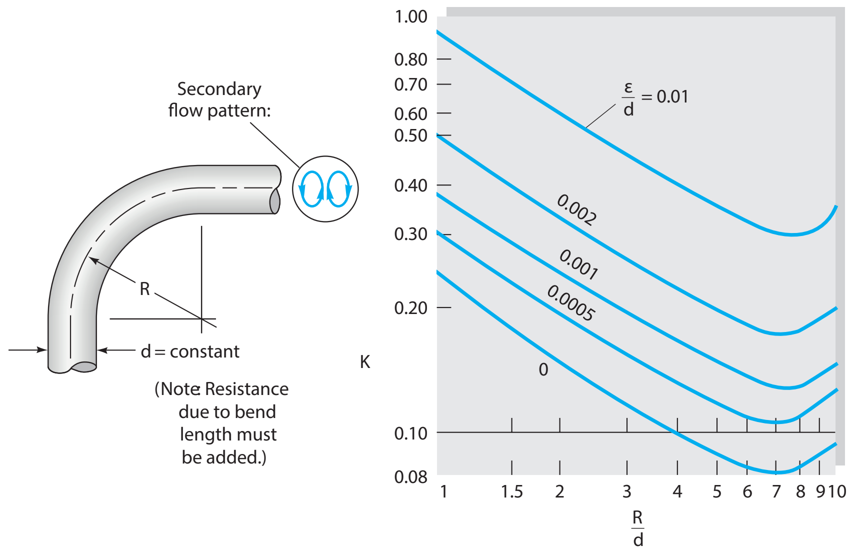


Figure J.4: Resistance coefficients for 90 degrees bends. [25]

The pipeline in the ship makes sharp corners which correspond with R/D rates of approximate 1.2. As the pipe is relatively smooth, it corresponds with the relatively roughness curve of 0. To be on the conservative side a resistance coefficient K_{bend} of 0.25 has been chosen.

J.2.2 Valves

All the valves in the system are butterfly valves. When opening the valve the disk stays in the pipe and rotates to a 90 degree angel. The disk splits the flow and causes head loss. The factor depends on the opening angle. Figure J.5 shows the friction factors for three different manufacturers. Assuming fully opened valves the factor K_{valve} drops till a value between 0.2 and 0.35, depending on the valve. The value of 0.35 is selected for the calculation to be on the conservative side.

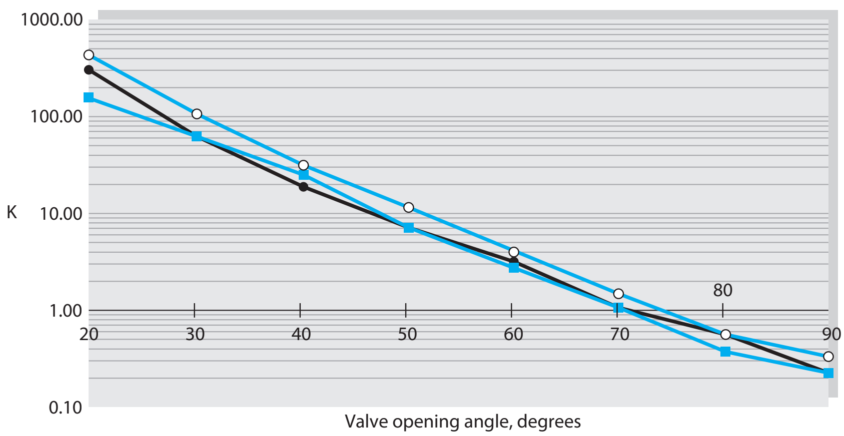


Figure J.5: Performance of butterfly valves: loss coefficients for three different manufacturers. [25]

J.2.3 Inlet and outlet loss coefficients

The transition zone between the ballast tank and the pipe is relatively sharp edged. Figure J.6 shows the friction factors corresponding to the proportional geometry of the transition zone. As the diameter is relatively large comparing with the "length" L of the inlet, the inlet is considered sharp. A friction factor K_{inlet} of 0.5 is selected.

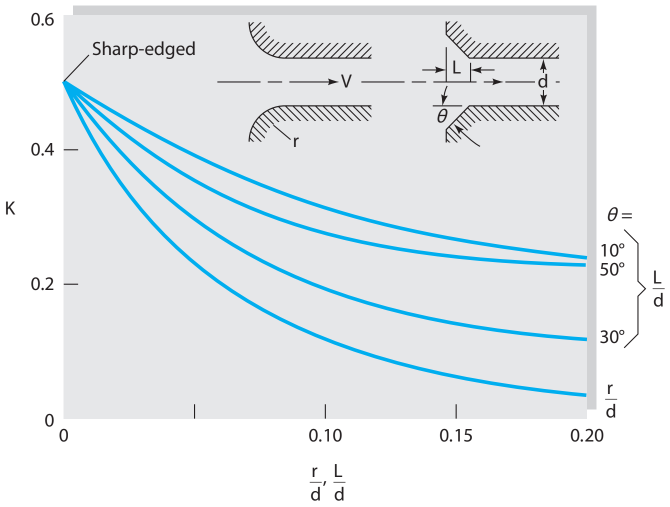


Figure J.6: Inlet and outlet loss coefficients for rounded and bevelled inlets. [25]

J.2.4 Tees

Due to the possibilities of different routing through the ballast system as described in 3.3, the system contain various tees. The head loss caused by a tee depends on if the flow is a line flow or a branch flow. See J.7.

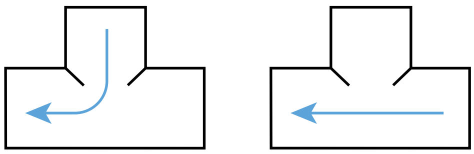


Figure J.7

The friction factor depends on the size of the tee and if it is screwed or flanges. See figure J.8. The tees in the system are 0.3 meter flanged tees. The factors K_{tee} are interpolated as being 0.09 and 0.53 for line flow and branch flow respectively.

	Nominal diameter, in								
	Screwed				Flanged				
	$\frac{1}{2}$	1	2	4	1	2	4	8	20
Tees:									
Line flow	0.90	0.90	0.90	0.90	0.24	0.19	0.14	0.10	0.07
Branch flow	2.4	1.8	1.4	1.1	1.0	0.80	0.64	0.58	0.41

Figure J.8: Resistance coefficients for tees

Appendix K

The centrifugal pump

In centrifugal pumps the fluid enters axially and passes through the pump blades. These blades rotate at an angular velocity and changes the velocity of the fluid. This change in velocity causes a pressure difference making the fluid flow. To understand how the head, power, efficiency and flow rate of a pump can be predicted, a simple one-dimensional approach is applied.

K.1 Elementary pump theory

The fluid enters the impeller at r_1 with a velocity component w_1 tangent to the blade angle plus the circumferential velocity ωr_1. So the absolute velocity is the vector sum of these two, shown as v_1 in figure K.1. In a similar way the v_2 if determined.

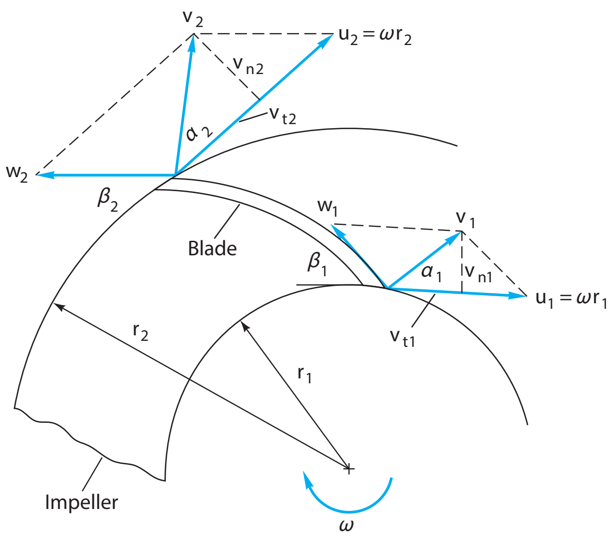


Figure K.1: Inlet and exit velocity diagrams for an idealized pump impeller. [25]

The *Euler turbomachine equations* for a idealizes centrifugal pumps show the relation between the parameters

$$P_w = \omega T = \rho Q \omega (r_2 v_{t2} - r_1 v_{t1}) \tag{K.1}$$

and

$$H = \frac{P_w}{\rho g Q} = \frac{\omega}{g}(r_2 v_{t2} - r_1 v_{t1}) \tag{K.2}$$

where v_{t2} and v_{t2} are the absolute circumferential velocity components of the flow, P_w power delivered to the fluid and H the ideal head.

Efficiency

As mentioned the equations K.1 and K.2 correspond to an ideal pump where $P_w = \omega T$. Traditionally the ωT is called the *brake horsepower* (bhp) and equals the power required to drive the pump. In practise the power delivered to the fluid is always lower than the brake horsepower. So $P_w < \omega T$. The relation between the two is determined by the efficiency of the system η is defined as

$$\eta = \frac{P_w}{bhp} = \frac{\rho g Q H}{\omega T} \tag{K.3}$$

K.2 Pump performance curves

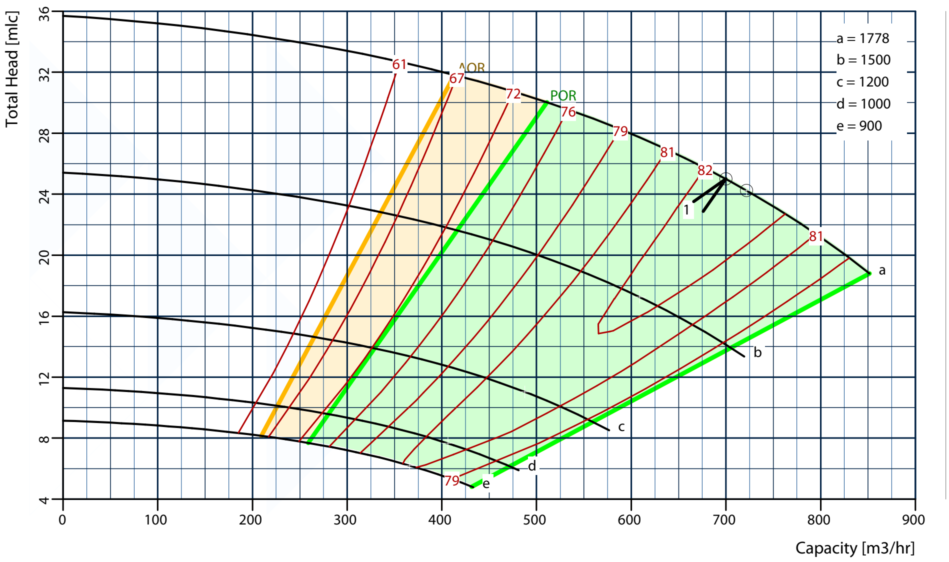


Figure K.2: Pump head curve

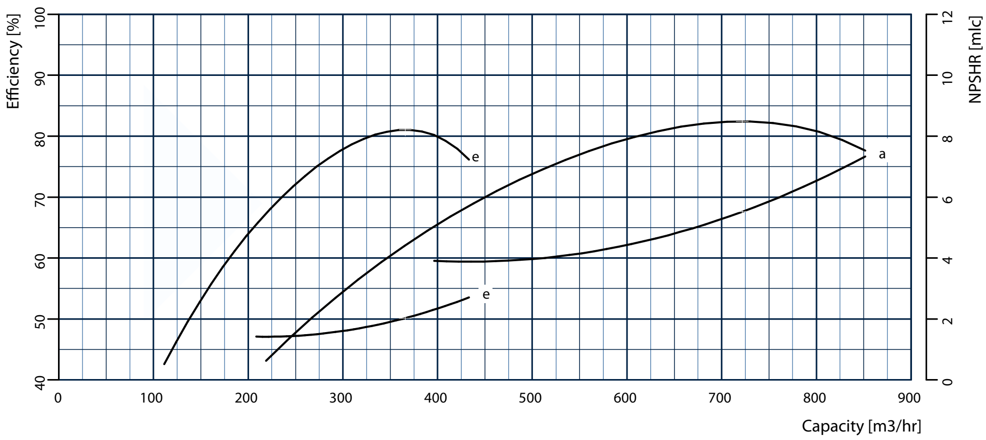


Figure K.3: Efficiency curve

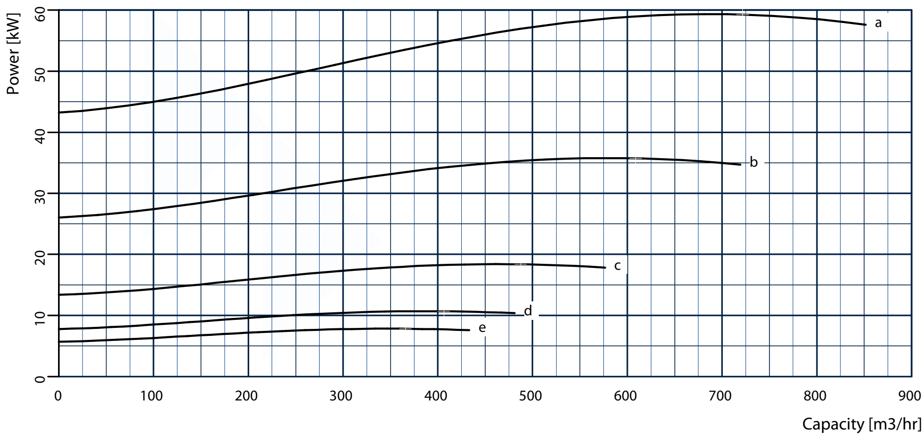


Figure K.4: Power curve

K.3 Operational limits

The operational limits of a centrifugal pump are given by the maximum rotations per minute given by the manufacturer of the pump. The range of rpm of the currently installed pumps are from 900 to 1785 rpm.

Appendix L

Pump model validation

The model validation is to check if the mathematics used are reliable and if the results are in the same order of magnitude as in operational practice. The model is roughly validated with three methods:

- Hand calculations in combination with the verified pump curves
- Pump log books
- Confirmation Captain

L.1 Hand calculations

Equation 7.6 is the governing equation of the pump model. To be able to check the result of the model, the equation can be simplified to find a direct relation between the head and the pipe flow. The assumptions are discussed below.

The tanks are open to the atmosphere so

$$P_{A,B} = P_{C,D} = P_{atm} \tag{L.1}$$

As the decrease of ballast water on port side should equal the increase of ballast water on starboard due to conservation of mass

$$v_{A,B} \cdot A_{A,B} = v_{C,D} \cdot A_{C,D} = v_{pipe} \cdot A_{pipe} \tag{L.2}$$

Filling the equations L.1 and L.2 in the equation 7.6 will give the expression

$$h_{pump} + \Delta h(\phi) - \sum_{i=1}^{n} h_i = \frac{v_{A,B}^2 \cdot k}{2g} \tag{L.3}$$

with $k = 1 - \left(\frac{A_{A,B}}{A_{C,D}}\right)^2$

The water surface area of the tank is much more larger than the cross area of the pipe. So the vertical velocity of water surface becomes negligible.

$$A_{A,B,C,D} \gg A_{pipe} \tag{L.4}$$

$$v_{A,B,C,D} \approx 0 \tag{L.5}$$

This will give the relation between the head and the pipe flow

$$h_{pump} + \Delta h(\phi) \approx \sum_{i=1}^{n} h_i \tag{L.6}$$

Because all frictional losses are in the pipeflow and the cross-sectional area is equal for all sections equation L.6 can be written as

$$h_{pump} + \Delta h(\phi) \approx K_{total} \cdot \frac{v_{pipe}^2}{2 \cdot g} \tag{L.7}$$

The friction coefficient K_{total} is the summation of all coefficients corresponding to the obstructions in the pipe and the straight pipe sections. Per pipe section between the connected wing tanks the K_{total} becomes

$$K_{total} = \left(f\frac{L}{D} + N_{valves}K_{valve} + N_{90bends}K_{90bend} + N_{120bends}K_{120bend} \right) \tag{L.8}$$

Equation L.7 can be rewritten as a function for v_{pipe_i} and when the product of the cross-sectional area A_{pipe} is taken the volumetric flow rate is given as

$$Q_i = \sqrt{(h_{pump} + h_{static})_i \cdot \frac{2 \cdot g}{K_{Total_i}}} \cdot A_{pipe} \tag{L.9}$$

where

$$\sqrt{\frac{2 \cdot g}{K_{Total_i}}} \cdot A_{pipe} = constant \tag{L.10}$$

Filling in the constants in equation L.9 and the volumetric flow rate in m^3/hr for the three ballast tanks pairs can be simplified as:

$$Q_1 \approx 269 \cdot \sqrt{h_{pump_1} + h_{static_1}} \tag{L.11}$$

$$Q_2 \approx 257 \cdot \sqrt{h_{pump_2} + h_{static_2}} \tag{L.12}$$

$$Q_3 \approx 248 \cdot \sqrt{h_{pump_3} + h_{static_3}} \tag{L.13}$$

When the total volumetric flow rate is known the moment rate in $ton.m/s$ can be found as

$$\dot{M}_{ballast} = \frac{(r_{SB} + r_{PS}) \cdot \rho}{3600} \cdot 10^{-3} \cdot \sum_{i=1}^{n} Q_i \approx 6.25 \cdot 10^{-3} \cdot \sum_{i=1}^{n} Q_i \tag{L.14}$$

L.1.1 Example

Figure L.1 shows the results of a 300 ton lift in 300 seconds. At t = 150 seconds the ship is approximately at upright position so the static head can be assumed to be the difference in tank levels. The pump head is given in the lower right graph. Below the volumetric flow rates per tank pair are calculated

Volumetric flow rates

For t = 150 seconds:

$h_{static_1} = 3.6 - 1 = 2.6m$
$h_{pump_1} = 12.3$

$Q_1 = 273 \cdot \sqrt{2.6 + 11.7} = 1017 m^3/hr$

$h_{static_2} = 3.5 - 0.9 = 2.6m$
$h_{pump_2} = 13.1$

$Q_2 = 257 \cdot \sqrt{2.6 + 12.4} = 995 m^3/hr$

$h_{static_3} = 3.5 - 1 = 2.5m$
$h_{pump_3} = 13.7$

$Q_3 = 248 \cdot \sqrt{2.5 + 13.1} = 979 m^3/hr$

The volumetric flow rates are in the same order. Therefore we can conclude that the model script works correctly.

Moment rate

The moment rate is 6.25 times the total volumetric flow rate so:

$\dot{M}_{ballast} = 6.2 \cdot 10^{-3} \cdot (1017 + 985 + 966) = 18.55\, ton.m/s$

The moment rate of the ballast system is in the same order. Therefore we can conclude that the model script works correctly.

Moment

In the model the ballast moment is calculated as the integral of the moment rate over time. In the example of L.1 the pumps are pumping at constant rpm. If we take the average moment rate to be approximately 18.55 $ton.m/s$ for 300 seconds the total moment will be 5.5 $ton.m$. This also corresponds with the graphs.

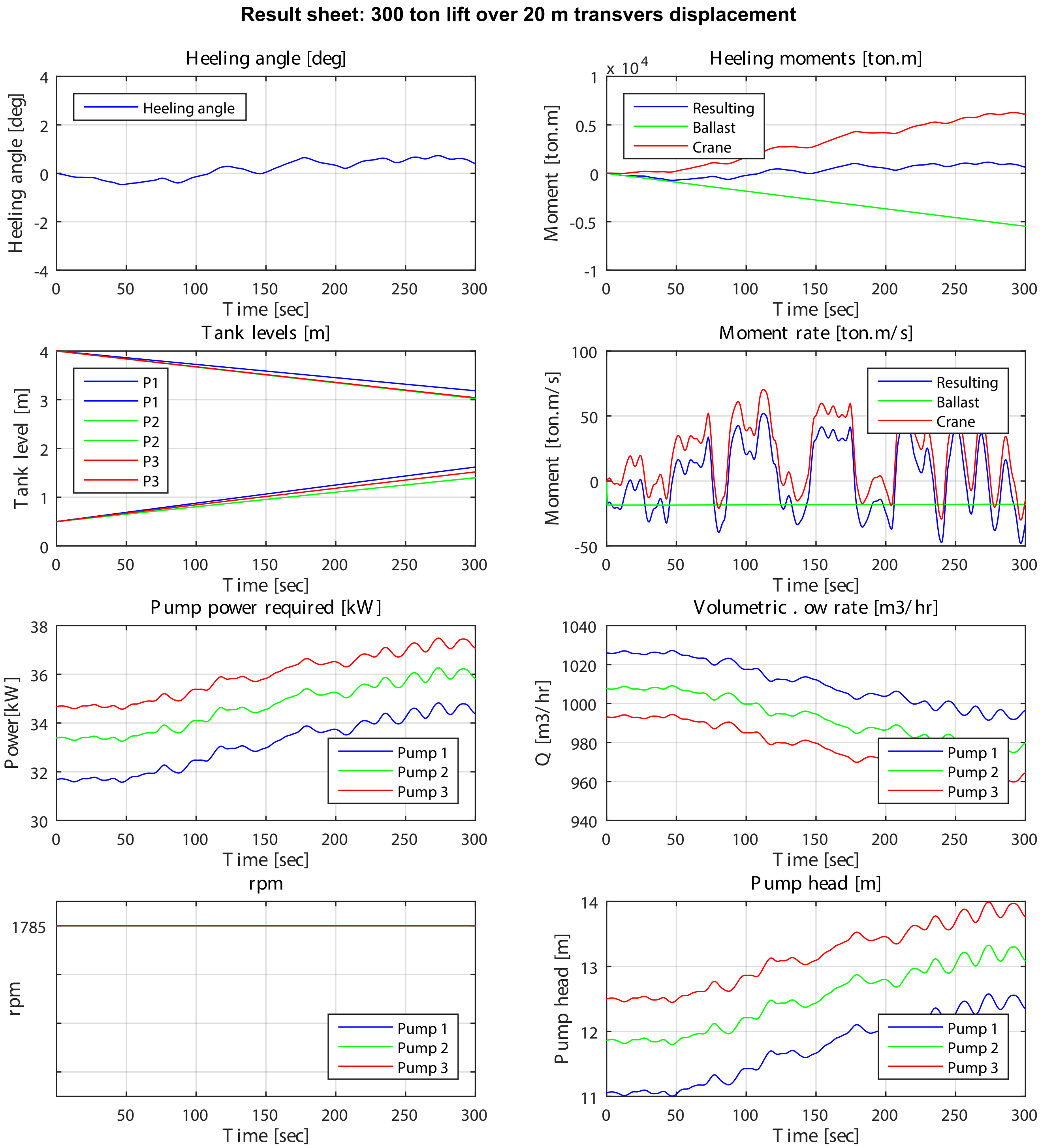


Figure L.1: 300 ton lift over 20 meters transverse displacement
Power required: 32 MJ

Appendix M

Dynamic ship response

This Appendix show results of the ship roll motions when the load is transverse displaced over 1 meter. The distance between the crane tip and the centre of gravity is 30 meters and the —$\overline{GM}_{eff}$ = 2.25 meter. The test is done for loads of 300, 600, 900 and 1800 ton.

300 ton

Figure M.1 shows the roll excitation when 300 ton is displaced in 5 seconds. The graph shows that the ship will roll to a maximum of 0.9° and will slowly damp out to the static equilibrium of 0.4°.

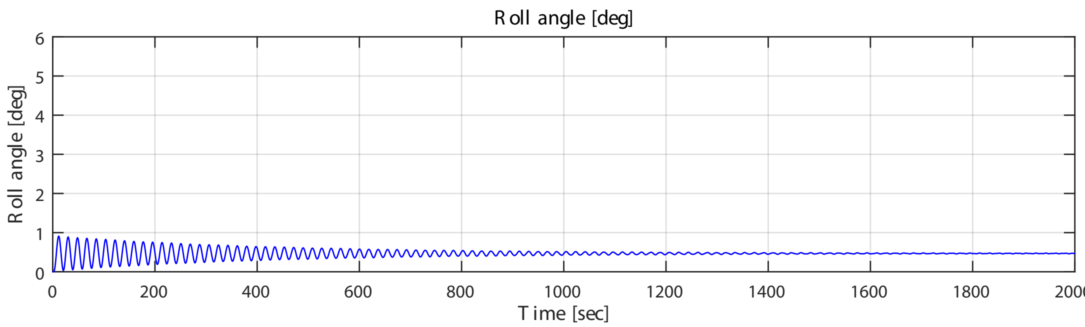


Figure M.1: 1 meter transverse cranetip displacement in 5 seconds

600 ton

Figure M.2 shows the roll excitation when 600 ton is displaced in 5 seconds. The graph shows that the ship will roll to a maximum of 2.3° and will slowly damp out to the static equilibrium of 1.3°. The amplitude of the oscillating roll motion starts with 1° around its static equilibrium. This is about twice the static heeling angle. This shows that the vessel is very sensitive to the impulse caused by the load.

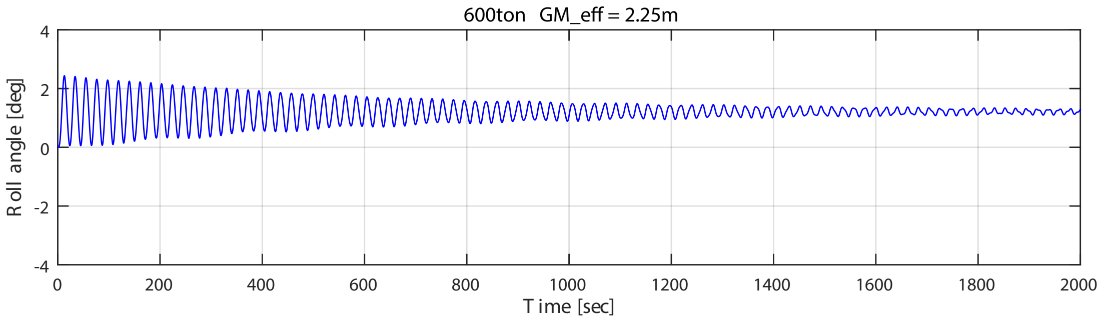


Figure M.2: 1 meter transverse cranetip displacement in 5 seconds

Figure M.3 shows the roll excitation when 600 ton is displaced in 20 seconds. The graph shows that the ship will roll to a maximum of 1.7° and also will slowly damp out to the static equilibrium of 1.2°. The maximum excitation is 0.6° lower than that the load is displaced in 5 seconds.

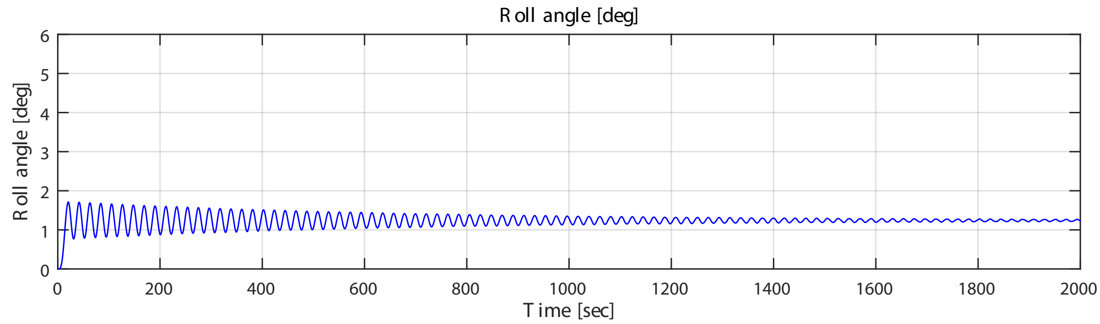


Figure M.3: 1 meter transverse cranetip displacement in 20 seconds

Figure M.3 shows the roll excitation when 600 ton is displaced in 30 seconds. The graph shows that the ship will roll to a maximum of 1.4° and also will damp out to the static equilibrium of 1.3°. The maximum excitation is 0.9° lower than that the load is displaced in 5 seconds.

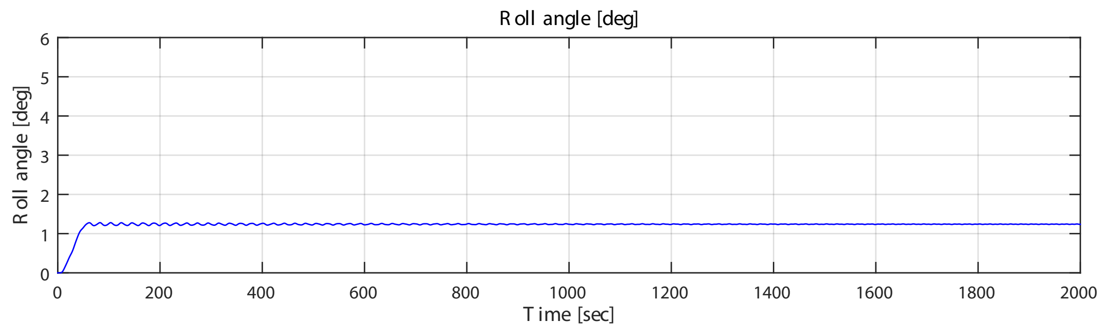


Figure M.4: 1 meter transverse cranetip displacement in 30 seconds

The figures above show that the ship will have larger roll excitations when the crane load induces higher moment rates.

900 ton

Figure M.5 shows the roll excitation when 900 ton is displaced in 10 seconds. The roll excitation goes over 5°. This clearly shows that a $\overline{GM}_{eff}$ of 2.25 meter is not sufficient.

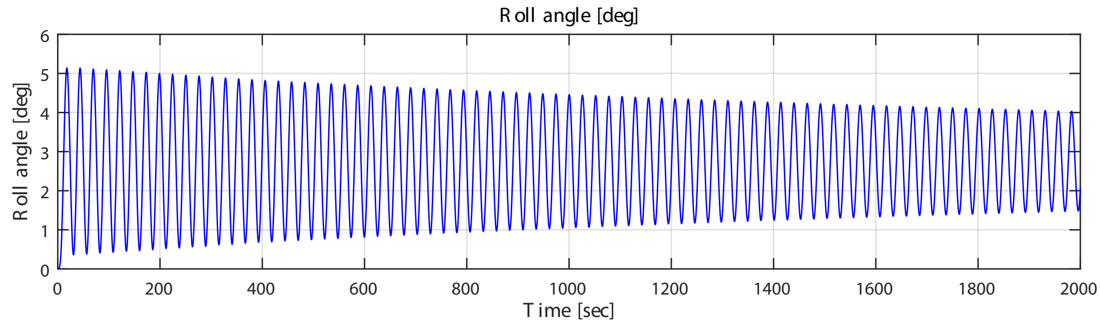


Figure M.5: 1 meter transverse cranetip displacement in 10 seconds

Figure M.6 shows the roll excitation when 900 ton is displaced in 30 seconds. The maximum roll excitation is 3.3°. This is 1.9° smaller than when the 900 ton is displaced in 10 seconds. But with a static heeling angle of 2.8° the ship is very sensitive to the hanging load. The stability properties have to be improved to ensure a safe lift.

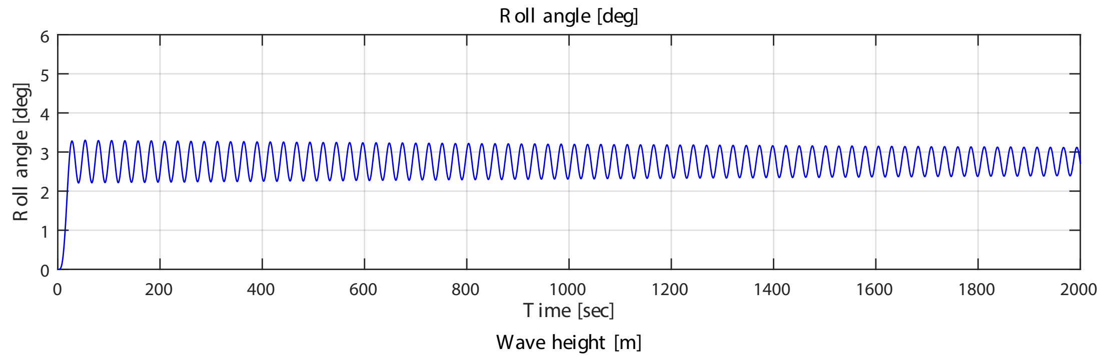


Figure M.6: 1 meter transverse cranetip displacement in 30 seconds

1800 ton

Figure M.7 shows the roll excitation when 1800 ton is displaced in 10 seconds. Obviously the ship is not suitable for 1800 ton lifts without stability improvements.

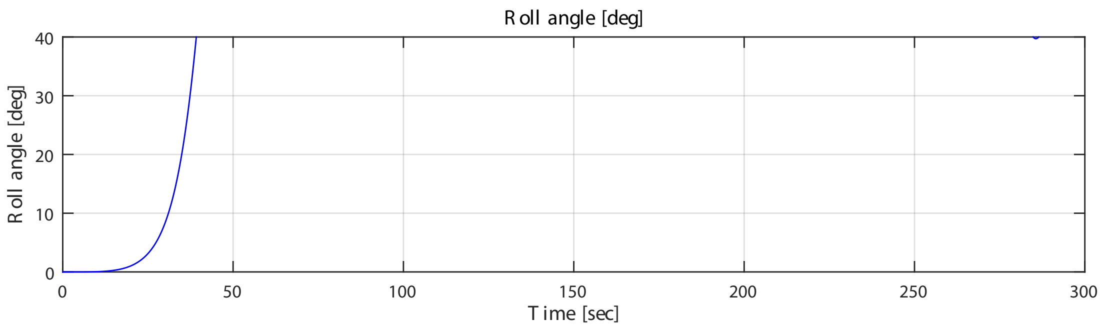


Figure M.7: 1 meter transverse cranetip displacement in 10 seconds

Appendix N

Dynamics of crane vessels

The information provided in this appendix is based on chapter 7 of Compliant Offshore Structures by Minoo H Patel and Joel A Witz [17]. In this appendix the complete vectors and matrices of the equation of motion introduced in 7.3.3.

The equation of motion with 9 degrees of freedom of the coupled motions of vessel and crane load can be written as

$$M'\ddot{Z'} + B'\dot{Z'} + \left(K' + K_c\right) Z' = F' e^{i\omega t} \tag{N.1}$$

With the corresponding vectors and matrices:

$$Z' = \begin{bmatrix} X & Y & Z & \theta_x & \theta_y & \theta_z & X_c & Y_c & Z_c \end{bmatrix}^T$$

$$F' = \begin{bmatrix} R(\omega) \\ 0 \\ 0 \\ 0 \end{bmatrix}$$

$$M' = \begin{bmatrix} (M + M_A) & 0 & 0 & 0 \\ 0 & m_L & 0 & 0 \\ 0 & 0 & m_L & 0 \\ 0 & 0 & 0 & m_L \end{bmatrix}$$

$$B' = \begin{bmatrix} (B_R + B_V) & 0 & 0 & 0 \\ 0 & b_1 & 0 & 0 \\ 0 & 0 & b_2 & 0 \\ 0 & 0 & 0 & b_3 \end{bmatrix}$$

$$K' = \begin{bmatrix} (K + K_V) & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}$$

$$K_c = \begin{array}{cc} & \begin{array}{ccccccccc} X & Y & Z & \theta_x & \theta_y & \theta_z & X_c & Y_c & Z_c \end{array} \\ \begin{array}{c} X \\ Y \\ Z \\ \theta_x \\ \theta_y \\ \theta_z \\ X_c \\ Y_c \\ Z_c \end{array} & \begin{bmatrix} k_x k_t & 0 & 0 & 0 & z'k_t & -yk_t & -k_t & 0 & 0 \\ 0 & k_y + k_c & 0 & -z'k_c & 0 & x'k_c & 0 & -k_c & 0 \\ 0 & 0 & k_z + k_t & y'k_c & -x'k_t & 0 & 0 & 0 & -k_t \\ 0 & -z'k_c & y'k_t & r_x + z'^2k_c + y'^2k_t & -x'y'k_t & -x'z'k_c & 0 & z'k_c & -y'k_t \\ z'k_t & 0 & -x'k_t & -x'y'k_t & r_y + (x'^2 + z'^2)k_t & -y'z'k_t & -z'k_t & 0 & x'k_t \\ -y'k_t & x'k_c & 0 & -x'z'k_c & -y'z'k_t & r_z + x'^2k_c + y'^2k_t & y'k_t & -x'k_c & 0 \\ -k_t & 0 & 0 & 0 & -z'k_t & y'k_t & k_t & 0 & 0 \\ 0 & -k_c & 0 & z'k_c & 0 & -x'k_c & 0 & k_c & 0 \\ 0 & 0 & -k_t & -y'k_t & x'k_t & 0 & 0 & 0 & k_t \end{bmatrix} \end{array}$$

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